

Engine Downsizing - An Analysis Perspective

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MAHLE Powertrain (MPT) is constantly exploring new ways to improve the efficiency and performance of engines to meet the demanding objectives Automotive OEM's are faced with today, i.e. to reduce fuel consumption and emissions. MPT's key expertise lies in the development of high performance engines with low emissions and excellent fuel economy through the optimisation of gas exchange, combustion, friction and durability.

This strategy is being demonstrated by the development of MAHLE's own state of the art three-cylinder 1.2-litre downsizing technology demonstrator engine which has been designed, built and tested at Northampton in the UK.

One of the objectives of the project was to design a compact engine with high specific power output by using a turbo charger combined with state of the art direct injection technology and variable valve timing. This ensures that vehicle performance targets can be met using the smallest capacity engine thus minimising throttling losses which otherwise leads to high fuel consumption.

With such a high specific power output predictive analysis has played a key role in guiding, validating and optimising the design. This paper highlights the use of Abaqus to perform structural analysis of the main engine: connecting rod, crankshaft and cylinder block bottom end as well as thermo-mechanical analyses of the head and block assembly and exhaust manifolds.

1. Introduction

Since the agreement to reduce average new car CO₂ emissions to 140g/km by 2008, fuel consumption improvement has been one of the main drivers for engine development within the automotive industry. Current status of the EU fleet indicates that significant work is still required to achieve this target, particularly in the field of gasoline spark ignition engines. Furthermore recent proposed legislation in California indicates the US market will follow the EU's lead.

In recent times various technologies have been applied to gasoline engines to improve operating efficiency. The most significant of these being:

- Direct injection (homogenous and stratified lean)

- Variable valvetrains (profile switching, cylinder deactivation, fully variable lift and duration)
- Controlled auto-ignition or homogenous charge compression ignition
- Engine downsizing

Gasoline engine downsizing is the process whereby the speed / load operating point is shifted to a more efficient region through the reduction of engine capacity whilst maintaining the full load performance via pressure charging. It has long been known as one of the most effective technologies for immediate implementation [Atkins 2003]. This has led to a significant increase in the number of downsized engines on the market today [Welter 2006; Groff 2006; Nishida 2005 and Krebs 2004] with more engines to be introduced in the near future.

Further improvements in fuel economy have been shown to be possible through increased levels of engine downsizing [Fraser 2004 and Bandel 2006]. However, as specific output increases so too do the technical challenges; the foremost of these being:

- A robust combustion system that allows a high compression ratio to maintain part load efficiency
- Good low speed torque and transient performance
- Real world fuel consumption benefits through a reduction in full load fuel enrichment
- Base engine robustness and durability

These challenges require unique engine design solutions to enable the significant efficiency improvements of an aggressively downsized engine to be realised.

This paper summarizes the analysis work carried out to ensure the robustness and durability of MAHLE Powertrains new 1.2l 3 cylinder high specific output gasoline downsize engine [Fraser, 2008].

2. MPT Downsizing Engine

The design targets for this engine were initially to produce an engine capable of replacing a typical 2.4l V6 PFI engine used in a C or D class European vehicle platform (circa 1600kg), delivering comparable performance with a fuel consumption reduction of circa 30% on the NEDC driving cycle. The engine would be compliant with EU5 / ULEV2 emissions from conception. The concept approach was based on the use of technology which would be ultimately available for mass production techniques.

As a downsizing demonstrator the engine was designed to deliver the best possible levels of friction & weight but also a high level of robustness to support the future requirements as an R & D platform. The design was independent of any carry over requirements from manufacturing facilities or existing engine architecture, however, it was intended to demonstrate a maximum content of MAHLE group technology. Table 1 shows the basic performance and fuel economy

targets for the I3 twin turbo engine (shown in Figure 1). The I3 engine was also designed with a high level of protection for a V6 version.

Table 1. Performance targets.

Engine Performance			
Torque	153 Nm	At 1000 rpm	(16 bar BMEP)
	286 Nm	2500-3000 rpm	(30 bar BMEP)
Max Power	144 kW	At 6500 rpm	(Specific Power = 120 kW/l)
	(193 bhp)		(160 bhp/l)
Fuel Consumption at Part Load			
Optimum	bsfc = 235 g/kWh		
At 2000 rpm, 4 bar BMEP	bsfc < 295 g/kWh		
Hydrocarbon Emissions			
At 2000 rpm, 2 bar BMEP	< 5.4 g/kWh		
At 2000 rpm, 4 bar BMEP	< 4.4 g/kWh		



Figure 1. Twin turbocharged downsize engine.

3. Cranktrain Analysis

At the heart of the engine is the cranktrain (see Figure 2) which turns the energy from combustion into the rotary motion required to drive the vehicle. With the very high specific output required from the downsize engine, 1D cycle simulation suggested combustion pressures as high as 140 bar could be expected. These pressures are very high for a gasoline engine and structural analysis of the connecting rod and crankshaft was therefore essential in order to achieve durability at these loads whilst keeping mass and friction (bearing sizes) to a minimum.



Figure 2. Cranktrain layout.

3.1 Connecting Rod

Analysis of the connecting rod can be done relatively simply as the load cycle is dominated by the compression loads from cylinder pressure and tensile loads from piston inertia at top dead centre exhaust. As a worst case scenario the connecting rod was analysed for maximum gas pressure at minimum speed, giving minimum inertia relief, and maximum inertia loading at maximum engine speed. Thus including an assembly step only 3 steps were analysed. As the connecting rod is symmetrical just half of the rod assembly was analysed (see Figure 3). The analysis model included the rod, cap, bolts, bearing shells, small end bush and piston pin. Small sliding contact was defined between all of these parts except the bolt thread which was attached to the rod with tied contact. Interference fits were defined for the bearing shells and bush and clearances for the piston pin and crank pin. The crank pin was modelled as an analytical rigid surface.

The stress distribution in the rod can be seen in Figure 3. Deformation of the rod and contact pressures across the cap and rod interface were assessed and a fatigue analysis carried out using FEMFAT to determine fatigue safety factors. Based on the results of the analysis 2 design

iterations were made in order to ensure the rod and cap didn't separate under tensile loading and to achieve acceptable safety factors whilst keeping mass of the rod as low as possible (see Figure 4).

