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Master Thesis

Structural Optimization of a Cylinder Head Segment of the Layout of the Stiffeners and of the Walls Thickness With Focus on Weight Optimization Including the Evaluation of Different Design Variants

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Abstract

In this thesis work, a structural optimization activity of the cylinder head structure focused on the weight reduction was performed. The analysis revolved around a single cylinder section of a V8 bi-turbo gasoline engine, equipped with direct injection, currently developed In Porsche Engineering. First, a benchmark activity with other 12 competitor cylinder head segments was run, in order to assess its positioning on the market. Result showed that, despite the larger dimensions, the reference cylinder head was one of the lightest. Additional informations about particular features and design solutions were gathered. Subsequently, the reference segment was optimized using *Hypermesh* and *Optistruct* softwares, after conducing an optimizing investigation were weight and stiffness targets were defined. From the optimized shape, three different design variants were created, with weight reductions achieved up to 14.7%. The design revisions underwent then mechanical and thermomechanical analysis, to assess their performance compared to the original component. As performance indicators, the displacement of four important nodes were selected. The results of the analysis showed a general worsening of the deformations, up to 6%, for a cold condition, while reductions up to 26% were achieved under hot conditions. Finally, the guidelines for the design of future cylinder heads were given, considering the results of the benchmark activity and of the revisions analysis.

Chapter 1

Basics of the cylinder head design, analysis and manufacturing

IN this first chapter, a general discussion about the cylinder heads will be developed. A first introduction about the emission regulations and the role of the weight on the regulatory framework will be discussed. A review on the history of the design of this component, on the description of its parts and on some technical choices that can be implemented will follow. The analysis of this component will be then discussed, highlighting the different types of loads and simulations that can be performed, while a general overview on the manufacturing processes and constraints of the cylinder head will follow. A brief summary of the cylinder head development timeline together with the description of this thesis work will conclude this chapter.

1.1 Introduction

The concerns about important environmental issues and about the effects of air pollution on health have pushed in the last years the public opinion and the governments to set more and more stringent limits on vehicle emissions. For what concerns the air quality, *EURO* standards introduced by European Union in 1992 managed, in 22 years, to achieve great results. Reductions from 70 to 80 percent of all noxious emissions with peaks of 95% like for the particulate matter for diesel engines have been achieved [1]. At the moment, the focus is not any more on new technologies to further reduce the emissions but on the adoption of more realistic test cycles and test environments, more representative of the actual driving conditions. In fact, as of September 2017, the new *WLTP*[1] cycle has been phased in and will substitute officially the NEDC[2] at the end of 2020 in all tests for the vehicle homologation type

[1]Worldwide Harmonised Light Vehicle Test Procedure

[2]New European Driving Cycle

approval procedure. The NEDC was starting to be obsolete, since its last modification was in 1997. This new cycle differs a lot in different aspects: the testing time is increased from 1200 to 1800 seconds, implies larger vehicle accelerations, the average speed is higher and the vehicle weight is larger because also optional equipments are taken into account. Those factors lead to more representative vehicles which reflect in higher values of the noxious emissions and fuel consumption.

Since a great reduction of pollutant emissions has been achieved, the focus is now shifting more and more towards the CO_2 emission, which is not considered as a noxious one but it is the main actor in the *greenhouse effect*. Passenger cars alone are responsible for the 12% of the total carbon dioxide emissions in Europe, a percentage which rises to one fifth if the whole road transportation sector is considered. Despite the improvements in the last years of the engine technology and management, which helped to reduce fuel consumption, the CO_2 emissions continued to increase, due to the constant growth of the traffic and of the vehicle size. For this reason, the European environmental commission together with the ACEA[3] has implemented a new legislation, with different targets and different deadlines. At this moment, the limit is set to 130 grams per kilometre, but in 2021 it will be lowered to 95 $\mathrm{g/Km}$, which is a value 20% lower if compared to the average CO_2 emissions level of a new car sold in 2016 (118 $\mathrm{g/Km}$).

Differently from the noxious emissions, where all vehicles are forced to comply to the limits in order to be sold, the legislation has implemented a system based on *premiums*. In 2021, every OEM that will not meet the 95 $\mathrm{g/Km}$ limit for the average of the vehicles sold will have to pay a fine of 95€ for each CO_2 gram exceeding the limit multiplied for each car sold [2]. It is clear that this system can lead to large penalties for OEMs that fail to stay within the limits, so car manufacturers are spending a lot of money in research and development to find new technologies to reduce CO_2 emissions.

Carbon dioxide is strictly connected to fuel consumption and there is no after-treatment device which can limit its emissions; the main parameters on which it is possible to act are the following:

- **Electrification**: while full electric vehicles are useful to reduce the fleet average CO_2 emissions and to have local zero emissions, the hybridization enables more degrees of freedom which allow the engine to work in the optimal operating points, to recover energy from braking and in some cases to move the vehicle in full electric mode, thus reducing effectively the carbon dioxide emissions. This is the most effective and promising strategy, but the development of these technologies is really expensive, especially for the full electric vehicles.

- **Fuels**: fuels with a lower carbon to hydrogen ratio, like methane, lead to lower

[3] European Automotive Manufacturers Associations

emissions without negatively affecting the engine performances.

- **Vehicle characteristics**: aerodynamic efficiency and weight are two key factors. The first one reduces the power needed to motion at speeds above the characteristic one[4], while the second one reduces the inertia forces, thus decreasing the rolling resistance and the power needed to accelerate the vehicle. These two factors can be really useful to reduce the CO_2 emissions, because they are less expensive than the electrification strategy and require just a better design in the early stage of the product development.

In this thesis work, the goal will be to *reduce the weight of the cylinder head segment* of a V8 biturbo engine, by analysing the geometry and the wall thickness of a reference component, to save as much weight as possible without compromising its performances.

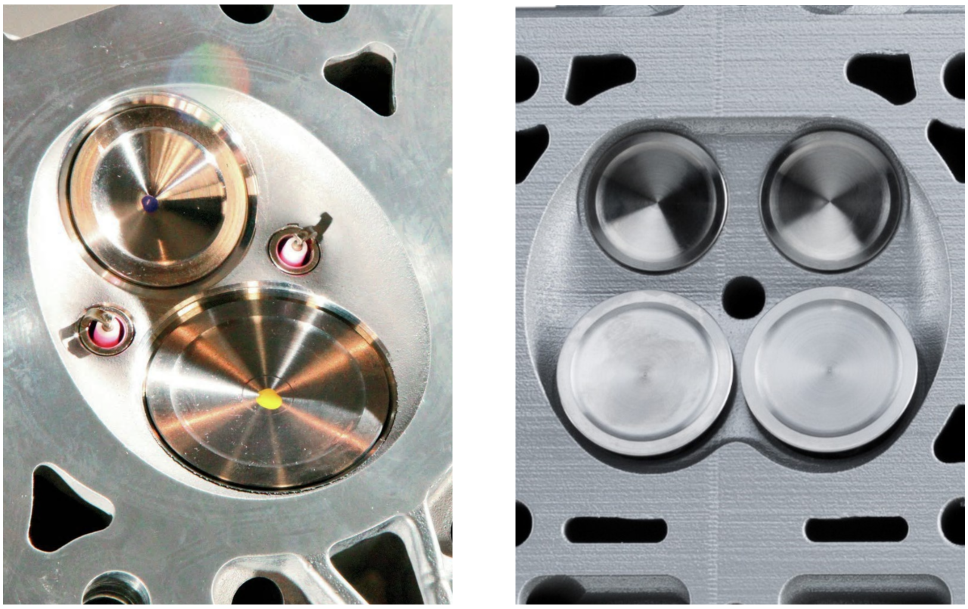
1.2 Design of cylinder heads

The cylinder head is a key component of the engine and it seats on the top of the cylinder block. It is separated from it by a *gasket*, which seals the gaps avoiding any leak of oil, coolant or, even worse, combustion gases. Its shape is similar to a box one, with an open top, which walls are then closed by an engine cover, usually made of aluminium or plastic. An additional rubber seal on the cover interface reduces the risk of leaks. It is in charge of the gas-exchange processes and for this reason the design of its internal parts affects the engine performances a lot. Exhaust ducts and intake ducts, which are integrated in it, need an optimal design and a proper validation, in order to allow an optimal exchange of the intake mixture and of the exhaust gases with the surroundings. It also supports the valvetrain elements, which are the responsible for the valve opening and closure, and its lubrication, in order to reduce parasitic losses and cool the components. The performance of this component is difficult to define with a single characteristic and usually different parameters are used. The most used ones are the *mass*, because a lightweight component can actively contribute to reduce fuel consumption, the *stiffness*, because misalignments in the drivetrain components should be avoided as much as possible, *optimal gas exchange* and *cooling capacity*, in order to reduce temperature in the most critical spots.

1.2.1 Components

Since this thesis work will deal with a gasoline engine, the following section will describe only spark ignited engines.

[4]It's the speed at which the aerodynamic resistance prevails over the rolling resistance



(a) *Hemispherical [3]* **(b)** *Pentroof [4]*

Figure 1.1: *Different typologies of combustion chambers*

This component can be subdivided in two parts, the *lower one* and the *upper one*. The lower one is composed by:

- **Combustion chamber**: this component is fundamental for a good combustion of the air-fuel mixture. The clearance volume is obtained almost entirely on the cylinder head, so pistons have a flat face or a domed one. The position of the spark plug and its distance from the cylinder walls affects the flame front speed. After the ignition of the mixture, a spherical flame front propagates and its rate of speed directly influences the energy release, thus affecting the efficiency, the power output and the noxious emissions. In addition, its design, together with the valve position and layout, affects the turbulence level which is a key factor for a good flame propagation.

  The shape which guarantees the best performances is the *hemispherical* one, displayed in Figure 1.1a, which allows the flame front to reach the mixture faster. This means that the energy is released in a shorter time, providing thus more power and better efficiency. In this case, the piston has a large dome to have the proper compression ratio (otherwise very long stroke would be required) and the combustion chamber at the TDC has a crescent moon shape. However, only two valves per cylinder can be fitted with this system because more valves on the same side would require different valve axes. This means a more complex distribution system and more expensive and difficult machining operations. For this reason, the most used shape nowadays is the *pent roof* one, shown in Figure 1.1b: it is a good approximation of the hemi one, allows more valve per cylinder because the valve axes can be aligned and the manufacturing costs are lower.

Particular attention must be paid also to the spark plug and its size. No threaded part should protrude into the combustion chamber, because the thread can be a perfect spot for some carbon deposits, which could then act as hot spots for the knocking effect and will make the removal of the spark plug more difficult too.

- **Intake and exhaust systems**: these components are in charge to control the gas exchange process. The mixture enters in the cylinder during the intake stroke through the *intake ducts* and the *intake ports*, which are then sealed during the other phases by the *intake valves*. After the expansion has occurred, the burnt gases exit the combustion chamber through the *exhaust ducts* and *ports*, flowing then downstream the *exhaust manifold*.

  The valves are usually made of high temperature strength steel alloys. They are composed by a *valve head*, which should be as large as possible to allow more air to enter in the chamber, and a *valve stem*, which diameter should be as low as possible in order to not compromise the gas flow in the ducts. The exhaust valves have a lower head diameter to allow the design of larger intake ports, so fresh air mixture can more easily enter in the combustion chamber. In fact, when the exhaust valves open, the pressure difference between the combustion chamber and the exhaust manifold is quite pronounced, allowing the exhaust gases to flow out without major problems. However, their stem diameter is larger, because they have to withstand higher temperatures. Sometimes hollow valves which contain sodium powder are used, because the powder motion inside the valve guarantees a more uniform temperature and a larger heat transfer from the valve stem.

  Valve seats are press fitted into the cylinder head to reduce wear due to impact and friction. It is also a critical component for the exhaust ports, because at least 75% of the heat transfer between valve and cylinder head happens through this interface, so they are made of the same valve material because the aluminium alloy would not be able to bear all these loads. Also *valve guides* are pressed-fit, in order to guarantee a perfect alignment of the valve stem and to increase the heat exchange. They are usually made of steel alloyed with some low friction metals like copper, in order to reduce the friction losses during the valve motion.

- **Water jacket**: Since some components face directly the combustion chamber, an adequate cooling must be guaranteed to avoid any excessive temperature

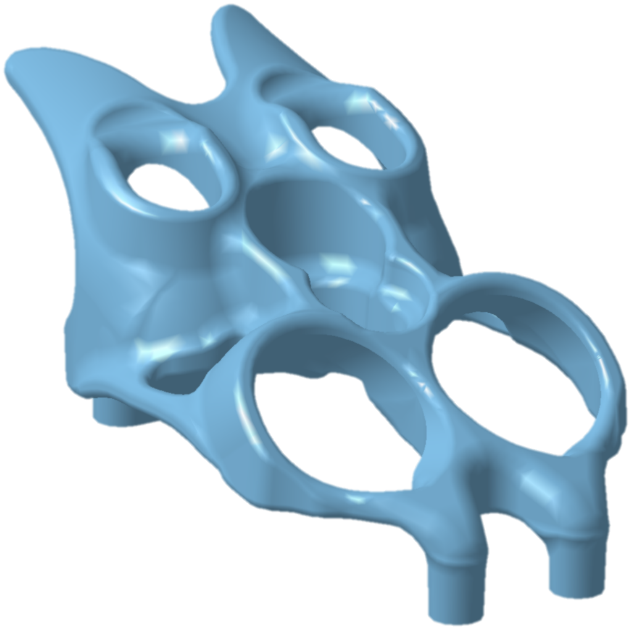


Figure 1.2: *CAD drawing of a cylinder head water jacket, seen from the intake ports*

increase which can decrease the mechanical properties of the material. For this reason, a flow of coolant surrounding those components which then transfers the heat to the engine radiator is mandatory (Figure 1.2). The study of the water jacket is really important for a proper engine performance and is done before the other components. Since the average temperature of those parts is well above the coolant boiling temperature, an accurate *CFD analysis* must be performed to avoid any flow stagnation and thus coolant evaporation. At the same time, overcooling must be avoided, because it's detrimental for engine efficiency. In addition, a proper coolant venting system must be designed, in order to avoid any coolant vapour bubbles to be entrapped in the water jacket, which can cause hot spots and big temperature rises. In the worst cases, this bubbles can reach the coolant pump, causing the flow to stop or the component to fail. In most engine designs, the coolant enters the cylinder block from a big opening on the side, flows into the cylinder head from some openings on the intake side and then returns back in the cylinder block on the exhaust side, before going to the coolant radiator and being pumped again in the cylinder block. In order to have an even temperature between the cylinders, the inlet should be positioned in order to have a ”parallel” flow layout between the cylinders, thus guaranteeing a more uniform cooling. This solution however is more expensive and comes with a penalty in terms of engine weight, so for small engines a ”series” flow layout is usually used.

The upper part instead is made by the following components:

- **Valvetrain components**: those parts are the responsible for the valve motion and they control their *lift* and their *timing*, like the camshaft and the hydraulic tappets. The different valvetrain layouts and mechanisms will be shown more in detail later, in subsection 1.2.3.

- **Camshaft supports**: the camshaft seats are machined directly on the cylinder head and need an adequate lubrication to reduce friction. For this reason, ducts are machined from the oil channels to the lower surface of the camshaft bearing, while a *bridge* running transversally between the two external walls supports it. The force necessary for the valve opening cause the camshaft to lift; that's the reason why main caps are bolted to constraint it during these phases. The main cap position is usually between the cams of every single cylinder, to have an even distribution of the forces and to reduce the bending moment on the shaft and on the cap itself, but sometimes it can be found shared *between* the cylinders, mainly for packaging reasons.

- **Oil ducts and drains**: In order to work flawlessly, all the moving parts need to be lubricated in order to reduce friction and wear. Oil is usually fed from one side of the cylinder head and reaches all the components through some channels, which position depends on the layout and on the choice of the drivetrain, but are usually included in the external walls or in the middle section. Sometimes some small orifices can be found close to those ducts, which spray oil where needed.

  The oil is then collected in the *oil deck* or *lower oil core* and drained towards the crankcase thanks to some ducts. Their position is fundamental for a proper oil recirculation, because any oil stagnation can cause oil thickening, ageing and slush, which decrease the lubricating performance of the system. The duct diameter must be carefully designed, taking into account also the unsteady air flow coming from the crankcase generated by the oil motion. That's why two separated ducts are usually designed, one for the air and one for the oil, with the air one ”inside” the V for V layout engines.

1.2.2 Design evolution

The history of the design of the cylinder head was not so straightforward as someone could imagine, but it evolved dramatically in the years thanks to the continuous improvements of the materials and of the manufacturing technologies.

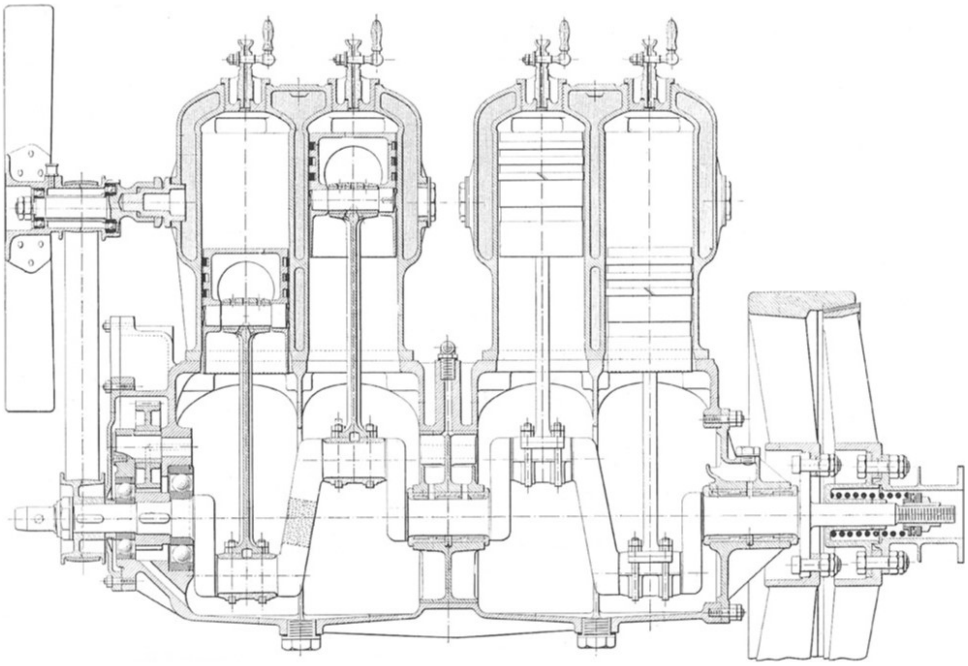


Figure 1.3: *A four cylinders engine of the 1910s featuring two blocks, with cylinders and head obtained in a single cast piece. [6]*

Good sealing materials did not exist at the end of XIX century, thus the problem of leakages was really important and difficult to solve. For this reason, first cylinder heads were integrated with the engine block in an unique casting. The hole for the igniting device and for the machining of the valve seats was obtained by drilling the component on its top, closing then the holes with some plugs. The sand core technology, still widely used nowadays, required small cores to be used because the absence of strong binders (only a mixture of silica sand, water and clay was used). This caused the cylinders to be often grouped in 2 or 3 per casting, so a single crankcase could have two or three cylinder blocks bolted on. In addition, all sand cores were oversized in order to add more strength to the elements and did not allow a good surface finishing. The first engine to use a separated cylinder head was the one fitted in the revolutionary *Lancia Lambda* in 1922.

The Lancia Lambda was also one first engines to display an *overhead camshaft* with *overhead valves*, enabling thus a more efficient design of the combustion chamber. In fact, intake and exhaust valves were usually implemented on the side of the cylinder head, actuated by some long rods which were connected to a camshaft located usually in the crankcase. This camshaft was put in rotation directly by the crankshaft by a set of spur gears, so the reason to locate it as closer as possible to the crankshaft is understandable. However, when the benefits of having overhead valves were

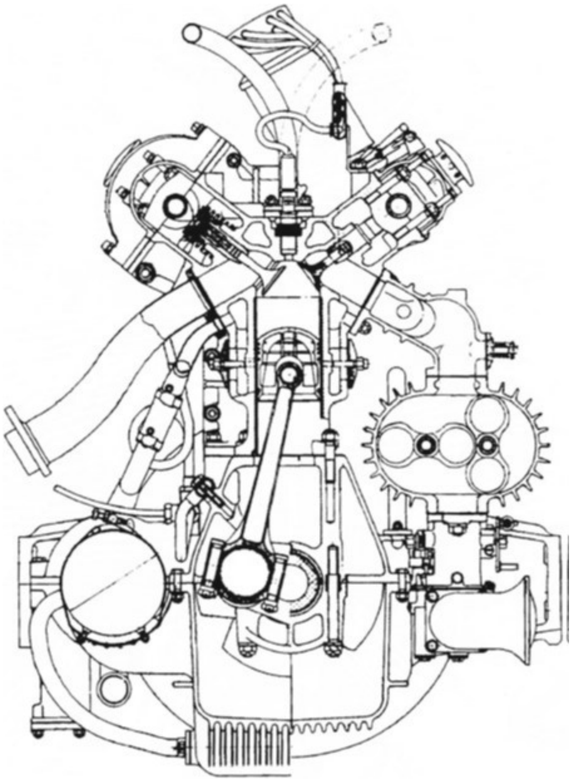
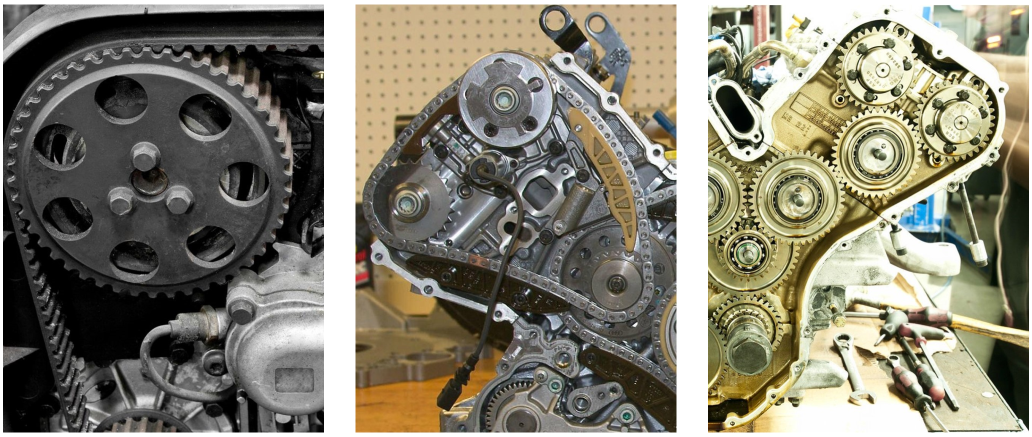


Figure 1.4: *Alfa Romeo 8C 2900 B engine, with inclined overhead valves and hemispherical combustion chamber [7]*

widely documented, every car manufacturer started to include overhead camshafts, connecting them to the crankshaft via spur gears or via chains.

The next major evolution step consisted in the adoption of inclined valve axes. Until the 20's, the valves were positioned parallel to the cylinder axis, both in the sides or over the cylinder. Later those years, after some researches conducted on engine combustion, it was discovered that a hemispherical combustion chamber would have increased the combustion efficiency, increased the power output and reduced the knock behaviour. In order to do this, the poppet valves should have been inclined to better follow the combustion chamber shape. First applications were solely on aircraft engines, but later it was applied also for race engines and then to high power sports cars. This allowed the power output to increase importantly; in 1937, the *Alfa Romeo 8C*, equipped with an 8 cylinder in-line supercharged engine shown in Figure 1.4, was able to reach about 62 HP/l, an astonishing result for that time. Parallel poppet valves however continued to be used until the 80's, but for low performance mass production engines only. A particular solution which was adopted in the early '900 to reduce the noise generated by the valvetrain was the one so called *Avalve* engine,



(a) *Belt [8]* **(b)** *Chain [9]* **(c)** *Gears [10]*

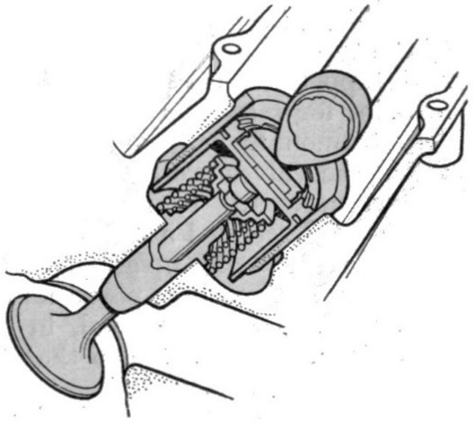
Figure 1.5: *Different types of distribution systems*

patented in 1912 by Panhard and Levassor. Avalve is a french word which means ”without valves”. This particular engine relied on a sliding sleeve around the piston, which acted also as a cylinder liner instead of a standard valvetrain system. This sleeve was connected with a set of gears to the crankshaft and moved at half of its speed. During its motion, the sleeve opened and closed some ducts which acted as intake and exhaust ports, allowing thus the gas exchange. This engine was claimed so quiet that it was impossible to hear it idling, but at the cost of great friction losses. It was used until the late 20’s, when the overall improvements on the valve machining processes and on the component tolerances reduced drastically valve tappet noise, nullifying thus the positive aspects of the sleeve system.

1.2.3 Valvetrain

The term *valvetrain* is used to refer to the assembly of those parts which are in charge of the valve motion. Everything is ruled by the *crankshaft*, which makes the camshafts rotate by different means, shown in Figure 1.5:

- **Toothed belt**: made of elastomeric material and reinforced with steel or kevlar wires, is the most common solution because of its lower price. Requires a periodical maintenance (usually every 100 thousands kilometres) (Figure 1.5a)
- **Metallic chain**: is usually used in more expensive cars because it is a costlier solution. It doesn’t need any maintenance and controls better any thermal dilatation of the system, but it’s noisier than the chain (Figure 1.5b)
- **Set of gears**: used in high performance engine or in racecars, where a perfect



(a) *Hydraulic tappet [11]*

(b) *Roller finger follower with hydraulic lash adjuster [12]*

Figure 1.6: *Valve clearance control mechanisms*

control of the drivetrain system and a high level of reliability are required (Figure 1.5c)

The camshaft needs a rotational speed half of the crankshaft one, since the valves must be actuated every engine cycle and not every engine rotation, so the number of teeth is twice the crankshaft gear one. In order to convert the rotational motion of the cam into a linear one, different mechanisms can be used; the most simple one consists in the cam hitting directly a *tappet*, but this system does not allow a proper control of the contact clearance, needed for thermal dilatation, and contact shims which are on the top of the tappet must be periodically replaced because of the constant wear. There are some systems which allow a complete control of this clearance; the most important ones for this thesis work, shown in Figure 1.6 are:

- **Hydraulic tappet**: the working principle is the same of a standard tappet, but a hydraulic mechanism takes care of the clearance control, pushing always the tappet against the cam. The tappet can be divided in two chambers: the upper one, which is constantly fed with oil and the lower one, which is connected with the first with a non return valve. When the cam moves the tappet, the check valve is closed and the oil acts like a rigid body since it can be approximated as incompressible, moving then the piston and the valve stem. Because of small leakages, the pressure in the second chamber decreases when the valve returns in the rest position, thus the check valve opens and refill the second chamber, pushing the tappet interface against the cam. In fact, despite the oil pressure being high, the force it generates is not enough to overcome the valve spring preload, which keeps the valve closed, and reflects the motion on the tappet

- **Roller finger follower**: this system is completely different with respect to the previous one and relies on a lever system instead of a direct contact between valve and cam. An *hydraulic lash adjuster* acts as a fulcrum of the lever, which is constituted by an element with a "finger" shape at which opposite end the valve tip is connected. The cam presses this lever between the fulcrum and the valve tip; in order to reduce friction as much as possible, a *roller bearing* is fitted on the contact point in order to have a rolling contact instead of a sliding contact, which dramatically reduces the friction losses. This mechanism enables different advantages, like the use of lower preload values for the valve springs thus improving the mechanical efficiency (the force is applied between the fulcrum and the valve, which reduces the force needed by the camshaft to open it). In some cases, the roller bearing is not present for packaging or weight reduction reasons and a machined contact interface is used, which however needs more lubrication (sometimes additional oil jets are used to reduce the friction).

At the tip of the valve a *spring retainer plate* is fitted, held by a *valve keeper*, on which the preload of the spring acts. An *oil seals* surrounds the valve stem close to the spring interface, avoiding any oil leak into the intake and exhaust ports.

1.2.4 Layout and technical choices

The choice of some technical solutions with respect to some others can lead to major differences in the cylinder head layout.

Direct injection This technology, which is used more and more in new engines, changes the layout of the component. In the most recent ones, the direct injector is placed in the central channel close to the spark plug, positioned towards the intake valves for packaging reasons. While the advantages in terms of mixture formation and in packaging are evident, the presence of another element in the central channel tends to increase its thickness, thus leading to a possible weight penalty.

Another common position for the direct injector is below the intake valves. Usually found in engines which also have a port fuel injection system, this position requires space below the intake ports, leading to issues in the intake design and in the engine packaging. In both cases, on order to respect the new particulate emissions standards for gasoline engines, a GPF[5] must be fitted.

[5] Gasoline Particulate Filter

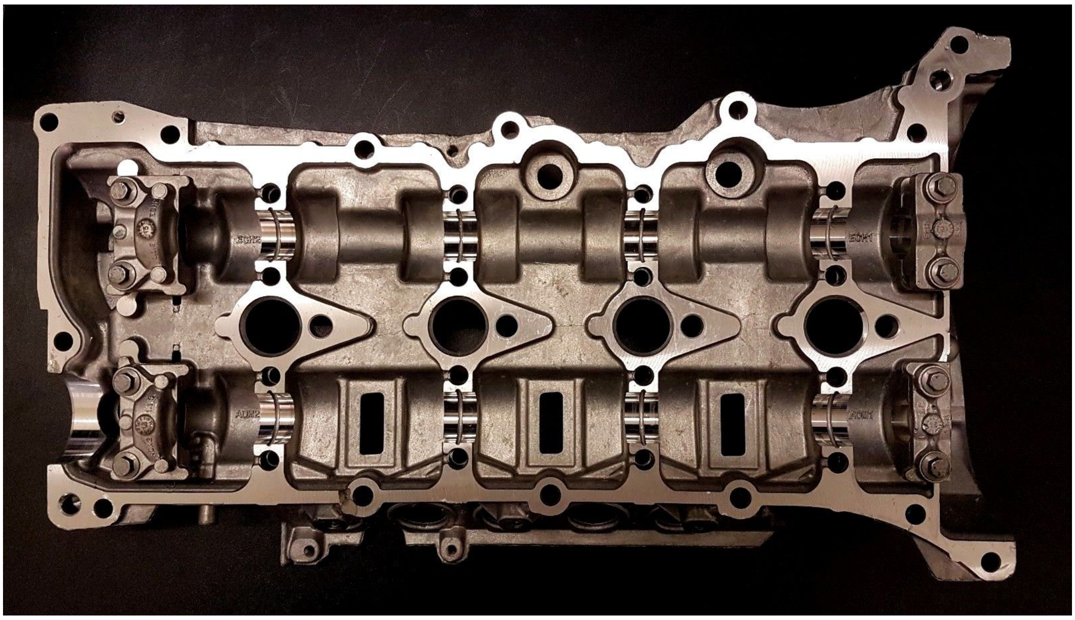


Figure 1.7: *Top view of a cam carrier of a 2.0 litres diesel engine*

Cam-carrier solution As explained before in the subsection 1.2.3, the cylinder head supports the camshaft with bridges which connect transversally the external walls. Another solution is to *split* in two parts the cylinder head, whith one part responsible for the gas exchange process (valves, ports, combustion chamber, ...) and the other one, called *cam carrier*, which supports the camshaft and its bearings (visible in Figure 1.7). This solution offers huge benefits during the assembly process, with the cam carrier which can be assembled on a different line and then clamped on the cylinder head at the end of the processes, shortening thus the total time for the assembly operations, the handling of the component and the number of operations. In addition, working on a single component instead of a complete engine assembly is much easier. This technical choice enables also a reduction in the cylinder head size: the head bolts which are responsible for the coupling of the cylinder head with the cylinder block can be positioned closer to the combustion chamber and the camshafts can be aligned over them without any particular issue, enabling a large flexibility in the assembly processes; at the contrary, for a standard layout, this position of the camshafts would require a more rigid workflow, with the cylinder head forced to be clamped before the valvetrain assembly.

The major drawback of this solution is a lower rigidity of the component; the stiffness usually decrease, since the part is not cast in a single instance, so careful attention on the geometry must be applied, with the placement of stiffeners and ribs.

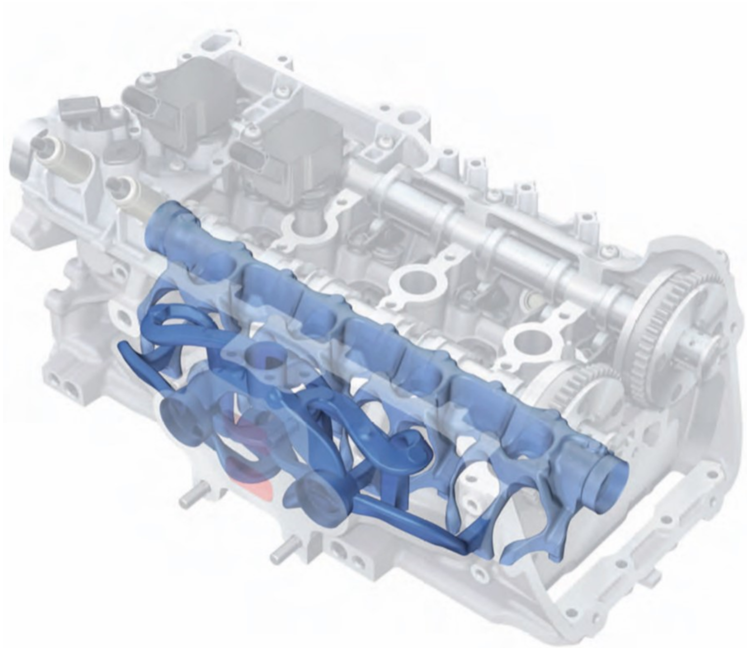


Figure 1.8: *Integrated exhaust manifold with the dedicated cooling channels highlighted in blue [13]*

Integrated exhaust manifold The more and more stricter CO_2 emissions regulations have forced the OEM to use more and more downsized engines with a large boost pressure. In order to keep the turbine components working, an acceptable temperature must be guaranteed throughout all engine operating points, which should never exceed 950 °C[6]. In order to do so, air to fuel ratios below the stoichiometric one are used for high load conditions, in order to cool down and protect the turbine components. To reduce the fuel enrichment, the *integrated exhaust manifold* is becoming more and more used. The exhaust ducts are integrated in the cylinder head, surrounded by the engine coolant which helps to subtract heat (Figure 1.8). For this reason, the turbine is flanged directly on the cylinder head external wall. The advantages are numerous: less components are used (no more exhaust manifold and less high temperature resistant screws are needed), the machining operations are reduced and the mixture enrichment is less frequent. In addition, despite the bulkier cylinder head, some OEMs claim a weight reduction of 3 kg with respect to a standard steel cast manifold and of 1 kg if compared to a standard steel sheets manifold.

For what concerns the drawbacks, the cooling circuit must be oversized in order to waste the additional heat, the water jacket dimensions are larger, since it has to surround also the exhaust ducts, and the casting process complexity of the cylinder

[6]The latest turbines can reach a maximum temperature of 1050 °C

head increases. [14]

1.3 Loads and analysis

The understanding of the different loads to which the cylinder head is subjected to is fundamental for the design of the component and the simulation of its performances. In this section, a short description of both is done.

1.3.1 Loads description

The loads acting on the cylinder head can be subdivided in two major categories: the *mechanical* ones and the *thermal* ones

Mechanical loads

- **Bolts**: they are used to clamp the cylinder head to the engine block and it's the major static force which is acting on the component. The compression force is distributed on the bolt interface, which is usually connected with ribs to the external walls. Some components use studs instead, and in this case the force is no more a compression one but a tension one instead.

- **Combustion pressure**: the combustion, together with the compression phase, obviously generates a pressure load on the combustion chamber and on the valves, which is then distributed on the valve seat. The values are not so large as the bolt ones, but they are varying in time depending on the crank angle and especially on the engine load, which creates problem for the fatigue life of the component.

- **Gasket**: this element is interposed between the cylinder head and the cylinder block; for this reason it is the responsible for all the forces exchanged between those two components. Usually the gasket is made of thin metal sheets, coated with a rubber-like material, that have some bead patterns on them. When the cylinder head bolts are tightened, the beads are pressed together and their contact seal the components from a gas or coolant leak. These patterns can be of two types: *half bead*, when just one side has a bead pattern and the other is flat, and *full bead*, where the two layers are beaded. In order to avoid any gasket failure during the assembly phase, a *stopper* element is fit in between the two sides. The stopper is made of a thick steel ring, which thickness equals the maximum allowed one for the gasket correct functioning, which stops the cylinder head during the assembly phase. For this reason, the stopper is the element on which more pressure is carried, followed by the full bead and then the half bead.

- **Valvetrain**: the loads deriving from this assembly are numerous and are varying in time depending on the camshaft angle, since it is a dynamic system. The most important ones are the force generated by the *valve spring*, the force of the hydraulic lash adjuster on its seat and the reaction force on the main bearing cap of the camshaft. However, as a first approximation, the spring and the lash adjuster forces should balance with the main cap one, leading to a null contribution on the constraint forces.[7]

Thermal loads The effects of the temperature loading are the most important ones. The main heat source is the combustion, which happens at the end of the second piston stroke and through the third one. Instantaneous combustion temperature values can reach up to 2000 °C, but the average temperature value in the cylinder is much lower, since the exhaust gases are expelled in the exhaust phase and fresh ”cold” mixture is fed during the intake phase. The situation is more different for the exhaust manifold: this component is fed with hot gases only, thus keeping its temperature really high. However, while the peak temperature is much lower than the combustion one, the average value is usually higher (maximum 950 °C as said in section 1.2.4).

In order to avoid any damage to the component, the coolant flows in the water jacket around the most critical sections (valve bridges, exhaust ducts and valves, spark plug channel, ...). This allows to lower the temperature to values which are less critical, but high thermal gradients lead to large stresses, so a lot of attention must be paid. Additional cooling is provided by the oil in the oil deck surface, even if its temperature and cooling power are lower than the coolant one.

Those high temperatures have mainly two detrimental effects. The first one is a general worsening of the mechanical properties of the material, which Young modulus and ultimate tensile strength decrease with a temperature increase. Secondly, the material expands with temperature, which leads to high stresses and plastic deformation in the most critical areas, like sharp edges, fillets, chamfers and close to constraints. Additionally, cycles of high temperature followed by ambient one lead to variable strains which are the initiators of a surface crack leading to a fatigue failure of the component.

