New Monobolt Piston Design for Large Engines

High-load pistons for modern, powerful large diesel and gas engines require increasingly more compact and lightweight designs. Due to the low installation height, less room remains for the required screw length in traditional large-bore piston designs, where the piston crown is screwed to the piston skirt. With the new development of the compact monobolt piston design, the Mahle Large Engine Components Profit Center succeeded in satisfying the requirements for an extremely low installation height of a composite piston, while allowing high mechanical and thermal loads.

1 Introduction

Due to their sizes and stresses, pistons for modern four-cycle large diesel and gas engines ranging from 160 mm to 640 mm in diameter are no longer configured in single-piece designs. Depending on the load, composite large-bore pistons are provided with a forged steel crown, combined with a forged steel or aluminium skirt. In addition, piston skirts made of nodular cast iron can also be used. The piston crown and skirt are screwed together either via a central anti-fatigue bolt, or via two, four, or more screws.

In an effort to comply with future emissions limits, while increasing performance and efficiency at the same time, the demands placed on pistons as the central engine component are on the rise as well. Optimized combustion processes result in new piston bowl geometries, which in conjunction with a required low weight and drastically reduced piston height or compression height, have made a complete redesign of the piston screw assembly necessary.

2 The Monobolt Piston Design

With a drastically reduced compression height, the installation space remaining between the conrod small end and the inside piston form is no longer sufficient to use conventional length anti-fatigue bolts. In order to implement a screw assembly, a design solution was required, which used the tight space between the conrod small end and the bowl dome, while guaranteeing the same functionality of an anti-fatigue bolt. The requirement regarding the new screw assembly concept included the option of a forged steel or forged aluminium skirt, depending on the component load.

By integrally forming a central threaded stud directly onto the piston crown and in conjunction with an anti-fatigue sleeve supported on the piston skirt, the new monobolt screw assembly concept described in **Figure 1** was ultimately implemented. The very short threaded stud alone does not have the required elongation length. The additional required length is provided by the anti-fatigue sleeve. The thread is cut to the lower end of the threaded stud, and the assembly is screwed together with a special nut. Four oil outflow grooves are provided in order to ensure that the cooling oil can effuse from the internal cooling gallery.

Two examples of monobolt piston developments that have already progressed to the production stage are shown in **Figure 2** and **Figure 3**. The extremely compact design resulting from the reduced compression height becomes especially apparent in a direct comparison with the previous version. Figure 2 shows the monobolt piston designed for a maximum peak cylinder pressure of 190 bar for a diesel engine with forged steel crown made of 42CrMo4 alloy and forged aluminium skirt made of the M124P alloy with pin bore bushing. In this configuration, the compression height was reduced by 23 % from 128.8 mm to 105.1 mm.

A monobolt piston with forged steel crown made of 42CrMo4 and forged piston skirt made of 38MnVS6 designed for a maximum peak cylinder pressure of

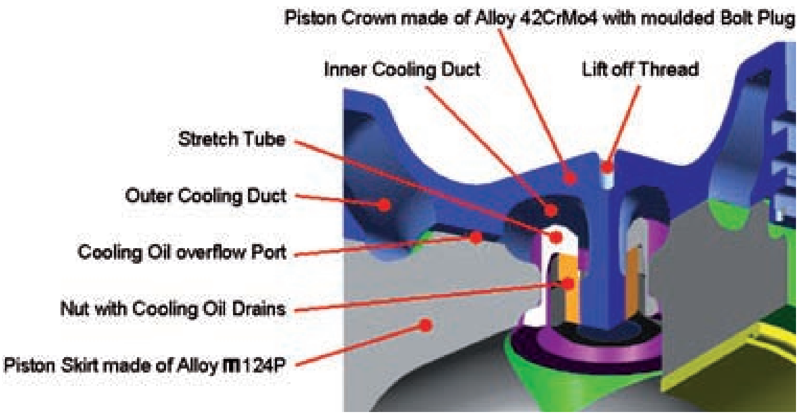

Figure 1: The monobolt screw assembly concept

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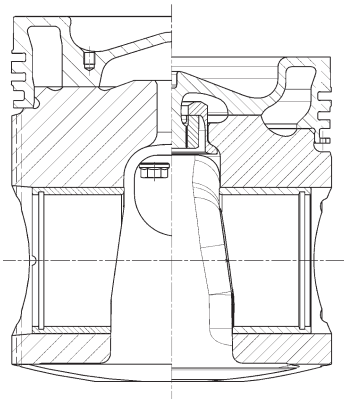

Figure 2: Comparison of the previous four-screw version with the newly developed monobolt design

250 bar in a gas engine achieves an 11 % lower compression height compared to the previous version, Figure 3.