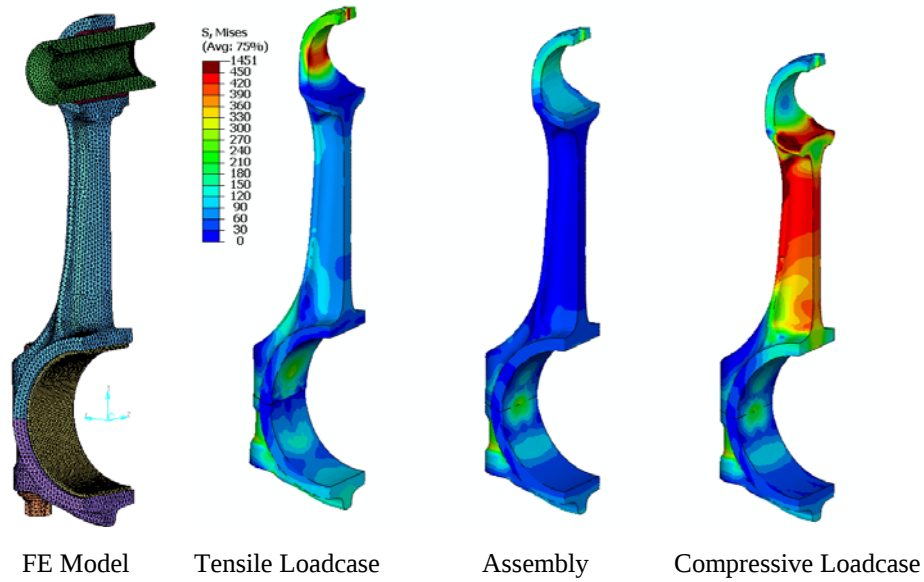


Figure 3. Connecting rod FE Model and stress distribution.

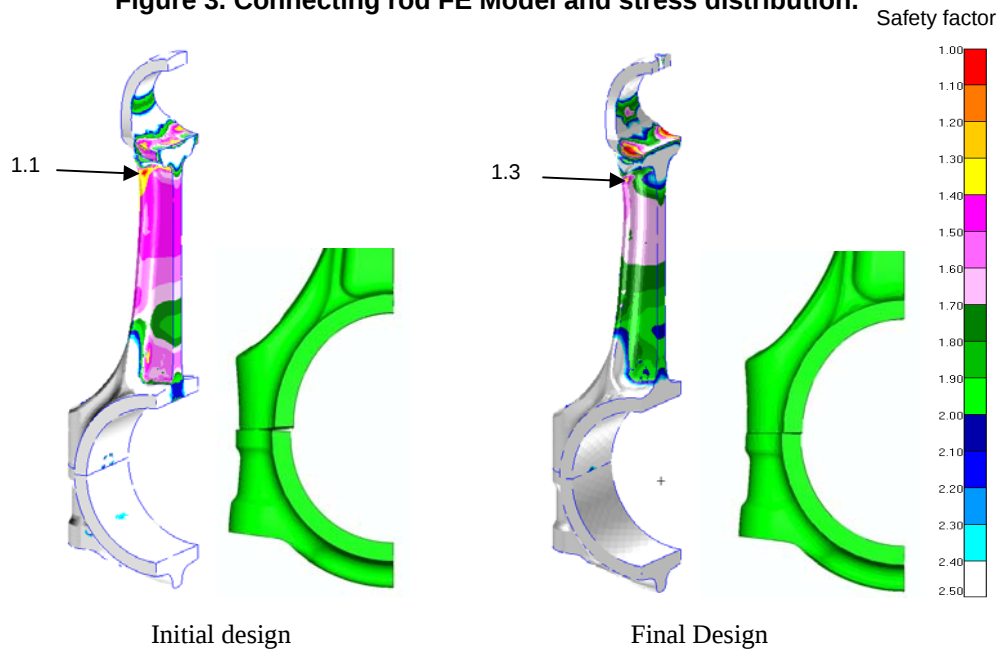


Figure 4. Connecting rod factor of safety.

3.2 Crankshaft

In order to capture the dynamic behaviour of the crankshaft (particularly torsional oscillations) a flexible multi-body dynamic analysis of the crankshaft was carried out. This dynamic simulation was performed using AVL Excite. AVL Excite uses a condensed stiffness (super-element) model of the crankshaft. With Abaqus' partnership with AVL the setting up and running of the condensation analysis was very straight-forward. Once the dynamic analysis had been completed the deformation results, at the retained degrees of freedom, were then used to drive the full Abaqus model of the crankshaft in order to recover the stresses. This was performed for a full 720° cycle. An example of the stress distribution is shown in Figure 6. A FEMFAT analysis was then performed to determine the fatigue safety factors for the crankshaft (Figure 7).

Although a relatively fine mesh was used for the crankshaft it was still only practical to have 3 elements across the crankshaft journal fillets. Thus once the fillet with minimum safety factor had been ascertained a sub-model of this region was generated with the compound fillet radii captured in much more detail (see Figure 8). The sub-model analysis was then run for the 720° cycle and the fatigue analysis repeated.

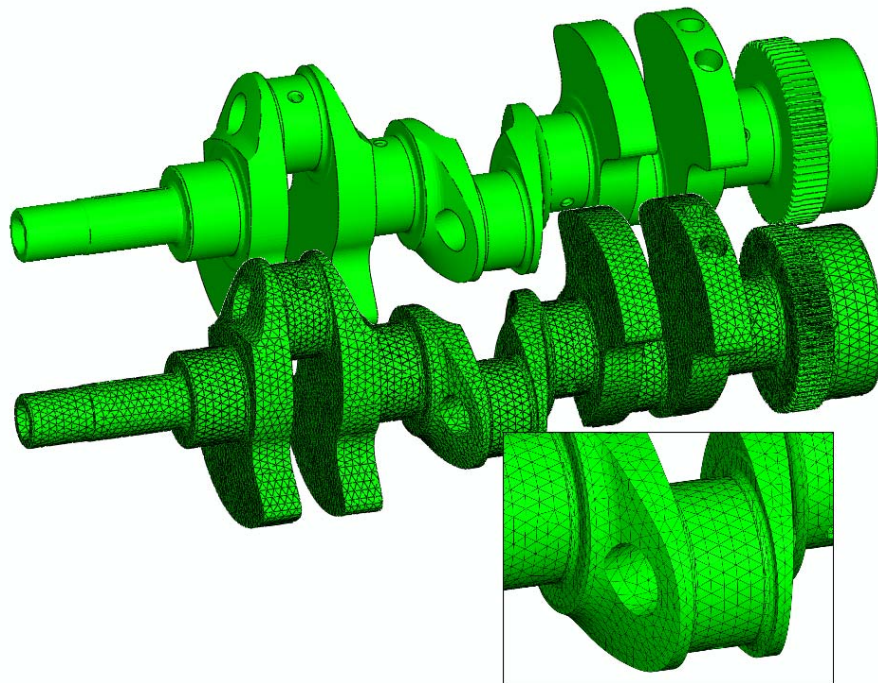


Figure 5. AVL Excite model and FE Model of crankshaft.