1.3.2 Analysis overview

The finite element analysis of the cylinder head is a fundamental step in the product design and testing, to shorten the time to market and to drastically reduce

[7]This is true for static analysis, because for a dynamic analysis also the inertia and the motion of the components must be considered

costs. There are mainly *three* different types of simulations which are done during the component design:

- **CFD[8] simulation**: in order to obtain the maximum performance from the engine, an accurate analysis of the fluid motion inside the engine is mandatory. First of all, a CFD analysis of the intake and exhaust ports are mandatory to have the *optimal gas exchange process* which heavily impacts on the engine power output. Intake port design should provide the maximum possible volumetric efficiency and an adequate turbulence level inside the combustion chamber, while the exhaust ports have to be designed to allow an ideal exit for burnt gases, in order to reduce the amount of residuals.
  Secondly, but not less important, CFD simulation has a fundamental role in the evaluation of the *thermal behaviour* of the component. Indeed, the fluid motion and its turbulence level heavily affect the heat transfer coefficients and thus the temperature field of the component. The water jacket shape, for example, is determined after different iterations between simulations and design changes. Optimal cooling must be provided considering as input the amount of heat generated by the combustion and the one to be dissipated through the heat exchangers. Additionally, oil flows are analysed for cooling reasons (they extract heat from the oil deck interface) and for lubrication purposes, because a certain pressure level must be guaranteed at the camshaft contact interface. Important output results of these simulations are the *convective heat transfer coefficients*, which are very useful for the next type of analysis.

- **Mechanical and thermal loads simulation**: an evaluation of the effect of the mechanical forces on the component is mandatory. The primary goal is to retrieve informations about *stresses and strains*, to find weak spots on the geometry and to reinforce them, in order to preventively avoid any failure. This kind of analysis is also important for estimating the *gasket* behaviour during engine running conditions. Simulations with different load patterns varying in component temperature, loading conditions and firing cycles are performed together with a contact analysis between the gasket and the cylinder head, to see if the minimum contact pressure is always above the minimum threshold sealing value.

  Using as input the results of the CFD analysis, *thermal expansion* effects are analysed. Material dilatation is a critical factor which has to be carefully evaluated, since it can give rise to large local stresses which could diminish the component reliability. In addition, bolt preload analysis has to be run for the ”hot” condition, since a large cylinder head deformation could increase the bolt preload force above the yielding point, generating thus reliability issues.

[8] Computational Fluid Dynamics

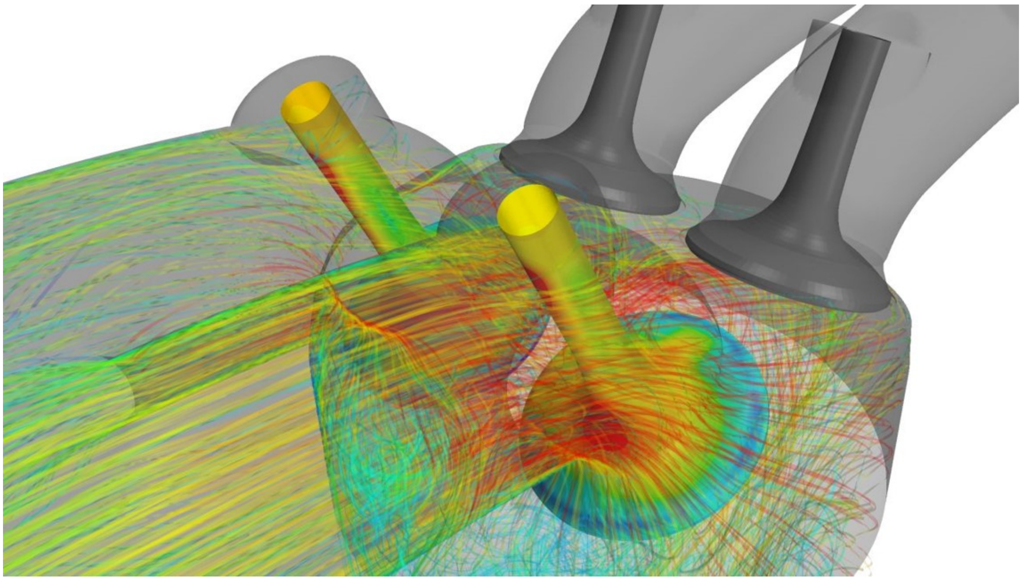


Figure 1.9: *CFD analysis with streamlines view of intake ports [15]*

Another fundamental aspect to be analysed is the *fatigue behaviour* of the component. This kind of analysis can be subdivided in two big sections: the *high cycle fatigue* and the *low cycle fatigue*. The first one is done to understand the effects of the cyclic mechanical loads on the component which are induced by the drivetrain: the spring forces, the loads on the camshaft bearing caps and the ones on the hydraulic lash adjusters are crank angle dependent and vary in time. The magnitude of these loads is not so large but the number of cycles the component has to withstand is really high, so a high cycle fatigue with an infinite life target is performed, showing as a results the areas where design changes are required in order to achieve the minimum safety factor. To conclude, a low cycle fatigue analysis is also performed. It is run in order to understand the effects of the recurring thermal dilatation on the component, so from a cold condition to a steady state hot one and back. This continuous change in temperature causes the material to expand and then to shrink, leading to cracks formation and propagation on the surface of the component. To have a complete investigation, a *thermomechanical fatigue analysis* can also be performed, which introduces in the analysis also the effects due to creep and oxidation. This last analysis however is very expensive and requires an accurate *damage model* and precise material properties, which are difficult to obtain. In both cases the analysis is run with a life target of finite number of cycles, which is however more than enough for the effective life of the vehicle.

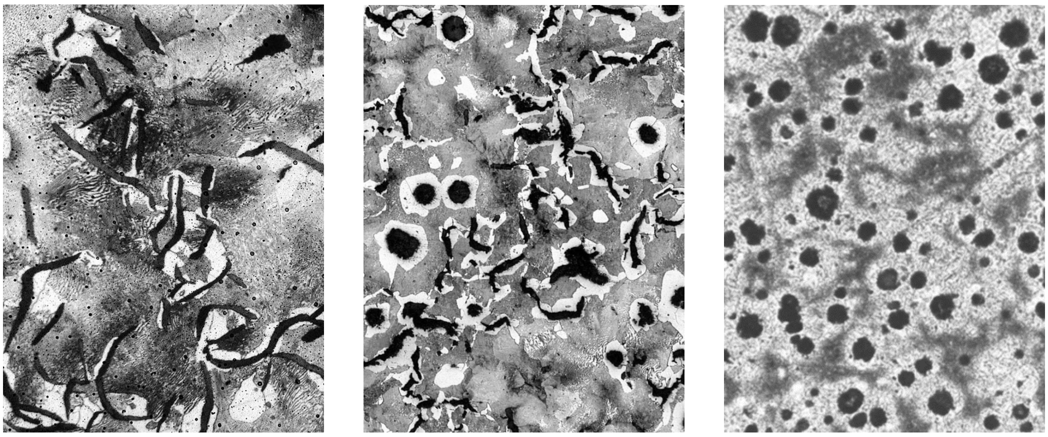
- **NVH simulation**: the cylinder head is subjected mainly to the drivetrain forces and the combustion pressure force, which do not contribute a lot in the noise generation. However, its ”box shape” together with the engine cover and the manifolds creates a perfect structure which amplifies vibrations and radiates noise. In order to reduce these negative aspects it is possible to act at two different levels. The first one is on the *packaging* of the component, so on the position of the bolt supports, on the spring interfaces and on the oil ducts position. Those components are defined in the alpha stage of the product development, so previous experiences and previous researches activities have to be applied in order to correctly define the component layout, which cannot be modified later in the project. The second level in which it is possible to act is the *local* one, with the addition of ribs and with a wall thickness increase, in order to locally stiffen the critical areas. [16]

1.4 Materials

The numerous requirements in terms of weight, mechanical properties and thermal behaviour of the component put stringent demands on the material choices. The ideal material for cylinder head manufacturing should be lightweight, with a high tensile stress, with a high thermal conductivity coefficient, easy to cast and machine and resistant to high temperatures. Obviously this material does not exist and the materials mainly used nowadays are *cast iron* and *aluminium* alloys, which embed some of the requirements listed above

1.4.1 Cast Iron

Different types of alloys can be used, with carbon content percentages ranging from 2, called usually *ductile iron*, to 4 percent, called *grey iron*. Anyway, the mechanical properties depend not only on this percentage but especially on the material *micro structure*. In both cases, the carbon present in the alloys precipitates under form of *graphite* instead of *carbide*, as it happens for steels, but this precipitation can be controlled with the use of different additives. For what concerns gray iron, the presence of silicon enables the control of the graphite precipitation in small flakes (figure 1.10a): the higher the content, the finer is the precipitation. The resulting alloy is cheap, has great compression and shear strength thanks to the flake structure and easy to cast and machine; unfortunately those flakes act also as crack initiation sites, thus decreasing the tensile strength of the material. For the ductile iron instead, an addition of magnesium to the alloy allows the graphite to precipitate in small spheres (figure 1.10c); this results in higher tensile strength, but at a cost of more difficult machining operations and of a more expensive material.



(a) *Gray [17]* **(b)** *Compacted [18]* **(c)** *Ductile [19]*

Figure 1.10: *Different types of cast iron for cylinder head production*

Another factor affecting the mechanical properties of the alloys is the *solidification process*. Iron alloys should be solidified in a temperature window, which is between the *graphite eutectic temperature* and the *carbide eutectic temperature*. If the temperature of solidification drops below this last limit, carbide particles are formed instead of graphite ones: while those last particles are able to deflect a crack, thus increasing the tensile strength of the material, the carbide ones can be passed through. Several alloying elements, like copper, nickel and cobalt, are added in order to improve castability and increase this temperature window. However, some other elements which are added to increase the mechanical properties, like chromium and molybdenum, shorten this window and increase machining difficulties.

Another type of iron alloy widely used is the *compacted graphite* one. Thanks to the addition of magnesium, it is possible to control the shape of the graphite precipitations. The particles shape can be defined as an intermediate one between the gray iron and the ductile iron ones: it is usually described as a worm shape or a noodle structure, with the graphite particles which resemble to the flake ones but with a shorter length and a much larger thickness (figure 1.10b). This pattern gives a great mechanical strength, which is the double of the gray iron one, but it was not widely used in the past for casting process control reasons: too much magnesium and the graphite precipitates as small spheres, too little and the graphite structure reverts to the flake one.

1.4.2 Aluminium alloys

Aluminium alloys have become more and more relevant in the automotive field, especially for the internal combustion engine components, for their lower weight and good mechanical properties. Nowadays, cylinder heads for automotive applications are almost all made in aluminium, relegating cast iron alloys to heavy-duty and some diesel application only. Despite the fact that purchasing price of aluminium is larger than steel, it is less expensive to handle and work. Its lower melting temperature requires less energy to cast it and enables the use of permanent molds and die casting techniques, reducing thus also the mass production costs. Additionally, the machining costs are much lower, tool life is increased and aluminium alloys can be recycled to a high degree. In addition, it has a much larger thermal conductivity, making it ideal for cylinder head applications, where it is necessary to dispose of a lot of heat coming from the combustion process. The main drawbacks are lower mechanical properties, lower stiffness, poor wear characteristics and higher tendency to creep and oxidation phenomena. Additional attention must be paid also in coupling steel or iron components with aluminium ones (like a cylinder head with steel bolts): steel has a thermal expansion coefficient which is more or less the half of the aluminium one, which can cause interferences and additional stresses if tolerances and deformations are not considered during the design phase.

Almost all aluminium alloys relevant for automotive engines contain *copper* (up to 5%) and *silicon* (up to 18%). If alloy castings are let cool down slowly, the aluminium grains are separated by large silicon areas, which decrease the mechanical properties a lot. For this reason, the castings should be rapidly cooled down (*quenched*); in this way the silicon remains finely dispersed, thus increasing hardness. Later, the components are heated again to a certain temperature, in order to let the copper exit from the solution but to remain still finely dispersed. Those thermal treatments and their effects will be explained later in this section. Other elements usually added in aluminium alloys are *lithium*, which is the most effective way to improve the Young modulus, *Scandium*, which helps increasing toughness and thermal properties.

1.4.3 Thermal treatments

Aluminium alloys usually display poor mechanical characteristics if compared to steel ones. For this reason, *thermal treatments* are mandatory in order to increase strength and toughness of those alloys, making them attractive for automotive industry. The main ones to which they usually undergo are *solutionizing* and *precipitation hardening*.

Solutionizing is very simple: the purpose is to put in solution as much as possible alloying elements that are present in the alloy. In order to do so, after the casting process, the component must be heated at a very high temperature, in order to have

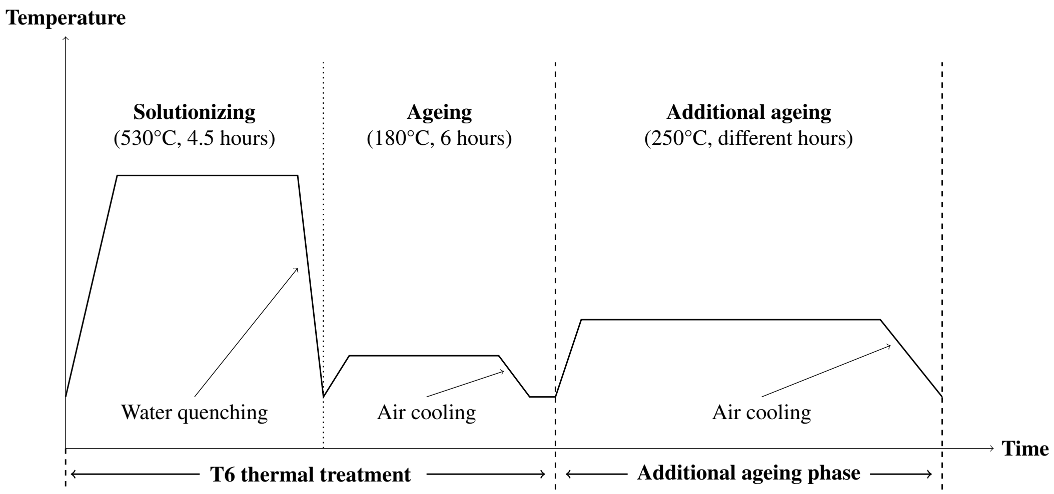


Figure 1.11: *Time evolution of a T6 thermal treatment plus an additional ageing phase. The additional ageing time depends on the final desired material properties [20]*

a saturated solution in a single phase only. In order to keep this level of saturation without any precipitation, the component has to be cooled down quite fast, in order to "freeze" the crystalline structure in this single phase; if the cooling is too slow, the alloying elements will precipitate, nullifying the positive aspects of the solutionizing. This last process of rapid cooling is called *quenching*. The only problem with this process is that the solutionizing temperature is quite close to the liquidus one, where aluminium grains can start to melt. For this reason, an accurate control of the process is required, with systems which can guarantee an uniform air temperature in the furnace. In addition, if the part has varying cross-sectional thickness, it is necessary to have an uniform temperature through the section, in order to avoid only local solutionizing or, even worse, local grain melting.

Precipitation hardening is another process which is done after the solutionizing. Also known as age hardening, it consists in the precipitation of a second phase exploiting the phenomenon of the supersaturation. After the quench, the components are usually heated again to a high temperature, but well below the liquidus one. Since the alloy is in a supersaturated condition, this re-heating causes part of the alloying elements to precipitate in an amount which depends on the ageing temperature and on the phase diagram of the considered alloy. These small precipitated particles have the beneficial effect to limit the movement of the dislocations and of the defects in the alloy crystalline structure, which are the main carriers of the plastic deformation. For this reason, precipitation hardening is really effective in hardening the material, increasing its elastic working window. This is why a quenching process after the solutionizing is needed. If the component was let to cool down slowly, all the alloying elements would have already precipitated; quenching and ageing allow to control the amount of alloying elements trapped in the crystalline structure and the ones precipitating, thus achieving the best mechanical properties possible. However, ageing time should be evaluated carefully, because over-ageing causes a loss of strength of the material. Some recent researches have revealed that an additional ageing process after a T6[9] thermal treatment is very effective in increasing the fatigue life of the aluminium alloys. This happens because the alloy enters in a over-ageing status, but a ”stable” one, which reduces thus the over-ageing effects due to the varying temperature cycles. It has to be noticed that this cause a reduction of strength and, thus, an increase in deformation. [20]

1.4.4 Polymers and magnesium: future challenges

The use of exotic materials is becoming more and more common in order to pursue the maximum weight reduction as possible. The one which is actually under the focus of numerous researches is *Magnesium*. The production use of this metal has been limited up to today to some small engine components only, like manifolds or covers, or has been limited to racing engines. In this last case, the high cost and the low resistance to corrosion of this material are not a concern. The interest it is generating as cylinder head material is due to the fact that has a great weight saving potential without impairing mechanical strength. Some estimates have suggested that a reduction of up to 25% in weight can be achieved, assuming a cylinder head entirely cast with this material [21]. The main limits of Magnesium are currently corrosion resistance, casting control, structural stiffness, machining, little knowledge about its fatigue properties and temperature usage limits.

A lot of interest is also building around *polymers*. Plastic materials have already revolutionized the engine manufacturing, substituting metals for components like engine covers, manifolds, casing and ducts. However, they never found a wide use as structural components. Several attempts were made in the past; one of the most famous is the *Polimotor* engine created by Matti Holtzberg in the 80’s. His first attempt was on a Ford 2.3 litres engine, where engine block, cylinder head, valve spring retainers, timing gears and intake valves with ceramic heads were entirely made of fibre reinforced composites. Reciprocating parts however were still made out of metal. A second attempt was made in 1984, where also timing gears, piston skirts, rings connecting rods, intake valve stems, push rods, rocker arms, wrist pins, cam followers, turbo casings and impellers were made out of plastic material [24]. In order to prove the effectiveness of this project and that polymers could be used as a substitute of metals, the engine was fitted on a Lola T616 HU04 race car (Figure 1.12) and run some races on the IMSA championship, scoring also a

[9] Solutionizing followed by an artificial ageing

Figure 1.12: *Lola T616, fitting the Polimotor 1 [22]*

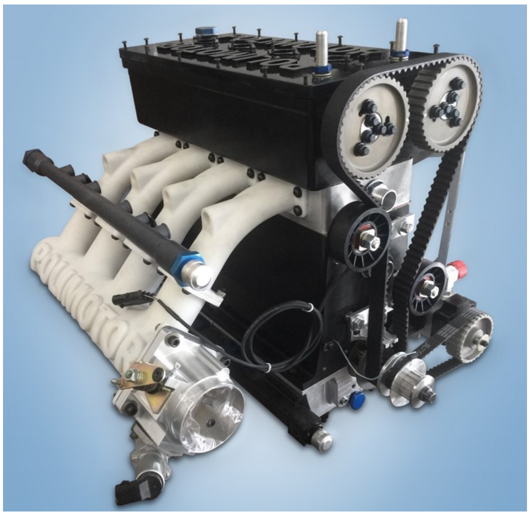


Figure 1.13: *Polimotor 2, now under development [23]*

podium. That engine weighted only 76 kg compared to its ”metal” counterpart which weighted 158 kg [24]. Holtzberg later also patented different production processes and techniques, showing that those polymeric parts could not only withstand the heavy loads to which they were subjected to, but also that those parts could be also easily manufactured. Nowadays, 3D printing technology and the accumulated knowledge on plastic materials could push the boundaries of the weight reduction even further without sacrificing mechanical resistance of the components, especially for the cylinder head. In fact, Holtzberg is currently developing a new version of the Polimotor, shown in Figure 1.13, called *Polimotor 2*, with the support of numerous chemical companies [23].

1.5 Casting processes

1.5.1 Sand casting

The main process through which cylinder heads are obtained is *sand casting*. Despite being an old technology, this is still the most widely used, because it allows to obtain complex shapes with undercuts and hollow cavities. A schematic view of the parts used for this process is shown in Figure 1.14. Once the drawing of the component to be cast is defined, a steel pattern recreating its exterior features is obtained. Then a *mold* is created. This mold is the negative part of the component to be cast and is made of a mixture of sand, silica, clay and water. This mold is usually divided in two parts, called *cope* and *drag*, which allow to remove the metal pattern

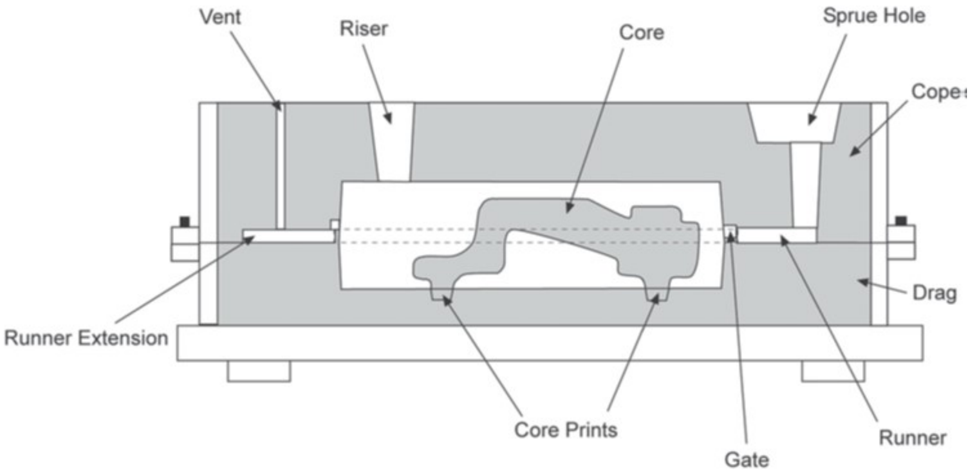


Figure 1.14: *Schematic of the mold for sand casting and its parts [25]*

used to create the mold itself. This division of the mold is called *parting line*.

Interior cavities are then obtained using *cores* obtained from sand and an adhesive binder. For what concerns cylinder head, cores are used to obtain *intake* and *exhaust ducts*, *water jacket*, *combustion chamber* and *oil deck surface*. This last core is usually split up in more parts (usually two, the *upper* and the *lower* oil cores) which division is ruled by the parting line of the cope and drag. Sometimes steel cores can be used if they can be easily drafted from the component. Steel cores can be used numerous times and provide better local cooling of the cast, increasing thus the mechanical properties; however they are much more expensive, so they are only used if strictly necessary. For example, for the cylinder head under exam in this thesis work, during the prototyping phase only the combustion chamber core was made out of steel, while all the other ones were obtained through sand.

The molten metal is poured into the mould through a *sprue* and reaches it thanks to some *runners*, which geometry is fundamental for a high quality casting. Some vent holes prevent air bubbles to remain entrapped into the mold, while a riser accumulates additional molten material to compensate for *shrinkage* and *porosity* as the casting cools down. Because of the molten metal entering the mold, particular attention must be paid to cores clamping, to avoid any core floatation which may cause casting defects and, in worst cases, a part to be scrapped. To avoid this, sometimes water jackets are held in place by some pins which are connected to the top side of the cope, which constraint any vertical floatation of the cores. Those pins leave some holes, which are then plugged later during the machining phase. Once the cast has solidified, the sand is broken away from the cast and the sand cores removed. This can be quite a difficult step for very complex geometries like the one of the water jacket; cleaning completely the cavity is fundamental to avoid any possibility of failure caused by

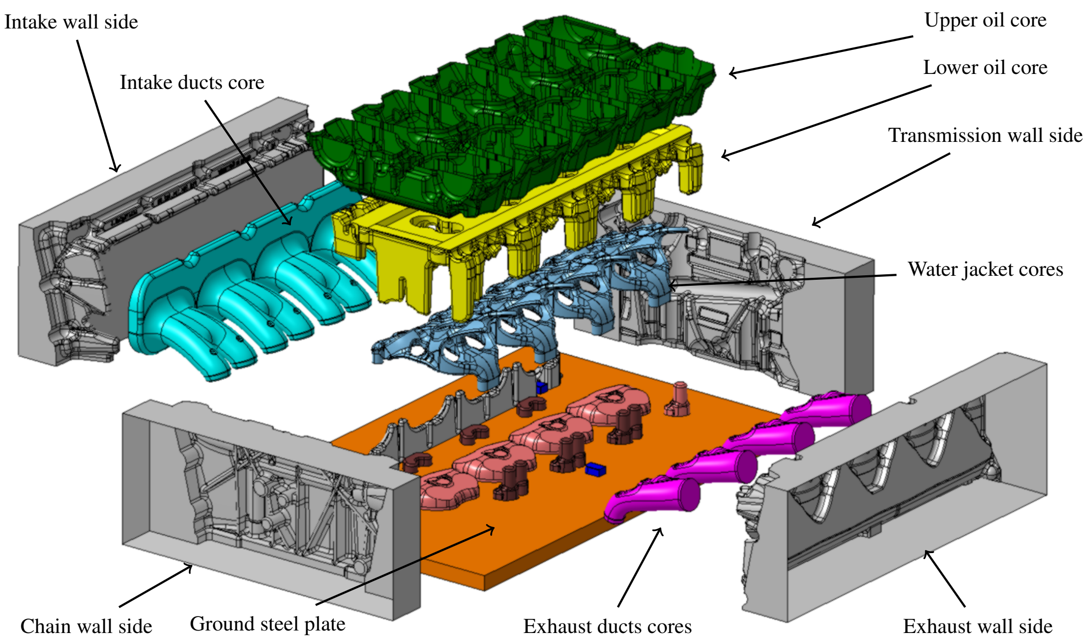


Figure 1.15: *Exploded view of the cylinder head casting cores and components of the cylinder head protagonist of this thesis work*

some sand remainders, like of the water pump in this example.

Casting process and the cooling of the part must be well designed and controlled. The gradual cooling of the cast induces in the component some thermal gradients, which can cause large residual stresses. Those stresses can be detrimental for the mechanical performances of the component, especially in critical areas. Counteractive actions consists in better design of the mould, change in the shape of the part, better position of the cooling channels and better design of sand cores. This last aspect not only affects residual stresses, but also overall mechanical properties. The sand binder, when in contact with the high temperature of the molten material, burns and creates noxious fumes, which are dangerous for the health of the workers and can reduce superficial strength of the component. A new inorganic binder has been developed by BMW, which guarantees faster cooling, almost zero noxious emissions, 50% increase in the tensile elongation before fracture and a 10% increase on the yield strength and ultimate tensile stress of the material [26].

1.5.2 Alternative casting processes

Some new casting processes have been developed in the recent years, but the principle and the functioning remains more or less similar to the sand one.

Permanent mold casting is very useful when production volumes increase a lot. If the cylinder head is designed to be in aluminium, the external walls of the cope and drag can be made out of steel, so they can be reused. Unfortunately this technique does not allow any undercut geometry and requires the parts to be drafted, but the surface finishing is very good and cooling ducts in the plates can be added to improve strength of the component in critical regions. This technique will be used for the mass-production of the cylinder head which is investigated on this thesis work (Figure 1.15), while for the prototyping phase only sand cores have been used.

High pressure die casting is another form of permanent mold casting. The molten material is usually gravity-fed, while in this process is injected under high pressure by an injecting machine. With this approach it is possible to achieve wall thickness which are really thin (up to 2.5 mm) and it is especially used for aluminium castings. Unfortunately disposable cores cannot be usually used, because the molten metal pressure would destroy them. In addition, the cast tends to cool really rapidly, causing porosity under the surface of the component, worsening thus the component mechanical properties.

Lost Foam casting is a new process which is rapidly growing for cylinder head manufacturing. An expendable polystyrene model of the cylinder head is produced, also by gluing multiple parts together for the most complex parts. This model is then coated with a refractory ceramic layer and then fitted in the mould where it is covered by sand. Finally, the molten material is poured on the polystyrene model, which causes its vaporization, replacing its volume with the metal cast. This process has the big benefit to have very good dimensional stability, which reduces costs of machining operations. The costs of this process can be lower than the ones of a standard sand casting if the casting is well designed and controlled. The largest disadvantage is that it is not possible to have cooled plates or cooling channels to improve mechanical properties of the parts.

1.6 Workflow

1.6.1 Generic cylinder head design

The design of a new cylinder head can be summarised in these following steps.

1. **Combustion chamber and valve shape definition**: First of all, after the first engine specifications in terms of bore diameter, performances, layout and packaging are defined, the first shape of the valves and of the combustion

chamber is determined. These could be carried over from older projects or can be designed from scratch, using best practices and research activities.

2. **Valvetrain**: the main characteristics of the distribution system are defined, like the camshaft position, the distribution system, the valve position and the valve angles with respect to the cylinder axis.

3. **Flange and ports design**: next step is to design ducts and ports, starting from the valve design defined earlier.

4. **3D CFD of gas exchange**: analysis of the gas exchange is then performed, to validate the geometries of the elements or to feedback some results to the designers to optimise them. In this last case, the workflow starts back again at point 1, until the required performances are not met.

5. **Bore diameter**: after the geometries are validated through CFD, some simulations are run to identify the ideal bore dimension, which provides the adequate performances and the necessary compression ratio while guaranteeing the layout dimensions defined earlier.

6. **Water jacket**: the water jacket is sized for maximum cooling power, together with the position of the spark plug and the fuel injector (if present) for the best combustion efficiency and performance.

7. **Walls position definition**: the cylinder head walls are positioned according to the packaging of these elements, offsetting their external shape.

8. **Inner package definition**: the position of the drivetrain elements is then defined, together with the distribution type and the oil feeding channels. Also the bolt position is chosen at this stage only, taking into account also the cylinder block and the crankcase geometry.

9. **Oil cores design**: starting from the elements defined before, the shape of the oil cores is obtained, considering also the addition of stiffeners and ribs. They will generate the geometry of the cavities of the cylinder head.

10. **External design**: finally, the external walls are designed by offsetting the oil cores and the elements defined before. In this stage, also the transmission and the distribution sides are designed, considering the packaging requirements.

1.6.2 Thesis work

The aim of this thesis work is to perform a structural optimization of the cylinder head structure, focusing on the wall thickness and on different layout choices with the goal of weight reduction. All the analysis and optimization studies of this work will be developed on a *single cylinder section*, in order to have the work the most focused as possible. The component to be analysed is part of an engine which is currently in its beta stage. The geometry of the ducts and of the water jacket are already the optimized ones, since they underwent to a study similar to the one of this thesis. For this reason, their shape will not be changed. In addition, also the drivetrain concept and layout has been already validated, so any change in the internal element position is not possible. Taking into account all these premises, the modifications can be brought only at the *oil cores* level, which geometry define the external wall thickness, the thickness between the water jacket and the oil deck, the shape and dimensions of the camshaft bridge and the supports of the different drivetrain elements. For these reasons, this work positions itself almost at the end of the cylinder head design workflow listed in the section before.

The workflow of this master thesis activity instead can be subdivided in the following phases:

1. First, a benchmark activity on some competitor cylinder heads will be run. The reasons for this activity are two:

    - **assessing** the characteristic of the component analysed with the ones of the competitors, to see its positioning on the market
    - **gathering** of technical solution and packaging layouts which could be useful to pursue the weight reduction objective

    The results of this analysis will be used to compare also the progresses and the final results obtained on the *reference* component with the various design variants.

2. Later, the optimization phase will start. In the first phase it will be necessary to understand the role of the boundary conditions and which ones to adopt, considering that the work will be developed on a single section. For this reason, some assumptions will be made. After the boundary conditions definition, the proper structural optimization with the softwares *Hypermesh* and *Optistruct* will start.

3. Once the optimized component shape will be obtained from the software, the result will be analysed, gathering as much informations as possible on the possible optimization spots. In this phase, manufacturing constraints will be considered, in order to obtain a final component which could be theoretically mass produced.

4. From the informations obtained in the previous phase and in the benchmark one, different *design revisions* of the component will be produced and subsequently

analysed. The results of the analysis will be used to evaluate if the modifications implemented are fine or are detrimental for the component performance.

Chapter 2

Benchmark activity of the cylinder head structures

EVEN if the design of the component has to take into account a lot of parameters and requirements, due to cooling, optimal inlet and outlet flows and manufacturing constraints that are listed in the previous chapter, there are a lot of degrees of freedom which enable the designers to find and try different layouts and solutions. In any case, the final goal is the reduction of as much as possible weight, maintaining however the structure stiff enough. Lower weight improves the fuel consumption and also cost reduction in raw material, which leads to indirect cost reduction in the manufacturing processes. Stiff cylinder heads are required in order to lower misalignments and deformations, which could decrease engine performance and life, and reduce the system vibrations, which could impact negatively on the vehicle NVH[1].

The biggest influence on the final design is given by the oil cores, since the other components are already defined. In fact their geometry and shape are optimized for the engine on which they will be operating, in order to provide the maximum possible performance. For example, on the cylinder head object of this thesis, the water jacket was completely designed from scratch and underwent to a lot of studies and simulations, with the goal of having an optimal cooling. The engine packaging and layout was not considered as a constraint and, for this reason, the oil core had to ”follow” the water jacket shape, without any possibility to have some optimization spots in that area. The oil core is the component which rules also the presence of stiffeners and ribs, that if well placed and designed, they can lead to a substantial stiffness increase. Oil cores also controls the thickness of the walls between the ducts, the water jacket and the side walls; it is the main parameter on which is possible to act in order to reduce weight, since the material removal is broadband

[1] Noise, Vibration and Harshness

instead of being localized in certain specific regions. However, this parameter is heavily influenced by the manufacturing process considered; usually a smaller wall thickness requires a higher quality casting procedure and control, which definitely increases the overall costs of the production, necessitating of special techniques and knowledge which not all suppliers have. For these reasons, a compromise between weight savings and manufacturing costs must be found. In the past, the geometry of this component was based solely on experience and best practices, without any analysis on the role different design parameters. The aim of this work is to investigate in which areas material is less needed, in order to reduce the component weight for the future projects.

2.1 Benchmark description and factors

Benchmarking is an activity which consists in comparing the standards of the company with the ones of a business which is considered to be the best in class, analysing its products and processes. For what concerns the products, it is usually done to improve their performances or quality, by testing the competitors ones, to understand in which areas is possible to improve; this usually happens with some performance analysis and, especially in cases like this one, with some destructive analysis, to have better understanding of the product manufacturing, geometry and assembly. In any case, companies should not rely solely on these kind of activities, because otherwise they will be always following the standards of the leaders, but they should implement it as a tool to support research and development of new designs and technologies.

The activity was carried out on 12 different cylinder heads mounted on different engines of the VW group. In order to have a ”proper” comparison with the component subjected to our analysis, all the engines considered are gasoline fuelled, by means of a direct injection or coupled also with an indirect one. In fact, compression ignition engines have to face different requirements, which are a larger peak combustion pressure ($250\,\mathrm{bar}$ against $130\,\mathrm{bar}$) but lower thermal loads and, thus, cooling power; they also differ in the geometry, since the combustion chamber is almost entirely obtained on the piston surface instead of being on the cylinder head. For these reasons, any comparison would be useless.

The benchmark was performed on CATIA V5 software, on a single cylinder, using the original drawings of the components. The workflow consisted in these following steps:

1. Import the geometry of the final machined part
2. Create two transversal planes, perpendicular to the engine bank, each one intersecting two cylinder head bolts axis, to use them as a reference for the

section cuts. The cylinder was picked always in the middle of the bank, to avoid any influence of the design changes at the drivetrain or at the gearbox sides

3. Compute the weight, considering the same material for all the components (AlSi7MgCu0.5 T6, the one used for the reference cylinder head casting, with a density of $\rho = 2.65\,\mathrm{Kg/dm^3}$)

4. Create some dynamic section planes to understand the geometry of the components and retrieve the informations about the thickness and the layout

The informations retrieved are the following:

- **Weight**: this is the focus of this thesis activity

- **Packaging**: necessary to analyse the effect of the overall dimensions on the final weight; all the dimensions will be referred in millimetres:

  – *D*: cylinder bore
  – *h*: section height, from gasket to engine cover interfaces
  – *l*: section length, from one engine cover side to the other
  – *w*: section width, distance between section planes
  – d_{cs}: distance between camshaft axis
  – d_f: minimum distance between intake and exhaust flanges
  – d_b: transversal distance between the bolt axis

- **Wall thickness**: necessary to learn more about their role on the final weight and the manufacturing processes used; all dimension will be referred in millimetres:

  – d_{bridge}: distance between the camshaft support windows and camshaft bushing
  – $t_{wj,c}$: wall thickness between the combustion chamber and the water jacket
  – $t_{wj,o}$: wall thickness between the water jacket and the oil core
  – t_{avg}: average thickness of the component, usually measured on the external walls, on the different ribs and on the various flanges
  – t_{ducts}: thickness of the walls of the oil ducts

2.2 Reference cylinder head

The cylinder head which has been taken as reference for the benchmark activity and on which the subsequent work of structural optimization and wall thickness analysis will be developed is shown in Figure 2.1. It is mounted on a 4.4 litres, V8, gasoline direct injected biturbo engine, with an angle of 90 degrees between the cylinder banks and with the air intake located in the middle of the V. The material used for the manufacturing of this component is an aluminium-based alloy called AlSi7MgCu0.5 and the castings are subjected to a T6 thermal treatment. The molten material is cast in a mould with steel side walls and cooled ground plate, while all the other cores are made out of sand. In the following pages it will be referred simply as the *reference* component.

The valve motion is guaranteed by a system of roller finger followers, which are actuated by two camshafts, which stay between the bolt axis and the outer walls; two longitudinal oil ducts feed the camshaft bearings and the hydraulic lash adjusters, which seats are machined between the ducts and the external walls.

The water jackets are not interconnected and have an optimized shape; in fact, the optimal shape was obtained through 3D CFD simulations starting from scratch, using as constraint the intake and exhaust ducts geometry only, so the cylinder head design revolved around these components more than the usual. The inlet is located close to the intake valves and is split in two circular ducts; the outlets are positioned on the exhaust side, with the major one shaped as an elongated hole and two smaller ducts, which allow the coolant to flow out from the upper flaps surrounding the exhaust ducts.

An additional longitudinal duct, called *secondary air duct*, running below the exhaust ducts and connected with it with some small channels, can be seen in Figure 2.3 in the bottom right. It is used in cold start conditions, when it is not possible to rely on the engine heat to evaporate the gasoline injected on the cylinder (or on the intake duct, depending on the injection type) and it is necessary to inject more fuel in order to have more volatile hydrocarbon molecules that can guarantee an acceptable mixture formation. In order to not increase too much the pollutant emission in the starting phase, which is the most critical phase during emission test, an electric pump feeds air to the exhaust manifold through this duct, guaranteeing in this way a stoichiometric mixture at the three-way catalyst inlet, which is a fundamental requirement to make this component work properly.

In between one cylinder and the adjacent one, it is possible to find two oil draining channels on the exhaust side and two crankcase ventilation pipes on the intake side. On each side, the channels are separated by a thin wall, which then forms a small rib connecting the external wall to the bolt channel (Figure 2.2).

The gasket interface is a standard one with a rectangular shape, only two pockets close to the vent channels are present.