3 Piston Development

In the beginning, development was focused on optimizing the component geometry through comprehensive finite element computations. The safety factors on the highly loaded locations on the piston crown and skirt as well as on the anti-fatigue sleeve were determined using the load combinations of temperature, bushing overlap (for the version with aluminium skirt and pin bore bushing), screw pretensioning, peak cylinder pressure, inertia force, and lateral force. For the piston with the aluminium skirt and pin bore bushing, the relative theoretical service life of the locations subject to high load was determined with respect to the required life. Based on the VDI 2230 Directive, the safety margins at the upper and lower anti-fatigue shaft transition and at the thread of the threaded pin were determined for high and low cycle loads, **Figure 4**. Furthermore, the clearance and force distribution between the inner and outer support surfaces of the piston crown to the skirt and the screw tightening values were optimized based on calculations.

The projected screw stress was analyzed by means of DMS measurements on the threaded pin and on the anti-fatigue sleeve outside of the engine, i.e., experimentally under dynamic load in the hydropulsator. The results obtained from the experiments were used for the subsequent validation of the computational results.

On the cooling oil shaker test bench, the catching efficiency of the cooling oil free jet and the filling level of the cooling channels were analyzed. By balancing the volume flows between the external and internal cooling galleries, it was possible to optimise the cooling oil-conducting cross-sections with respect to the

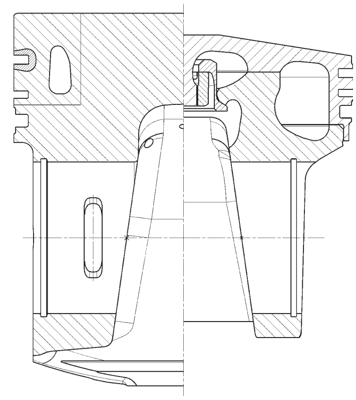

Figure 3: Comparison of the previous electron beam-welded aluminium piston with the newly developed monobolt design

rated power through an appropriate rotational speed and the associated cooling oil volume flow.

The temperature load projected based on zero-dimensional cyclic process parameters was validated by a temperature measurement in the engine. For this purpose, a temperature measuring piston equipped with thermocouples was used. The thermocouple pairs positioned on the internal and external cooling gallery allowed the computation of the heat transmission coefficient on the cooling gallery side and consequently an assessment of the cooling effectiveness. Calibration and endurance tests on the test bench and in field engines concluded the development of the new piston design. The successful design was then ready for series production.

4 Summary

With the new development of the monobolt piston design for large engines, Mahle succeeded in satisfying the increased requirements for a compact, lightweight design, while allowing higher mechanical and thermal loads.

In addition to the extremely compact design, the monobolt piston provides the ability to combine a variety of combustion bowl geometries for different applications with the same piston skirt. Depending on the load, the piston skirt can be configured in aluminium or in steel, for peak cylinder pressures that clearly exceed 200 bar. ■

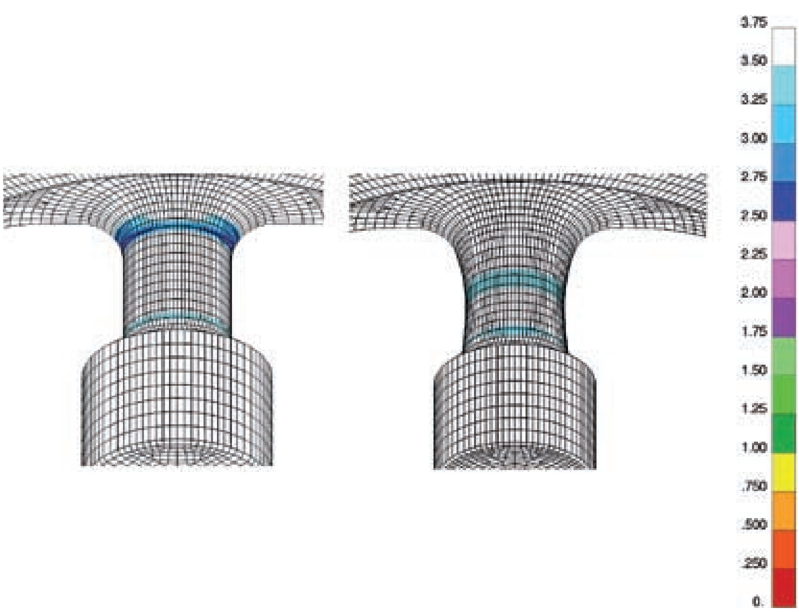

Figure 4: Comparison of the safety margins on the threaded pin between the initial design on the left and the optimized design on the right

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