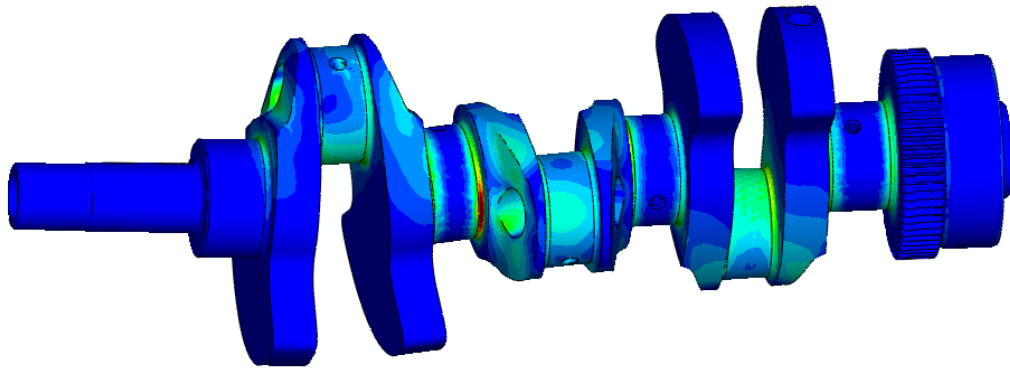


Figure 6. Crankshaft von Mises stress distribution (Cyl1 Firing).

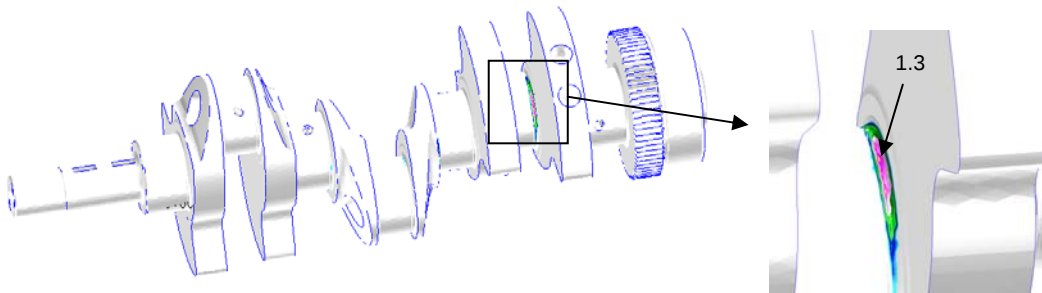


Figure 7. Crankshaft fatigue safety factors.

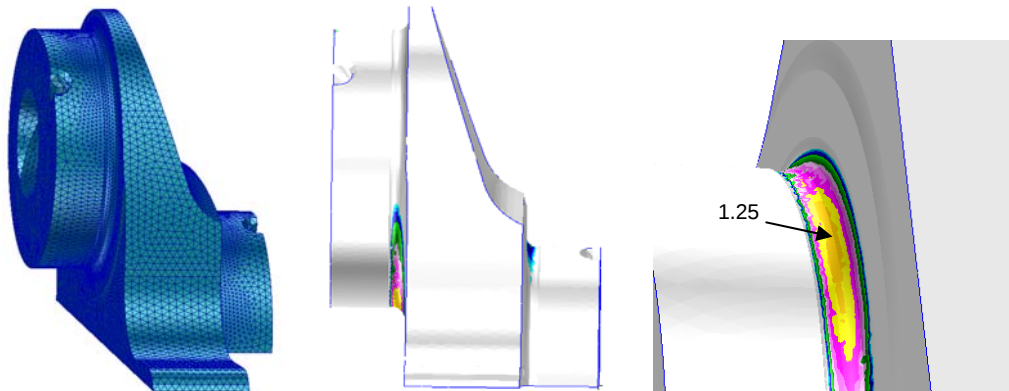


Figure 8. Crankshaft fillet sub-model.

4. Crankcase (Bottom End) Assembly Analysis

During the concept phase of design, a variety of crank case architectures were considered, including:

- Monoblock crankcase and cylinder head
- Cast-in liners (aluminium or iron)
- Bedplate verses deep skirt construction

Consideration was given to the best overall solution for function, weight and cost. The eventual choice of an all aluminium construction with Nikasil coating for the cylinder bore was driven by the combination of excellent heat transfer properties and minimum bore spacing.

Through bolts run from the cylinder head through the crankcase and bedplate assembly into a separate nut plate to allow all of the aluminium components to be held in compression. This allows the use of large bay to bay breather passages to reduce pumping losses and also results in excellent bore distortion characteristics. The nut plate provides good load distribution for the main bearings giving low main bearing distortion. A cross section through the crankcase assembly is shown in Figure 9.

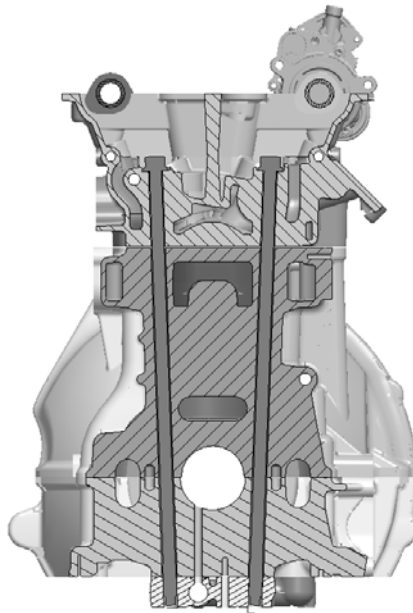


Figure 9. Cross section through crankcase and cylinder head.