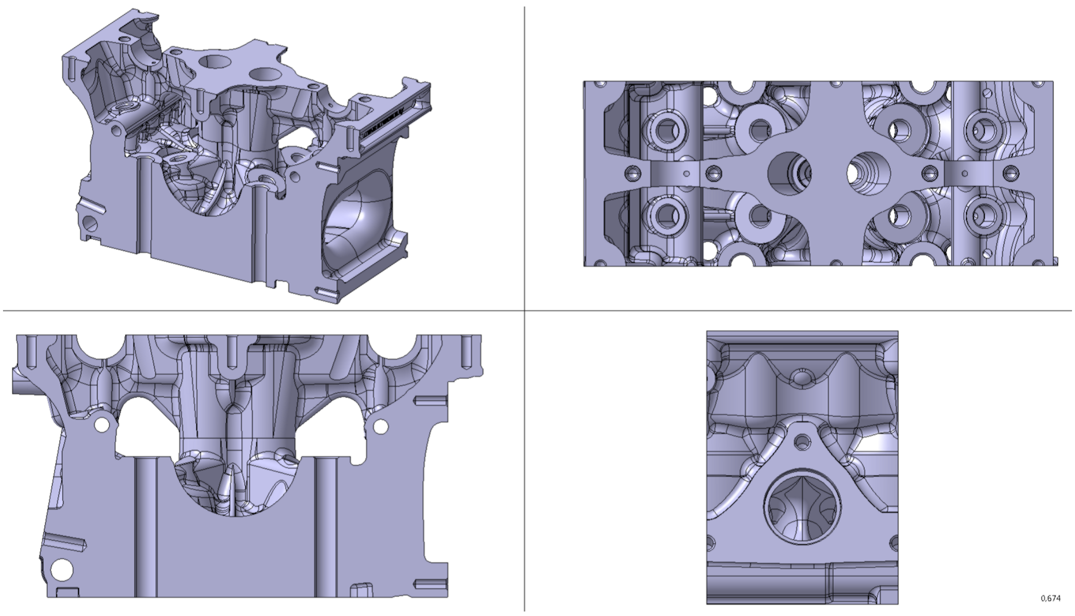


Figure 2.1: *Reference cylinder head*

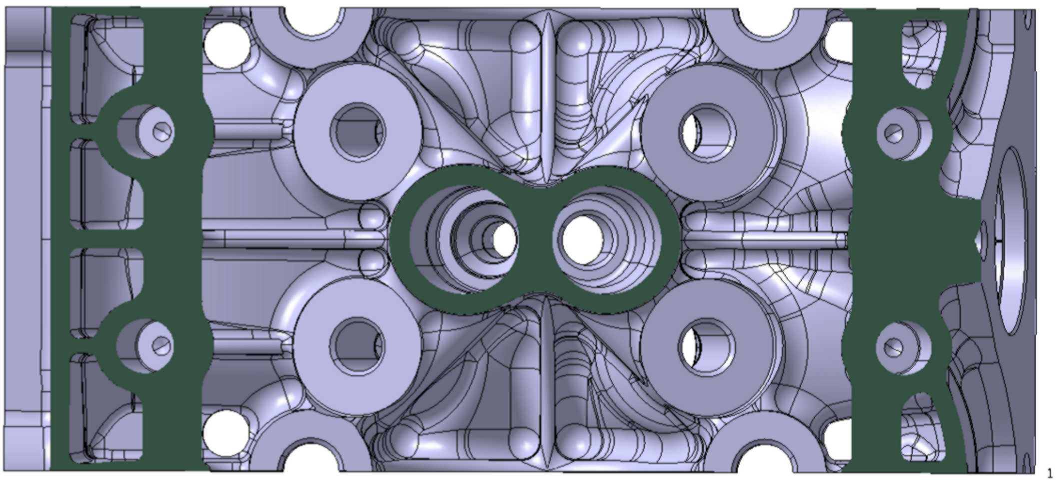


Figure 2.2: *Reference ribs layout*

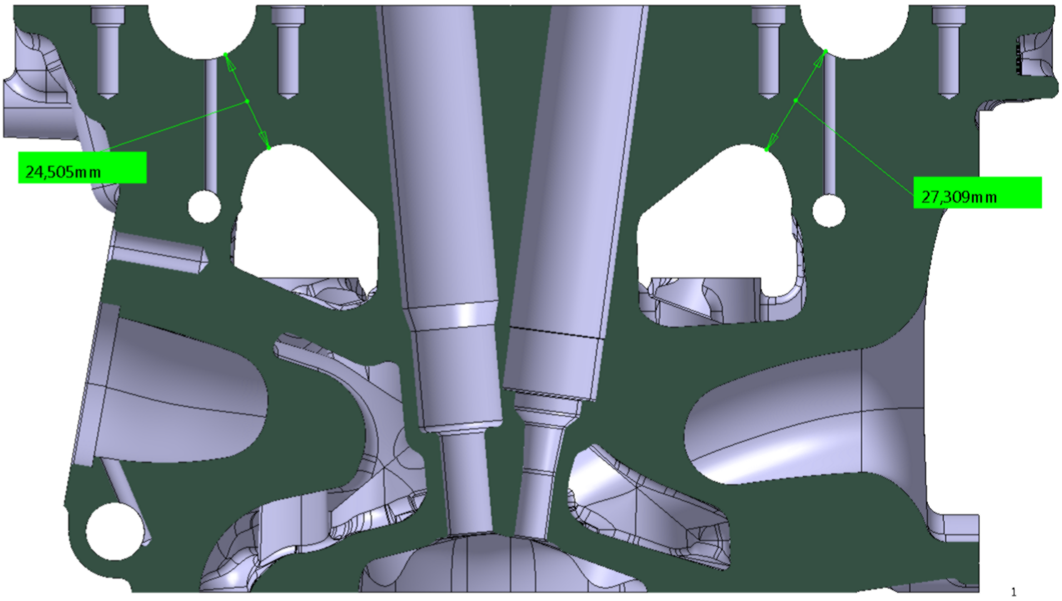


Figure 2.3: *Reference camshaft bridge section view*

For what concerns the rib layout, visible in Figure 2.2, the central channel is connected to the external walls by means of a thin rib, while very thick ones join it with the valve spring interfaces and then with the bolt ones, forming an X shape. To reinforce these last elements, additional material is added in this region, close to the central channel, increasing the stiffness but inevitably the weight too. A longitudinal thin rib connects all the central channels of the cylinders, while the bolts of one side are connected with the ones on the other side with transversal ribs, which are shared among the two adjacent cylinders. In the upper part of the component, it is possible to see an additional longitudinal bridge which connects all the cylinders and has a flat surface in order to host the cylinder head sealing and the engine cover. Some threaded holes are present too allowing the clamping of this last component.

Additional ribs can be found supporting the hydraulic lash adjusters. On the intake side, two of them are located at the sides of this component, while a single one connects the bottom surface of it with the intake manifold and then to the spring interface; on the exhaust side instead, two thick ribs act as support while there is no third thin rib since the component seats directly on the exhaust duct wall. Additional stiffeners are present also inside and outside the component close to clamping screws, like on the intake and exhaust manifold flanges, and below the oil duct on the intake side, following all its length.

The camshaft bridge section view is shown in Figure 2.3, where the distance between the camshaft bearings and the windows is displayed. It is possible to notice how bulky is the region around the central channel and also how thin is the wall

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
4.4 l	V8	88	140	248.3	98	155.6	205.4	96	2685 g

Table 2.1: *Main characteristics of the Reference cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
25	7.5	5	4	4

Table 2.2: *Thickness values of the Reference component*

separating the injector from the spark plug (0.6 mm), which requires a precise and careful machining. In the bottom right, under the intake flange, there seems to be a lot of material, but instead it's just a section of a rib stiffening a large pocket used to remove material in a region where it is not needed, supporting also the intake manifold flange.

For what concerns the wall thickness, the values are a bit cautious for the water jacket and oil core separation, while the average thickness is pretty low, requiring thus a high quality casting. In any case, despite the big dimensions of this component (Table 2.1), the weight is quite low.

All the component characteristics, dimensions and thickness will be compared with the values of the other components in section 2.4.

2.3 Competitors' component

2.3.1 Competitor A

The component analysed, represented in Figure 2.4, which characteristics can be found in Table 2.3, is mounted on a 6-cylinder turbocharged boxer engine with a total displacement of 3 litres, supplied with a direct injection system. This engine is equipped with hydraulic tappets instead of roller finger followers; in order to guarantee a correct motion of the tappet, this component must be fitted in a guide, directly machined on a wall, which has to be constantly lubricated in order to reduce the friction and to have a perfect functioning of the hydraulic cam clearance control mechanism. This solution dramatically changes the upper layout of the cylinder head, because it requires an "additional" wall on the *xy* plane (parallel to the gasket interface) just below the camshafts and, usually, more oil channels directly at the tappet level. This solution is obtained by splitting the oil core in two parts during the manufacturing, the upper and the lower one, allowing thus the creation of this "plane"

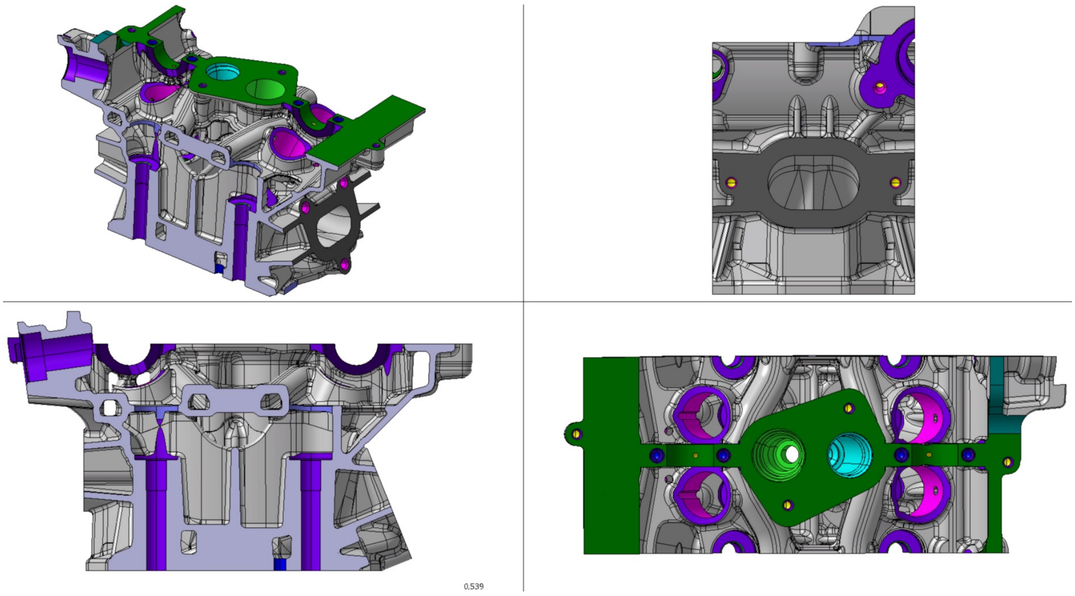


Figure 2.4: *Competitor A cylider head*

wall.

For what concerns the layout, displayed in Figure 2.5, each cylinder has two transversal ribs which are slightly curved in their middle towards the section planes, passing between the head bolts and the spring seats, joining then with the external side walls. The spring interface and the bolt one are connected to the central spark plug and injector channel using small but thick ribs, forming a X shape. A longitudinal rib then stiffens the structure, connecting all the central channels with the transversal ribs.

In the upper level, it is possible to see 3 oil channels, one for the exhaust side and two for the intake, one feeding the tappet only and the other one feeding the camshaft bearing too. The two ducts close to the central channels are not straight but are bended, in order to allow some clearance for the mounting of the cylinder head bolts during the engine assembly. The wall surrounding the tappets displays some holes, mainly for oil draining, like in the exhaust side, and for weight reduction, like between the central oil ducts. The engine cover interfaces are quite big, especially in the exhaust and in the central channel, where the position of the clamping bolts increases a lot this area, increasing inevitably the weight.

The water jackets are connected longitudinally with two channels, while there is an additional connection with the cylinder block on the exhaust side close to the bolts holes, probably for cooling reasons. The gasket interface is quite particular: there are some grooves close to the inlet and outlet of the water jacket, for weight reduction, which are obtained directly in the ground plate of the casting mould. In addition, the

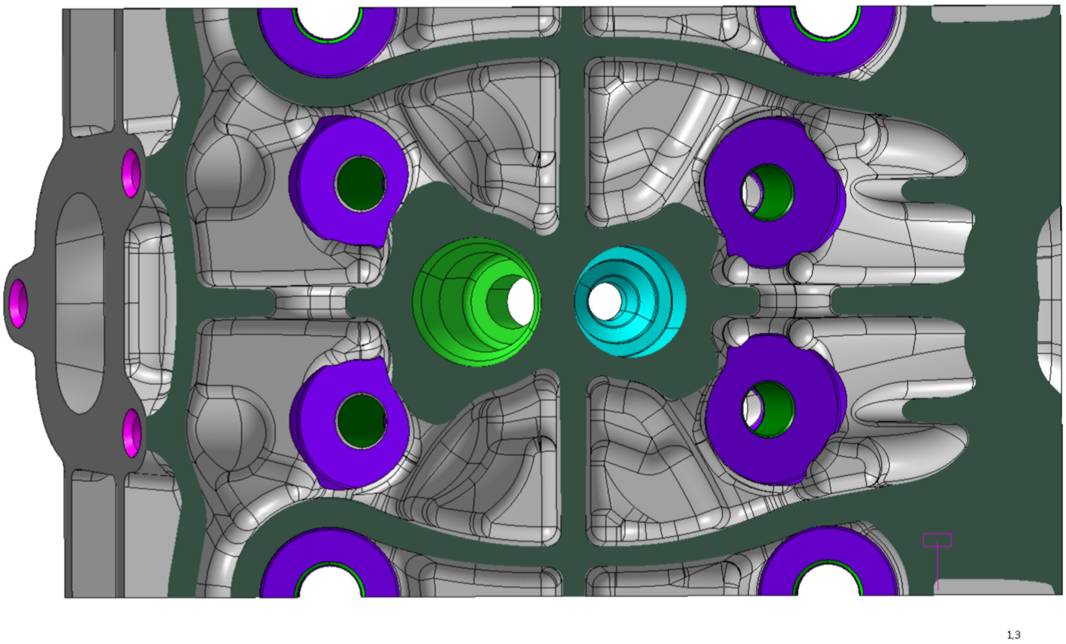


Figure 2.5: *Competitor A ribs layout*

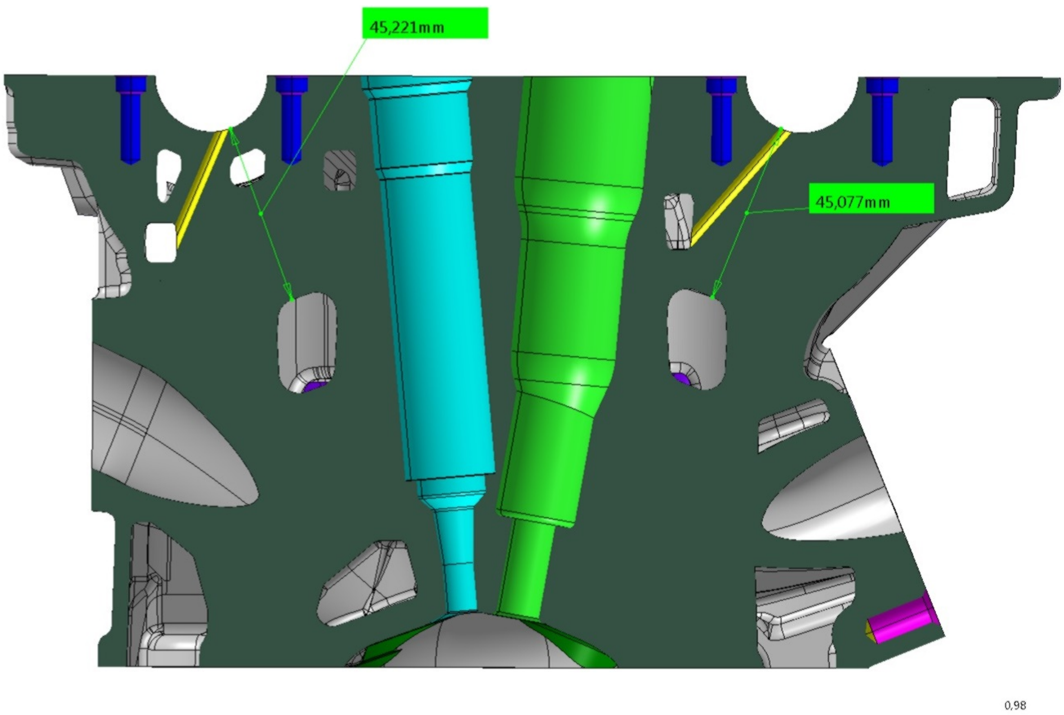


Figure 2.6: *Competitor A camshaft bridge section view*

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
3.0 l	Boxer 6	91	147	251	118	147	183.5	100	3710 g

Table 2.3: *Main characteristics of the Competitor A cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
45	7	4.5	4	6

Table 2.4: *Thickness values of the Competitor A component*

profile of the interface is not rectangular but is more complex, following the shape of the combustion chamber and the inlet/outlet of the water jacket.

For what concerns the camshaft bridge, visible in Figure 2.6, the section view shows that there is a lot of material and just two small windows are present, which are quite far from the camshaft bearing (45 mm). This layout was dictated by the fact that there are the oil channels which run just below the camshafts supports, thus reducing the space for those windows, affecting negatively the final weight.

To conclude, the component walls thickness are displayed in the Table 2.4. While the overall one is 4 mm, which is quite good, $t_{wj,o}$ is really thick, probably due to a conservative approach for the casting processes, to reduce the probability of having defective parts. In addition, also t_{ducts} is quite thick, which contributes in increasing the weight, especially considering the fact that this component has three oil ducts.

2.3.2 Competitor B

This cylinder head, shown in Figure 2.7, is the evolved version of the component shown previously. It shows a total different layout from its precursor, due mainly to the use of roller finger followers instead of hydraulic tappets. This technical solution does not need the plane wall to be present, thus reducing the weight and the casting complexity. Anyway, the overall dimensions, visible in Table 2.5, do not change too much.

The two transversal ribs which are curved in the middle are still present (Figure 2.8), but they are smaller than the ones in the previous component. Another rib design is implemented, with the ribs which connect the spring interface and the bolts that connect with the ones of the other side, forming like a "circular" rib layout between the cylinders. Those ribs have a 3 mm thickness and are higher than the others, since they also have the function to act also as a engine cover interface (the top of the ribs is flat and wider than the rib itself). In any case, the area of the spark plug channel remains quite bulky, probably even larger than the original model. A

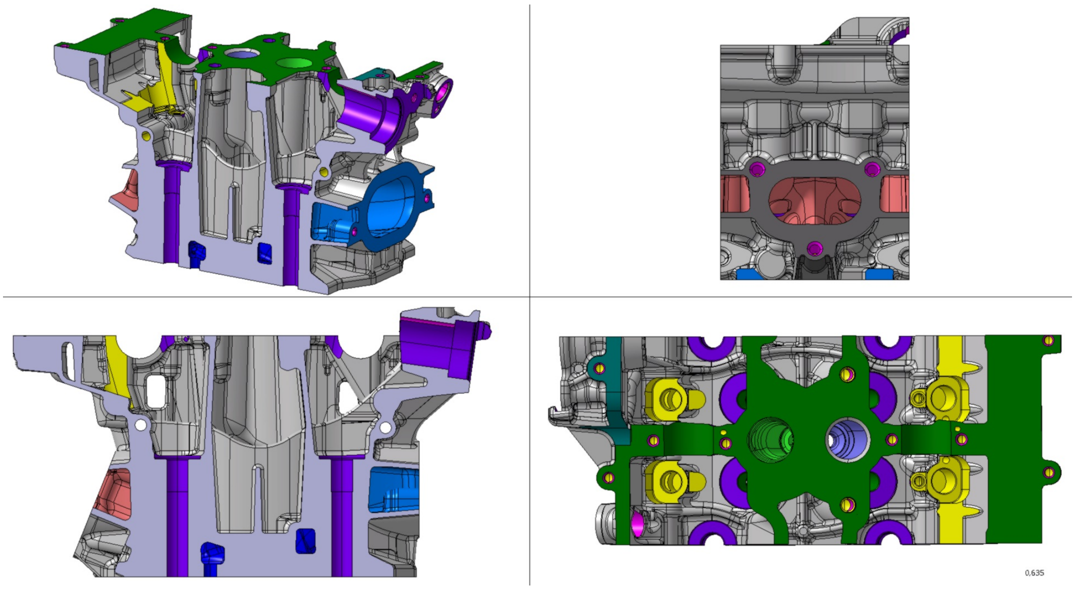


Figure 2.7: *Competitor B cylinder head*

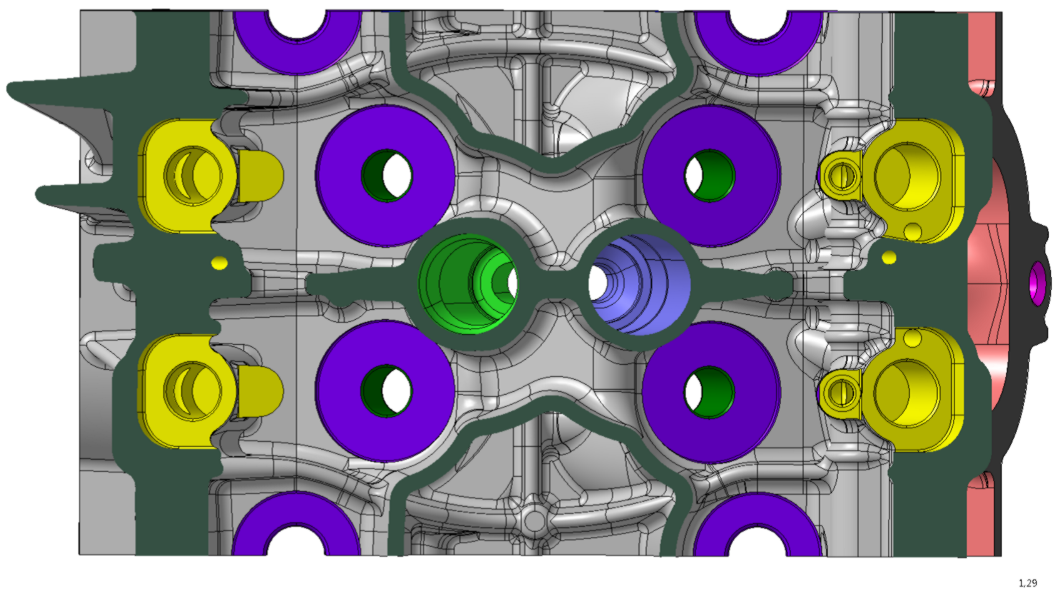


Figure 2.8: *Competitor B ribs layout*

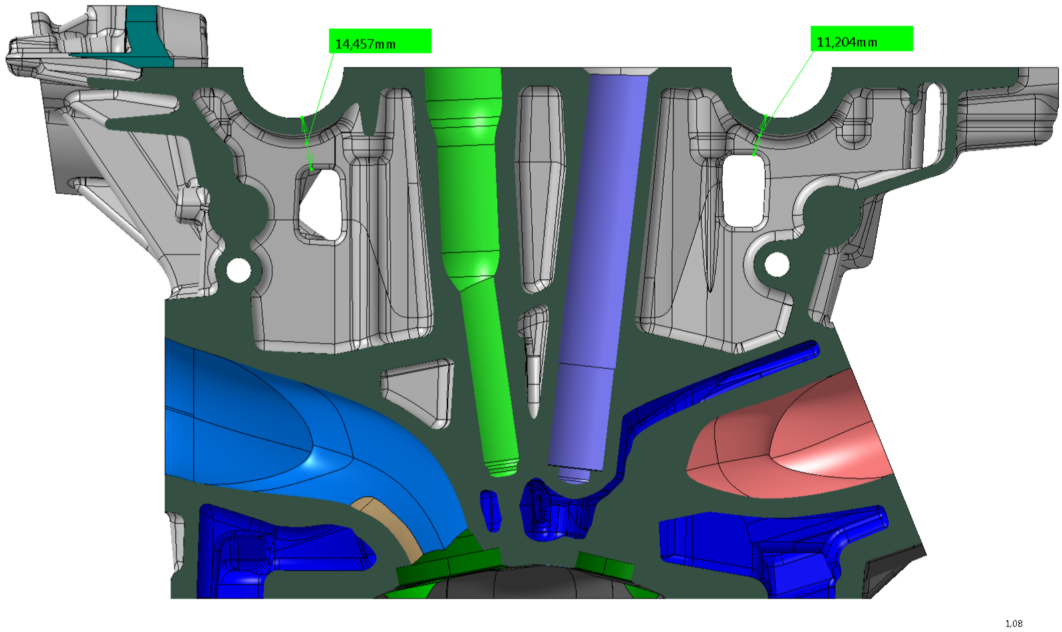


Figure 2.9: *Competitor B camshaft bridge section view*

longitudinal small rib at the bolt interface level connects all the central channels, stiffening the structure even more.

Since the valvetrain system has changed, the longitudinal oil ducts are only two and are no more implemented in the middle of the component, but they are positioned in the lateral walls of the cylinder head, where the seats of the hydraulic lash adjuster are located. The reduction in the longitudinal stiffness is however balanced by the engine cover interface ribs.

For what concerns the other elements, they remained the same: the water jackets are all connected as the original component, the gasket interface has the same shape and grooves and even the camshaft bridge has the same shape (Figure 2.9); despite the absence of the central oil channels, the windows have the same size, even if they are much closer to the camshaft bearing.

For what concerns the walls thickness (Table 2.6), they are generally lower with respect to the original component, especially on the water jacket, where it ranges from 3 to 4 mm with the oil core and from 3 to 7 mm with the ducts and the combustion chamber. The overall thickness ranges from the 3 mm of the rib to the 4 mm of the external walls. These values are quite low for a mass production vehicle and require a good quality and control of the casting.

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
3.0 l	Boxer 6	102	146.7	242	118	135	184.6	100	3228 g

Table 2.5: *Main characteristics of the Competitor B cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
11	7	3.5	3	3

Table 2.6: *Thickness values of the Competitor B component*

2.3.3 Competitor C

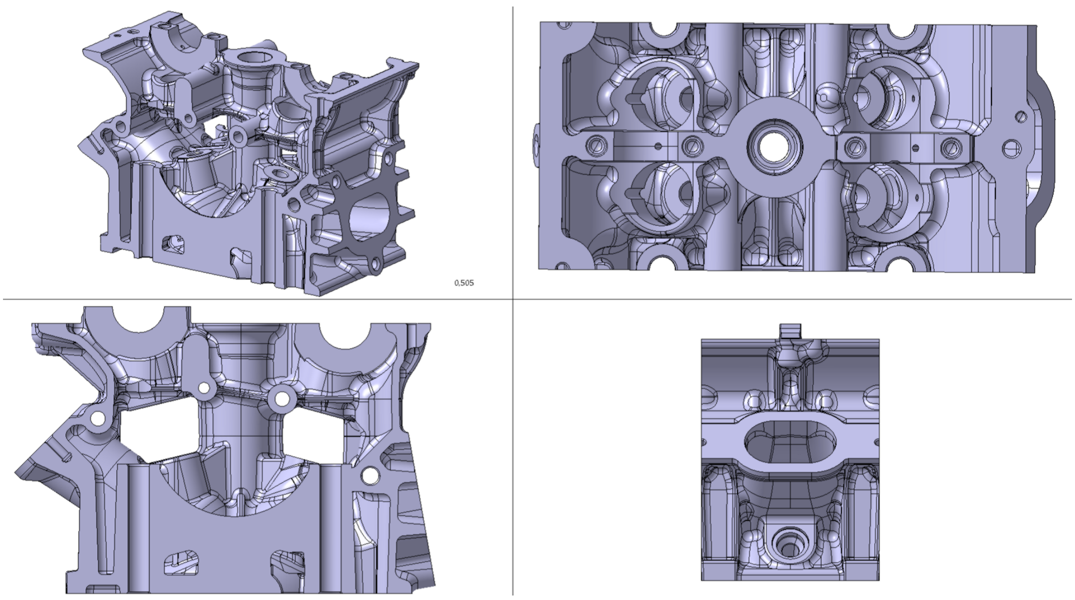


Figure 2.10: *Competitor C cylinder head*

This component, shown in Figure 2.10, is assembled in a V6 turbocharged engine, with indirect and direct gasoline injection (the injector housing can be seen below the intake flange). This cylinder head is quite compact (Table 2.7), with a reduced distance between the two camshafts and a small overall length; the valve motion is guaranteed by means of hydraulic tappets, but differently from the previous components, the tappet guides are obtained on the longitudinal oil channels instead of a plane wall, thus reducing the total mass of the component and the complexity of the casting. The oil is fed with three ducts, one in the exhaust side and two on the intake, the smaller of the two feeding the hydraulic tappet only. It is possible to see also the secondary

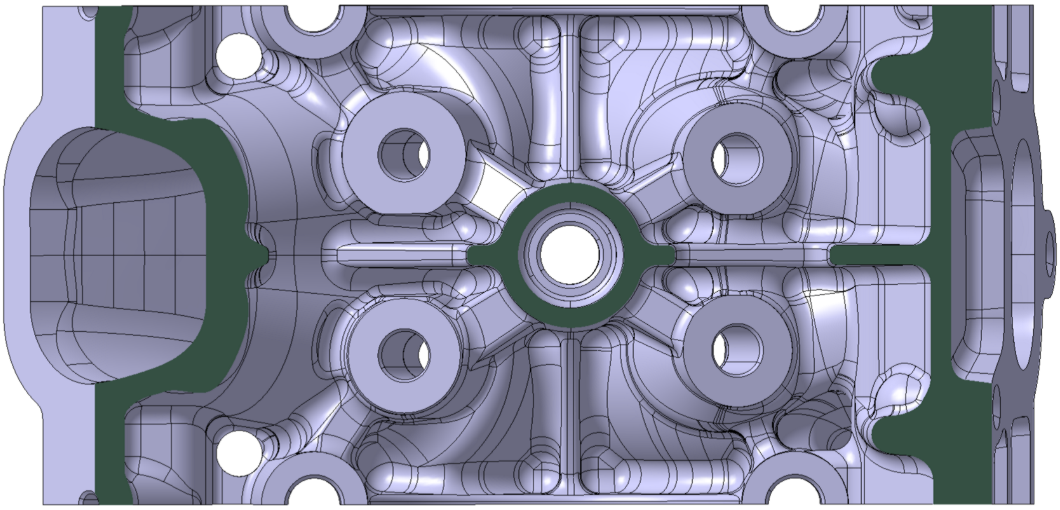


Figure 2.11: *Competitor C ribs layout*

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
3.0 l	V6	96	155	213	108	112.1	177.98	104	2698 g

Table 2.7: *Main characteristics of the Competitor C cylinder head*

air duct running longitudinally above the exhaust manifold.

Also in this case the water jackets are connected, but their shape is different due to the space needed to fit the fuel injector. A duct is machined close to the outlet of the water jacket, between the exhaust valves (visible on the bottom right of Figure 2.12), guaranteeing thus a better cooling of the exhaust valves bridge, which is a critical area from the thermal point of view. For what concerns the gasket interface, the shape is a standard one, with just some slots close to the crankcase ventilation ducts and to the water jacket inlet and outlet.

For what concerns the rib layout, visible in Figure 2.11, the valve spring seats are connected to the spark plug channel by small but very thick ribs (around 11 mm), while the bolts are connected with those seats by means of thinner ones. A longitudinal rib connects the central channels of each cylinder, while the bolts are connected to the one in front by a transversal one. The overall layout is similar to a squared shape with the diagonals connecting the opposite vertexes. The thickness of the thinner ribs is around 4 mm and they have a curved profile, with a larger height on the extremities. It is worth noticing that on the intake side the bolt channels are not connected with the external walls, while the exhaust ones are connected just at their top, allowing thus more space for the oil drain channels. This solution has been

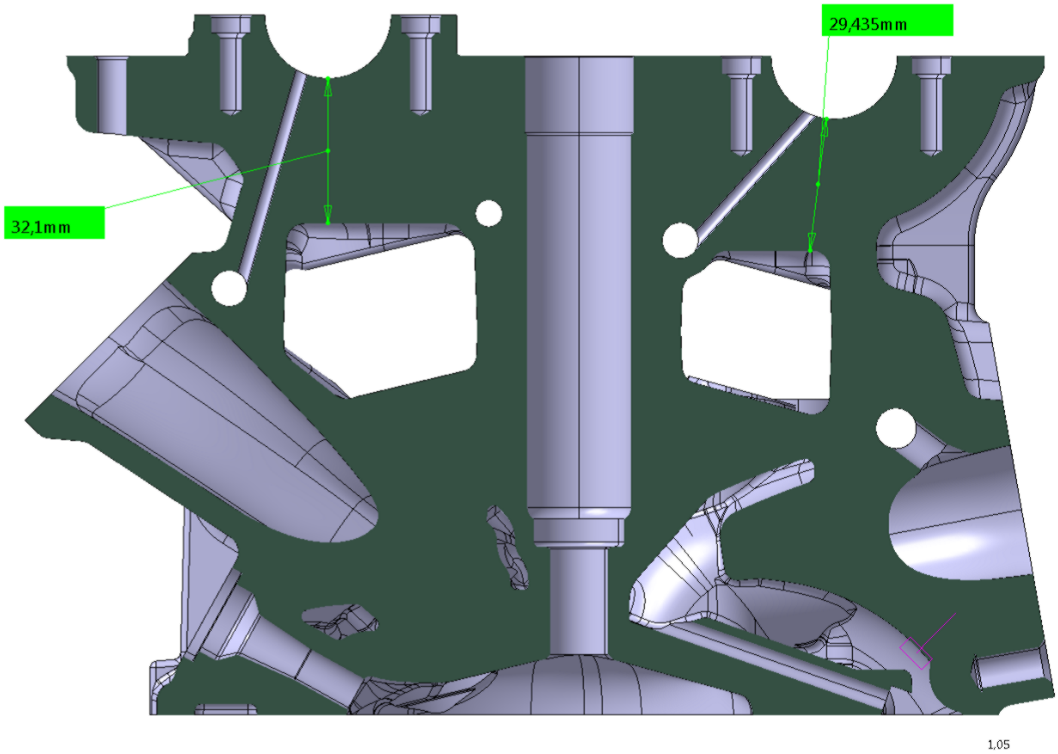


Figure 2.12: *Competitor C camshaft bridge section view*

probably driven by the compact size of this component, but it may worsen the overall stiffness.

It is worth noticing also that the two camshafts are not positioned at the same height but the one in the intake side is located higher than the other one. This design choice was introduced in order to allow more space for the intake group and instead of increasing the height of the overall component, only the intake camshaft has been raised. The engine cover interface remains anyway at the same level.

The camshaft support section, visible in Figure 2.12, displays two large windows, thanks to the fact that the injector is not positioned close to the spark plug but below the intake valves, thus freeing a lot of space in this area, but the amount of material around the top of the bridge and at the lower oil core part remains anyway high, due mainly to the thick ribs and the small transversal ones. Their average distance to the camshaft bearing is around 31 mm, since the arrangement of these last components is different.

The wall thickness is quite high (Table 2.8), with an average value of 4.2 mm which is used between the water jacket and the oil core, on the external walls and on the spark plug channel, while it increases in the more critical areas ranging between 5 and 7 mm, like close to the bolt channels and to the press-fitted valve guides.

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
31	11	4.2	4.2	4.2

Table 2.8: *Thickness values of the Competitor C component*

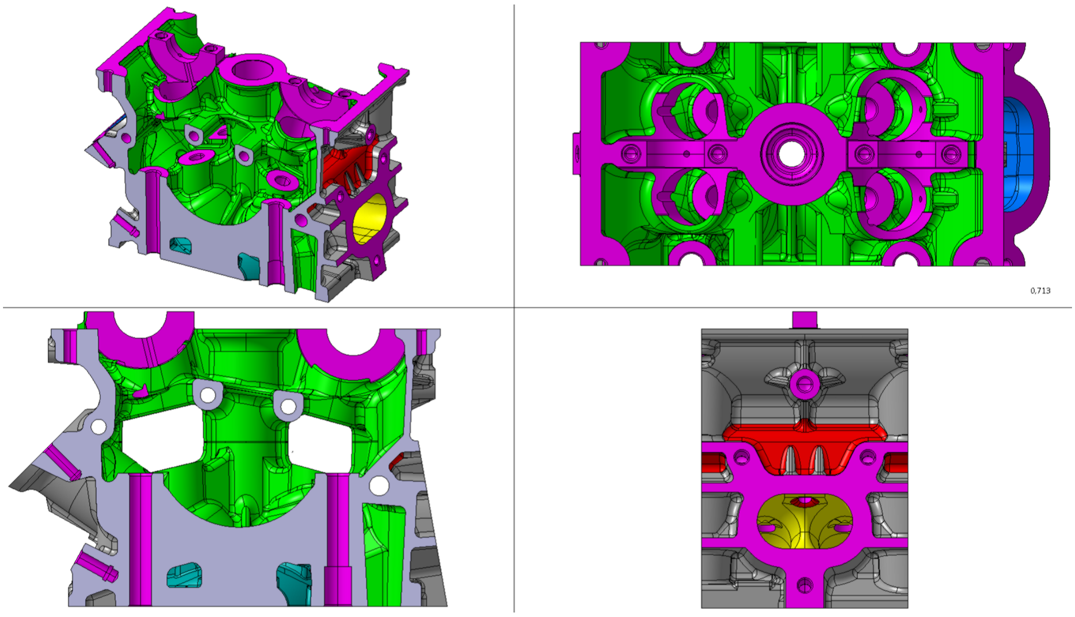


Figure 2.13: *Competitor D cylinder head*

2.3.4 Competitor D

This component, show in Figure 2.13, shares the same design with the one discussed in subsection 2.3.3, but is assembled on a V8 4.8 l turbocharged engine. In fact, as it is possible to see in Figure 2.14 and Figure 2.15, the ribs layout is the same, the position of the oil channels and of the secondary air duct did not change and also the camshafts still have a height difference. The major differernce stays in the intake bolt channels, which are now connected directly to the external wall.

For what concerns the walls thickness (Table 2.10), they are overall similar. The only differences are in the position of the camshaft bridge windows, which are farther and also smaller in size, and on the oil ducts, which are larger in diameter (4 mm

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
4.8 l	V8	96	155	205	108	112.1	176.5	104	2828 g

Table 2.9: *Main characteristics of the Competitor D cylinder head*

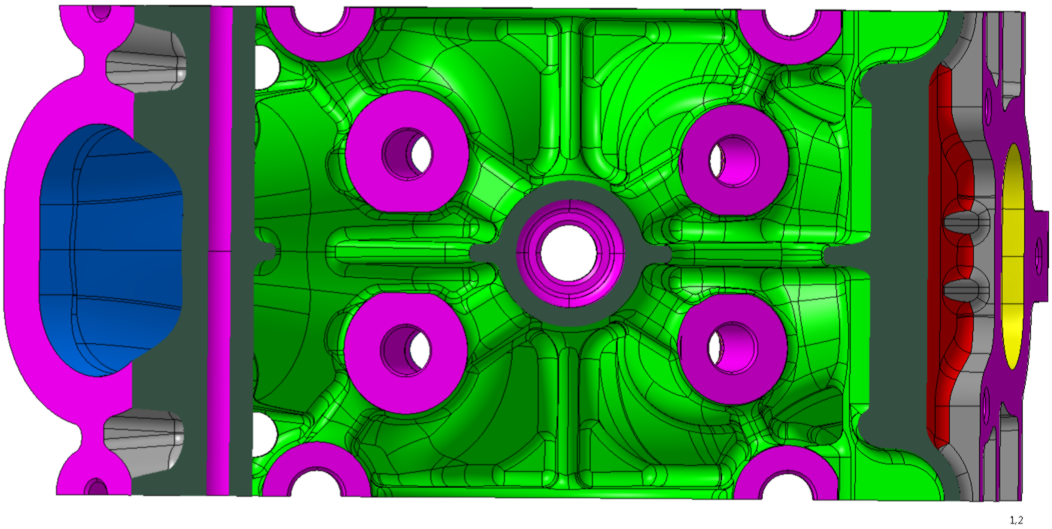


Figure 2.14: *Competitor D ribs layout*

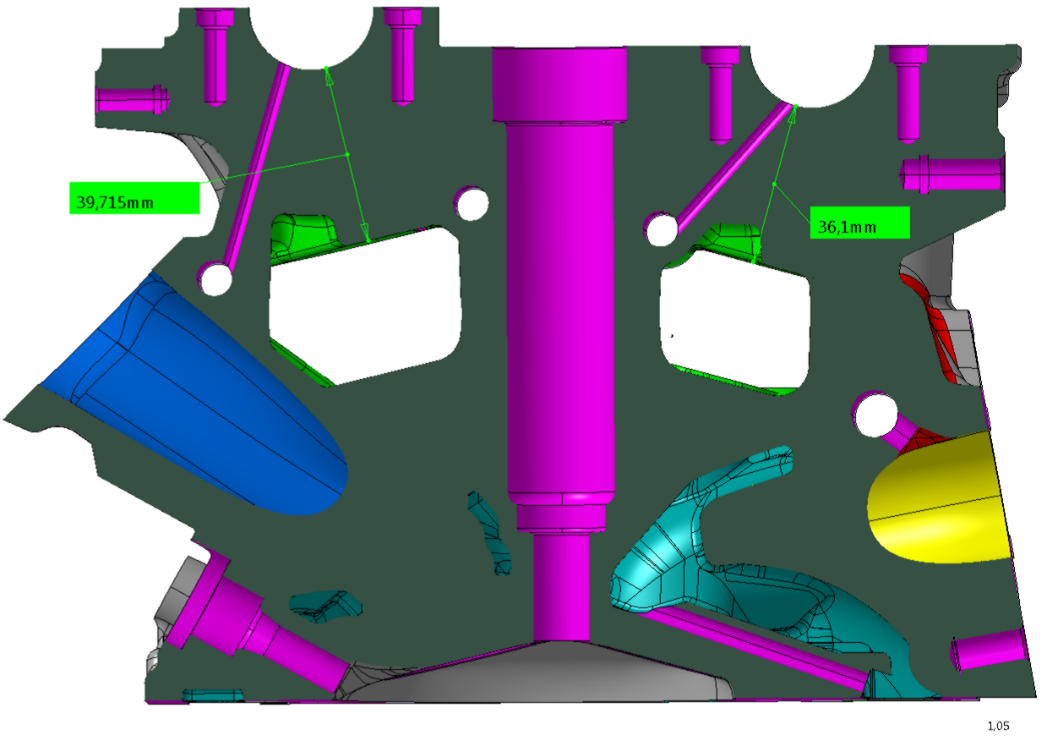


Figure 2.15: *Competitor D camshaft bridge section view*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
37	11	4.2	4.2	4

Table 2.10: *Thickness values of the Competitor D component*

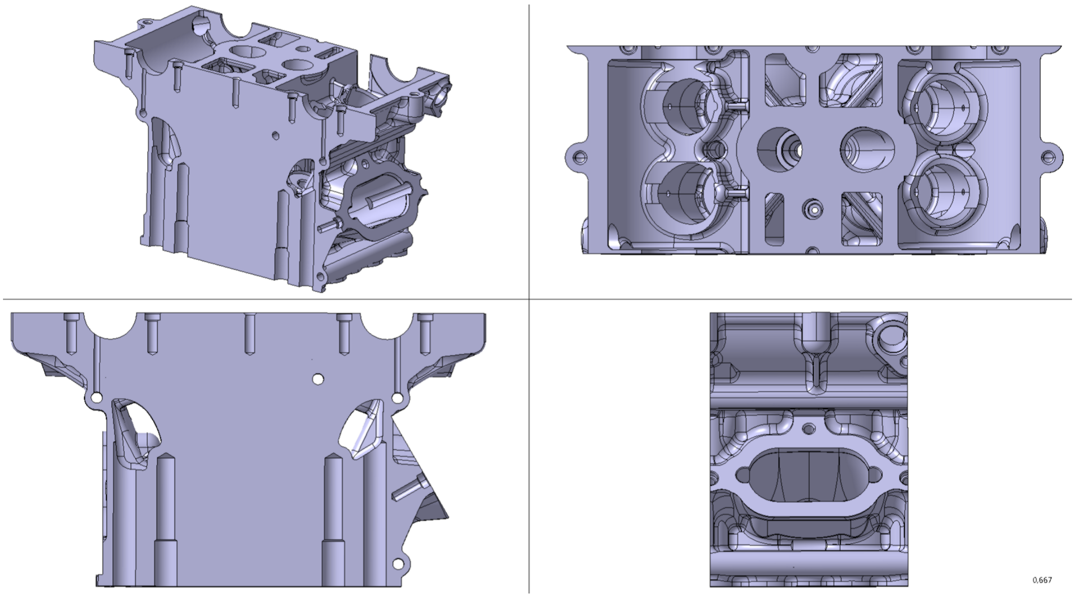


Figure 2.16: *Competitor E cylinder head*

against 3 mm) and lower in thickness. However, the weight has increased a little bit.

2.3.5 Competitor E

This cylinder head, shown in Figure 2.16, is part of a 3 cylinder in-line turbocharged engine of 1.9 litres, with a gasoline direct injection system. The valve motion is guaranteed by means of hydraulic tappets, which guide is obtained using a plane wall which then connects with the engine cover interface. The oil is fed with three ducts, two on the intake side and one in the exhaust, and drains to the oil core thanks to a hole located in between the tappets. Those ducts are located on the external walls, differently from the previous components, where they were located in the middle of the transversal section. The water jackets are independent one from the other, the only connection happens on a longitudinal duct running below the intake flange, but it is used only to allow the air trapped in the cooling circuit to exit.

This cylinder head differs completely in the layout from the ones seen before for two reasons. The first one is the camshaft bearing position, which usually is located between the tappets, while in this case is positioned on the side of the cylinder section,

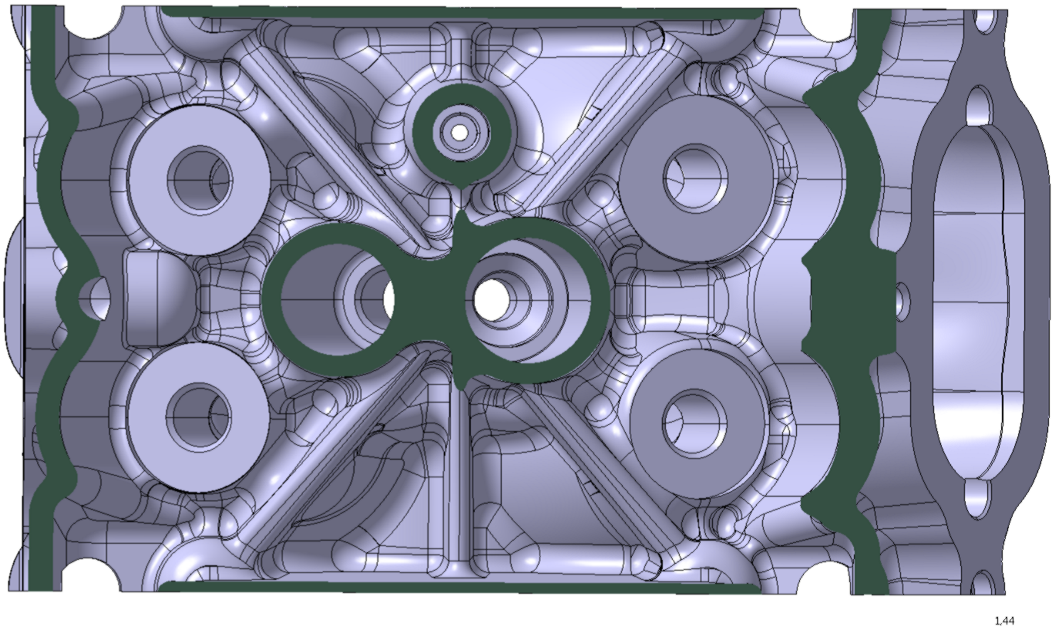


Figure 2.17: *Competitor E ribs layout*

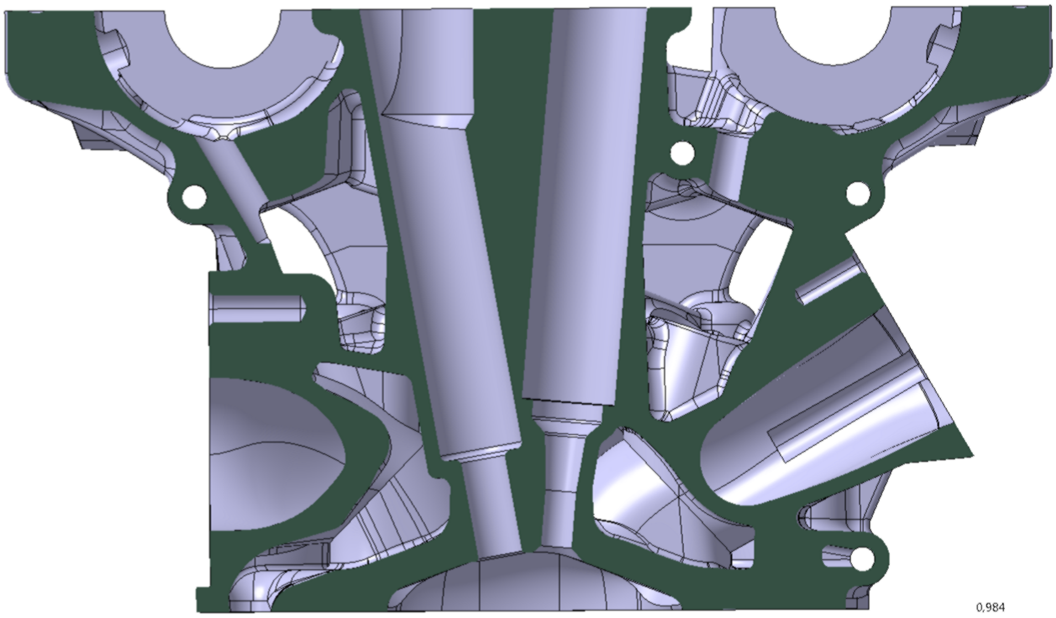


Figure 2.18: *Competitor E mid-section plane view*

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
1.9 l	In-line 3	94	145	229	105	146.9	153.3	89	2748 g

Table 2.11: *Main characteristics of the Competitor E cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
29.6	8	4	3.5 - 5	3.5

Table 2.12: *Thickness values of the Competitor E component*

probably for packaging reason since there is not enough space to fit a bearing between this tappets. Unfortunately this solution causes a larger bending of the camshaft associated with a bending moment on the camshaft support with respect to a standard solution, since there is an unbalance of forces between the sides of the cap due to the phase shifting of the cylinder firing sequence.

The second main difference can be noticed in the clamping mechanism of the component. Differently from the previous components, this cylinder head does not rely on bolts to be fitted on the bolt channels, but on studs, which are fitted first during the assembly operation and then are tightened using some nuts at the engine crankcase level, after having inserted the engine block and the crankcase itself. This solution has the advantage to free some space at the oil core level and to avoid large stress concentrations which are present with a bolt-nut layout, but require a longer machining of the part, including also the threading of the stud hole, and also more complex operations during the assembly phase.

For what concerns the rib layout, visible in Figure 2.17, the stud holes are connected with the central channel, which includes also the direct injector, with 5 mm thick ribs, which are coonected also to the valve spring seats by means of fillets. A longitudinal rib, quite thick (5 mm), adds some longitudinal stiffness. At the camshaft bridge, visible in the bottom right of Figure 2.16, the studs are connected by a wall, which has two small windows close to the external walls and distant from the camshaft bearing. Those windows are connected to the oil drains and to the crankcase ventilation channels, which are located between the stud holes and the external walls.

The wall thickness are generally low, especially between the water jacket and the oil core, which is close to 3.5 mm, considered as the minimum safe value to be adopted in this region for a high quality casting.

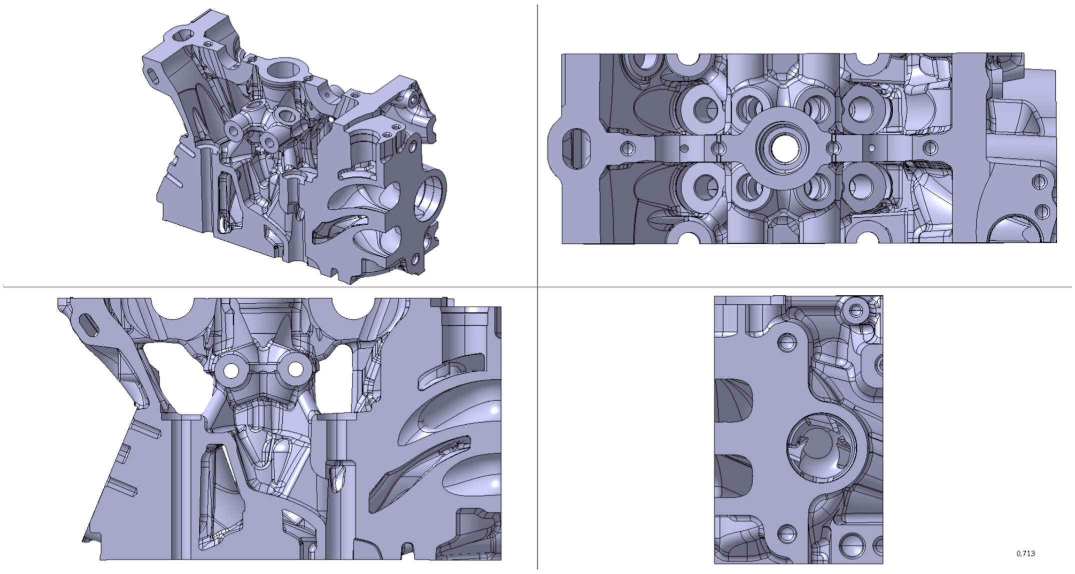


Figure 2.19: *Competitor F cylinder head*

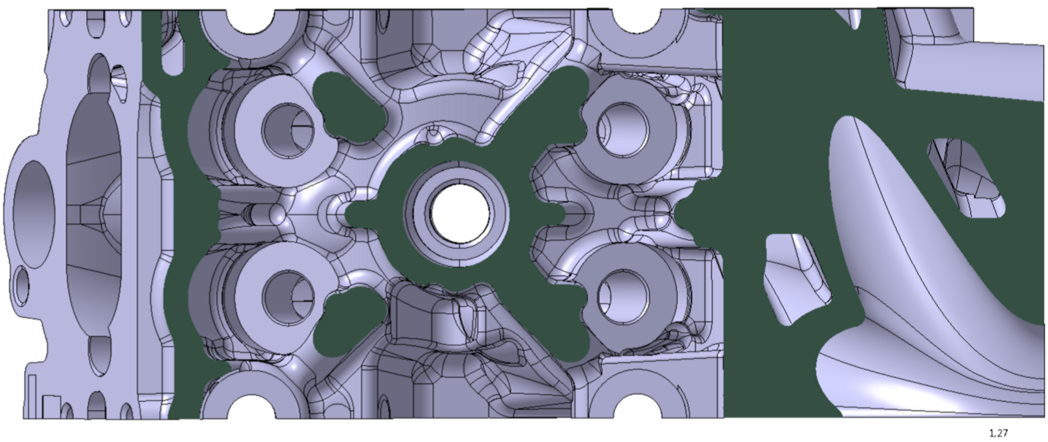


Figure 2.20: *Competitor F ribs layout*

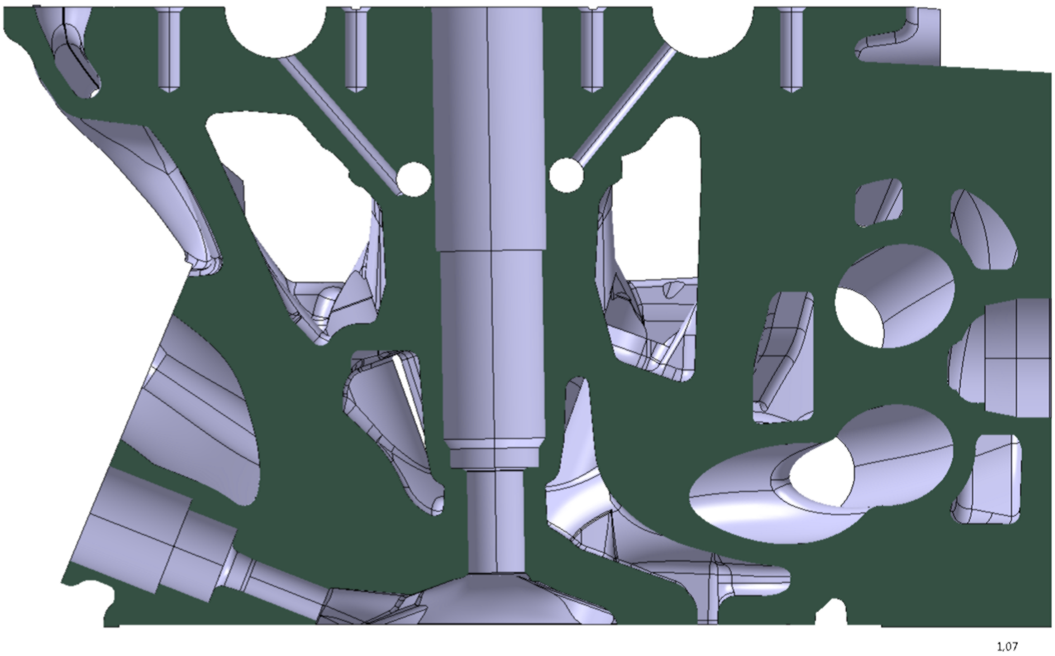


Figure 2.21: *Competitor F camshaft bridge section view*

2.3.6 Competitor F

This cylinder head, shown in Figure 2.19, is used in a in-line 4 cylinder turbocharged engine, with direct and indirect injection (the direct injector is located below the intake valves). The peculiarity of this component is that it has an *integrated exhaust manifold*, which is a technology and a design solution which is more and more adopted by car manufacturers; for this reason the benchmark in terms of layout and wall thickness has less meaning, but it is anyway interesting for a weight comparison.

The valve motion is guaranteed by means of roller finger followers, which hydraulic lash adjuster are located in the center of the transversal section, fed by two oil ducts, one for side. The water jackets are interconnected with some very large openings, and their shape is really complex, since they have also to cool the exhaust manifold. The gasket interface has a standard rectangular shape, but big grooves are located on the water inlets, in order to save weight. Another particular technical solution can be seen in the bottom left of Figure 2.19, where the bolt channel is exploited also as an oil drain channel and for crankcase ventilation, thanks to two large openings, one towards the center of the oil core and the other towards the spring valve interface. In fact, the oil is allowed to drain only in this part of the cylinder head, while the intake side is completely dedicated to the water jacket inlet. The exhaust side of the component is quite bulky, due to the exhaust ducts coming from

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
2.0 l	In-line 4	74	139.6	197*	88	96.5	193.8	85	2816 g

* 220 if considering the integrated exhaust manifold

Table 2.13: *Main characteristics of the Competitor F cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
13	10.5	5.5	5	5

Table 2.14: *Thickness values of the Competitor F component*

the adjacent cylinders, which are surrounded then by some water channels, to keep the temperature of the components downstream the valves to an acceptable value. The exhaust flange is wide and flat in order to allow the mounting of the turbo unit. The position of the camshaft bearings is not the optimal one for the assembly operation, since it covers completely the clamping bolts, thus forcing the drivetrain to be assembled after the clamping of the cylinder head.

For what concerns the rib layout, visible at Figure 2.20, four thick ribs connect the bolt interface with the spark plug channel, which then are connected to the spring valve interfaces with some fillets. The ribs on the intake side have a small window, probably to allow a better oil drain more than for weight reduction purposes. There are no longitudinal ribs, probably because the two oil channels and the wall separating the water jacket from the oil core, which location is quite high with respect to the components seen previously (Figure 2.21), can guarantee already a satisfactory stiffness value.

The wall thickness of this component (Table 2.14) were quite difficult to evaluate because they vary a lot on the geometry; on certain spots t_{avg} was around 4 mm, while in others it was around 8 mm. In any case, they are a bit conservative, but probably because of the integrated exhaust manifold. Only exception is between the oil channels and the spark plug one, where the values are definitely too low to respect any manufacturing constraint; probably this drawing is part of one of the first design releases of the engine and this value has been later corrected. However, despite all the factors listed above, the weight of this cylinder head is comparable to the one of the other components.

2.3.7 Competitor G

The following component, shown in Figure 2.22, is part of a 4.0 litres, 6 cylinder boxer naturally aspirated high performance engine, used also for motorsport appli-

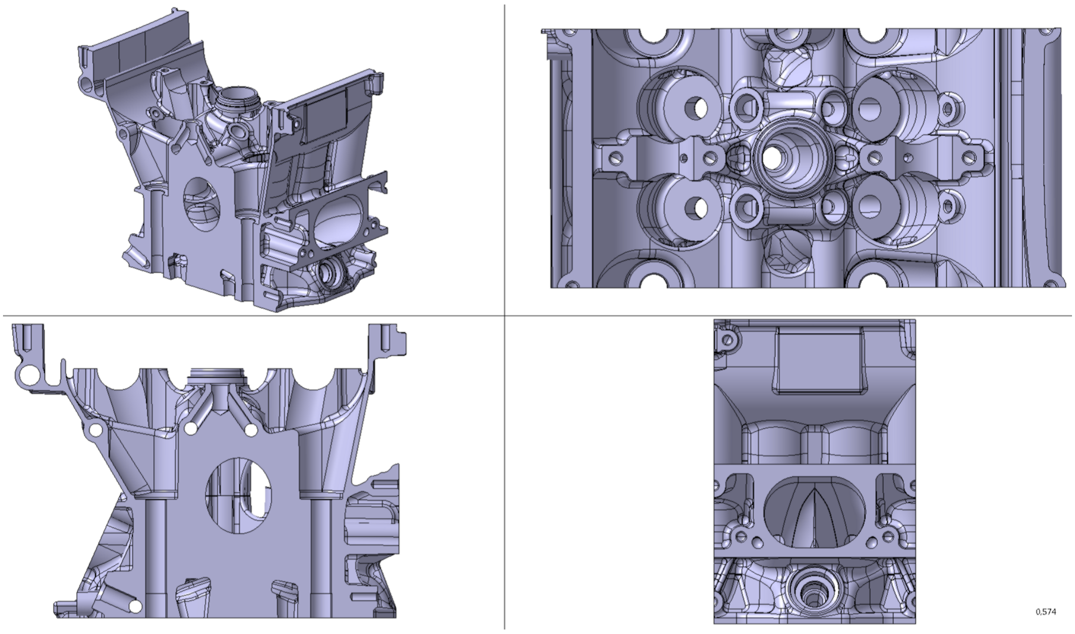


Figure 2.22: *Competitor G cylinder head*

cations, with both direct and indirect injection. The valvetrain relies on a finger follower system, which hydraulic lash adjusters are located at the sides of the central channel, in which is fitted the spark plug, while the direct injector sits on the intake side, just below the intake ports. Two longitudinal oil ducts in a central position feed respectively the valvetrain on the two sides, while a third one sprays oil directly on the finger followers of the exhaust valves, probably avoid any friction issue.

The ribs layout, visible in Figure 2.23, is quite different from the others seen before. Instead of relying on ribs, this component guarantees the stiffness by means of walls, with a very big circular opening in their middle to save weight; all the bolts are connected to the central channel, while additional walls perpendicular to the cylinder head length connect the intake bolts with the exhaust ones. The bolt interface area is stiffened up by surrounding it with some circular walls, which connect then to the side walls and to the oil channels (C-shaped section cuts around the bolts in Figure 2.23). This special arrangement of the walls guarantees a lot of free space around the camshaft bridge support, visible in Figure 2.24, which enables the use of very large windows which are not so distant from the camshaft bearing. It can be also noticed that the external walls are a lot inclined towards the center of the cylinder head, which is a solution aiming at the reduction of the overall dimensions, leaving the intake flange overhanging from the external wall.

For what concerns the wall thickness, shown in Table 2.16, the values are quite low, especially for t_{avg} where the values are of 3 mm in the external walls and on the

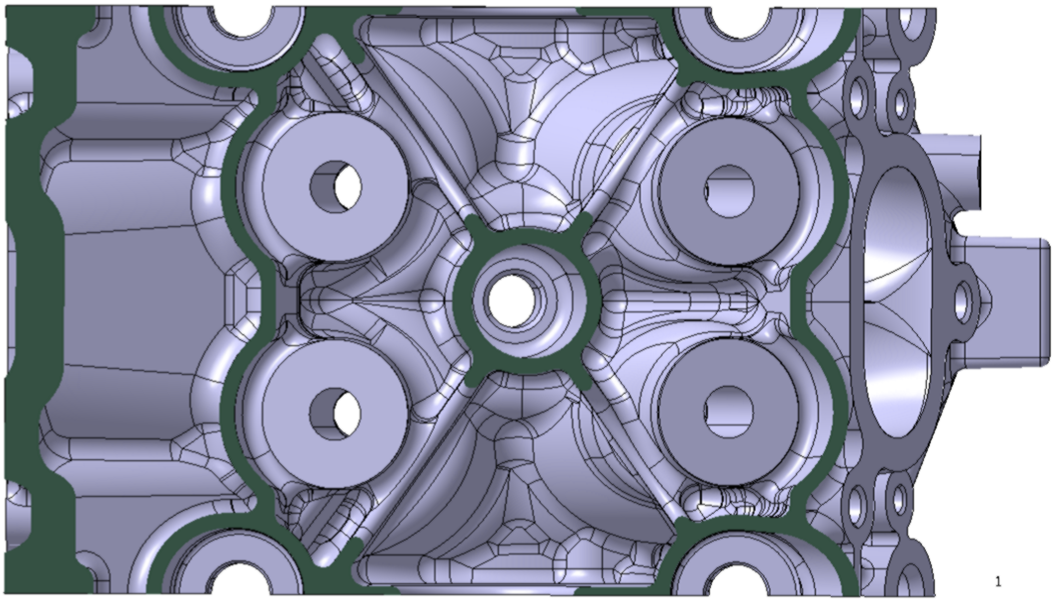


Figure 2.23: *Competitor G ribs layout*

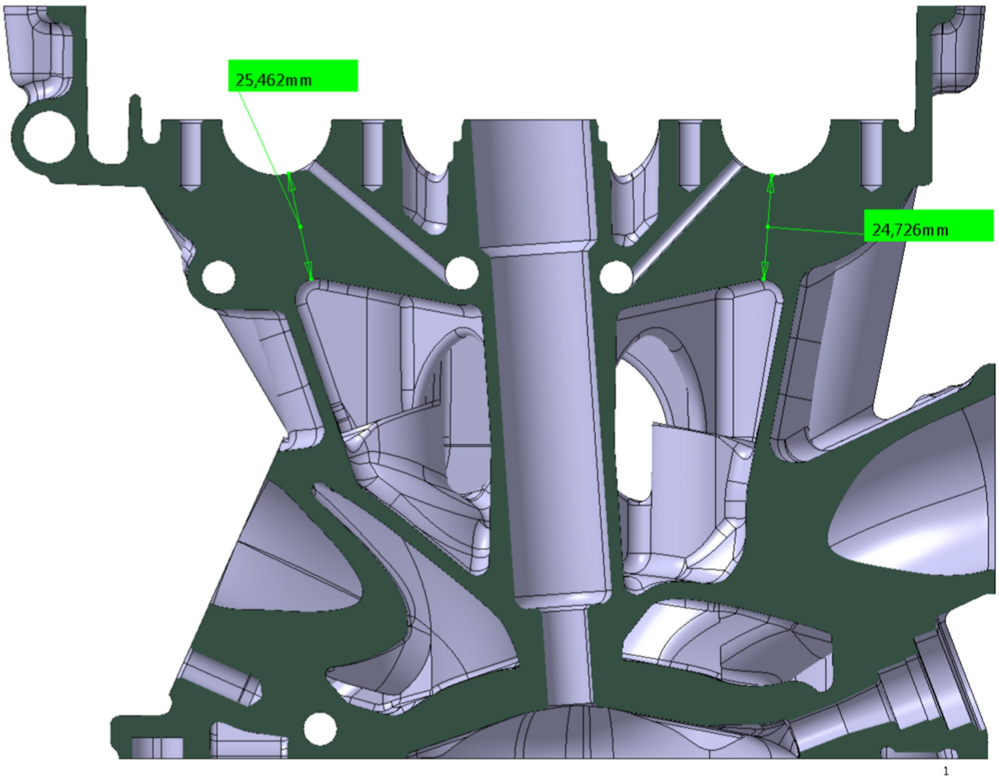


Figure 2.24: *Competitor G camshaft bridge section view*

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
4.0 l	Boxer 6	102	178	214.9	118	118	168.1	100	2714 g

Table 2.15: *Main characteristics of the Competitor G cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
25	10.5	4	3.5	3.5

Table 2.16: *Thickness values of the Competitor G component*

ribs. Despite this factor, the weight is comparable to the one of the other components, probably because of the X walls used to reinforce the structure, which however seem, at a glance, to provide larger stiffness. It can be concluded that this cylinder head can be taken as an example of a very good trade-off between lightness and stiffness.

2.3.8 Competitor H

This cylinder head, shown in Figure 2.25, is mounted on a V6 3.0 litres turbocharged engine, with a direct injector placed on the central channel together with the spark plug, and an indirect one placed on the intake manifold, between the intake flange and the intake wall. Two major characteristics are the integrated exhaust manifold, already discussed before, and the absence of camshaft bearings and supports, as it is possible to see in Figure 2.27. The camshaft is assembled on a cam-carrier or directly on the engine cover, thus allowing more flexibility in the design and easier operations during the assembly phase, but at the cost of lower component performances; in fact, splitting the cylinder head in two components causes a reduction in terms of stiffness. The valve motion is governed by a roller finger follower system, which hydraulic lash adjusters are fitted attached to the central channels. Another difference can be found on the oil feeding system, which is fed from the engine cover interface instead of being fed from channels running longitudinally on the cylinder head. The water jacket has a very complex shape because of the integrated exhaust manifold, has a large area below the intake flange because the coolant is fed from the cylinder head instead of the cylinder block, as it usually happens, surrounds completely the exhaust pipes, has some connections with the cylinder block at the combustion chamber sides and is connected to the ones of the other cylinders (mandatory if an integrated exhaust manifold is used).

For what concerns the rib layout, shown in Figure 2.26, the solution adopted is quite particular: instead of relying on a ”classical” X shape, the bolts are connected just with an inclined wall, the one separating the oil core from the water jacket, higher

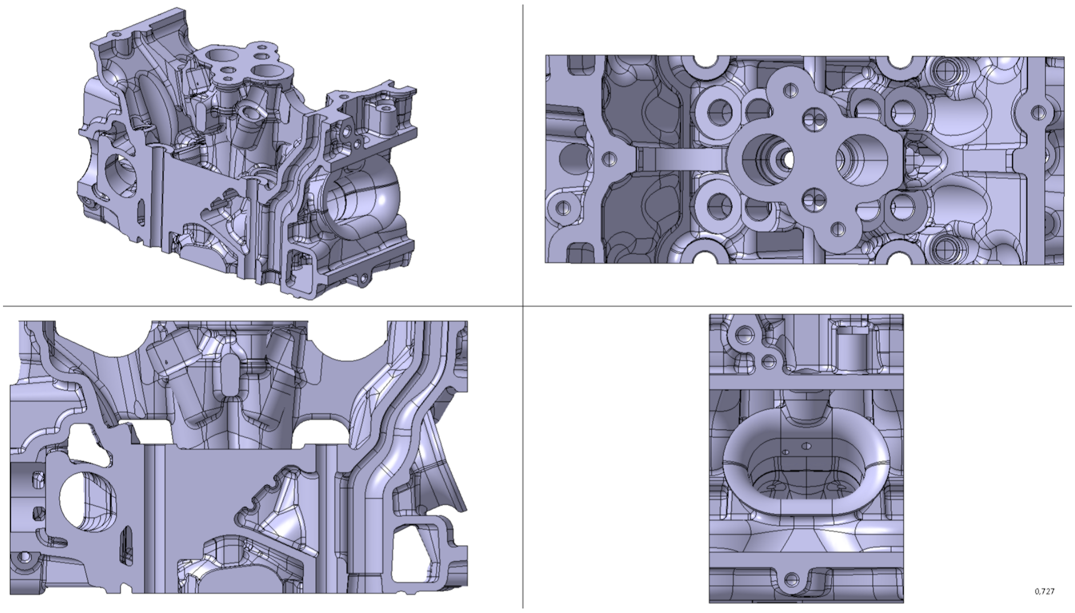


Figure 2.25: *Competitor H cylinder head*

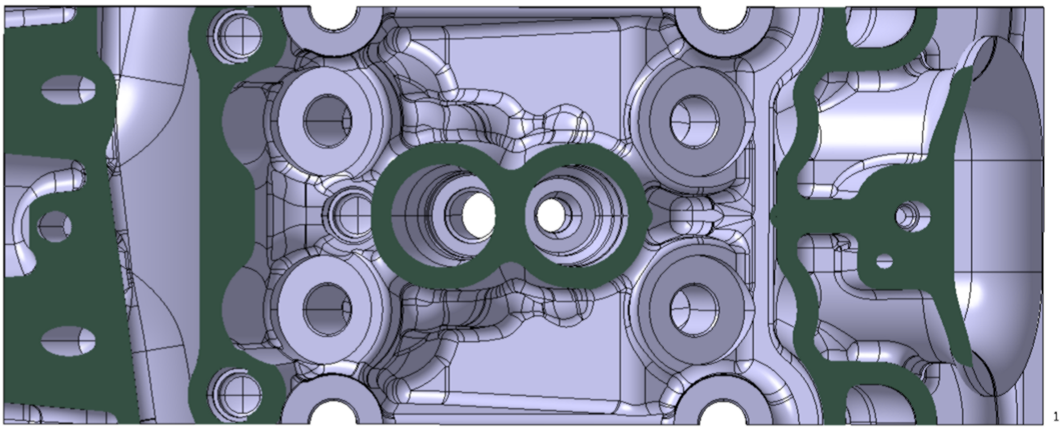


Figure 2.26: *Competitor H ribs layout*

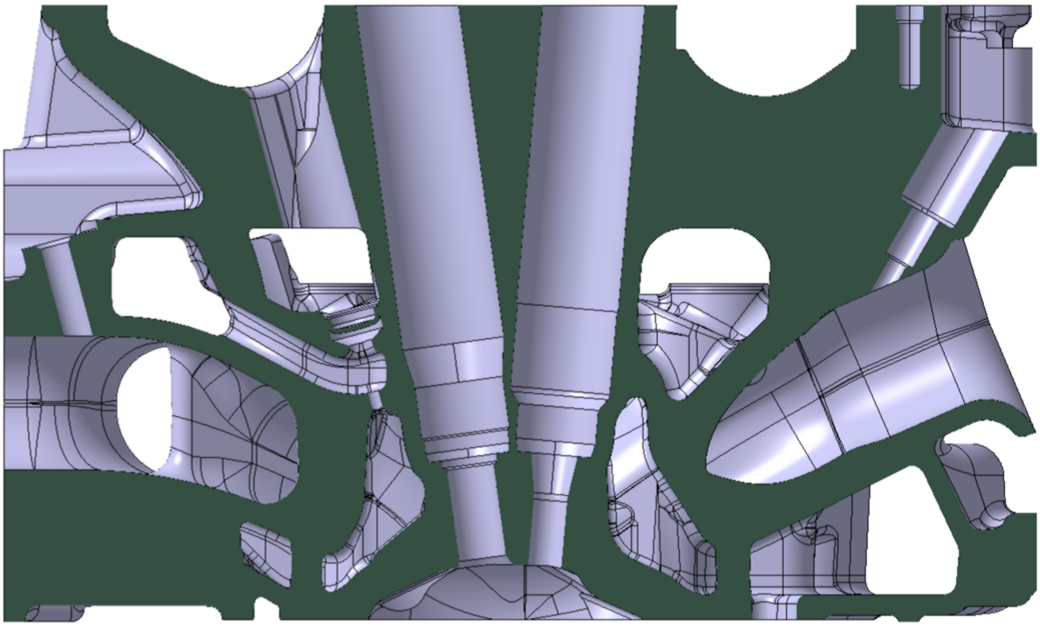


Figure 2.27: *Competitor H mid-section plane view*

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
3.0 l	V6	84	138	223.1	93	-	232	87	2946 g

Table 2.17: *Main characteristics of the Competitor H cylinder head*

at the intake side and lower at the exhaust one, reinforced with some longitudinal ribs on the side facing towards the combustion chamber; only a longitudinal rib is present, connecting the bolts transversally, which is shared between two adjacent cylinders. The valve spring seats and the bolts are connected with those walls with large fillet radii. The oil draining is guaranteed by the bolt channel on the intake side, since the inclined wall mentioned before has an angle of 45 degrees, but in assembly condition it will be at zero degrees since the amplitude of the V angle is of 90 degrees.

The thickness values, displayed in Table 2.18, are comparable to the ones seen before, but the weight is however quite high (Table 2.17), even though the dimensions are not so big and the camshaft section is not present. The explanation relies probably on the fact that the central channel and the exhaust manifold are really bulky.