In order to assess the durability of the crankcase and bed-plate assembly structural and fatigue analyses were carried out. A finite element model of the complete engine block bottom end

assembly was created which included the block, bed-plate, nut-plate, bolts and bearing shells (see Figure 10). Full sliding contact, with friction, was defined between all mating parts. Bolt loads were applied using pre-tension sections and bearing shell interference fits applied using *CONTACT INTERFERENCE.

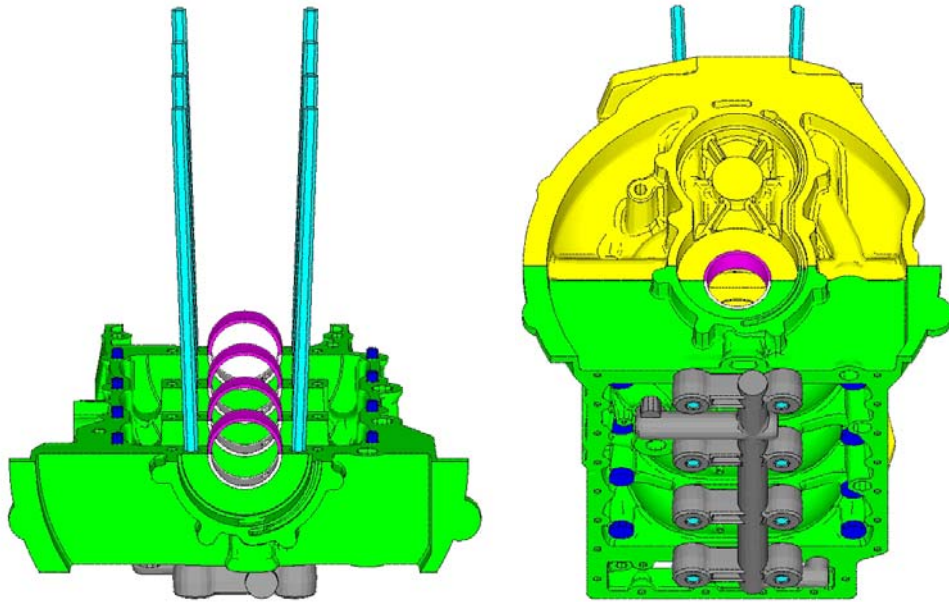


Figure 10. Crankcase assembly FE model.

From the dynamic crankshaft analysis main bearing loads were assessed over the operating engine speed range of 1000 to 6500rpm. 4 loads were selected for each bearing panel; max firing load (4000rpm for all), max side loads and max vertical load (6500rpm for all), see Figure 11. From the 16 loadcases selected some are duplicated thus it was only necessary to analyse 10 of them to capture the full range of loads on all panles. For each of these loadcases the loads at each of the 4 main bearings was applied as a sinusoidal pressure distribution on the bearing shell. 12 analysis steps were run in total; an initial assembly, a uniform temperature rise and the 10 bearing loadcases.

The stress distribution in the assembly when the maximum firing load for cylinder 2 is applied are shown in Figure 12. FEMFAT was used to assess the durability of the assembly and the calculated fatigue safety factors are shown in Figure 13.

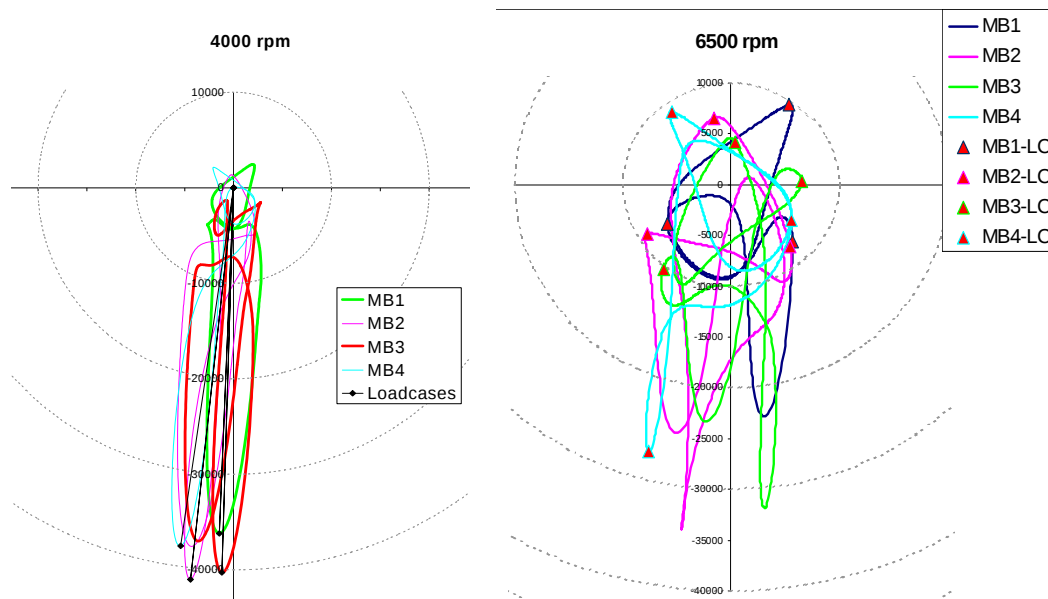


Figure 11. Main bearing loads.

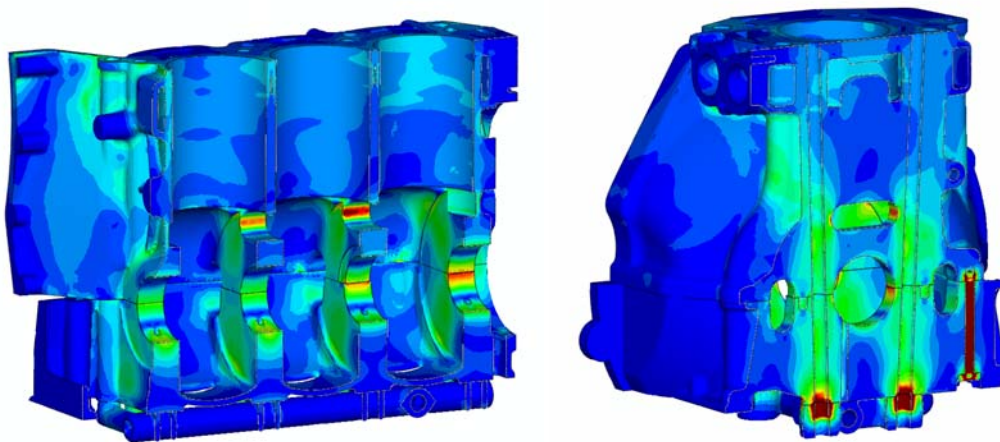


Figure 12. Crankcase assembly stress distribution.

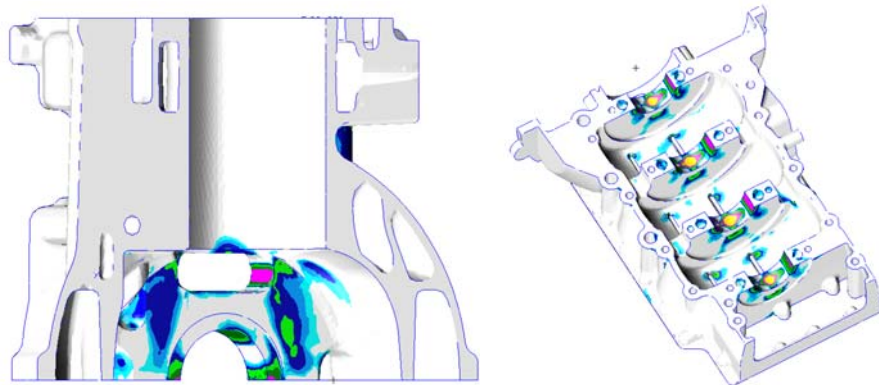


Figure 13. Fatigue Safety factors for crankcase and bed-plate.

5. Cylinder Head and Block Assembly Analysis

The MPT downsize engine utilises the latest direct injection injector technology and has a centrally located injector. With both the injector and spark plug in the centre of the combustion chamber adequate cooling of this region is difficult. The complex cooling jacket design which wraps around the plug, injector and ports was optimised using CFD analysis. A cross flow configuration was also used in order to achieve uniform cooling for each of the cylinders. A cross section of the combustion chamber layout is shown in Figure 14. In order to ensure the cooling jacket provides adequate cooling a thermal analysis of the head and block assembly was carried out. This was followed by a thermo-mechanical analysis of the assembly to verify head and block durability and adequate head gasket sealing performance.

The head and block analysis model (Figure 15) comprised of the block, bed-plate, nut plate, head bolts, cylinder head, head gasket, valve guides, valve seats and valves.

For the thermal analysis convection coefficients (htcs) were applied to all surfaces of the model to simulate the heat input into the engine from combustion and hot exhaust gases and the cooling from oil and coolant flow, from intake air and from the external surfaces. The different htc regions used for the cylinder head are shown in Figure 16. The htcs for the cooling jacket (Figure 17) were mapped from the CFD analysis.

The temperatures calculated from the thermal analysis (Figure 18) were then assessed to ensure that critical areas such as the cylinder head exhaust valve bridge and the region around the injector and between the cylinder bores remained within acceptable levels.

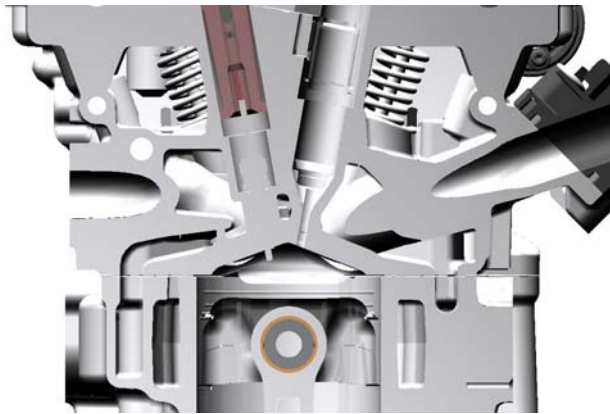


Figure 14. Engine assembly, section through combustion chamber.

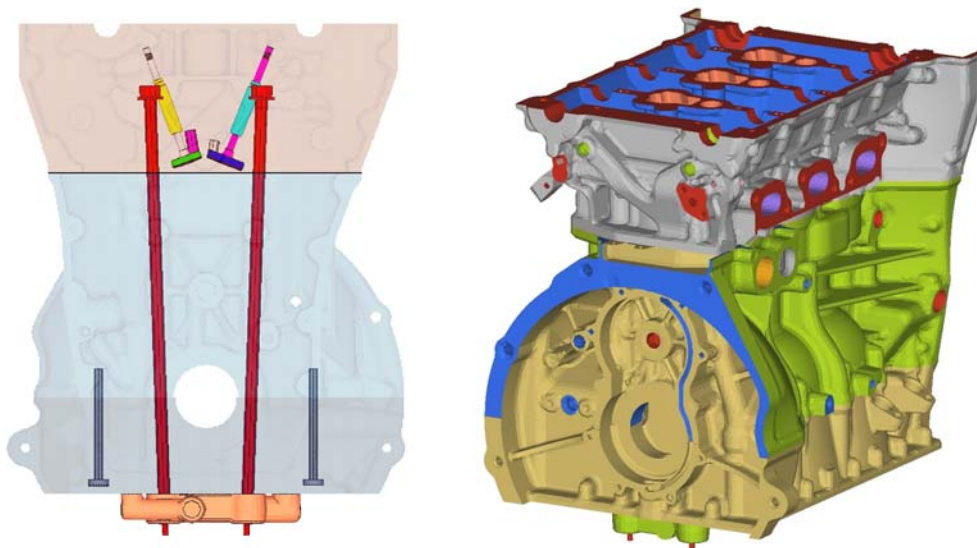


Figure 15. Head and block assembly FE model.

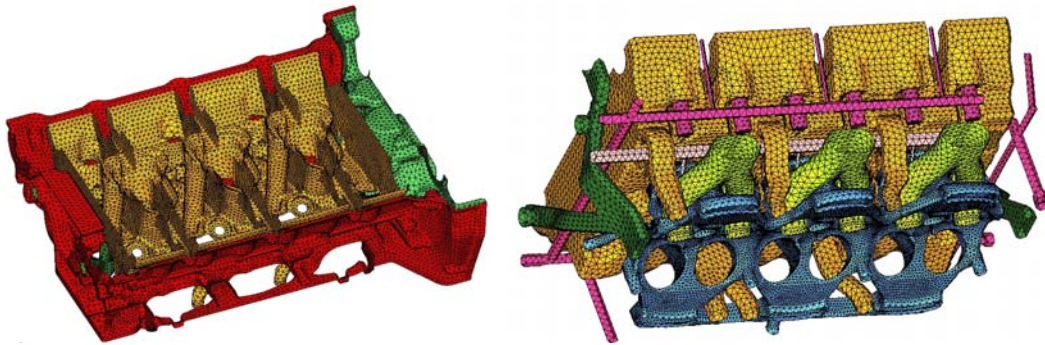


Figure 16. Wetted surfaces used for application of htcs.