2.3.9 Competitor I

This cylinder head, shown in Figure 2.28, is mounted on a 4.0 litres V8 turbocharged direct injected gasoline engine, with the injector placed in the central

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
-	7	4.5	4	-

Table 2.18: *Thickness values of the Competitor H component*

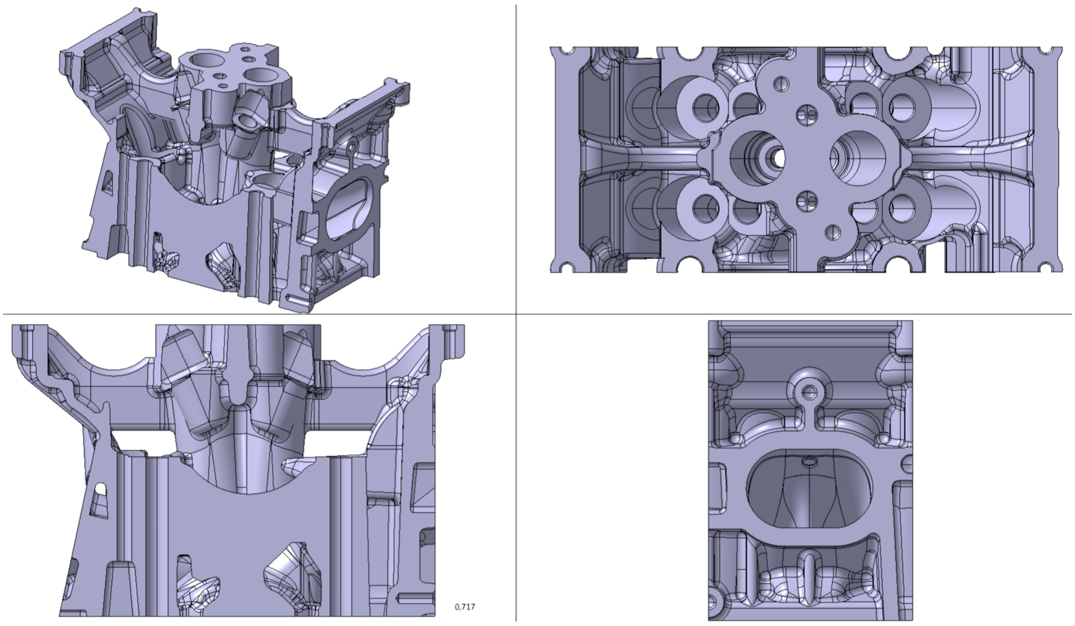


Figure 2.28: *Competitor I cylinder head*

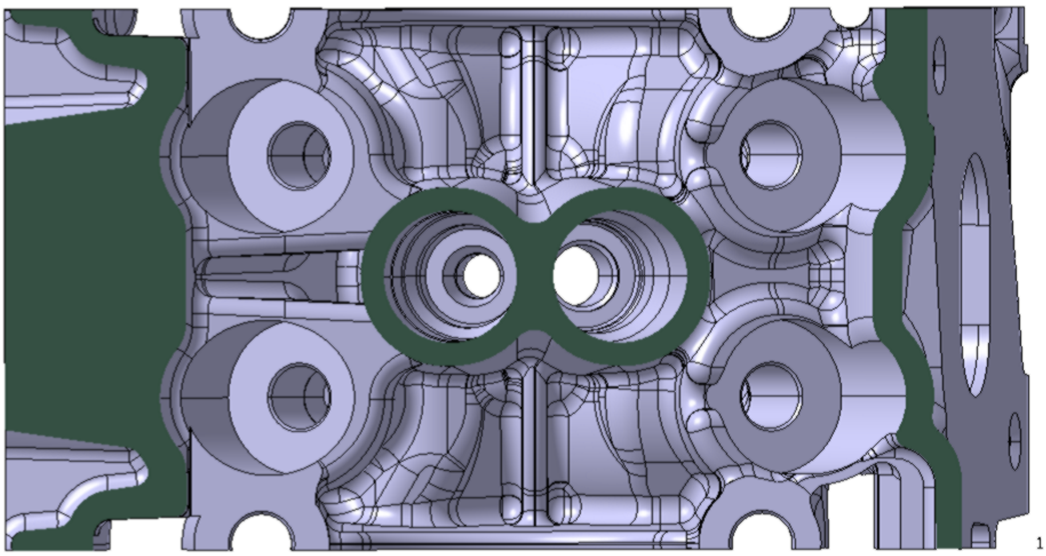


Figure 2.29: *Competitor I ribs layout*

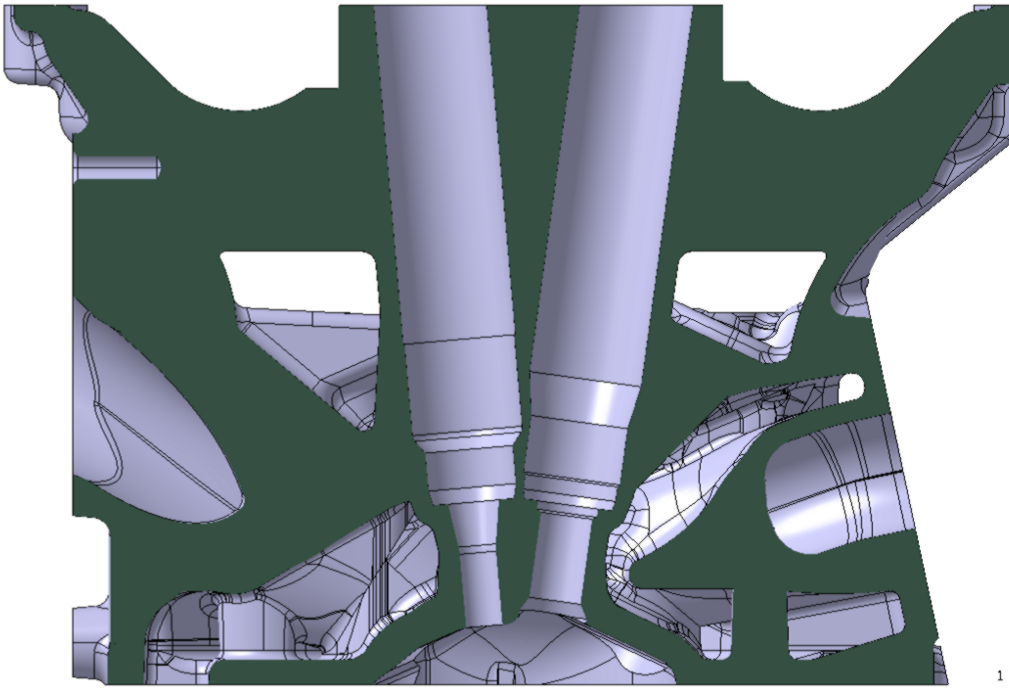


Figure 2.30: *Competitor I mid-section plane view*

channel together with the spark plug. From a technical point of view, this cylinder head does not display any camshaft support, like the component discussed in subsection 2.3.8 (they are part of the same engine family), but the integrated exhaust manifold solution has been dropped for a "standard" design, with the exhaust manifold independent and flanged on the designated interface. The valve motion is provided by means of roller finger followers, which hydraulic lash adjusters are located at the sides of the spark plug channel; the lubricating oil is fed from the engine cover, where the oil circuit feeds the oil ducts located at the cylinder head mid-line on the spark plug channel, necessary for the correct functioning of the HLA. The water jackets are interconnected, with large windows between one cylinder and the adjacent one, and their dimensions are larger than the ones seen previously, especially around the exhaust manifold. The gasket interface is a standard one, with no grooves or different shapes in order to reduce weight.

The layout, shown in Figure 2.29, is an unusual one, with the bolt interfaces directly connected to the spring seat ones, which then are connected directly to the central channel. There is no rib between these elements, they are all connected using the wall between the oil core and the water jacket, which is raised a bit from its usual position, especially around the exhaust springs, enlarging thus also the volume occupied by the coolant (Figure 2.30). The only exceptions are the intake spring

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
4.0 l	V8	84	138	210	93	-	160.1	87	2262 g

Table 2.19: *Main characteristics of the Competitor I cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
-	6	4.5	5	-

Table 2.20: *Thickness values of the Competitor I component*

interfaces, which are additionally connected to the central channel with an additional bridge, which allows the oil present in the lower part of the oil core and on the spring interfaces to better flow to the oil drain channels and the crankcase vent ducts. Two additional ribs, one longitudinal and one transversal connecting the bolt interfaces, add some stiffness.

For what concerns the wall thickness, displayed in Table 2.20, they are on the average and are similar to the ones of the other components. Nevertheless, this is one of the lightest cylinder head analysed until now, but it has to be considered that the absence of any camshaft support jeopardize the benchmark analysis result.

2.3.10 Competitor J

This cylinder head, displayed in Figure 2.31, is assembled on a 4.6 litres V8 naturally aspirated engine with gasoline direct injection, on a high performance car, a detail which will justify some design and technical choices discussed later. The valve motion is guaranteed by means of finger followers, which hydraulic lash adjusters are placed at the sides of the central channel. The oil is fed with three ducts, one placed in the center line of the cylinder head, crossing all central channels between the spark plug and the injector, which feeds the hydraulic lash adjusters only, while the other two, which are located on the external walls and have a lower diameter (6 mm instead of 8), feed the camshaft bearings and spray oil on the finger follower surface in order to reduce friction. It is then drained thanks to two ducts behind the intake bolts (Figure 2.31 bottom left). The water jackets are independent one from the other and have some additional connection with the cylinder block between the bolts and the combustion chamber. The gasket interface has a particular shape which follows the profile of the combustion chamber and of the ducts, in order to save as much weight as possible

For what concerns the structure (Figure 2.32), the lower part of the oil core is completely free of ribs or stiffening elements; every bolt interface is surrounded by

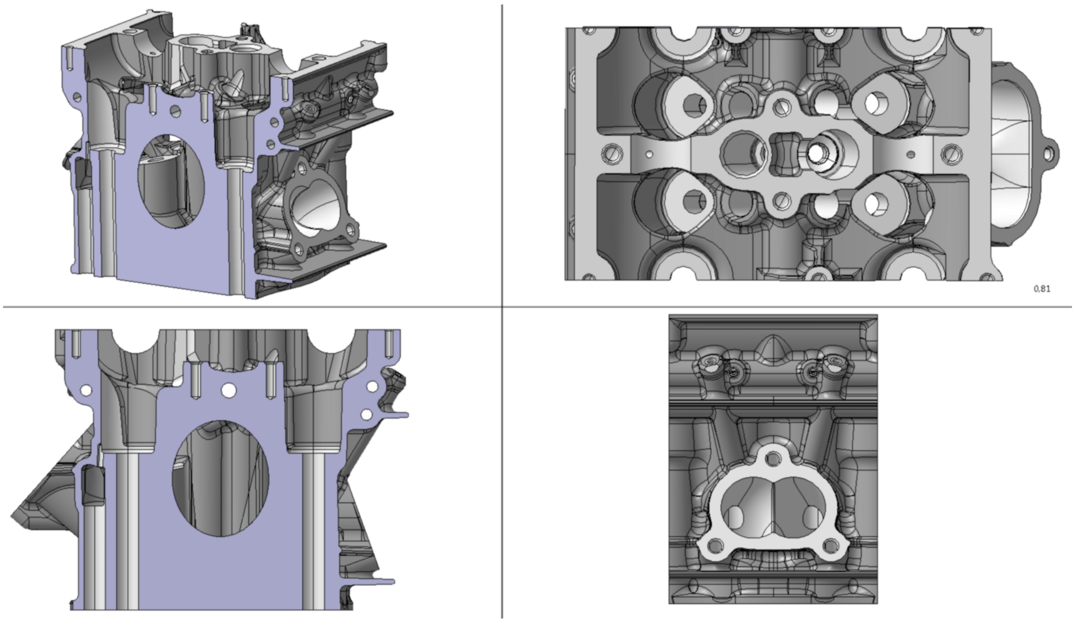


Figure 2.31: *Competitor J cylinder head*

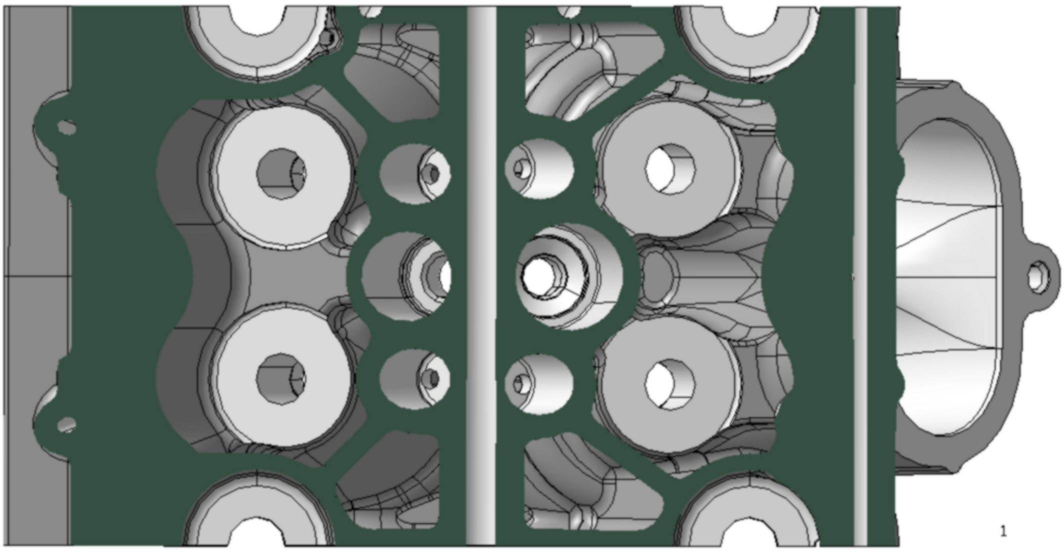


Figure 2.32: *Competitor J ribs layout*

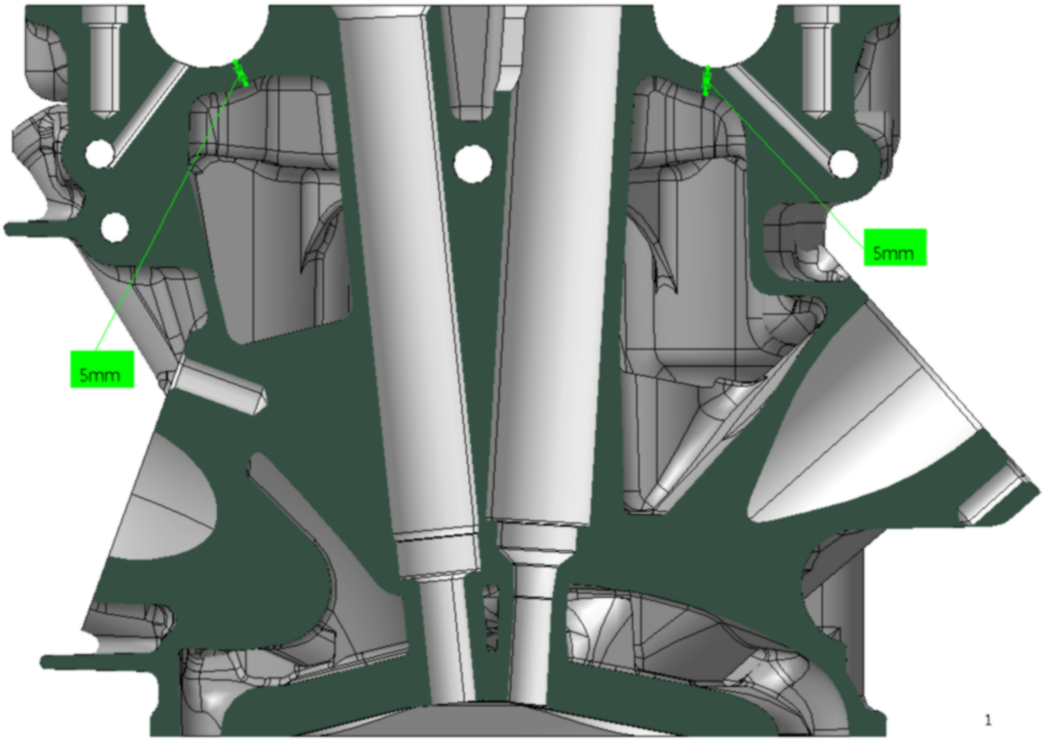


Figure 2.33: *Competitor J camshaft bridge section view*

a vertical cone wall which then connects to a horizontal wall at the hydraulic lash adjuster level which stiffens the whole structure. The bolt interfaces are connected with the ones on the opposite side by means of a wall, which has a large circular window (Figure 2.31 bottom left) while four ribs, located between the cone walls and the central channel, support the horizontal wall, stiffening the area around the hydraulic lash adjuster (visible in Figure 2.33).

The camshaft bridge section view (Figure 2.33) shows three interesting aspect: the compact size of this component, the machined groove on the central channel between the spark plug and the injector to reduce the weight and, most important, the structure of the camshaft support. In fact, as it is possible to see, the distance between the window and the camshaft bearing is really short, because instead of having a thick rib with a classical rectangular section, like the components seen previously, the support is guaranteed by two thin ribs of 3 mm of thickness, minimizing the cross sectional area and thus reducing the use of material.

For what concerns the wall thickness, shown in Table 2.22, the values are really low, especially for the side walls and the ribs, which is of 3 mm, close to values which are used just for motorsport applications. To reach these values, very good quality casting processes and controls are necessary, something which can be justified only

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
4.6 l	V8	95	145	173.7	105	101.1	129.1	96	2106 g

Table 2.21: *Main characteristics of the 918 cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
5	10	4	3	4.5

Table 2.22: *Thickness values of the Competitor J component*

by the low number of parts produced, by the performance level of the engine and by the total cost of the car. All these factors lead to an astonishing final weight of 2106 g, which is the lowest of all the components analysed in this benchmark activity.

2.3.11 Competitor K

This cylinder head, shown in Figure 2.34, is mounted on a 1.4 litres 4 cylinders in-line engine, with direct and indirect gasoline injection and has both an integrated exhaust manifold and a structural engine cover with the two camshafts fitted in. The valve motion is based on roller finger followers, which hydraulic lash adjusters are fitted at the sides of the central spark plug channel and they are fed by two oil ducts, which run longitudinally and are connected with each other by a wall, forming thus a bulky body which has also the role of engine cover support. The water jackets are interconnected and have very big dimensions, especially in the part close to the lower oil core which stays at the same level of the bolt interfaces, where can be divided in two parts, the upper one which is close to the oil core itself and the lower one, which is in contact with the ducts and the combustion chamber. Even if the two parts are connected with some openings close to the central channel, visible in Figure 2.35, they are almost everywhere separated by a horizontal wall.

The mid-section plane view in Figure 2.36 shows that the bridge section of this component is totally different from the ones seen before, because the central channel is not connected to the external walls and there is just a stiffeners on the intake side connected to the wall separating the oil core to the water jacket. However, in the cylinders which are closer to the gearbox and to the transmission sides (cylinders 1 and 4), a bridge with a rectangular section area and an average thickness of 7.5 mm is present, joining the central channel to the exhaust wall. Probably in the others cylinder the bridges are absent because the wall dividing in two parts the water jacket is already enough to guarantee the necessary stiffness.

The gasket interface displays big pockets to reduce the weight, especially on the

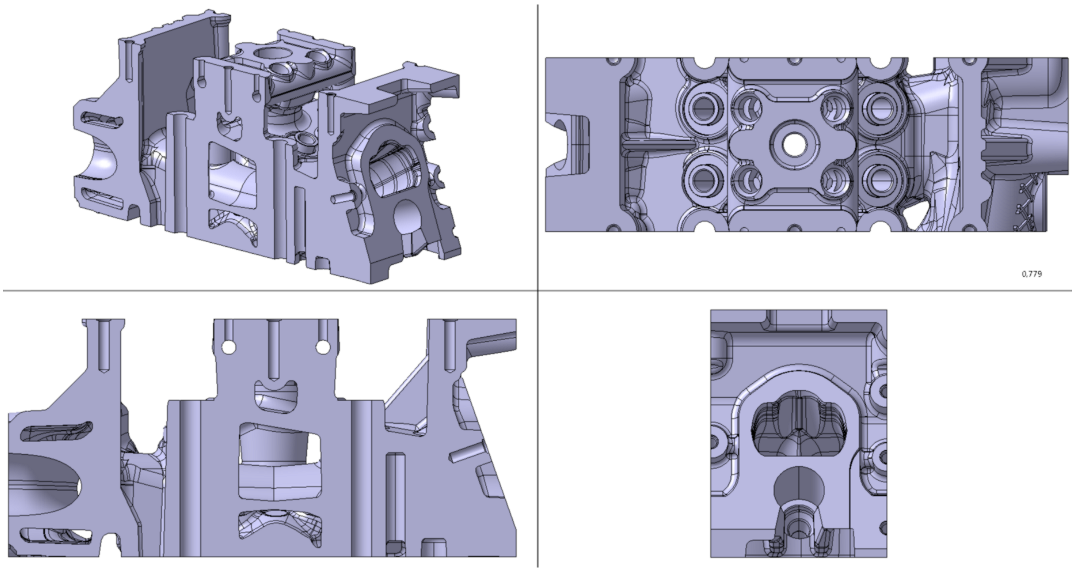


Figure 2.34: *Competitor K cylinder head*

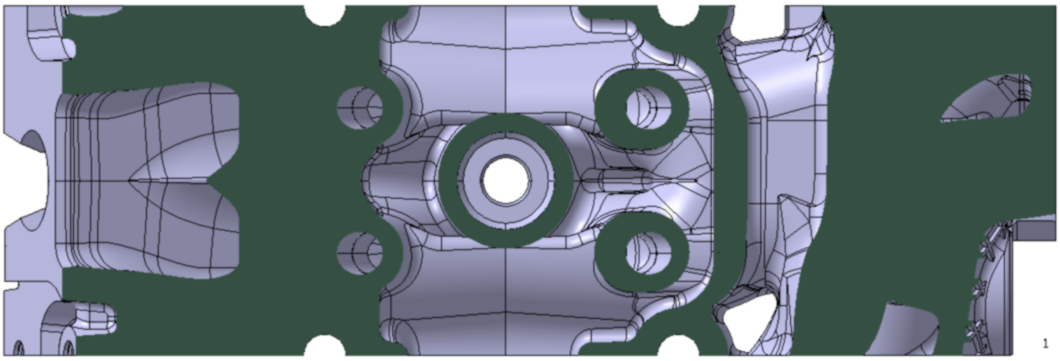


Figure 2.35: *Competitor K ribs layout*

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
1.4 l	In-line 4	76.5	115.4	220.5	105	82	221	83	2440 g

Table 2.23: *Main characteristics of the Competitor K cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
-	8.5	4.5	5	4.5

Table 2.24: *Thickness values of the Competitor K component*

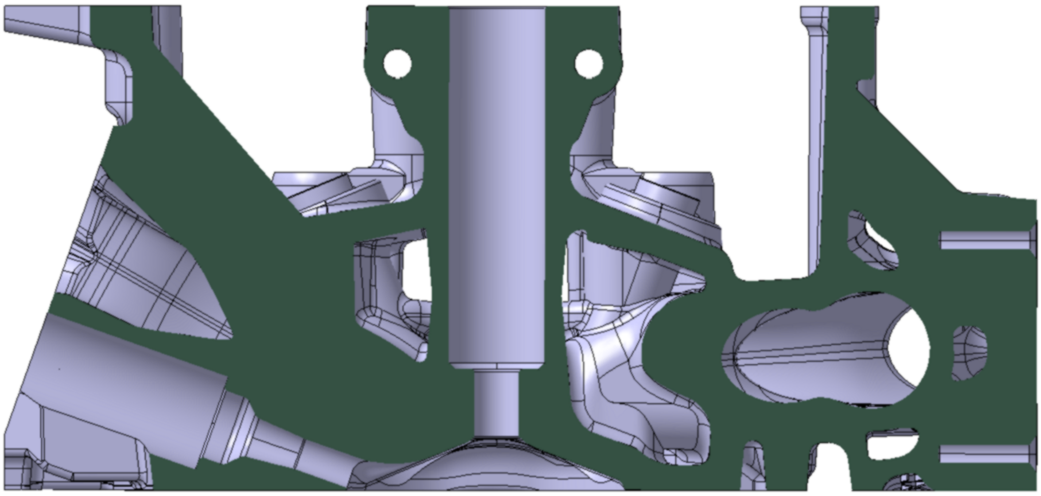


Figure 2.36: *Competitor K mid-section plane view*

intake side, behind the bolts and around the direct injector, some of which include also the water jacket inlet.

For what concerns the wall thickness values, displayed in Table 2.24, the values are on the average, also considering that this is a mass-produced engine which fits in different cars and models. The weight is lower than the average, but definitely a lot lower with respect to the other models analysed which fitted the integrated exhaust manifold.

2.3.12 Competitor L

This cylinder head, shown in Figure 2.37, is mounted on a V8 naturally aspirated engine, with direct and indirect gasoline injection. It is part of the same engine family of the component described in subsection 2.3.11, but differently from that one, it does not have an integrated exhaust manifold, while the camshaft supports are integrated in the cylinder head itself and not on an additional cam-carrier. The valve motion is controlled by a system of roller finger followers, which hydraulic lash adjusters are located at the sides of the central spark plug channel, which are fed by two oil ducts which run longitudinally on the partition plane of the two oil cores, as it's clearly visible in the bottom left of Figure 2.37. The water jackets are interconnected, especially at the exhaust side, probably to increase the cooling capacity. This cylinder head is compact, as it is possible to notice from Table 2.25, with a short distance between the flanges (like a hourglass) that, as discussed in subsection 2.3.10, helps a lot in the weight reduction. The gasket interface is really complex, with a lot of grooves for the water jacket, pockets for the weight reduction and a particular

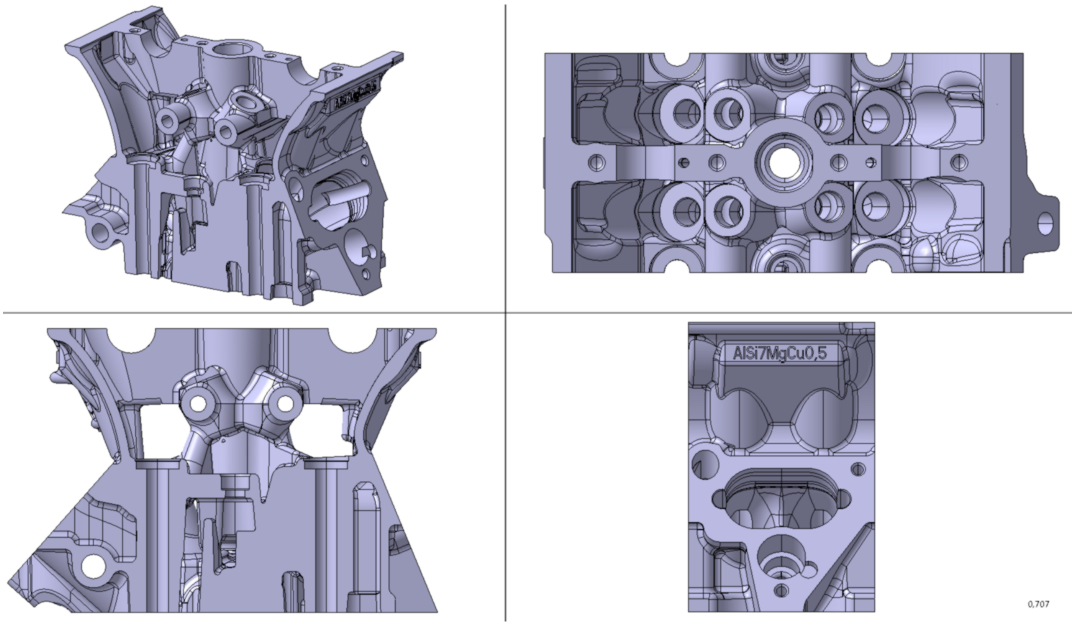


Figure 2.37: *Competitor L cylinder head*

geometry close to the side walls to minimize the overall surface and, thus, the final weight.

For what concerns the rib layout, visible in Figure 2.38, is it possible to see that no longitudinal ribs are present, probably because the oil ducts already guarantee the necessary level of stiffness. The bolt interfaces are machined directly on the lower oil core wall, like the valve spring interfaces and are connected with a thick rib on the intake side and with a bridge on the exhaust one. The camshaft bridge section view (Figure 2.39) shows two big windows which are quite close to the camshaft bearings and the oil ducts which have to lubricate those supports. Differently from all the

Displacement	*Layout*	D	h	l	w	d_{cs}	d_f	d_b	M
4.2 l	V8	84	139.5	191.6	90	109	123	83	2177 g

Table 2.25: *Main characteristics of the Competitor L cylinder head*

d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}
26	10	4.5	4.5	4.5

Table 2.26: *Thickness values of the Competitor L component*

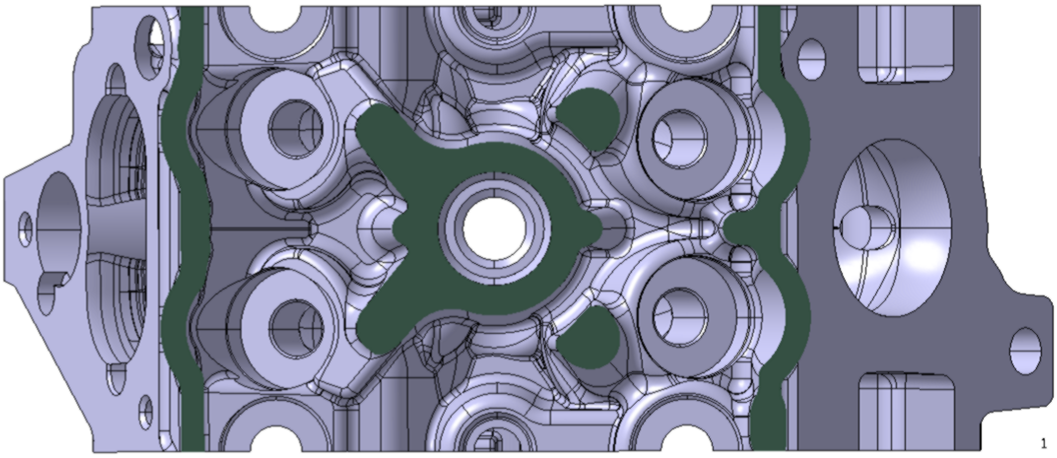


Figure 2.38: *Competitor L ribs layout*

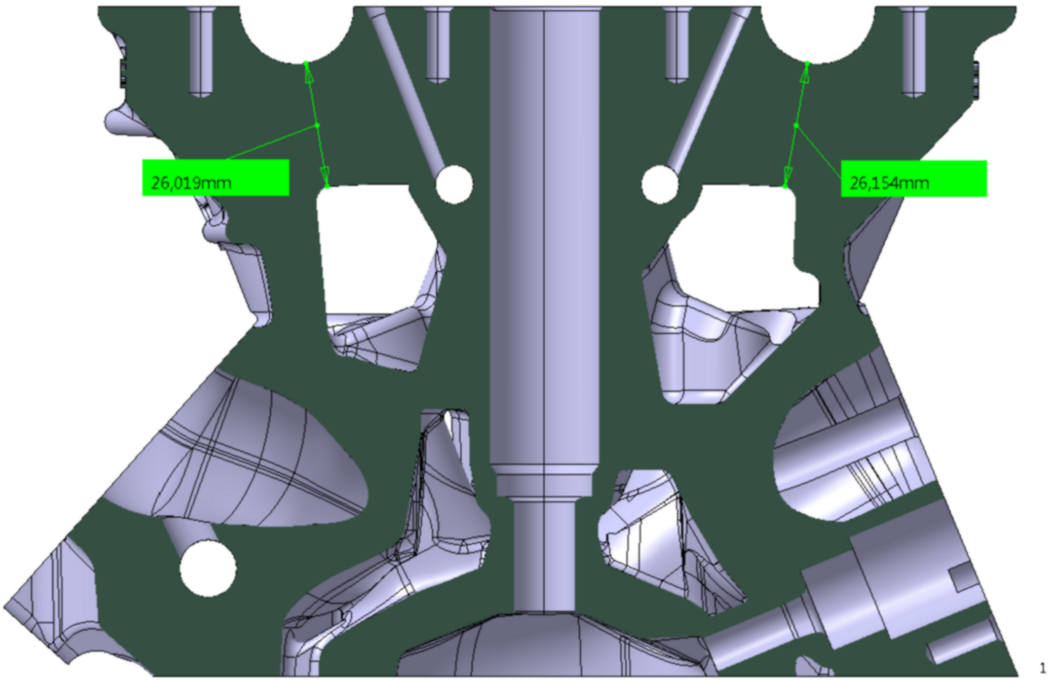


Figure 2.39: *Competitor L camshaft bridge section view*

models seen previously, the oil channel outlet is at the side of the bushing, feeding the camshaft from the side instead of the bottom, like it always happened in the previous elements. For what concerns the wall thickness, they are on the average, since this is a mass produced vehicle. However, the weight of this component is really low, slightly higher than the one of the component discussed in subsection 2.3.10, which is mounted on a limited edition vehicle where the final price was not so relevant as in this case.

2.4 Considerations and final remarks

The results obtained from the benchmark activity are shown in Table 2.27. For what concerns the overall dimensions, the reference cylinder head is the bulkiest one analysed, with the largest distance between the camshafts, between the two manifold flanges and the second longest overall length. This is due mainly to three design choices which have a negative impact on the packaging:

- **Camshafts**: their position has an offset with respect to the cylinder head bolt interfaces, which increases the overall dimensions and should lead to a weight penalty due to the additional material necessary to have more distant external walls. This solution is used only in the components explained in the Subsections 2.3.1 and 2.3.2, which have a similar overall length. However, the advantages in the assembly phase are really consistent, since it is possible to assemble the cylinder head in a separate station and then clamp it later on the engine block and gasket, which is a process impossible to do with the more compact layout and can save a lot of time, especially high volume production engines. The ideal technical solution would be to have a separate cam-carrier from the cylinder head, like the components described in Subsections 2.3.8, 2.3.9 and 2.3.11, but a careful analysis of the final stiffness of the component must be performed, since the division in two parts of the cylinder head might lead to an overall increase of compliance, causing small clearance errors in the valvetrain system.

- **Hydraulic lash adjusters**: differently from the majority of the other cylinder heads equipped with a finger follower system, the *REFERENCE* has the hydraulic lash adjusters fitted on the external walls, which requires the thickness of this region to be larger to withstand the HLA loads and the position of the wall to have a larger offset. The only other component to share the same layout is the one sown in subsection 2.3.2.

- **Oil channels**: since the position of the hydraulic lash adjusters is unusual, the oil channels are located on the sides of the component instead of being in the central part. This position is more common for the hydraulic tappet solution, like the components described in Subsections 2.3.1, 2.3.3, 2.3.4 and 2.3.5, but in all those cases the channels are located inside the external walls. On the other hand, the ducts of the reference cylinder head are located internally, which requires more material with respect to the other solution and additional ribs and stiffeners to support it. In fact, it is possible to get rid of all those elements positioning the ducts on the walls, as it is possible to see in subsection 2.3.2.

For what concerns the wall thickness, the reference component shows one of the lowest thickness of the wall between the combustion chamber and the water jacket and the thickness of the oil ducts; the average one is pretty good but the one between the water jacket and the lower oil core is the second highest of all, showing that there is still room for improvements. Despite the packaging and the design choices described before, the REFERENCE shows to be one of the lightest cylinder head analysed, being third on the weight comparison[2].

From this activity we can conclude that the weight is influenced by a lot of factors which are interacting between each other; the external dimensions give generally the largest contribution, followed then by the arrangement of the internal components (oil ducts, HLA, camshaft supports), which has to be defined really early in the project development. Also the wall thickness has a great importance, but since the values analysed were already quite low, the influence was lower than expected; a further decrease in the values would increase a lot the manufacturing costs and could not be applied in all areas, since the stresses and the strains could rise up to some values causing the failure of the component.

[2] Cylinder heads with separated cam-carrier were not considered

Name	*Disp [l]*	*Layout*	*Dimensions [mm]*							*Thickness [mm]*					*M [g]*
			D	h	l	w	d_{cs}	d_f	d_b	d_{bridge}	$t_{wj,c}$	$t_{wj,o}$	t_{avg}	t_{ducts}	
REFERENCE	4.4	V8	88	140	248.3	98	155.6	205.4	96	25	7.5	5	4	4	2685
Competitor A	3.0	Boxer 6	91	147	251	118	147	183.5	100	45	7	4.5	4	6	3710
Competitor B	3.0	Boxer 6	102	146.7	242	118	135	184.6	100	11	7	3.5	3	3	3228
Competitor C	3.0	V6	96	155	213	108	112.1	177.98	104	31	11	4.2	4.2	4.2	2698
Competitor D	4.8	V8	96	155	205	108	112.1	176.5	104	37	11	4.2	4.2	4	2828
Competitor E	1.9	In-line 3	94	145	229	105	146.9	153.3	89	29.6	8	4	3.5 - 5	3.5	2748
Competitor F*	2.0	In-line 4	74	139.6	197	88	96.5	193.8	85	13	10.5	5.5	5	5	2816
Competitor G	4.0	Boxer 6	102	178	214.9	118	118	168.1	100	25	10.5	4	3.5	3.5	2714
Competitor H*†	3.0	V6	84	138	223.1	93	-	232	87	-	7	4.5	4	-	2946
Competitor I†	4.0	V8	84	138	210	93	-	160.1	87	-	6	4.5	5	-	2262
Competitor J	4.6	V8	95	145	173.7	105	101.1	129.1	96	5	10	4	3	4.5	2106
Competitor K*†	1.4	In-line 4	76.5	115.4	220.5	82	-	221	83	-	8.5	4.5	5	4.5	2440
Competitor L	4.2	V8	84	139.5	191.6	90	109	123	83	26	10	4.5	4.5	4.5	2177

* Integrated exhaust manifold

† Camshaft integrated in the engine cover

Table 2.27: *Benchmark activity results*

Chapter 3

Structural optimization and analysis definition

IN this chapter, some basic knowledge about structural optimization will be given, with focus on how the results will be obtained and on their interpretation. Then, a description of the starting geometry for the optimization will be given. Next, the definition of the boundary condition will be addressed, with the different loads, constraints and the problems which had to be faced, which led to different simplifications and assumptions. Finally, the optimization phase results will be shown and commented.

3.1 Fundamentals of optimization

3.1.1 Introduction

When designing a product, it is obvious that the designer goal is to achieve the best design which can guarantee the maximum performance possible. In order to do so, the *optimum* design must be found. The word optimum can be defined as *"the greatest degree or best result obtained or obtainable under specific conditions"*. Let's notice for a moment that generic terms were used: *result* and *specific conditions*. It's up to the designer to define which is the *performance indicator*: it can be weight, volume, stiffness, use of material, total production cost, processing time, perceived quality, drag coefficient just to name a few. At the contrary, the *specific conditions* are defined by the environment in which the designer is working: can be budget, maximum dimensions, manufacturing constraints, process limitations, materials, *etc*. It is clear that the final design will be the result of a *compromise*; the task of the designer can be summarised then in *finding the best compromise*, i.e. *an optimal design*. The parameter which is used to quantify the result is called *objective*, which is limited by the specific conditions, which are called *constraints*.

Let's introduce some technical terminology now, which will be used later in this thesis work:

- **Design variable**: it is the structural parameter which is free to be modified during the optimization process. There are different typologies of variables, which can be the shape of a part, its topology and the set of layers of a composite material just to name some.

- **Design space**: it is the set of parts or elements which can be modified during the optimization process based on the design variable. To clear any doubt, the design variable can be defined as the shape of a part, while the design space is the set of elements which are part of the shape which can be modified to reach the optimum.