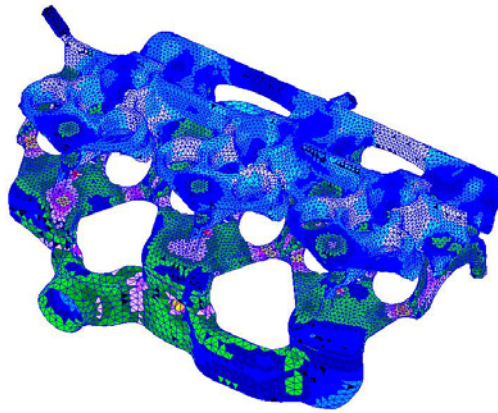


Figure 17. Cooling Jacket htcs mapped from CFD analysis.

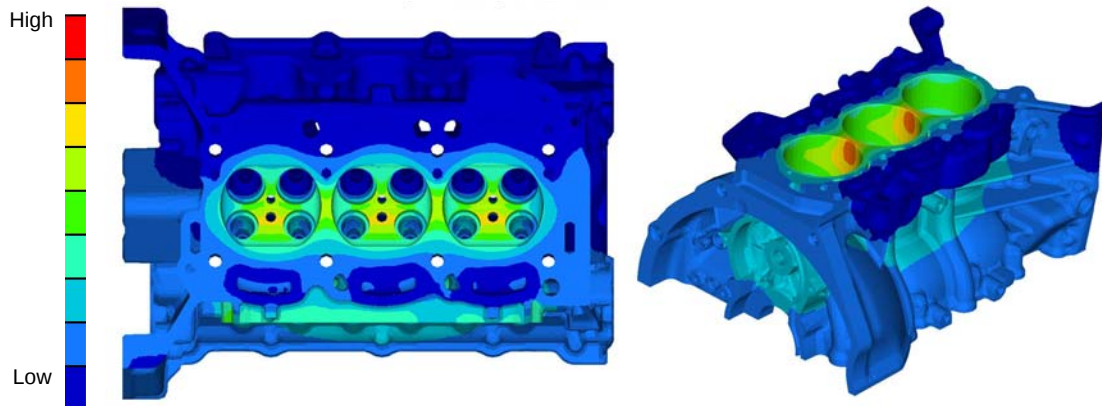


Figure 18. Head and block temperatures.

The structural analysis includes a full non-linear definition of the cylinder head gasket modelled using gasket elements (GK3D8N). Separate gasket properties (*GASKET BEHAVIOUR) were defined for each region of the gasket (stopper, full bead, half bead etc) as shown in Figure 19. The structural analysis model also includes material plasticity for the aluminium head and block. Small sliding contact (with friction) is defined between all mating components, and interference fits between valve guides and seats and the head are applied. The main cylinder head bolt loads are applied using pre-tension sections.

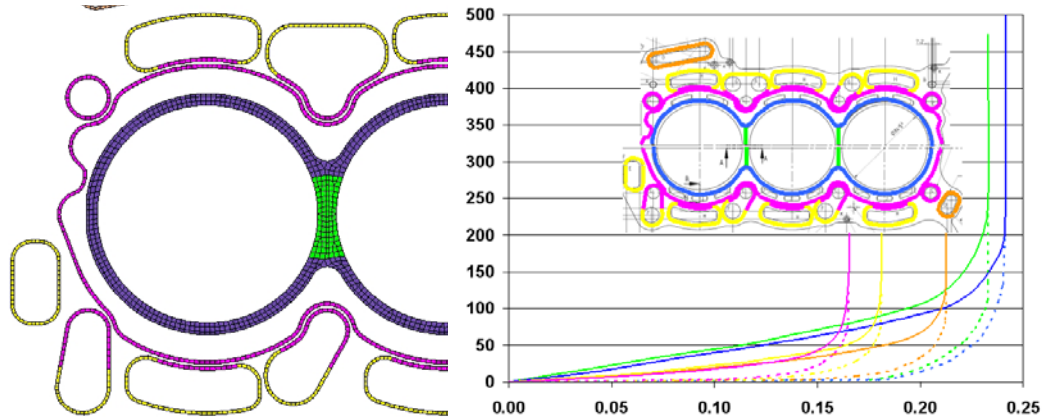


Figure 19. Head gasket definition.

10 structural analysis steps were analysed (see Figure 20) . A pre-assembly step was used to apply the interference fits before applying the bolt loads in the cold assembly step. The temperatures from the thermal analysis were then applied before gas pressures for each cylinder were applied in turn. To assess the gasket sealing the engine was cooled down and gas pressures applied to the cool engine to simulate firing of a cold engine.

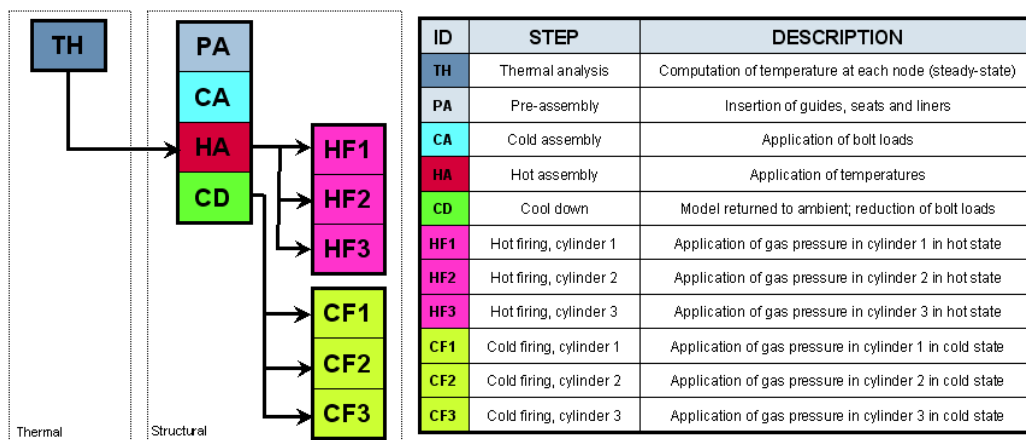


Figure 20. Analysis steps.