- **Response**: the measurements of the system performance, i.e. the parameters on which we are interested. It can be mass, volume, stiffness, stress, strain, fatigue life, safety factor, ...

- **Objective function**: can be defined as the goal of the optimization analysis. It represents the most important property of the design (defined by a response), which has to be minimized or maximized. Example: maximize the stiffness of a component.

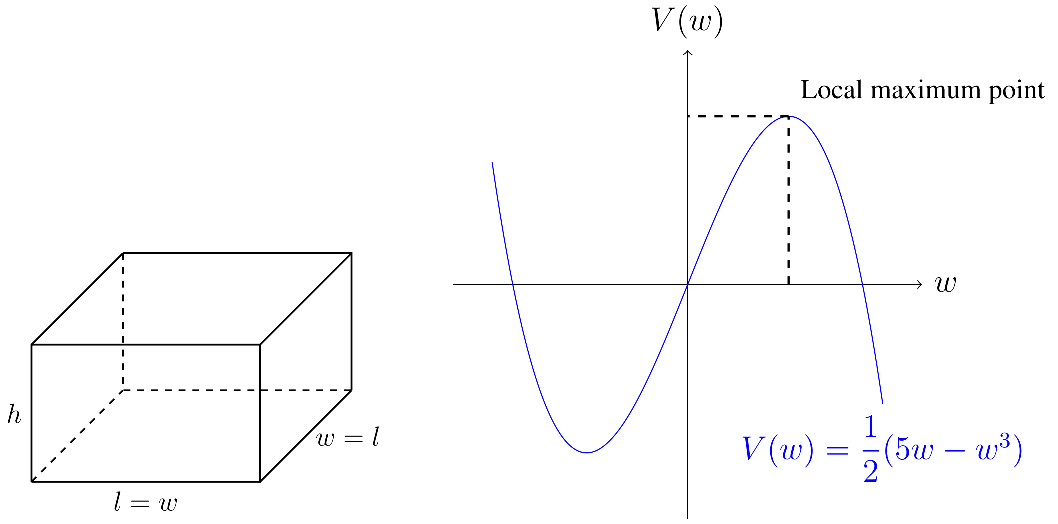
- **Design constraint functions**: they are the restriction placed on some responses of the optimization problem. Example: maximize the stiffness of a part but with a maximum weight 40% of the initial one.

3.1.2 A trivial example

In order to better understand how the optimization process work, let's consider the following simple example:

> *It is needed to construct a closed box with a squared base surface with only 10* m^2 *of material. Assuming that all the material will be used, the box dimensions in order to have the maximum volume must be found.*

Let's start with associating the terms listed in the previous section with the data of this problem. Looking at the figure 3.1a, there are two design variables (the edges w and h), two design space (the length of those edges), two responses (the maximum surface of the box and its volume), one objective (maximization of the box volume) and one design constraint (the maximum box surface).



(a) *Design variables* **(b)** *Result function*

Figure 3.1: *Example problem description*

The next step is to obtain some algebraic equations from the relationships between the data. Let's start with the maximum surface, which will give the *design constraint function*:

$$S_{box} = S_{base} + S_{sides} = 2lw + 2lh + 2wh = 2w^2 + 4wh = 10 \tag{3.1}$$

The *objective function* at the contrary is derived from the volume equation of the box:

$$V_{box} = lwh = w^2 h \tag{3.2}$$

If the equation 3.1 is solved for h, the constraint function becomes:

$$h = \frac{10 - 2w^2}{4w} = \frac{5 - w^2}{2w} \tag{3.3}$$

Replacing equation 3.3 in equation 3.2, the volume of the box as function of the base length w is obtained:

$$V(w) = w^2 \left(\frac{5 - w^2}{2w} \right) = \frac{1}{2}(5w - w^3) \tag{3.4}$$

Equation 3.4 is the *result function* of this problem and it is shown in figure 3.1b; finding the maximum volume of the box means *finding the point of maximum of this function* through its derivatives:

$$V'(w) = \frac{1}{2}(5 - 3w^2) \qquad \text{and} \qquad V''(w) = -3w \tag{3.5}$$

The stagnation points of the result equation can be easily found by setting equal to zero the first derivative:

$$V'(w) = 0 \quad \longrightarrow \quad w_{1,2} = \pm\sqrt{\frac{5}{3}} = \pm 1.2910 \tag{3.6}$$

Since it is impossible to deal with negative lengths, from equation 3.6 it can be concluded that the optimal base side length should be 1.291 m. Replacing this value on equation 3.3, the value of the height is found.

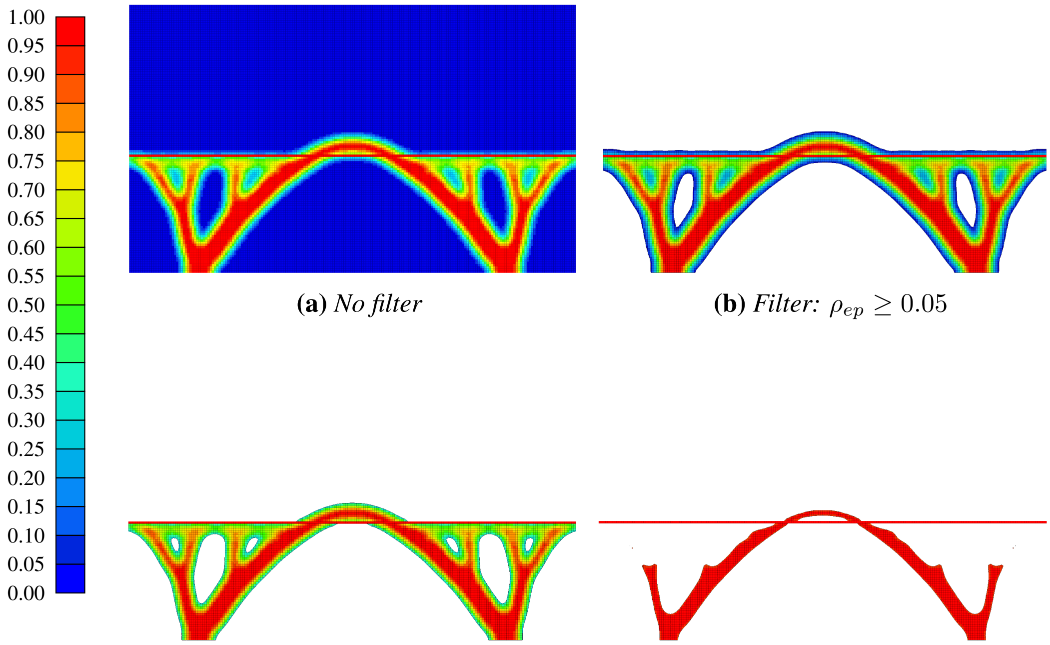
This trivial yet effective example helps to understand how the optimization solvers work. They retrieve the constraints and the objective functions from the geometry and the constraints defined by the user, and then solve them looking for the minimum or maximum points. Obviously the objective functions are much more complex than the one seen in figure 3.1b. They are usually defined in a 3D space and can have numerous local stagnation points; that's the reason why it is usually referred to them as *local* optimum points, because the design space definition rules the spectrum of the possible solutions.

3.1.3 Topological optimization

In all types of structural optimization, the topological one concerns the *material distribution and connection between members of a component*. Since the goal of this thesis work is to reduce the weight of a cylinder head section, the optimization strategy to be followed is the one that allows to *vary the material properties*, in order to save weight. For this reason, the topological optimization is the most suited type for this kind of work.

The characteristics of this optimization strategy do not differ from the examples shown before. In order to reach the objective, the software acts on the *density* of the elements included in the design space, using an *equivalent density*. This method is known in the literature as *SIMP*[1] and relies on a very simple assumption: *the Young modulus of a material is linearly dependent to its density*. This might sound as a strong hypothesis, but it is consistent with our understanding of materials [28]. To sum this theory up with an example, if an element has a 20% decrease in density it will have at the same time a 20% decrease in its stiffness. The equivalent density ranges from 0 to 1, where zero equivalences to a void material, while 1 equivalence to full material. Theoretically only the two extreme values should exist (zero and one), because an intermediate value would represent a porous metal, but an optimization with a large number of discrete variables is computationally prohibitive; for this reason, the *equivalent density* is represented as a continuous variable [29]. This

[1] Solid Isotropic Material with Penalization



(a) *No filter* **(b)** *Filter:* $\rho_{ep} \geq 0.05$

(c) *Filter:* $\rho_{ep} \geq 0.30$ **(d)** *Filter:* $\rho_{ep} \geq 0.85$

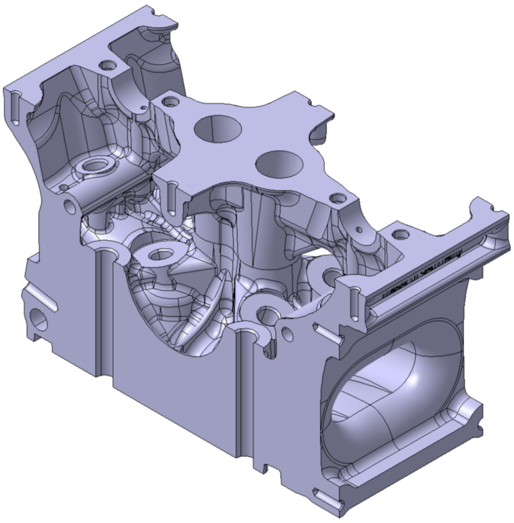
Figure 3.2: *Topological optimization example of a bridge; element density result. Objective: maximization of the stiffness - Constraint: final mass 15% of the initial one*

causes also the stiffness of an element to represented as a continuous equation, which is displayed below:

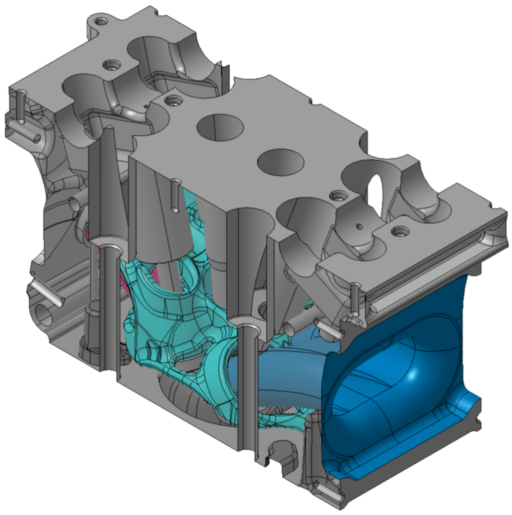
$$[K](\rho_{eq}) = \rho_{eq}^{p}[K] \tag{3.7}$$

where ρ_{eq} is the equivalent density, $[K](\rho_{eq})$ is the stiffness matrix of the optimized element as function of the equivalent density, $[K]$ is the original stiffness matrix and p is the *penalty coefficient*. Since the equivalent density ranges from 0 to 1, it is clear that the larger is the penalty coefficient the closer the density will be to its extreme values, leading thus to a material behaviour closer to the reality.

It is worth remarking that the result output by the software *is not the optimal shape itself, but the values of the elements density of the design space*. It's the designer's task to judge then which is the shape to be adopted, considering the element density. In order to better clarify this aspect, an example is shown in Figure 3.2, where a topological optimization of a bridge is performed. The force was distributed along a longitudinal line of the structure, to simulate the weight of the bridge itself. The elements on which the load was applied were considered outside the design space, also to have a ”straight line” where vehicles could transit. The result obtained is



(a) *Machined original section*



(b) *Geometry for the optimization*

Figure 3.3: *View of the sections of the reference cylinder head*

shown in figure 3.2a, where it is possible to see that the *final shape is the same of the starting one* and *no element has been removed* from the geometry. The only parameter which has changed is the element equivalent density (usually the lower threshold is set at 0.1%). Notice that the elements outside the design space are shown as a red line with the maximum equivalent density (1.00). In figures 3.2b and 3.2c the same result is shown but with different masking density thresholds. In figure 3.2d the minimum density value shown is really high. For this reason, the elements outside the design space are no more connected to the design ones: that's why, as mentioned earlier, is the designer's responsibility to choose the final optimal shape. The threshold value for the density to be used is not fixed and depends on the simulation, on the objective function and on the design constraints; in this phase, designer's experience and skills are fundamental.

3.2 Geometry and software definition

Form this point on, the discussions done before for the structural optimization[2] will be applied to the reference cylinder head, described in section 2.2.

Before starting with the optimization process, some words have to be spent on how this component is usually designed with the CAD softwares. Usually, a *manufacturing*

[2]From now on, the terms "structural optimization" will be used referring to the *topological optimization*

oriented design is used, which consists in drawing the component by considering the manufacturing process which will be used. In this case, the first step consists in designing the different parts used in the casting process: the base plate with the combustion chamber and the water jacket, the mold sides and the cores. Then, those parts are subtracted from a block of material using *boolean* operations, obtaining thus the part which is the result of the casting process, the so called *raw part*. Finally, bodies which represent the material removed with the machining operations are drawn and are then subtracted from the raw part to obtain the *machined part*.

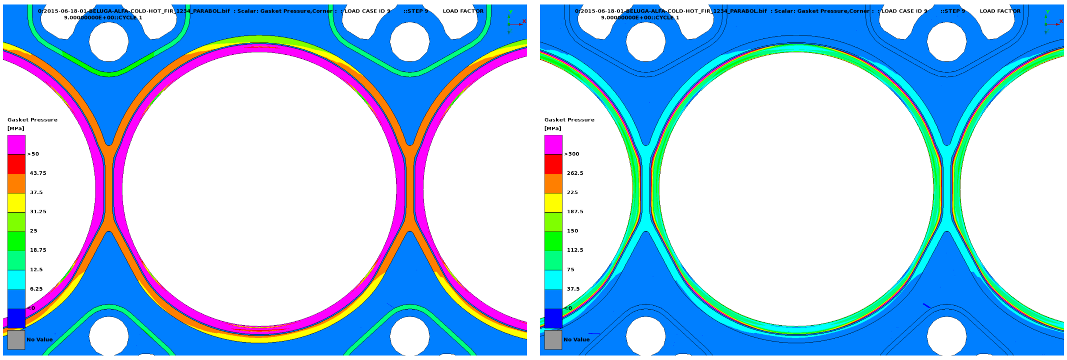
The machined part (Figure 2.1) is not so suitable for this kind of analysis. In fact the design space obtainable from this geometry is really small, leading to results which are not satisfactory. In order to obtain more meaningful results, a section of the raw part is considered, with the oil cores not present. This new geometry allows to exploit *all the space* occupied by those cores as design space, giving more freedom to the software and thus obtaining more optimal solutions. The machining bodies are then removed.

Another problem however rose. Since in the oil cores is included also the space for the valvetrain components, bodies of the envelope of the motion of the hydraulic lash adjusters and of the roller finger followers had to be created. These were then removed from the section, in order to avoid any interference. The final geometry used for the optimization analysis is shown in figure 3.3b, with the side walls transparent to highlight the absence of the oil cores.

For what concerns the software point of view, the workflow is split in general between three different programs:

- **Pre-processor**: is the one in charge of transforming the CAD part into a finite element model with its boundary conditions. The output of this software is a file which will be used by the next program.
- **Solver**: is the core of the process, the one which is in charge of performing the calculations given the input file from the pre-processor.
- **Post-processor**: the output of the solver is fed to this software, which allows to analyse and evaluate the results of the solver

It is obvious that a *coherent* software code must be used through all the process, i.e. that the output file of the pre-processor can be read by the solver and in turn that the post-processor is able to read solver results. The preprocessor used for the following analysis is *Hypermesh*, while the solver used is *OptiStruct* and the post-processor selected is *Hyperview*. Another software, *solidThinking Inspire*, which can be used as pre-processor and post-processor, was also considered at the start of this work. It was later abandoned because of stability issues and limited analysis potential.



(a) *Max legend value: 50 MPa* **(b)** *Max legend value: 300 MPa*

Figure 3.4: *Gasket pressure analysis of the reference cylinder head*

3.3 Boundary conditions definition

In this section the boundary conditions applied to the geometry to be optimized will be explained.

3.3.1 The gasket problem

As already explained in section 1.3.1, the gasket plays a fundamental role in the engine performance and behaviour. It is also the responsible of all the forces exchanged between the cylinder head and the cylinder block, thus its modelling is of paramount importance. However it is a highly anisotropic element, which mechanical properties change from point to point. Let's compare for example the full bead section and the stopper one. The first is the responsible of the sealing, which is guaranteed only when the two bead in contact are subjected to a *plastic deformation*, while the second bears the additional loads during the manufacturing, acting like a *high stiffness spring*. For this reason the behaviour of this element is highly *non-linear*, which leads to *non-linear analysis* and thus more computational power required. In addition, the loads on the gasket change depending on the combustion pressure, on the thermal expansion of the components and on other factors which are not constant over time.

These problem could be solved by performing a full engine analysis, with firing cycles for each cylinder, different temperature fields and an accurate description of the gasket behaviour[3]. This leads to accurate result, but requires high computational power and the whole geometries of the cylinder head, of the gasket and of the block. Since the focus of this work is just over a single cylinder section and optimization

[3]The behaviour is described with data for different *loading-unloading* cycles. In fact the gasket unloading curve (deformation over pressure) depends on the unloading point

analysis with non-linear contacts is computationally prohibitive, the model had to be simplified.

The values of the gasket contact pressures in different loading conditions were already known thanks to prior investigations and are shown in picture 3.4. Two different maximum values of the legend were used. In figure 3.4a, a maximum value of 50 MPa was selected, which allowed to see the contribution of the halfbead (green lines) and of the fullbead (second external ring surrounding the cylinder). It was decided then to model those elements by applying respectively a 20 MPa and a 40 MPa pressure on the gasket interface of the cylinder head in the interested areas. For what concerns the stopper contribution, visible better in figure 3.4b where a maximum value of the legend of 300 MPa was adopted, its contribution was quite difficult to assess. In fact, as it is possible to see, the pressure distribution varies along the diameter of the crown (the first one surrounding the cylinder) and in some diametrical spots. Some first optimization attempts with a stopper pressure of 150 MPa were performed, but with poor results. It was later decided to constrain the vertical nodal motion instead, for different reasons which will be explained later in subsection 3.3.3.

3.3.2 Loads and material

The loads which were applied to the model for this optimization run are now listed. It was decided to considered the worst case loading condition for each type of load, even if in a real case they do not happen together (i.e. intake valves and exhaust valves open in peak firing pressure condition).

- **CHD bolts**: the nominal preload force is 55 kN, but since the bolt interfaces are cut in half by the section planes, for each half bolt half of the force was considered, 27.5 kN. In order to better simulate the bolt pressing on the interface, the load was modelled as a *pressure*, by dividing the bolt force for the interface area. This led to a value of 146.9 MPa for the intake one and of 147.8 MPa for the exhaust one.

- **Fullbead** and **Halfbead**: As already discussed previously, the pressures were assumed as 40 MPa for the fullbead and 20 MPa for the halfbead.

- **Combustion pressure**: the peak firing one was assumed, which is 120 bar. It was modelled as a pressure orthogonal to the face of each element of the combustion chamber.

- **Intake valve spring**: 820 N for each valve, nodally distributed on the spring interface with a direction perpendicular to the interface, pressing on it.

- **Intake valve HLA**: 1600 N for each lash adjuster, nodally distributed on its seat with a direction perpendicular to the interface, pressing on it.

Description	*Symbol*	*Value*	*Measure unit*
Young modulus	E	$74\,000$	MPa
Poisson ratio	ν	0.35	-
Density	ρ	2.65×10^{-9}	$\mathrm{t/mm^3}$
Thermal expansion coefficient	A	2.45×10^{-5}	$1/\mathrm{K}$
Yielding stress	$R_{p0.2}$	200	MPa
Ultimate tensile stress	R_m	270	MPa
Elongation at fracture	$A5$	3.7297	%

Table 3.1: *Properties of the AlSi7MgCu0.5 T6 aluminium alloy*

- **Intake valve bearing**: 5120 N, assumed as split equally between the two screws holding the main cap and applied to the center nodes of their threaded parts. These nodes were then connected with rigid bodies[4] to the surrounding nodes of the thread. The force direction is parallel to the valve axis.
- **Exhaust valve spring**: 1050 N, modelled as the intake one.
- **Exhaust valve HLA**: 2240 N, modelled as the intake one.
- **Exhaust valve bearing**: 7460 N, modelled as the intake one.

A special mention should be made of the thermal loads. Despite the fact that they give the largest contribution to the component stress and deformation, they were not taken into account. Because the optimization strategy does not actually change the shape but varies the density of the design space elements, the interfaces for the convective heat exchange would have remained fixed throughout all the optimization cycle. Since the heat transfer with the surroundings is an important factor for the component cooling, that might have led to wrong result. They will be considered later, when the optimized revisions of the component will be designed.

The material used for the casting is the *AlSi7MgCu0.5*, an aluminium alloy, which castings are then subjected to a T6 thermal treatment (solutionizing plus artificial ageing). Its properties are shown in Table 3.1.

3.3.3 Constraints

The last fundamental aspect is the proper *constraining* of the model. For standard cylinder head analyses the component is held in place by the bolts, which preload

[4]Generally known as *rigid spiders*, or *RBE2 elements* using OptiStruct terminology

presses it against the cylinder block and the engine mounts fix this last component in the space. Since the thesis deals with a single cylinder only, an alternative to this approach had to be found. After different attempts, these constraints were adopted:

- **X direction**: This was the most critical constraint to model, since it lies on the section planes of the components. In order to have a constraint condition which could reflect a real case, the symmetry plane between the cylinders should be respected. This could be done by constraining the rotation of the nodes on those planes, but unfortunately it cannot be done on 3D element nodes. Different attempts were done to constrain it anyway, by creating a layer of 2D elements, by connecting the nodes with some rigid bodies to a layer of rigid shell elements and by trying an orthotropic material with infinite bending resistance. Unfortunately none of this approach has been successful, so only the displacement in the X direction was constrained. This assumption is quite strong, because does not allow any room for the material in this plane to deform. At the contrary, in a real case there is other material beyond the section planes, which allows for a small deformation. This obviously leads to an overestimation of the stresses in these areas, but since the mechanical deformations are quite small this approach was anyway chosen.

- **Z direction**: as discussed before during the modelling of the gasket, the Z direction was constrained by suppressing the vertical motion of the nodes at the stopper interface. The major benefit is that if the load on the bolts and on the other components changes, the reaction force exchanged at those nodes changes accordingly. This is similar to what happens in a real case: the pressure at the bead contact interfaces remains more or less constant, while any excess of pressure (larger bolt preload while assembling, thermal expansion of the cylinder head which increases bolt length, . . .) is borne entirely by the stopper. This will be useful for the thermal analysis, which will be used for the design revisions validation. In addition, this assumption is not even strong: the stopper is usually made of steel, which Young modulus is almost three times larger than the aluminium alloy used; for this reason the vertical displacement of those nodes would be really limited.

- **Y direction**: this constraint was the easiest to model. The choice was between locking the Y motion of a single node in the center of the combustion chamber or the Y motion of all the nodes lying on the xz mid-plane. Both approach ended with the same result, but the second was used in order to avoid any stress concentration on the single constrained point.

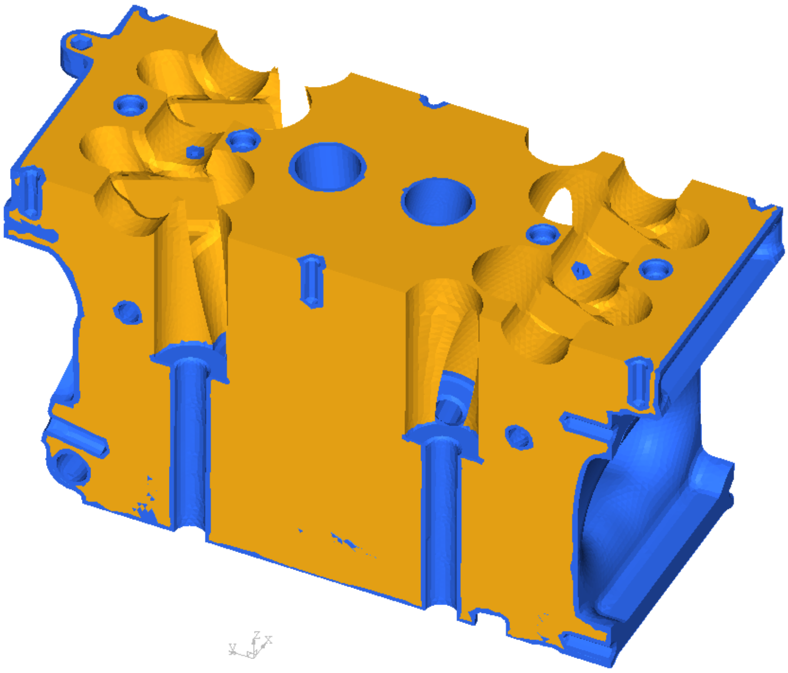


Figure 3.5: *Model after the meshing process and the design space definition (in yellow). One single layer of non-design elements*

3.4 Optimization results

3.4.1 Model set-up

After all this discussion, the model was finally ready to be optimized. Several optimization tests and analysis were run before in order to validate the aforementioned boundary conditions. Also a model with two half cylinders added to the original section planes was considered, but there was no difference in the results. This process was also useful to evaluate the computing power at disposal and to retrieve the model size limit which could guarantee accurate results in an acceptable time.

The model was meshed first using 2D first order shell elements (*tria elements*). An average size of 1 mm in the most critical part, like on the bolt interface, on the gasket ones and on the combustion chamber, was selected. A size of 1.5 mm in other regions where loads were applied was used, while the size for the remaining surfaces was 3 mm. From this skin mesh, a 3D mesh made of first order tetra element was obtained (*tetra elements*), setting a *grow ratio coefficient* as low as possible, in order to obtain a internal mesh size not so different form the external one. This is an important aspect in topological optimization, because the denser is the mesh, the more clear is the final result. The final model resulted in 159 000 2D elements and in

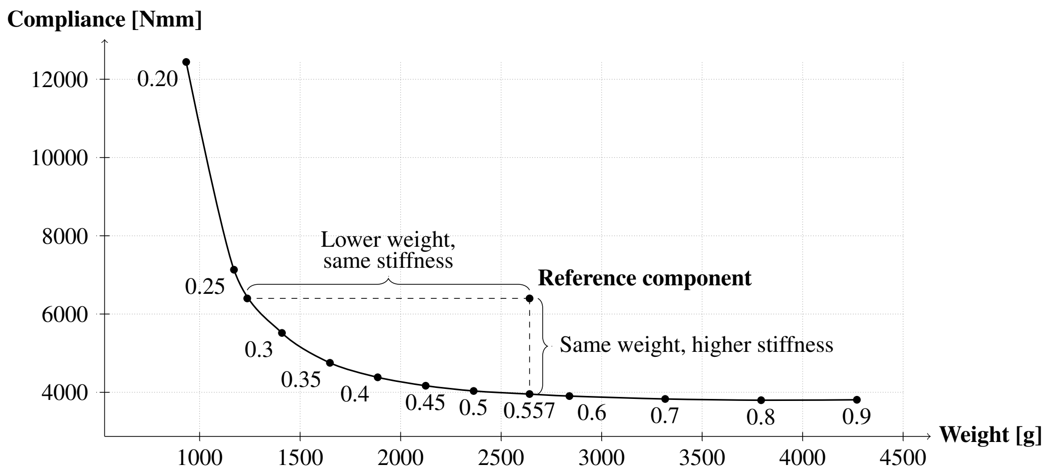


Figure 3.6: *Optimum optimization curve, with each point labelled with the maximum mass fraction of the overall component*

2 598 000 3D elements.

After this step, the design space was defined. Since no functional element could be modified, it was decided to assign as *non-design space* the first layer of elements surrounding the external shape and the main features, like the ducts, the water jacket and the bolt channels. In addition, every element which was directly loaded was assigned to the non-design space, like the spring interfaces, the HLA ones and the main cap screw bores. The final result is shown in Figure 3.5, where the design space is coloured in yellow.

3.4.2 Optimum curve definition

The last variable to evaluate is the design constraint for the *weight reduction*, i.e. how much material the software has to ”remove” from the design space. Best practices and personal experience suggest to *never* start directly with an optimization with an objective function aimed to minimize the weight. This is because of the difficulty to properly define the design constraints. While a maximum weight target is easy to set and understand, the definition of other constraints is more critical and not immediate to understand as the weight one (such as stiffness, natural modes, . . .). Additionally, setting a constraint on the maximum stress of the component is never a reliable choice, because of stress concentration due to nodal load and mesh irregularities, which could lead to ”wrong” results or infeasible optimizations. For these reasons, the mass was defined as the design constraint.

In order to evaluate which is the trade-off value which would have guaranteed the best compromise, a set of different optimizations were run on the same model but with

Weight constraint	*Compl. 1-el [Nmm]*	*Compl. 2-el [Nmm]*	*Variation*
0.45	4170	4501	+7.94%
0.4	4388	5150	+17.37%
0.35	4763	6892	+44.7%

Table 3.2: *Comparison of the compliance values for the different non-design element layer choices for different maximum final weight constraint*

different final weight target. The objective function in all cases was the minimization of the *compliance*. While in the literature this term refers to the reciprocal of the stiffness, in the finite element softwares it is defined as the *sum of the products between each nodal displacement and the force which is acting on it, namely the work of the loads on the component*. However, it is easy to see that a lower compliance means a lower overall deformation of the component, thus a larger stiffness. Twelve optimizations were run with different maximum mass constrained, which results are displayed in Figure 3.6. The value assumed by the mass constraint is labelled to the points appearing on the curve.

As it is possible to see, the variation in compliance is minimal until a value of the final weight of 45% is selected, while it increases abruptly when a value below 30% is chosen. This means that a value inside this window must be selected. All the optimization results were checked and analysed in their shape; after some discussions, the shape with a final weight of 45% of the initial one was selected. In addition, a final weight of around 2200 g seemed a quite realistic target for the future design revisions.

3.4.3 Optimal shape result

The optimal curve was obtained considering only *one single row of non-design space elements*, which means basically a wall thickness ranging from 0.8 to 1.5 mm for the functional parts. In order to have a result more meaningful in terms of manufacturability, some optimization runs were performed with *two layers of non-design elements*, a solution which guarantees more consistent result. The final weight targets selected were 0.45, 0.4 and 0.35. The results are shown in Table 3.2.

As it is possible to see, as the weight constraint becomes more and more stringent, the compliance of the components with 2-elements layer as a non-design space abruptly increases. This effect happens because for the same weight target the solver has less ”weight” at his disposal to distribute where needed. In any case, a 7.94% in increase in compliance for the 45% final weight target was considered more than acceptable, because it is well below the value of the original component. For these

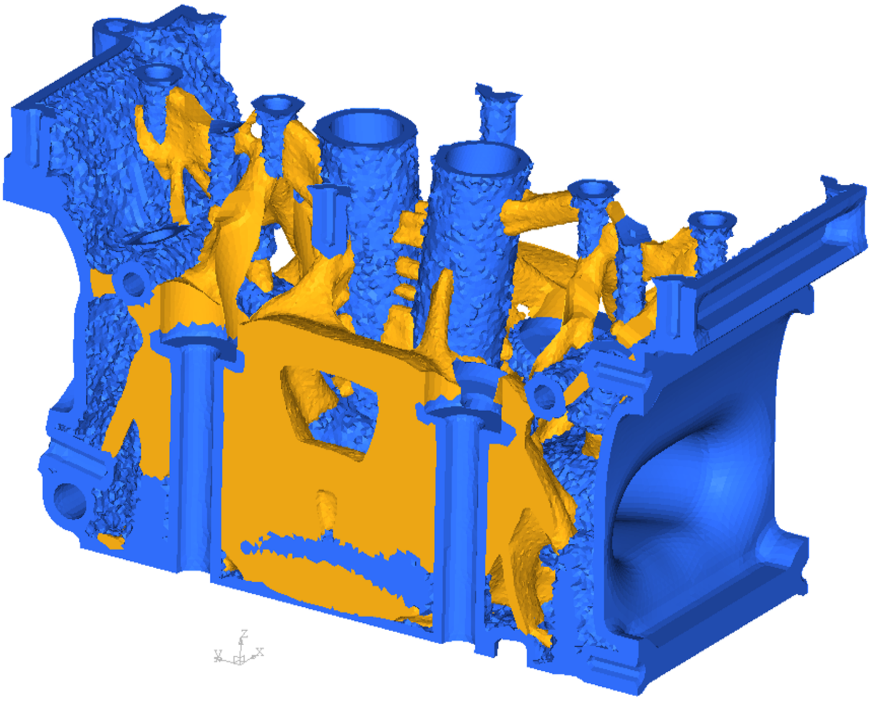


Figure 3.7: *Optimal shape result, with 0.35 density threshold view*

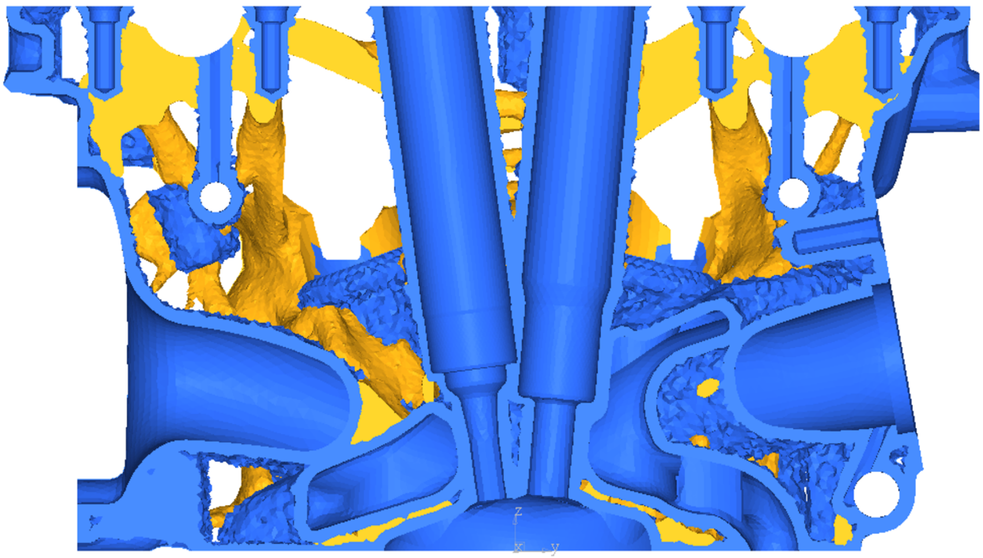


Figure 3.8: *Optimal shape result, with 0.35 density threshold view. YZ mid plane section view*

reasons, this optimization step was defined as the *optimal one* for the implementation of the design revisions. It is shown in figures 3.7, 3.8, 3.9 and 3.10, where only elements with a equivalent density value above 0.35 are shown.

As it is possible to see, the final result is quite complex and far from being manufacturable for different reasons. First of all there are pockets where the material is not needed which are not easily accessible, like between the water jacket interstices and around the secondary air duct. In addition, some parts are connected by very thin ribs or beams, like for the camshaft bearing supports or between the central channels for the spark plug and the injector. Lastly, but not less important, a proper oil cores design and production would be impossible with this actual geometry, because of its complexity and because of the absence of a defined parting line.

Despite the complexity of the design-space shape, some very useful informations can be retrieved:

- **Camshaft supports**: the region below the supports shows the highest weight reduction potential. The supports are connected with the central channels and the external walls only by some beams, with the intake ones being thinner because of the lower preload of the intake springs. A non symmetrical design could be then implemented.

- **External walls**: the HLAs are supported by a single rib only, especially on the intake side. This differs completely form the original component. In addition, the stiffeners that were placed on the external walls are not useful and are gone.

- **Bolt interfaces**: They are connected transversally by a thick bridge, which then connects with the central channel with two ribs. The large presence of the material in this area shows that it is crucial for the component overall stiffness, but the results are a bit distorted by the presence of the X displacement constraint in this area.

- **Oil deck**: the wall thickness of this region could be lowered, since in some areas the non-design space seems enough to withstand the loads applied.

These speculations must be obviously verified by implementing those features in the design revisions and with the relative analysis. In addition, it has to be said that *thermal loads were not yet taken into account* and for this reason conservative choices must be made when implementing the new features.

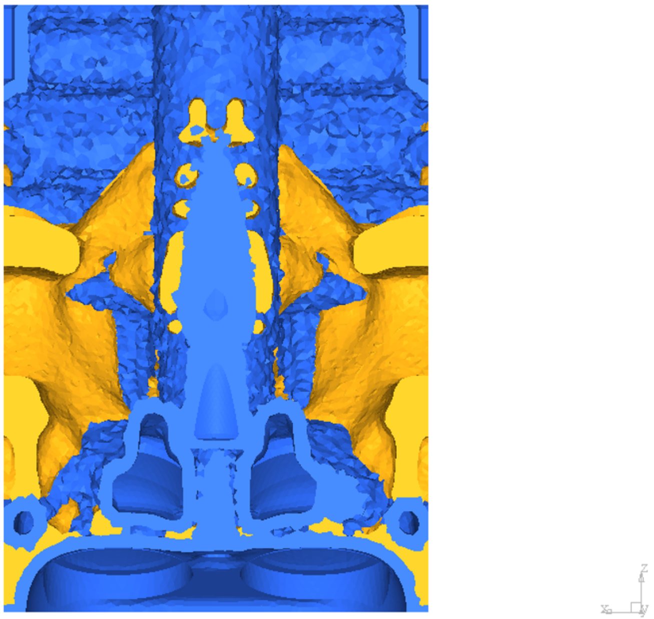


Figure 3.9: *Optimal shape result, with 0.35 density threshold view. XZ mid plane section view*

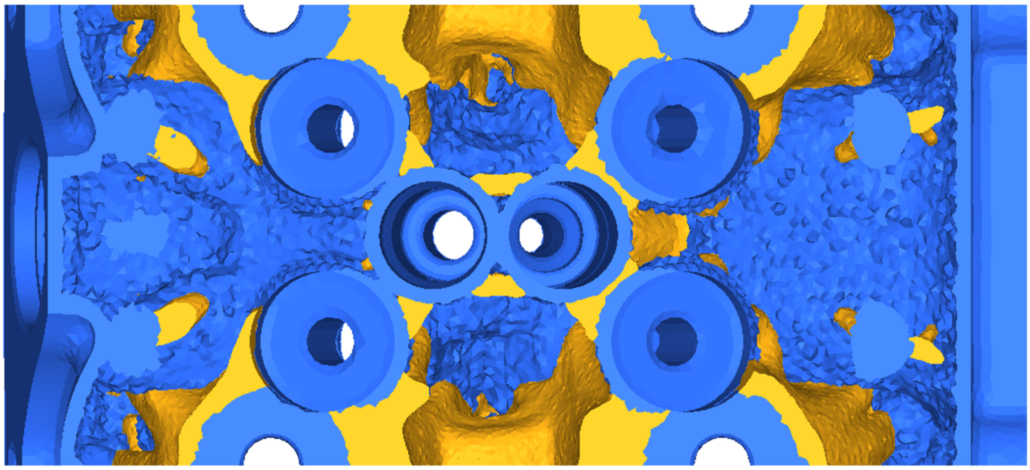


Figure 3.10: *Optimal shape result, with 0.35 density threshold view. XY mid plane section view*

Chapter 4

Design revisions implementation and validation

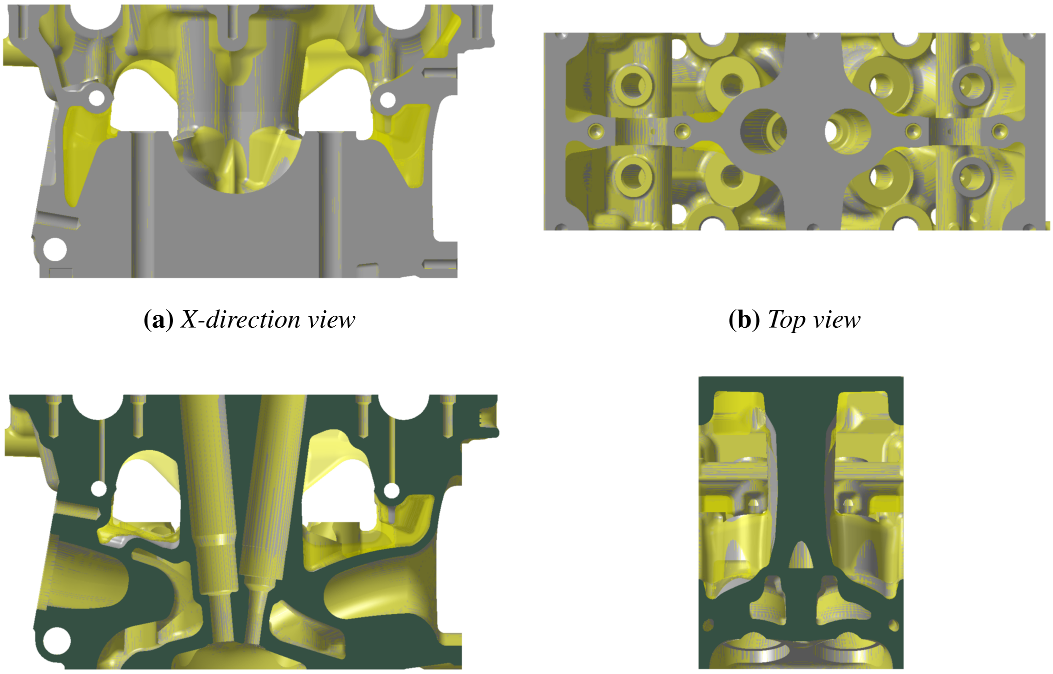
THE last part of the thesis work consisted in implementing the results from the optimization software into different design variants, validating them considering also the thermal loads. First, the modifications of the reference geometry were characterized, then applied to the reference component and subsequently simulated. Finally results were collected and analysed.

4.1 Optimization steps

The part obtained as a result from the software and shown in Figures 3.7, 3.8, 3.9 and 3.10 has been used as a guideline for the versions of the component. After overlapping the optimized shape on the original one, the optimization areas were found. Then it was decided to split the modifications in *different steps* and to group those modifications in *three different design variants*, which will be called from now on *revisions*.

- **Step I**: material was removed between the gasket interface and the intake ducts thanks to a pocket shaped by offsetting the intake duct and the water jacket (Figure 3.8). Additional material could have been removed by changing the gasket interface, leading to a larger weight reduction, with a shape following the cylinder bore and the water jacket.

- **Step II**: the focus was on the camshaft bridge. Considering the figures 3.7 and 3.8, the abundance of material of the original component is massive. For what concerns the top view, the connections between the bolts and the nearby walls were extremely tapered, especially on the intake side, while for the yz mid-plane section the dimensions of the bridge windows was definitely increased. The



(a) *X-direction view* **(b)** *Top view*

(c) *YZ section plane view* **(d)** *XZ section plane view*

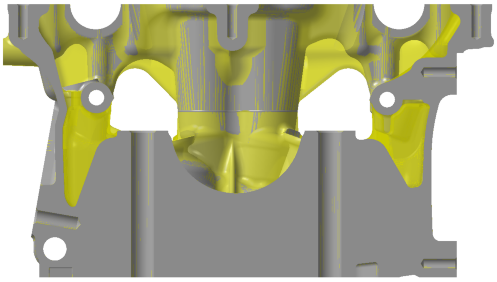
Figure 4.1: *Revision 01, with the reference component overlay in transparent yellow*

longitudinal bridge connecting the different central channels of the cylinders was reduced in thickness to a 8 mm distance to the engine cover interface, while the rib connecting the oil channel with the external wall at the bolt section plane was removed.

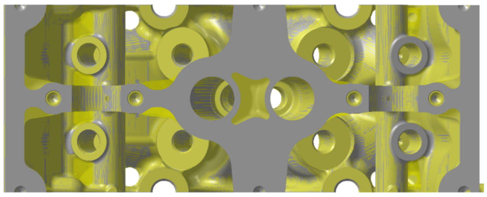
- **Step III**: the transversal ribs supporting the external walls to the core of the component have been removed. In fact, as shown in figures 3.7 and 3.8, there are no ribs connecting the external walls to the bolt interfaces, so they were removed. In addition, the transversal ribs over the intake duct and the exhaust one were removed, leaving only the ones supporting the hydraulic lash adjusters.

These three first steps, which were the easiest ones to implement, gave as a result the *Revision 1*, which can be seen in Figure 4.1.

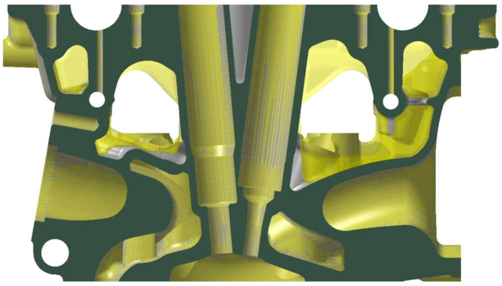
- **Step IV**: the geometry of the camshaft bridge was again modified. As it is possible to see from Figure 3.8, between the oil channels and the external walls



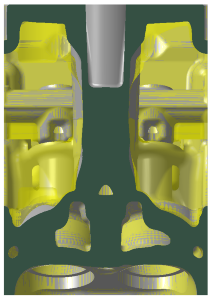
(a) *X-direction view*



(b) *Top view*



(c) *YZ section plane view*



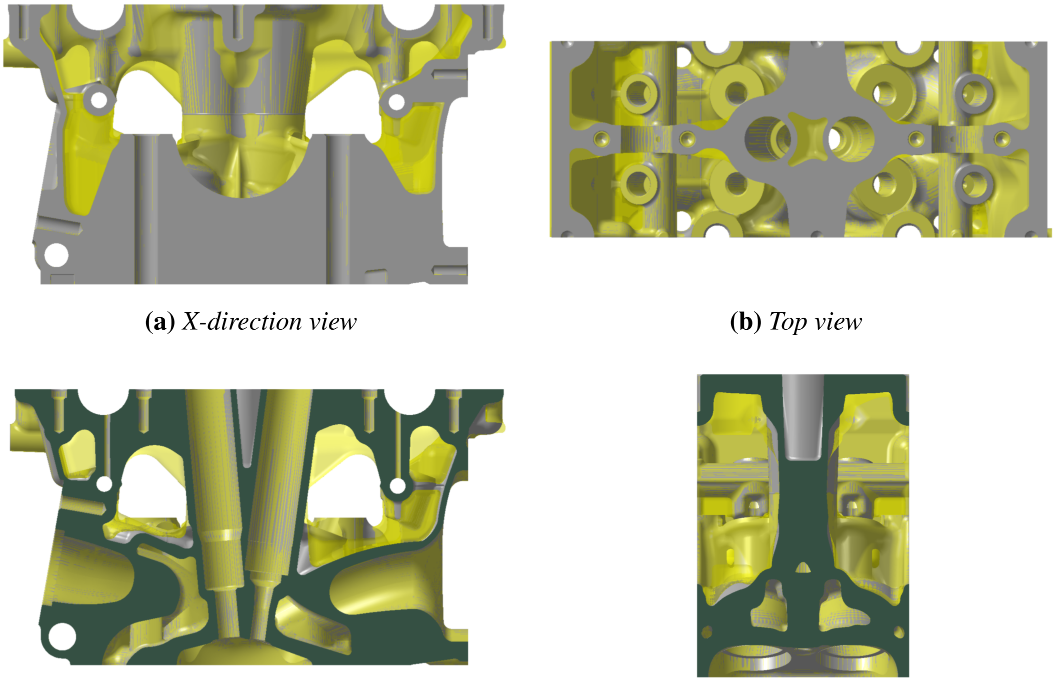
(d) *XZ section plane view*

Figure 4.2: *Revision 02, with the reference component overlay in transparent yellow*

there is no material. For this reason, a window in this area was implemented on both sides, with an offset of the oil ducts and of the external wall shapes.

- **Step V**: the oil deck surface has been heavily modified. First, the large beams connecting the spring interfaces to the central channel have been removed. Below the intake valves, some pockets were created, while below the exhaust ones a slot surrounding it was obtained. In addition, the longitudinal rib dimension was reduced as was also the wall thickness.

- **Step VI**: the central section with the channels for the fuel injector and the spark plug is originally very bulky. As the optimized shape suggested, there could be a lot of material to be removed. For this reason, a pocket was designed between the channels and on the longitudinal direction. Unfortunately, material removal was limited by two important factors. First, the rubber sealing shape of the head cover had to be respected, so the pocket shape had to follow the seal profile. Then, a connection between the two channels is not possible for a safety reason, because some fuel vapours coming from the injector could get in contact with



(a) *X-direction view*

(b) *Top view*

(c) *YZ section plane view*

(d) *XZ section plane view*

Figure 4.3: *Revision 03, with the reference component overlay in transparent yellow*

some ozone generated by the high voltage of the spark plug, which could then degenerate in a fire.

Those steps, which were more difficult to implement, together with the ones of the Revision 1, resulted in the *Revision 2*, visible in Figure 4.2.

- **Step VII**: in order to avoid excessive stress below the intake spring surface, the pockets size was reduced. In addition the machining geometry was modified in order to avoid sharp edges and small cut surfaces, which could result in stress concentration and in meshing problems.
- **Step VIII**: the material between the external wall at the intake side and the oil duct is removed, leaving the duct supported only by the hydraulic lash adjuster rib
- **Step IX**: wall thickness on the intake was made thinner, in order to compensate the weight added under the intake spring interfaces.

Component	*Steps*	*Weight [g]*	*Reduction [g]*	*Reduction [%]*
Reference	-	2685	-	-
Revision 01	I-III	2492	-193	-7.2%
Revision 02	I-VI	2360	-325	-12.1%
Revision 03	I-IX	2290	-395	-14.7%

Table 4.1: *Weight of the different design revisions*

These last three steps resulted in the last design modification of the original component, called *Revision 03*.

In Table 4.1 the values of the weight of the different design revisions are listed. As it is possible to see, Revision 01 is the one with the largest weight reduction contribution, because the geometry was modified in its easiest spots. The higher the step, the more difficult it was. Anyway, the *final weight reduction achieved was of the 14.7%*, which is a very good result; if the modifications were applied to all the other cylinder segments, there would be a *total reduction of 3160* g for this V8 engine. This achievement means a lower cost for the vehicle owner for the improved fuel consumption but also lower manufacturing costs for the manufacturer. In fact the lower weight does not reflect only in a lower price for the raw material purchasing but enables a series of secondary ”hidden” benefits, like lower energy required for the casting process, for the thermal treatments and lower machining operations.

4.2 Design revisions analysis

In this section the analysis of the different revisions will be discussed. First, the thermal analysis will be described, then the geometry modifications and the new boundary conditions that had to be applied are discussed.

4.2.1 Thermal analysis general workflow

The cylinder head, as already discussed in subsection 1.3.2, is subjected to large thermal loads generated by the exhaust gases and the combustion. The heat is dissipated by the water jacket, where the coolant flow subtracts a portion of the heat from the surrounding keeping the temperature of the component to an acceptable one. The excessive heat is then rejected in the environment through the engine coolant radiator.

A proper modelling and analysis of the thermal loads is of paramount importance in the cylinder head design, because it affects the engine performances and, even more

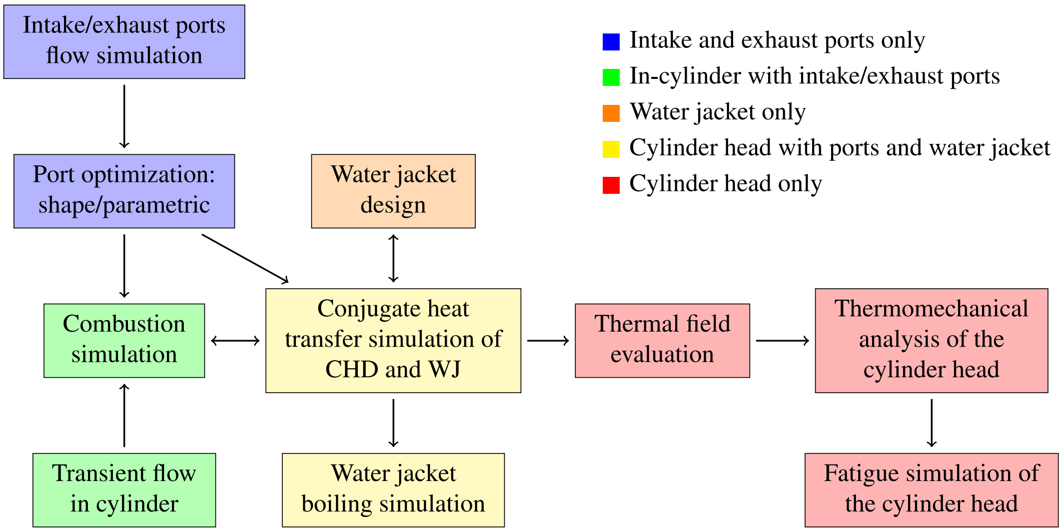


Figure 4.4: *Example of workflow for the thermal analysis of a cylinder head [30]*

importantly, the component life. Unfortunately it is not as trivial as it might seem; the complexity of the geometries involved and the large number of variables to be taken into account make it a very complex and long task, requiring numerous simulations repeated in an iterative process. In fact, the heat generated by the combustion raises the temperature of the walls, which then affects the water jacket design. The heat transfer is then affected by the coolant flow, its speed and its turbulence, which causes the wall temperature to change and decrease. However, the temperature of the wall changes the combustion dynamics, which once more affects every parameter once more. Additionally, other parameters like the turbulence levels of the intake and exhaust flows, the local boiling of the coolant and the combustion chamber design have to be taken into account, increasing again the complexity of the analysis.

In order to better understand this process, in Figure 4.4 a general workflow for the thermomechanical analysis of the cylinder head is shown. After the ports and the combustion chamber are designed in order to obtain the best performances, some combustion cycles simulations are run in order to maximize the power for different speeds and load conditions. In the mean time, the water jacket is shaped from previous designs and adapted to the new components. After this phase, iterative conjugate 3D CFD and FEM analysis analysis between the combustion chamber and the water jacket are run, with this latter component modified to meet the cooling requirements if necessary. Next step consists in mapping the temperature field of the cylinder head mesh nodes, which will be used later for a thermomechanical anaylsis and then for the fatigue simulation of the component [30].

	Convective coefficient [kW/(mm^2 K)]	*Temperature* [°C]
Ambient air	0.1053	25
Intake duct	0.3724	50
Water jacket	7.0	90
Combustion chamber	1.0	770
Exhaust gases	0.7	950

Table 4.2: *Thermal boundary conditions applied to the model*

4.2.2 Thermal loads modelling

For what concerns this thesis work, the workflow used is much more simple than the one described in the previous section. The reason for that is because CFD analysis are not in the scope of this subject, since the internal components have been already defined, and because of lack of time. For this reason, the analysis were limited only to FEM ones.

Since conjugate analysis were not considered, nodal temperatures of the component had to be retrieved in a different way. It was decided to resort to the *heat convection transfer theory*, because convective transfer coefficient and fluid temperatures could have been easily retrieved. The value used are shown in Table 4.2.

- **Ambient air**: the heat exchange was modelled both on the outside walls, except from the flanges and the ports, and on the internal ones, obtained with the oil cores. This leads to a large simplification of the heat transfer modelling at the oil deck level since these surfaces are continuously fed with oil sprayed by different nozzles. An accurate modelling would be really hard to obtain, because the surface in contact with the oil constantly changes, as also does the oil quantity and the anisotropy of the air-oil mixture. Even though the oil has a larger temperature than the ambient air one, it has a much larger cooling power. For these reasons this assumption was made. The convective transfer coefficient was retrieved from [31].

- **Intake duct**: was modelled on every element on the surfaces of the intake duct. The temperature was obtained from previous 1D CFD simulations of the global engine, which gave as a result a temperature of 50 °C. The convective heat transfer coefficient was retrieved from [31].

- **Water jacket**: for this contribution, a *best practice* was used. As approximation for the 1D CFD analysis, the water jacket heat transfer coefficient is assumed equal to the value of the engine speed expressed in rpm, 7 000 in this case. This approach was proved to provide very good results and for this reason was used for this analysis too. The temperature was chosen to be equal to the steady-state optimal one, 90 $°\mathrm{C}$.

- **Combustion chamber**: the temperature inside the combustion chamber varies depending on the crank angle, engine speed and engine load. For this reason, the most critical condition was considered. The data were obtained from previous 1D CFD analysis, which gave as a result an average temperature of 770 $°\mathrm{C}$ and an average convective coefficient of 1 $\mathrm{mW/(mm^2\,K)}$.
- **Exhaust gases**: the same approach used for the combustion chamber temperature definition was used for the exhaust gases temperature too. As a result, an average temperature of 950 $°\mathrm{C}$ and a convective coefficient of 0.7 $\mathrm{mW/(mm^2\,K)}$ were used.

Looking at Table 4.2 the aspect that stands out is the average exhaust temperature being larger than the combustion chamber one. This is due to the fact that this last component is filled with "fresh" mixture every cycle, something which lowers the average temperature a lot. On the other hand the exhaust duct is facing always hot gases, which can be from the analysed cylinder or from the adjacent ones during the exhaust phase. This causes the temperature to remain at high values, a condition which is critical for the turbine components and for the after-treatment devices.

These thermal boundary conditions defined until now were used for the thermal analysis of the cylinder head; however they led to different assumptions. First of all, the temperature of each contribution was considered uniform and fixed, while in a real case the point to point variation can be substantial, like for the water jacket and for the combustion chamber. Secondly, the *bolt preload* varies because of the different thermal expansion coefficients between steel and aluminium and changes from bolt to bolt; in this case, the maximum bolt preload under "hot" conditions, known thanks to prior investigations, was used for all bolts (68 kN). Lastly, the cylinder block plays an important role, not only for the deformation point of view, but also for the thermal point of view: since its temperature is lower than the cylinder head one, it deforms less and can subtract some heat from the gasket interface.

4.2.3 New boundary conditions and targets definition

The following step was to run some dummy thermomechanical simulations to evaluate the correctness of the boundary conditions and to validate them. Unfortunately three aspects had to be revised in order to better fit this kind of analysis: the *material*, the *x direction constraints* and the *evaluation targets*.

Description	*Symbol*	*Value*	*Measure unit*
Thermal expansion coefficient	A	2.45×10^{-5}	$1/\mathrm{K}$
Thermal conductivity	K	155	$\mathrm{mW/(mm\,K)}$
Specific heat	c_p	9.7×10^{8}	$\mathrm{mJ/(t\,K)}$
Yield stress	$R_{p0.2}$	200	MPa
Ultimate tensile stress	R_m	270	MPa
Elongation at fracture	$A5$	3.7297	$\%$

Table 4.3: *Thermal and non-linear properties of the material*

Material

Since a thermomechanical analysis was run, the material thermal properties, which are listed in Table 4.3, had to be added to the mechanical ones, shown in Table 3.1. Since the first analysis showed some spots with some stress above the yield stress of the material, it was decided to perform all the simulations with *non-linear* analysis. Differently from the linear analysis where the material properties are assumed constant and linear, the non-linear analysis allows to define some properties with a *non-linear behaviour*, but at the cost of more computational effort. The material *strain-stress* curve was then modelled as it follows. For stresses below the yield point, the value of the Young modulus was used, as it happens for a standard linear simulation; for stress values larger than the yielding one, the values are obtained by a linear interpolation of a straight line, where one point is defined as the yield stress at zero plastic strain and the second one as the ultimate tensile stress at the elongation at fracture. Obviously this is a simplified version of the stress-strain curve of the material, but it is sufficient enough to obtain meaningful results.

Constraints

Since the first analysis attempt it was clear that the constraints defined for the mechanical analysis were no more correct for the thermomechanical one. Until now the nodal displacement in the X direction of the nodes at the section interfaces was constrained, in order to simulate the presence of the other cylinders which resisted to the deformation. While this assumption was overestimating the stresses at these interfaces, it was acceptable since the deformations were small. Unfortunately, the deformations induced by the temperature are much larger, leading to meaningless results. As a matter of fact, despite the non-linear analysis, the plastic deformation assumed local values of about 8%, which is an unrealistic value considering that the elongation at fracture is equal to 3.73%. This phenomenon is called *thermal clamping* and it should be avoided as much as possible. This issue rises primarily when the component is not able to freely deform or, as in this case, when it has been *overconstrained*. As already discussed in subsection 3.3.3, the proper modelling of these boundary planes is really difficult because of different factors; for this reason, *the longitudinal displacement of the component was constrained on its yz midplane*, leaving thus these boundary planes free to deform. This assumption leads to an *underestimation of the stresses*, which has to be then taken into account during the post-processing of the results, but it's a compromise which has to be accepted in order to continue with this analysis.

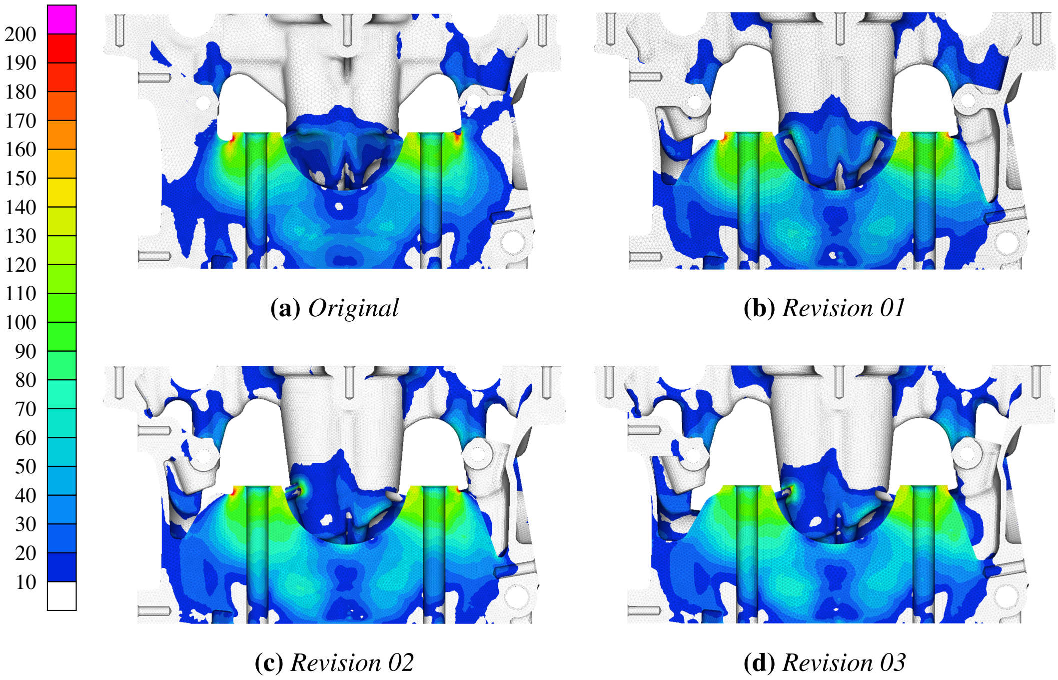
Targets

Until now, the performance of the optimized shape has been represented by its *compliance value*, which is, as already discussed, the product of each nodal displacement times the force acting on it. This parameter represents the *global* stiffness of the component, which can be useful as an objective function during the topological optimization process, but gives no indication of the local performances of the component. For this reason, the compliance index was set aside and the vertical displacement of some crucial points was considered instead. Four points were selected: the *intake and exhaust camshafts centres* on the camshaft supports and the *intake and exhaust bolt interfaces centres*. In order to obtain a precise result, every node of the interface considered was connected by means of rigid bodies[1] to the center node of the interface, designed as *slave node*. In this way, the displacement of each interface node was weighted and reflected to this single node. The values were then retrieved during the post-processing phase.

4.3 Design revisions results

In this section the results of the analysis performed on the different revisions will be shown. The discussion will be split in two parts, the first regarding the so called *cold* analysis, where only mechanical loads will be considered, and the second about the *hot* analysis, where also thermal loads are taken into account. In both subsection the Von Mises stresses will be shown, together with the values of the vertical displacement of the crucial nodes defined before. For all the revisions four views will be shown in order to highlight every critical aspect. The legend that will be used is divided in 21 steps, where all stress values below 10 MPa are indicated in white and the ones above the yielding strength of 200 MPa in magenta. It is worth stressing again the fact that these results are an *underestimation* of the real ones, so

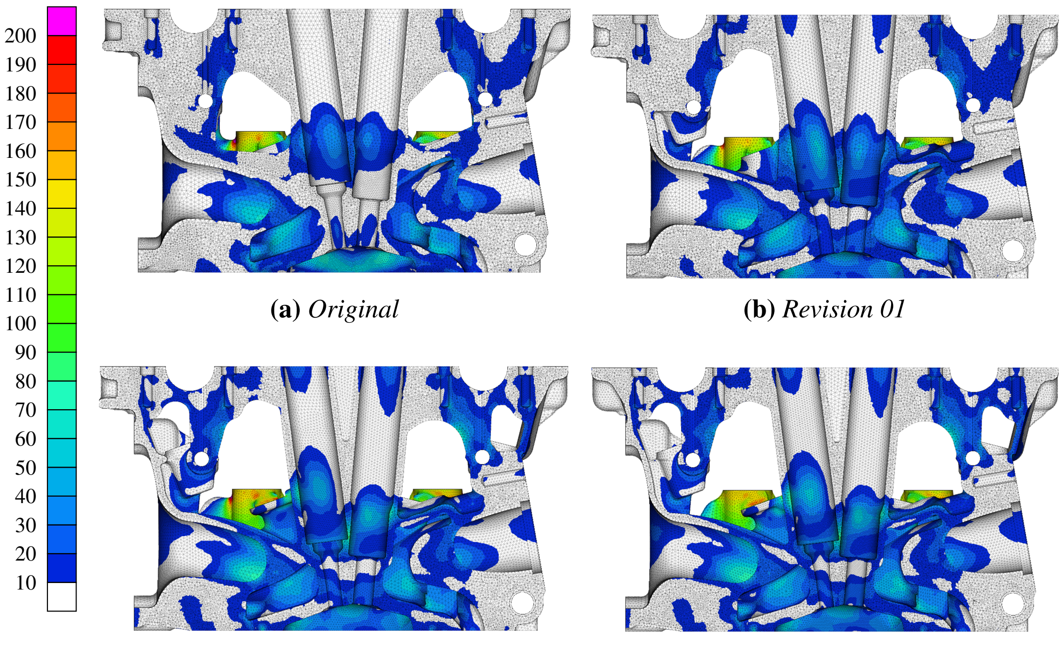
[1] A *spider* or *RBE3* elements using Optistruct terminology



(a) *Original* **(b)** *Revision 01*

(c) *Revision 02* **(d)** *Revision 03*

Figure 4.5: *Cold analysis results, bolt interface plane view*



(a) *Original* **(b)** *Revision 01*

(c) *Revision 02* **(d)** *Revision 03*

Figure 4.6: *Cold analysis results, yz midplane section view*

instead of relying solely on the numerical values it is better to compare them to the original geometry.

4.3.1 Cold analysis

Von Mises stress

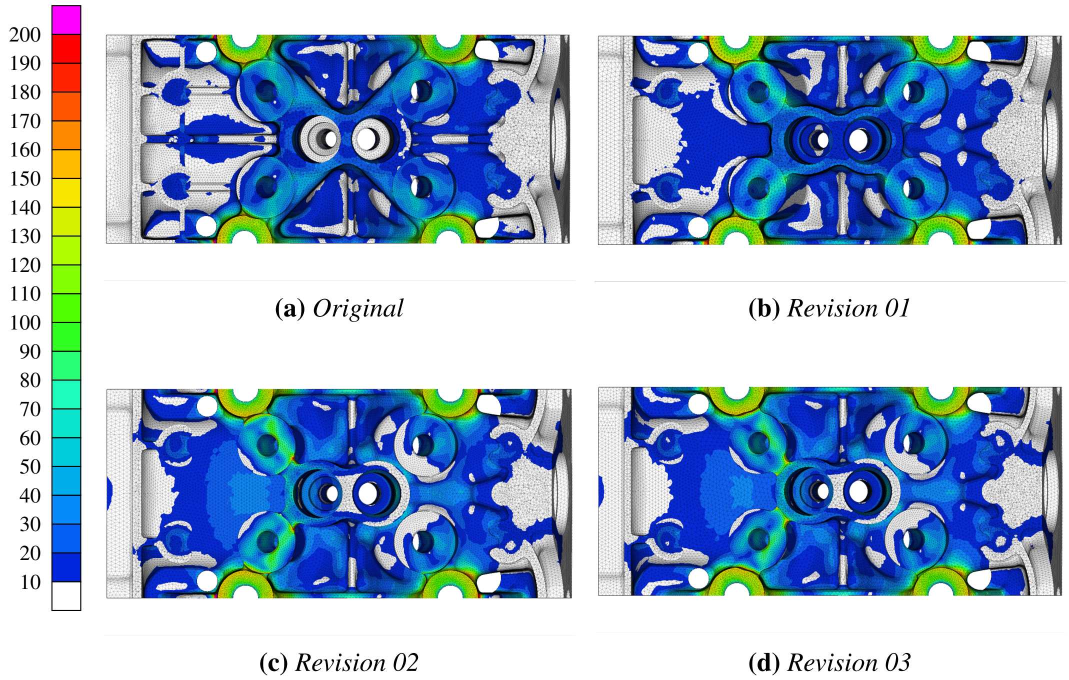
In Figures 4.5, 4.6, 4.7 and 4.8 the Von Mises stress for the different revisions and for different views is shown.

Starting from Figure 4.5, it is possible to see the influence of the geometry changes around on the bolt interfaces. On the original component the small fillet radius of the rib connecting the side wall to the bolt interface are cause of a large stress concentration, which locally exceeds the yielding value. While in Figures 4.5b and 4.5c the removal of the rib did not lead to any substantial change, the implementation of smoother transitions and larger radii led to a large stress reduction to values around 100 MPa. However, the stress slightly increases in the area located between the two bolt channels.

In Figure 4.6 it is possible to remark again the decrease in the stress on the aforementioned rib, but the stresses on the interior side of the bolt interface are substantially larger, especially in figures 4.6c and 4.6d. The reason behind this growth is the creation of the pockets around the spring interfaces, which reduce the amount of material resisting to the bolt and spring loads, leading thus to larger deformations. In addition, it can be noted a rise in the stresses in the region between the combustion chamber and the oil deck, mainly due to the removal of material on this lower part, but the values are anyway far from being alarming.

Moving to Figure 4.7, it is possible to see that the removal of the transversal ribs connecting the side walls to the central channels decrease locally the stiffness, thus allowing larger deformations, especially in the intake ducts area. Moreover a stress reduction on the exhaust spring interface can be noticed, due to the ”lip design” implemented in the 2nd and 3rd revisions.

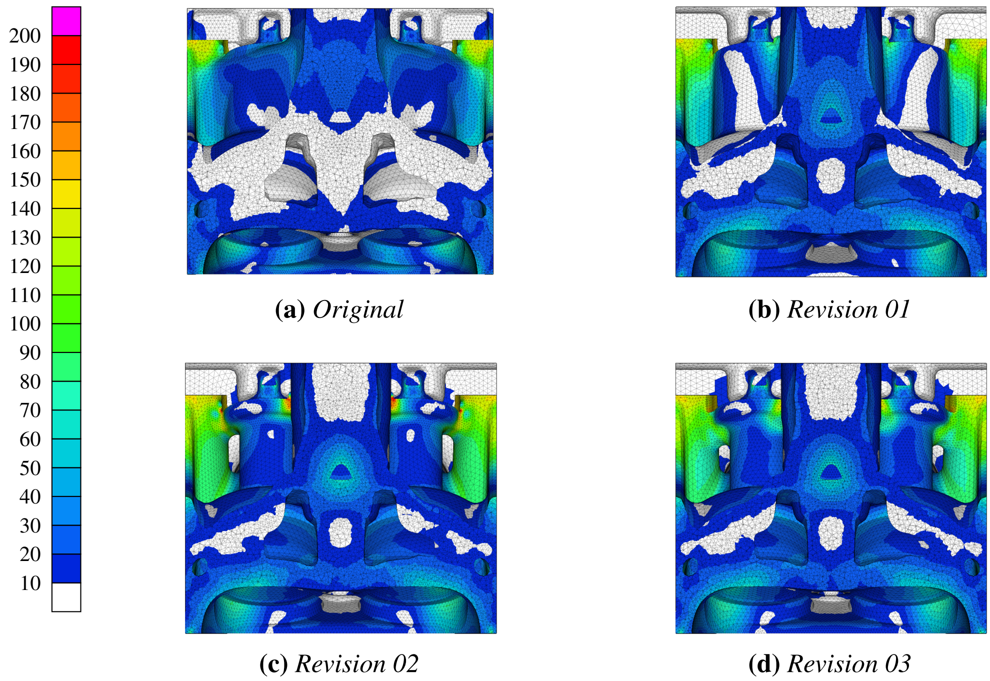
A more detailed view of the spring interface and of the bolt one of the intake side can be seen in Figure 4.8. The thicker bridge connection of the 3rd revision, visible in Figure 4.8d, reduces the stresses on the bolt channel while increasing the ones on the valve guide one. The spots with major stress concentration in Figure 4.8c are no more present in the following revision because the machining operations were changed in order to have less thin edges and geometry discontinuities. This also led to a locally higher quality mesh, which is a fundamental requirement in order to have good and accurate results.



(a) *Original* **(b)** *Revision 01*

(c) *Revision 02* **(d)** *Revision 03*

Figure 4.7: *Cold analysis results, xy plane section view*



(a) *Original* **(b)** *Revision 01*

(c) *Revision 02* **(d)** *Revision 03*

Figure 4.8: *Cold analysis results, xz midplane section view*

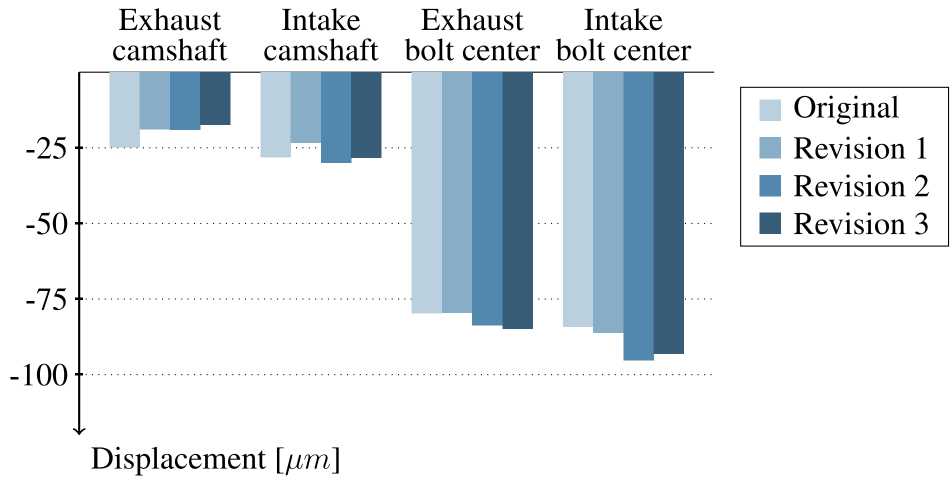


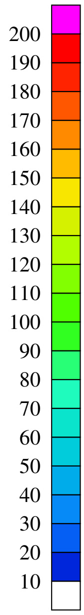
Figure 4.9: *Cold analysis displacement results*

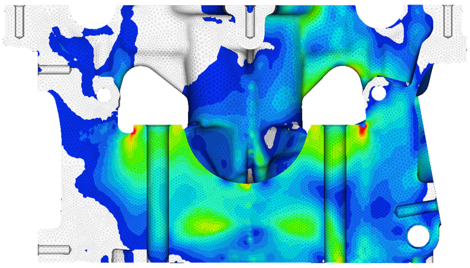
	Exhaust camshaft [μm]	*Intake camshaft* [μm]	*Exhaust bolt* [μm]	*Intake bolt* [μm]
Original	-24.84	-28.13	-79.82	-84.25
REV 01	-23.8%	-16.8%	-0.25%	+2.39%
REV 02	-23.4%	+6.86%	+4.82%	+13.2%
REV 03	-29.7%	+0.60%	+6.39%	+10.6%

Table 4.4: *Cold analysis displacement values compared to the original component*

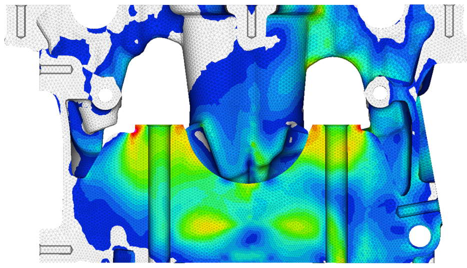
Nodal displacement

For what concerns the nodal displacement, the results are listed in Figure 4.4 and shown as a bar plot in Figure 4.9. As it is possible to see, the displacement of the bolt centres is much larger than the camshaft ones because of the larger forces acting on it. The best overall performances are obtained by the first revision, which has a comparable nodal displacement for the bolt centres and a consistent reduction for the camshaft ones. The worst ones are obtained by the 2nd revision, which has generally larger displacements compared to the original component. Another factor worth noticing is the displacement of the intake bolt center between the 2nd and 3rd revision. Despite the larger dimensions of the bridge connecting the spring interface to the bolt one of the last component, the displacement is however increased. It is possible to conclude that the pockets around the spring interface have a minor role

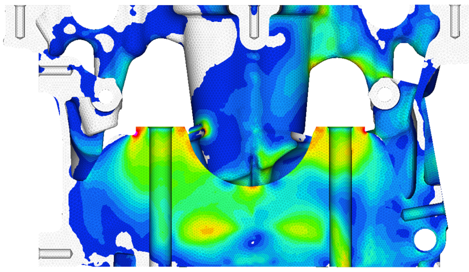




(a) *Original*



(b) *Revision 01*



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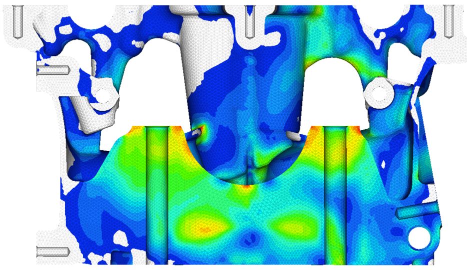
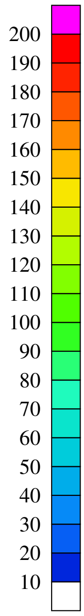
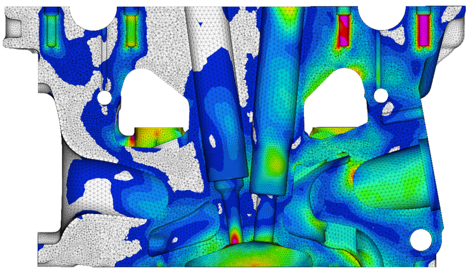


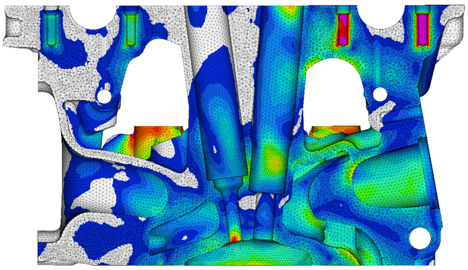
(d) *Revision 03*

Figure 4.10: *Hot analysis results, bolt interface plane view*





(a) *Original*



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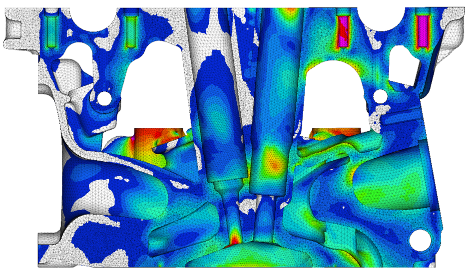

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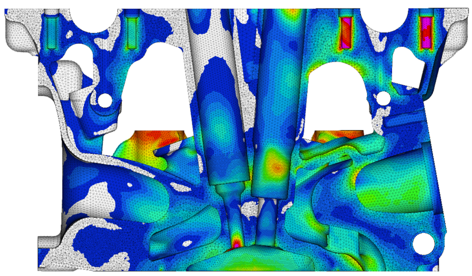


(d) *Revision 03*

Figure 4.11: *Hot analysis results, yz midplane section view*

in the bolt interface deformation, while the biggest contribution is given by the rib connecting it to the external wall. In addition, the removal of material in the lower oil core for the second revision led to larger deformation of the exhaust bolt interface.