Figure 21 shows the stress distribution in the head and block assembly for hot assembly. Durability of the assembly was assessed using FEMFAT for high cycle fatigue (hot firing cases) and low cycle (thermal) fatigue (cycling from hot assembly to cool down). Pressure in the gasket beads was assessed for all the firing cases to ensure adequate sealing, an example of gasket pressure distribution is shown in Figure 22. Cyclic deformation of the gasket bead is also assessed against manufacturers criteria to ensure that cracking of the gasket wont occur.

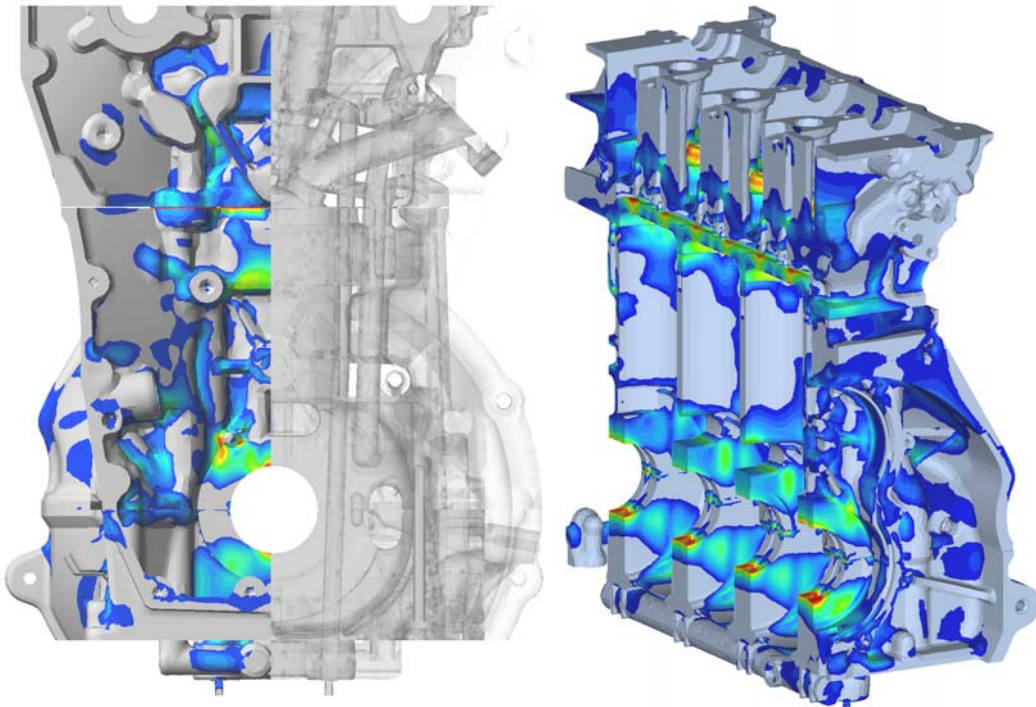


Figure 21. Head and block stress distribution.

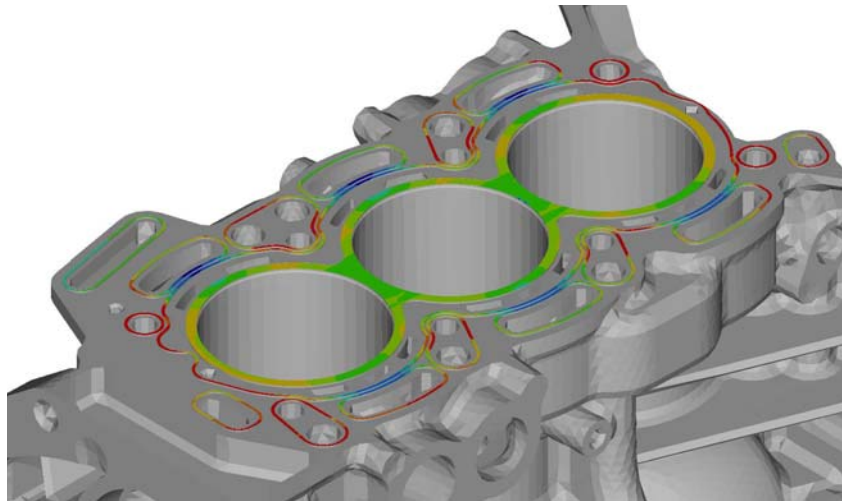


Figure 22. Gasket contact pressure.

6. Exhaust Manifold and Turbo Assembly Analysis

In order to achieve the performance, driveability and fuel consumption targets for the project in the most cost effective manner a twin turbocharger configuration was adopted. The twin turbochargers are configured in a sequential layout and are positioned as close to each other as possible to keep the gas path to a minimum. The twin turbo assembly is shown in Figure 23.

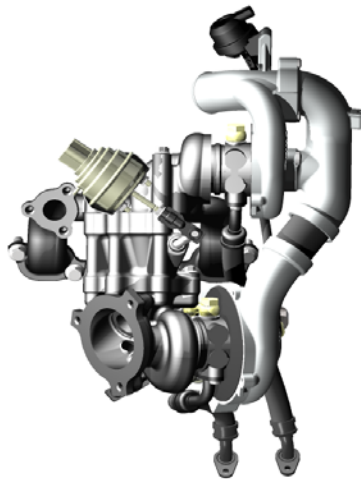


Figure 23. Twin turbocharger layout.

The housing for the first turbocharger, the high pressure one, is integrated into the exhaust manifold. A by-pass block is also mounted to the manifold to direct the gas to the low pressure turbocharger.

In order to assess the durability of the manifold and turbo housing assemblies a transient thermo-mechanical analysis was performed. The transient simulation mimics the exhaust manifold crack test which is a test designed specifically to test manifold durability. This involves running the engine at maximum power for approx 7.5 minutes and then motoring it at 3000 rpm for 2.5 minutes to blow cool air through the engine and thus rapidly cool the manifold, this cycle is then repeated.