For what concerns the camshaft bridge, the modifications caused contrasting results. For the exhaust side, the widening of the window with the removal of the stiffener led to a decrease of the deformation, while the creation of an additional window between the bearing lubrication duct and the external wall did not lead to any major difference. For the intake side, the 1st revision did bring lower deformation, while the second one, where material was removed between the external wall and the oil channel, led to a larger displacement. However with the last revision, where additional material was removed form the intake side, the deformation *reduced*, as opposed to what the previous revision showed. One explanation might rely on the lower thickness of the wall, which reduces the stiffness locally, reducing thus the effect of the bolt force on the camshaft center.

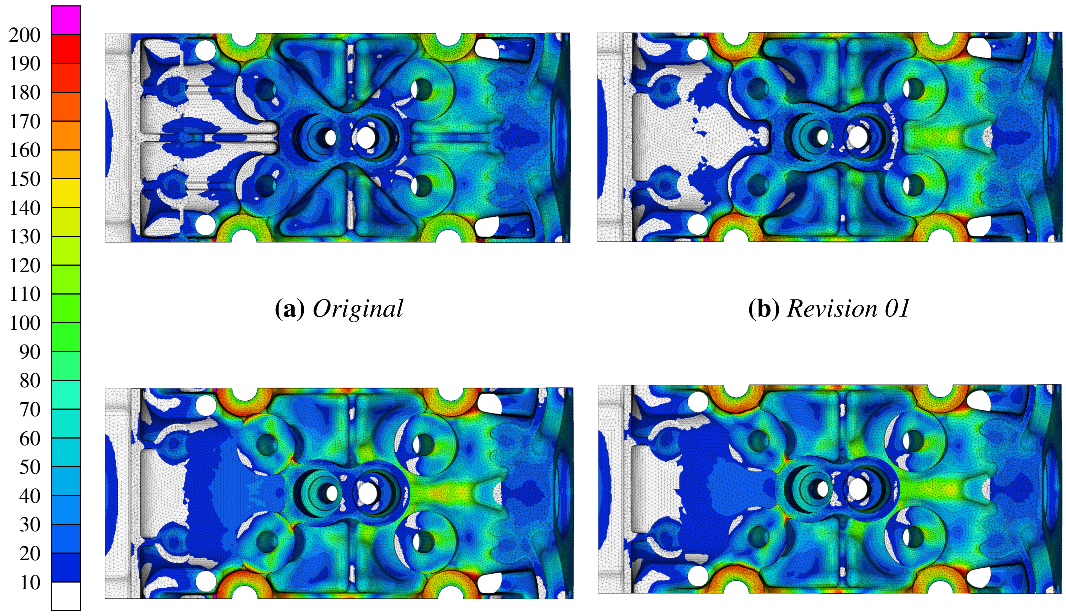
It is possible to conclude that the geometry modifications done have generally reduced the performance of the component, but only of a minimal fraction. Even though the percentages can be sometimes high (Figure 4.4), the *differences are however limited to cents of millimetres*. In any case, this downside is minimized by the weight reduction achieved, which is the core task of this thesis activity. In addition, the ”cold” condition is less relevant if compared to the ”hot” one, which is more critical for the stress point of view.

4.3.2 Hot analysis

Von Mises stress

The results of the thermomechanical analysis are shown in figures 4.10, 4.11, 4.12 and 4.13. As done in the previous section, different views and different section planes are displayed, to compare the difference of the design revisions versus the original component.

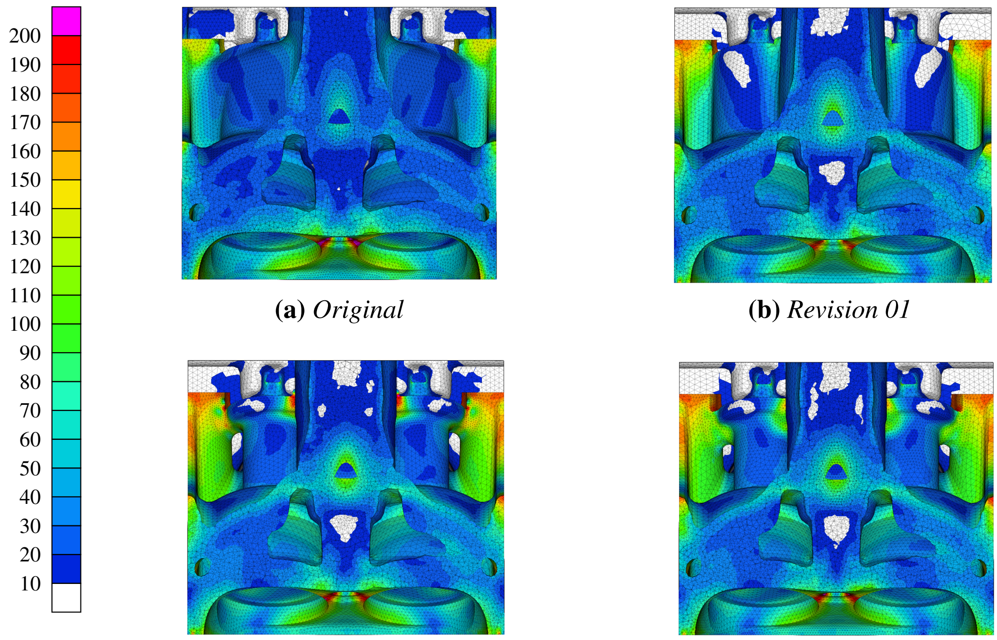
As it is possible to see from Figure 4.10, the stress pattern is very similar to the one of the cold simulation, displayed in Figure 4.5. The absence of the transversal rib decreases locally the stresses, especially in figure 4.10d, as already discussed before. The magnitude of the stresses is generally larger because of the thermal deformation induced by the temperature; however the area close to the exhaust ducts is not largely stressed, meaning that the water jacket is cooling down the part effectively. The major difference with respect to the cold analysis relies in the camshaft bridge. The part connected to the exhaust side external walls is definitely more stressed with respect to the intake one because of the high temperature of the exhaust gases. However the stresses due to the smaller radius between the bridge stiffener and the central channel are safely below the yielding point of the material. In any case, the design changes



(a) *Original* **(b)** *Revision 01*

(c) *Revision 02* **(d)** *Revision 03*

Figure 4.12: *Hot analysis results, xy plane section view*



(a) *Original* **(b)** *Revision 01*

(c) *Revision 02* **(d)** *Revision 03*

Figure 4.13: *Hot analysis results, xz midplane section view*

introduced in the revisions do not change the local stresses, which are comparable to the original component, displayed in figure 4.10a.

Moving to the yz mid-plane section displayed in Figure 4.11, some interesting comments can be made. The most relevant topic to be discussed is the change in the stress field of the bolt interfaces, both at the intake and the exhaust sides. While in the original component the stresses are under control (figure 4.11a), the situation gets already worse with the first design changes implemented in Revision 01 (figure 4.11b), where the stresses rise to values very close to the yielding strength. The situation worsens with the 2^{nd} revision, where the high stressed area increased and in some points the stress exceeds the 200 MPa threshold. The reason behind this is once more the pockets obtained around the valve seats, which decrease the amount of material able to resist to the loads. In the 3^{rd} revision the situation improves, with no plastic deformation spots and lower stresses overall.
In some spots the stress increases, like in the spark plug channel and on the lower part of the injector one, but in other crucial ones it decreases, like in the junction between the two exhaust ports and in the combustion chamber. Special mention must be done about the threaded holes for the bolts of the main caps of the exhaust camshaft bearing. The stresses are just above the yielding point of the material (maximum stress is around 210 MPa) mainly because of the *threaded connection modelling*; rigid elements were used to connect every node of the threaded part to the center node, which in turn increased locally the stiffness without allowing the hole to deform properly because of the higher temperature. The intake ones in fact are not affected by this phenomenon, displaying thus a stress much lower than the exhaust ones.

For what concerns the xy section view, displayed in Figure 4.12, is it possible to see that the stiffeners which have been removed over the ducts lead to a small stress increase, especially on the exhaust side. However the same behaviour was already noticed during the cold analysis review in Figure 4.7, so the thermal dilatation in this area has played a minor role. In addition, an increase of the stress in the fillet connecting the bolt interface with the intake ports can be noticed since the 1^{st} revision.

Finally, the xz section view is shown in Figure 4.13. The stress in the valve bridge between the intake and exhaust ones decreases, as also does the stress between the final part of the spark plug and the injector channels, but increases in the thinnest part between the two channels, which can be seen also in figure 4.11d and in the highest point of the water jacket. These growth however are not troubling, since the stresses are still well below the yielding threshold.
A better view of the effects of the thermal expansion on the spring interfaces can be seen. The increase in thickness of the bridge connection with the central channel drastically decreases the Von Mises stress in that area, which can be seen from figure 4.13c and 4.13d.

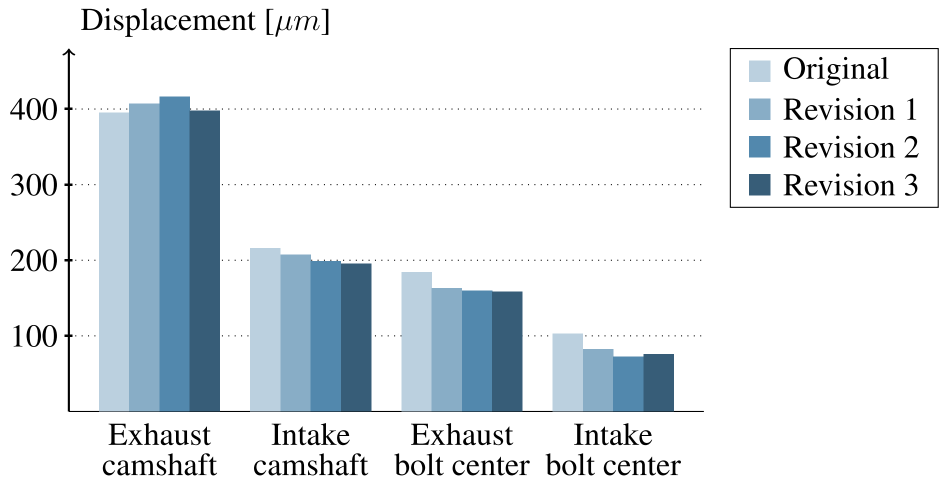


Figure 4.14: *Hot analysis displacement results*

	Exhaust camshaft [μm]	*Intake camshaft* [μm]	*Exhaust bolt* [μm]	*Intake bolt* [μm]
Original	395.0	215.8	184.2	102.6
REV 01	+3.01%	-3.93%	-11.4%	-19.6%
REV 02	+5.37%	-8.02%	-13.1%	-29.7%
REV 03	+0.68%	-9.36%	-13.9%	-26.4%

Table 4.5: *Hot analysis displacement values compared to the original component*

Nodal displacement

In Figure 4.14 the results of the vertical displacement of the important nodes are represented, which are then listed in Figure 4.5. With respect to the cold analysis, shown in Figure 4.9, the behaviour of the component has dramatically changed, from compressive values to expansion values only. In addition, the values of the camshafts centres are now larger or comparable to the ones of the bolts, while for the cold analysis they were definitely smaller.
Looking at the bar plot it can be said that, excluded the exhaust camshaft results, the more consistent the mass reduction is, the lower the deformations are. The reason behind this statement can be that for same given temperature conditions, lower mass means less material that can deform under the effect of the temperature.

For what concerns the intake part of the cylinder head, the deformations decrease

revision after revision; however the bolt interface seems to be influenced mainly by the first three steps of the design changes, while the creation of the pockets under the spring interface seem to not affect its thermal deformation; only a correlation between the bridge dimension and the spring interface can be made (3% of deformation increment between the 2^{nd} and the 3^{rd} revisions). A different behaviour can be described for the exhaust part of the component. For what concerns the camshaft center, every design change up to the Revision 2 made the deformation worse, but the reduced external wall thickness in the last revision has managed to reduce it to a value almost equal to the one of the original component. The bolt center however was not affected by this last parameter, showing the largest displacement decrease with the 1^{st} revision.
It is possible to conclude then that all the design modifications resulted in lower deformations under mechanical and thermal loading, which is the most critical and relevant condition.

Chapter 5

Guidelines for future cylinder head design

In this final chapter, the results obtained from the benchmark activity and from the analysis of the design revisions are collected and processed. As a result, general guidelines for future cylinder head design will be given, fulfilling the task of this thesis work.

Benchmark activity

The benchmark activity performed in chapter 2 has allowed to obtain some very useful information regarding the global structure of the cylinder head. In Table 5.1, the main characteristic of the most representative competitors are listed together with the reference component dimensions and its design revisions. From this activity, two important results can be derived. First, that the major parameters affecting the final weight are *the overall dimensions and the internal layout* and second that *the wall thickness has a limited role*. While the reasons behind the first result are quite trivial, the ones of the second need some discussions. It is indisputable that lower wall thickness leads to a lower weight, but taking a look in Table 2.27 and Table 5.1, it is possible to see that *Competitor B*, despite having a very low wall thickness (around 3.5 mm), its weight is much larger than *Component D* or *Component H* because of its larger dimensions. An additional comparison can be made with the *REFERENCE* component, that despite having larger dimensions and larger wall thickness, its weight is more than 500 g lighter. It can be concluded then that overall dimensions play a major role and efforts should be made in order to obtain a compact packaging. Additionally, the wall thickness is related to numerous manufacturing constraints, which limit its application only to certain features, and cost. At the contrary, a better component design can be obtained without any additional cost in the early stages of the engine development.

Name	*Dimensions [mm]*							*M [g]*	*Reduction*
	D	h	l	w	d_{cs}	d_f	d_b		
Competitor B	102	146.7	242	118	135	184.6	100	3228	
Competitor D	96	155	205	108	112.1	176.5	104	2828	
Competitor I†	84	138	210	93	-	160.1	87	2262	
Competitor J	95	145	173.7	105	101.1	129.1	96	2106	
Competitor L	84	139.5	191.6	90	109	123	83	2177	
REFERENCE	88	140	248.3	98	155.6	205.4	96	2685	-
REVISION 01	*(some as REFERENCE)*							2492	-7.2%
REVISION 02	*(some as REFERENCE)*							2360	-12.1%
REVISION 03	*(some as REFERENCE)*							2290	-14.5%

† Camshaft integrated in the engine cover

Table 5.1: *Benchmark table updated with the geometry revisions*

In order to obtain a very compact packaging, the following guidelines can be considered:

- One fundamental aspect is to have a *short distance between the intake and exhaust manifold flanges*. For this reason, the external side walls should be placed, if possible, in a vertical position, like in Figure 2.18, or with a hourglass shape, as in Figure 2.39. This last shape has numerous advantages, because can be used when a large gasket interface is needed, but keeping the overall dimensions limited.

- *Cylinder head bolts should be placed as close as possible to the combustion chamber*. This helps in increasing the stiffness in the gasket area, which is critical, and allows to shorten the transversal section of the component.

- *Oil channels and HLAs can be positioned along the central channels.* This allows to further reduce the distance between the walls, at cost of no major issues. In fact, almost all the cylinder heads are designed with this layout, with the REFERENCE one being one exception. This is probably the weakest point of the component analysed. In case it is not possible to use this layout, *oil channels can be machined directly on the external walls*, leading to a substantial weight reduction.

- *Camshaft distance should be low*, which can achieved by positioning them over the cylinder head bolts. A cam-carrier solution can be used, as it happens for Competitor I, for a high volume production engine of if easy operations are required in the assembly phase.

- *Oil drain and crankcase ventilation channels* can be coupled together with the bolt channel, as can be seen for Competitor F in Figure 2.21. This strategy allows to position the external walls even closer, but attention must be paid when designing and simulating the bolt interface.

- *Gasket interface shape* can be designed in order to better follow the core elements of the cylinder head. Instead of having a standard rectangular shape, the gasket profile can be obtained by offsetting the combustion chamber and the water jacket ducts profiles.

Design revisions analysis

Differently from the previous activity, the design revisions simulation phase has allowed to find *local spots* where weight reduction strategies can be implemented. These unfortunately are more related to the REFERENCE cylinder head, but however useful inputs can be used for new cylinder head design.

- *The camshaft bridge* revealed to be the area with largest weight reduction potential. Large windows on the yz plane section can be obtained, with the only major constrain of the camshaft main caps bolts position. *An asymmetric design* between intake side and exhaust one can be implemented, since the intake one is subjected to lower thermal loads and to lower valvetrain forces. *Pockets* can be obtained between the camshaft bearing and the external wall, without negatively affecting local stresses or deformations.

- *Pockets below the intake spring* are a viable solution to pursue weight reduction targets, as seen for Revision 02 and Revision 03. However the thickness of the bridge connection must be carefully evaluated, because the rise in local stress was not negligible and could lead to a component failure, especially under mechanical and thermomechanical fatigue cycles. Also the lip profile for the exhaust spring interface can be implemented, but the weight reduction achieved is less significant.

- An additional pocket can be obtained *between the spark plug and the injector channels*, which is not a critical area. Limits for this strategy are presented by the engine cover sealing, so a synergy when designing this component has to be applied. Additional weight can be reduced by increasing the machining diameter of the channels, but attention must be paid on their lower ends, which require large stiffness since they are close to the combustion chamber.

- *Hydraulic lash adjusters can be supported by one rib only* if they are placed close to the side walls, as it is the case for the REFERENCE component. Additional ribs were not useful for a structural point of view, but they might be necessary for the NVH behaviour of the component, especially for Revision 03 were the intake duct surface is clean from ribs and can create noise and resonance problems.

- *Pockets* on the gasket interface, between the intake flange and the water jacket, can be obtained without compromising structural integrity of the component. Unfortunately the same approach cannot be used for the exhaust side, since the complexity of the secondary air duct system makes the material removal very difficult.

Open questions

Since the thesis work has been developed considering one single cylinder section, some open points for future investigations are now given.

- *Gasket performance*, which was not considered in this thesis work, should be assessed. Its behaviour is important not only for the proper engine functioning but also for the cylinder head stress distribution. The design modifications have undoubtedly reduced the stiffness of the lower part of the component, so its behaviour must be again analysed.

- *Analysis on a full engine model* with the last design revision must be performed, in order to obtain more precise results, without worrying about the boundary conditions definition. In addition, the full engine model introduces some other aspects which in thesis work were neglected, like the smaller deformation of the cylinder block, which causes the cylinder head to slightly bend and assume a curved shape in the x and y directions.

- *Structural optimization of the chain and transmission sides*, which were not considered in this thesis work, can be analysed to further reduce the weight of the component.

Chapter 6

Conclusions

In this thesis work a study on the structural optimization of the cylinder head with the focus of weight reduction has been developed. The aim was to define guidelines for the lightweight design of this component.
In order to have a better knowledge about the study faced in this work, the first part was dedicated to a literature review of the component. An introduction on the legislative framework has been done in order to motivate the need for lighter automotive components. Subsequently the structure of the cylinder head has been explained, with the description of its principal components and their functions, like the ports, the ducts, the combustion chamber and the water jacket. Some knowledge about the history of the component has been given, in order to highlight how its structure has changed with the introduction of new technologies and with the increase in the processes quality. The description of the roller finger follower and of the hydraulic tappet distribution systems followed, since their knowledge was required then later in the thesis work, together with technical choices like the integrated exhaust manifold and the cam carrier solution. The mechanical and thermal loads to which the cylinder head is subjected to were then described, together with a description of the different simulation analysis that have to be performed. After this, a section related to the manufacturing aspect of this component has been addressed, with a description first of the materials used, namely steel and aluminium alloys, the thermal treatments to which they must be subjected, and then with a list of the casting process and their characteristics. The chapter was ended by a an explanation of the general workflow for a cylinder head design.

After this introduction, a benchmark activity was developed. After defining the characteristic dimensions and thickness values of a generic cylinder head, the REFERENCE component was analysed together with other 12 competitor products, in order to evaluate its positioning on the market. The results, which are displayed in Table 2.27, showed that despite its large dimensions, the cylinder head under analysis was one among the lightest ones; however the gap with the best in class was really consistent, mainly due for layout reasons. From this activity, some informations about the packaging and the structures were obtained, which could be then useful for the optimization phase and, in particular, for the future guidelines.

The following chapter was dedicated to the structural optimization of the REFERENCE cylinder head. After some technical background about the meaning of optimization and about the characteristics of the topological one, which was used in this work, the modelling of the cylinder head was described. The softwares used were *Hypermesh* for the pre-processing part, *Optistruct* as the solver and *Hyperview* for the post-processing. The load modelling was then illustrated, together with the issues faced while setting the proper constraints together with the modelling the gasket behaviour. In this phase, some assumptions were introduced to simplify the work. Finally an optimal curve, shown in Figure 3.6 was obtained in order to define which compromise between weight reduction and stiffness should have been used as a starting point. After this compromise was selected, the optimized structure was obtained.

Starting from this optimal shape, the following step consisted in implementing some design changes to the original component. Three design revisions were created, with three different degrees of complexity, with the final one bringing a 14.7% in weight reduction. Those revisions were then analysed in a ”cold” and in a ”hot” condition, the latter including the thermal loads and a larger bolt preload caused by the component thermal expansion. In order to do so, the constraint conditions had to be changed, because the assumptions made before during the optimization phase were no more acceptable, leading to inconclusive and meaningless results. The design changes brought a slightly loss in performance for the ”cold” condition, but the benefits for the ”hot” one were relevant, with reductions up to 26% in nodal displacement for the intake bolts center. These results showed that a weight reduction without a worsening of the performances can be achieved; however the results should be confirmed with a full engine analysis, in which more realistic boundary conditions can be implemented and the gasket performance can be evaluated too.

Finally, as a conclusion of this thesis work, the guidelines for the future cylinder head designs were given, taking as a results the outputs of the benchmark activity and of the optimization process.

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Appendix A

SolidThinking Inspire software assessment

A.1 Introduction and assessment reasons

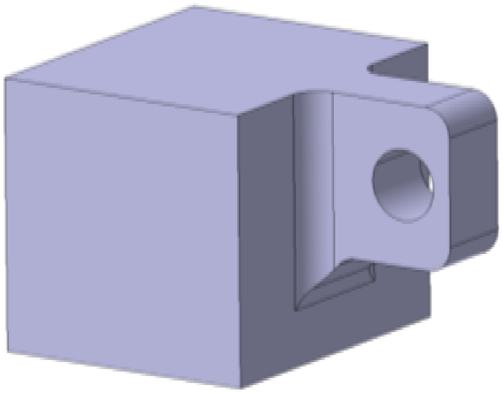
A short time to market is fundamental for a car manufacturer when launching a new vehicle, which means lower development costs and more possibility of success in the market. In order to reduce it, extensive use of CAD and CAE[1] tools is done, which allow to create, simulate and modify the products in the early phases of the design, dramatically reducing the costs in terms of prototyping and, even more, product specifications change during pre-production phase.

The general workflow during the design process consists in an iterative cycle, where designers create a 3D model of the part, simulation engineers analyse it and feed back to the designers some inputs and changes. This loop continues until a satisfactory performance from the component is obtained. In order to shorten this phase and to make it more efficient, Porsche Engineering is assessing the introduction of a new software, *solidThinking Inspire*, in the engine design department.

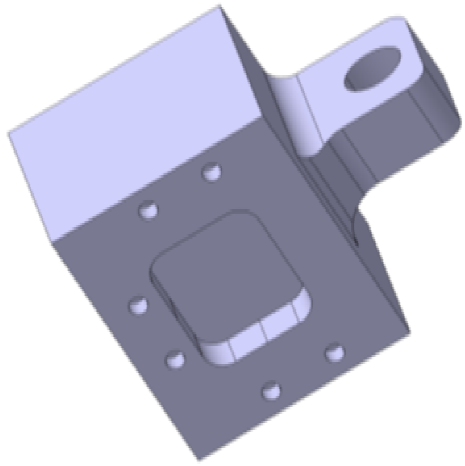
Inspire is a new software which allows to perform rapid simulations and topological optimization analysis. It is addressed to designers, in order to help them in creating optimized structures and to perform very fast analysis, in order to assess preventively the component performance without waiting the results from the simulation engineers. Ideally, this should shift some workload from the simulation department to the design one, with the simulation engineers in charge only of the complex simulations and of the validation of the geometries.

The assessment of this software will be done through a comparison with the softwares used for this thesis work, *Hypermesh* and *Optistruct*.

[1]Computer Aided Engineering



(a)



(b)

Figure A.1: *Starting bracket geometry*

A.2 Assessment process

In order to evaluate the software performances, both in terms of ease of use and final results, a structural optimization of a bracket, visible in Figure A.1 was run on both softwares. It has six holes with a 8 mm diameter and a 10 mm depth to support the component and a single 26 mm. The material considered is the steel *AISI 304*, with the following characteristics:

Young modulus	Poisson modulus	Density	Yield strength
$195\,\mathrm{GPa}$	0.29	$8 \times 10^{-6}\,\mathrm{kg/m^3}$	$215\,\mathrm{MPa}$

The bracket was then loaded with two forces on the center of the large hole, a vertical one of $10\,000\ \mathrm{N}$ and a lateral one of $5\,000\ \mathrm{N}$. Two optimization runs were then defined, with two different targets: the first one to *optimize the stiffness* for a given weight reduction and the second one to *minimize the mass* with a maximum stress constraint. In order to have higher quality results, for both runs some other constraints were defined: minimum member size of 15 mm and a symmetry plane perpendicular to the large hole axis, crossing it in its center. The component was constrained at the center of the smaller holes, while non design spaces were defined around the constraint and load regions.

For what concerns the model set-up, *Inspire revealed to be exceptionally faster* with respect to Hypermesh, with a modelling time needed of 10 minutes only. The user interface is really user friendly and intuitive, guiding the engineer all along the process, which allows also people with limited knowledge of optimization softwares to perform the analysis in a short time. A confirmation that this software is more "designer oriented" can be seen when simulation has to be started. In fact, the *component meshing is done automatically by the software*, leaving only few parameters to be tuned by the user, like the average mesh size and the mesh quality. In addition, there are numerous designing tools, which allow to modify the geometry on the fly, without re-importing the geometry in the CAD software.

The optimization runs were then launched with the following parameters:

- **First analysis**: maximize stiffness with a maximum final weight 35% of the initial one
- **Second analysis**: minimize mass with a minimum safety factor of 1.3

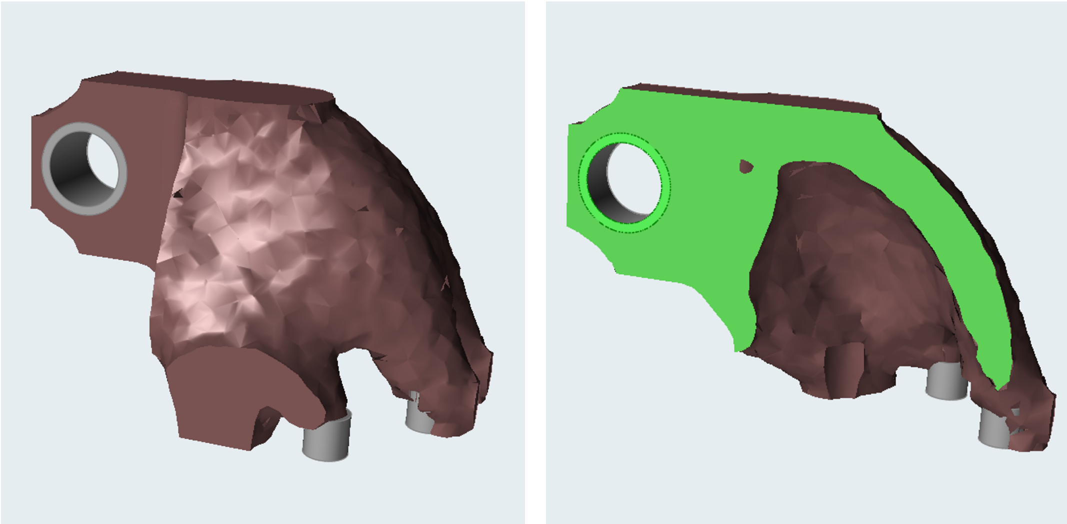
Despite the literature advices to not run optimizations with stress values as a design constraints, Inspire at the contrary forces the user to set this parameter, otherwise the optimization analysis cannot be started.

A.3 Optimization results

After some researches, it turned out that Inspire uses Optistruct too as the solving software, so *the final results will depend on the boundary conditions* (mesh size, mesh quality, constraint modelling, ...). For this reason, the optimization parameters in Hypermesh were set to be as close as possible to the ones of Inspire. In the following figures the results of the optimization analyses can be seen. As it was predicted, the final shapes are similar, but the post processing in Inspire revealed to be simpler. However, in my opinion, one major flaw was found: Inspire does not allow to set the masking threshold value for the equivalent density parameter; only a simple slider between "more material" and "less material" to be displayed is given. This is the reason why the resulting shape between Figure A.4 and Figure A.5 is different. However, the slider changes in real time the value of the mass of the component, assigning the real material density value to all the elements displayed in that specific moment. This feature, which Hyperview is missing, is really useful and gives an effective idea of the final weight of the component.

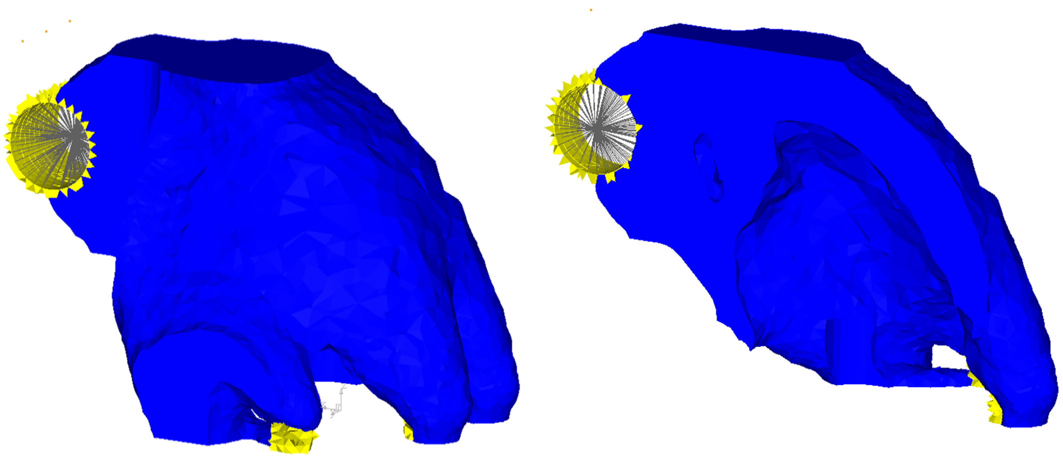
A.4 Conclusion

Before concluding this assessment with some final comments, the positive and negative aspects of this software are listed. Also some other personal experiences with the software for the structural optimization of the cylinder head of this thesis were considered.



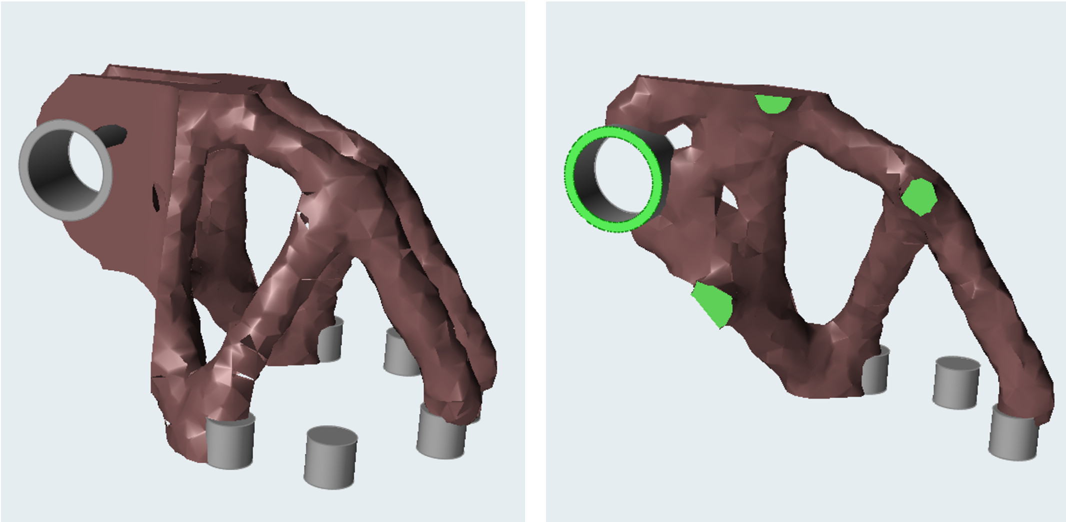
(a) **(b)**

Figure A.2: *Inspire results for stiffness optimization*



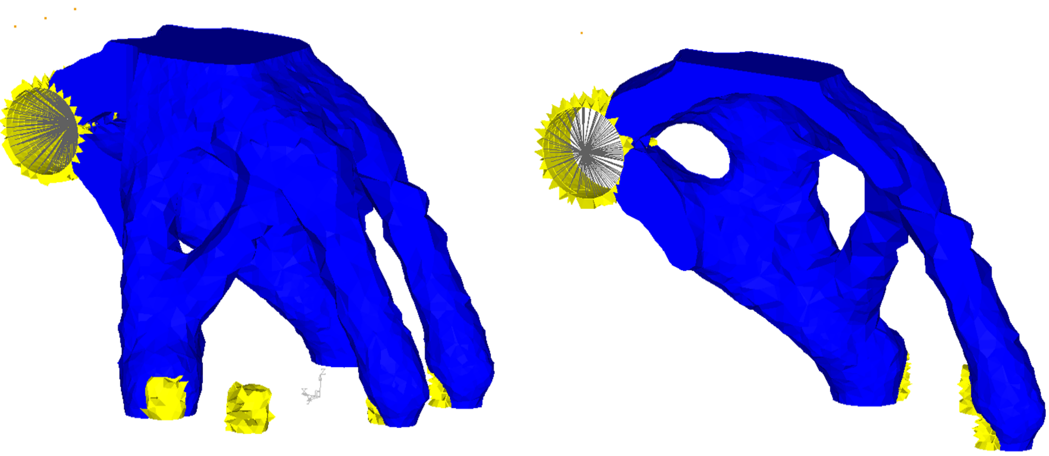
(a) **(b)**

Figure A.3: *Hypermesh results for stiffness optimization*



(a) (b)

Figure A.4: *Inspire results for mass optimization*



(a) (b)

Figure A.5: *Hypermesh results for mass optimization*

- **Positive aspects**:
  - The interface is really user friendly if compared to other CAE softwares like Hypermesh. The commands are easy to use, the boundary conditions are well organized and the overall experience is pretty straightforward. Hypermesh, at the contrary, has a lot of parameters, sub-menus, tools and capabilities which understanding and usage require a long training
  - The model set-up, for the reasons stated in the previous point, is really easy and especially fast. In my personal experience, pre-processing times were *at least* about three times shorter compared to Hypermesh
  - The meshing operation is done entirely by the software itself, requiring less knowledge from the user
  - Definition of design and non-design space is straightforward, as it is for fasteners, connectors and contacts
  - Post-processing operations are really simple and allow a fast comparison between different load cases
  - Modifications of the geometry on the fly allow to implement small modifications without resorting to CAD softwares
- **Negative aspects**:
  - Even though the simplicity of this software is a very good feature, for certain aspect it turned out to be *oversimplified*. This can be explained with a very simple example. When setting up an analysis, a choice between a ”fast” one or ”more accurate” one has to be made, which is quite misleading because it implicitly supposes that the fast analysis provides wrong results. After a long research, it turned out that this ”accurate results” option forces the software to use second order elements. Additionally, there is no control on the mesh quality, on the 3D mesh growth ratio and its maximum size (which is important for a topological optimization analysis).
  - Boundary condition definition is not clearly defined. Every time a part of the geometry is denoted as non-design, the software treats it as a *separate* body, defining then a contact surface with the design space. This could lead to some different results if compared to other softwares
  - Thermal loading cannot be simulated, which can be a limitation, especially for the analysis of engine components
  - Analysis typology is limited to the linear one, leaving no possibility for plastic strain evaluation

- The stability of the program is still poor. Software crashes were quite common, which forces the user to save the model after every single operation. In addition, the import of CAD files from CATIA is really slow and often results in geometry errors. This happened while importing the cylinder head section for example and it is absolutely not acceptable.

Despite those negative aspects, especially regarding the stability issues, I recognise this software to be a powerful tool with a good potential. Designers could obtain optimized shapes and performance responses from their designs in a short time, without the help of the simulation department. Its ease of use would not require even an intensive training, considering also that this product is developed for the designers and not for simulation engineers. For this reasons, I would advice its implementation after those issues are ironed out.