For the thermal analysis the heat input to the exhaust comes from heat transfer coefficients and a bulk fluid temperature calculated from a CFD analysis. Transient CFD analysis of an engine cycle is carried out in order to determine the heat transfer coefficients through the cycle for both maximum power and the motored cool down. This data is then time averaged to give an average value for the cycle that can be used in the FE analysis. These htcs are mapped from the CFD mesh onto the structural mesh as *FILM coefficients. Cooling of the manifold due to convection and radiation are also taken into account and the appropriate htcs and radiation coefficients are applied to the external surfaces.

The FE model of the exhaust manifold assembly is shown in Figure 24. Three steps were analysed. Step one starts with the manifold at ambient temperature with the boundary conditions for maximum power applied for 7.5minutes to heat the assembly up. Step 2 is cool down and boundary conditions for the motored condition are applied and the analysis run for 2.5minutes. The manifold is then heated up again in step 3 for 7.5mins. Temperatures of the manifold at the end of the cool down and heat up steps are shown in Figure 25.

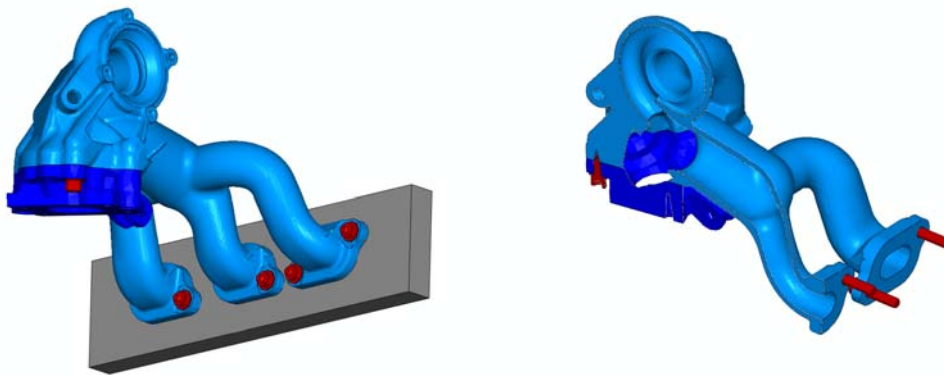


Figure 24. Exhaust assembly FE model.

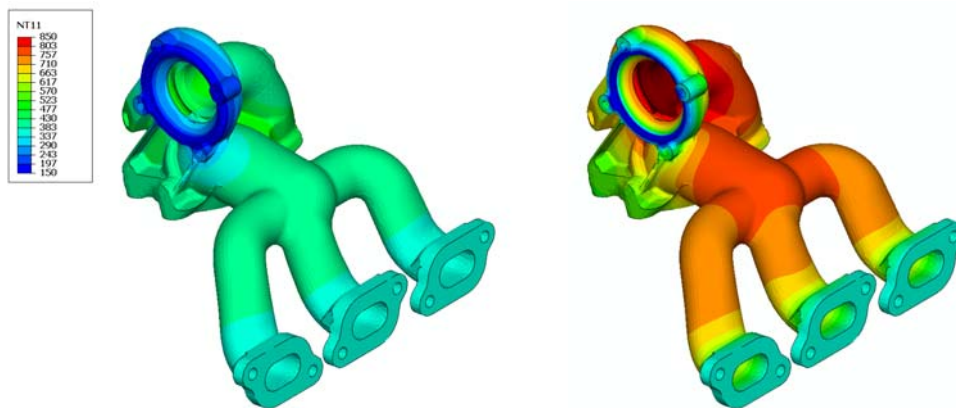


Figure 25. Exhaust temperatures.

Maximum stresses occur in the manifold approximately 30 seconds into the cool down and heat up steps (see Figure 26) when the components with low thermal inertia cool down or heat up rapidly but areas with high thermal inertia like the housing lag behind, producing maximum thermal strain. Durability of the exhaust manifold is assessed by comparing the equivalent plastic strain amplitude over one cycle to strain life data for the manifold material. Figure 27 shows the predicted equivalent plastic strain amplitude.

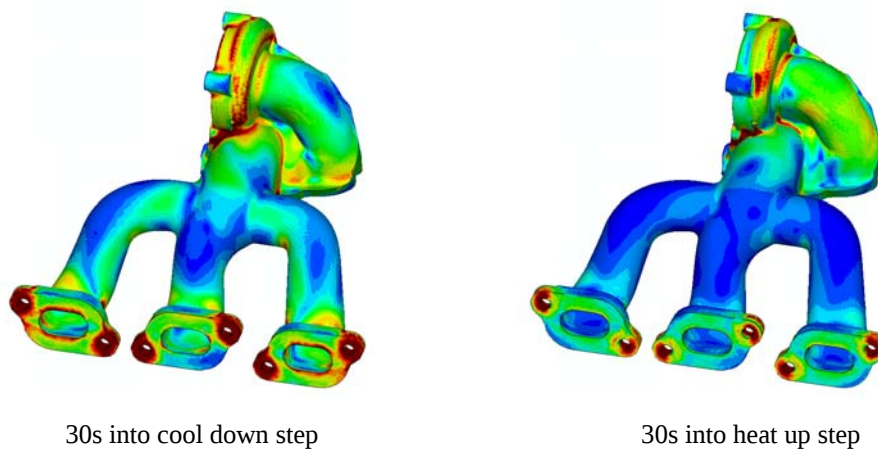


Figure 26. Exhaust manifold von Mises stress distribution.

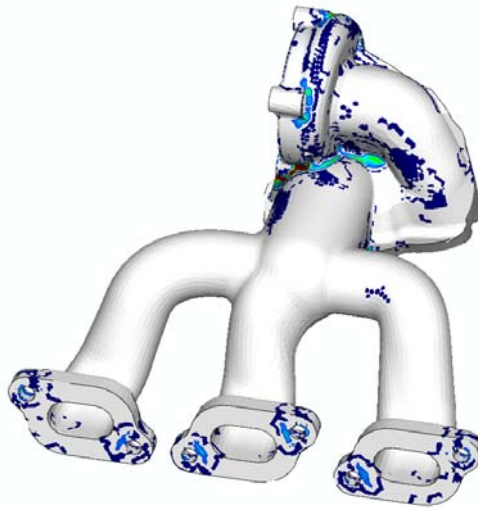


Figure 27. Exhaust manifold equivalent plastic strain amplitude.

7. Conclusion

The first I3 downsizing engines have now been built and initial test results show that all performance targets will be met. Analysis has played a crucial part in the design of this engine and will help ensure the durability of the engines throughout the extensive testing programme that has just commenced.

8. References

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