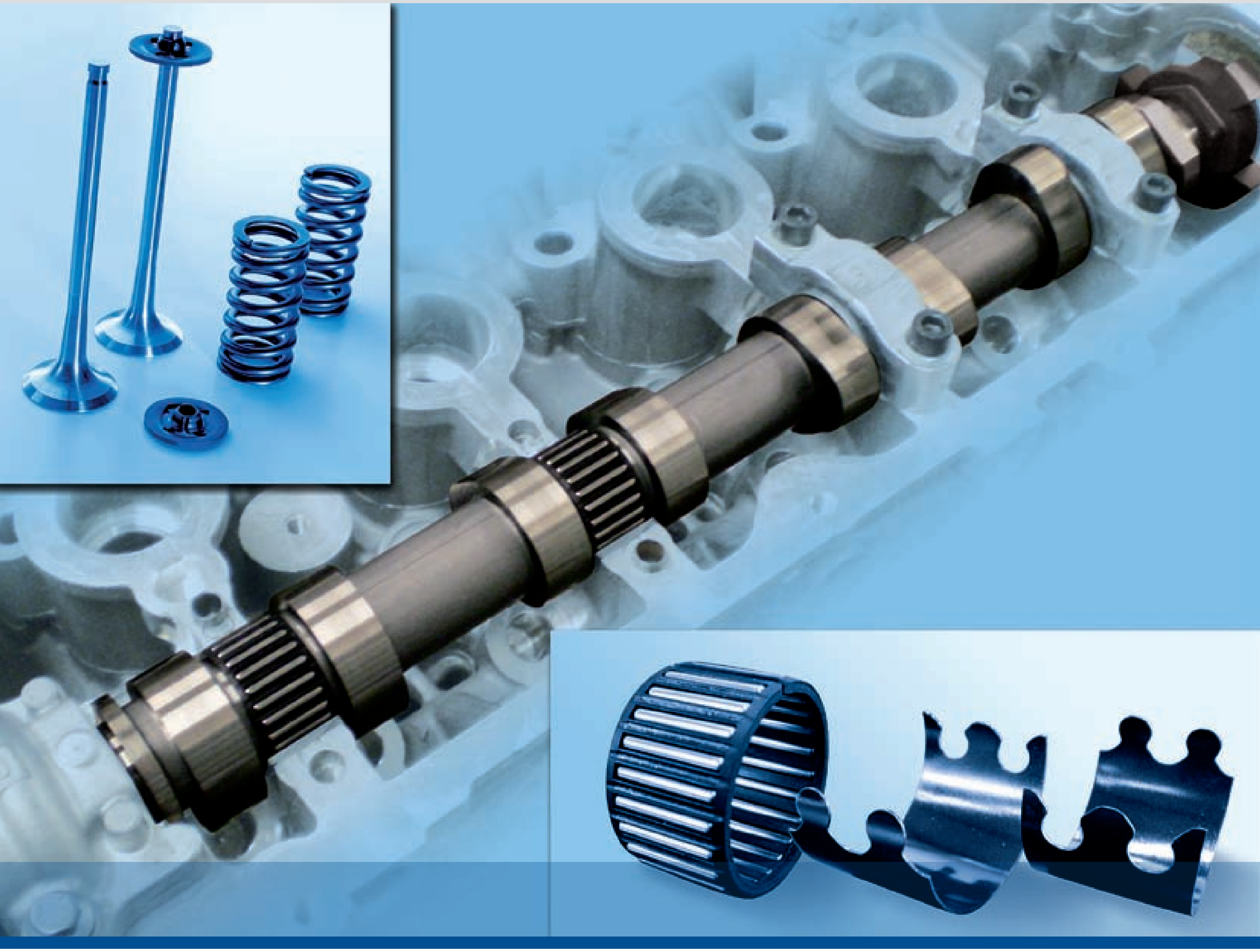

Friction and CO_2 Reduction through an Integrated Approach of Valvetrain Components

Increased focus and concern for automobile carbon dioxide emissions has recently led to a significant increase in the development work for suitable emission reduction technologies. The potential financial cost to the car manufacturer for not meeting strict emissions targets is considerable and such costs are driving this fervent development activity. This article should serve as an overview of the extent to which Mubea, as a manufacturer of valve train components – valve springs, retainers and camshafts – can support the OEM in reaching these stringent targets.

1 Introduction

The reduction of fuel consumption and carbon dioxide (CO_2) emissions of internal combustion engines can be achieved both through optimisation of system components and through an intrinsic design involvement at the engine concept stage. There are a number of key drivers that deliver clear reductions. The following optimisations and developments have been studied in detail with a combination of CAE methods and empirical measurements:

- optimisation of the valve spring with regards to installation length and weight
- reduction of the dynamic mass and valve side load through the use of a new keyless retainer
- reduction of the driving torque of cam shafts
- reduction of the oil-pump power consumption through reduced oil volume flow in the cylinder head.

In the following, the measurements are described in detail, examined and assessed.

2 Measurements

2.1 Optimisation of the Valve Spring

The use of a conical or beehive-shaped spring form leads to a strong progressive characteristic with a light and responsive valve head. The spring is naturally helix-shaped which results in the tension of the wire increasing through the cross-section to the inner of the spring. By using wire with an elliptical profile instead of the standard round wire, this tensile stress can be designed to spread in a homogeneous way throughout the cross-section, thus reducing the overall tensile level. Suitable material fatigue performance is ensured through the use of heat treated high-tensile wire. The length of the spring, the installation height of the cylinder head and also the valve stem length can be effectively reduced through this increase of tensile load capability in the spring [1].

2.2 Friction and Mass Reduction

Valve stem side load is responsible for frictional losses in the system. The symmetry of the valve spring and the structure and number of coils are the main drivers behind this side load [2]. To combat sideloading, a recently developed Keyless Retainer can be utilised, **Figure 1**. Due to its spherical beared behaviour the spring generates less side load. The retainer has previously required a key to fix it to the valve stem, a design that allows the retainer to rotate around the valve stem in use. The Keyless Retainer gives the spring end sufficient freedom at the valve stem connection to prevent transmission of bending moments. In comparison to the standard solution, the keyless retainer is able to reduce mass up to 60 %, the number of components up to 67 % and the axial tolerance is reduced by up to 60 %.

2.3 Reduction of the Driving Torque of Camshafts

It is obvious that the friction reduction of a hydrodynamic supported cam shaft at low speed increases by mixed friction. An appropriate coating (DLC), **Figure 2**, reduces that effect and transmission can be cancelled through the use of a roller bearing. The installation of the roller needles and bearing shells can be simply and directly integrated into the camshaft design and assembly. Additionally, this leads to a reduction in the cylinder head oil requirement, in turn reducing the loading on the oil pump itself.

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Figure 1: Keyed retainer in comparison to the Keyless Retainer

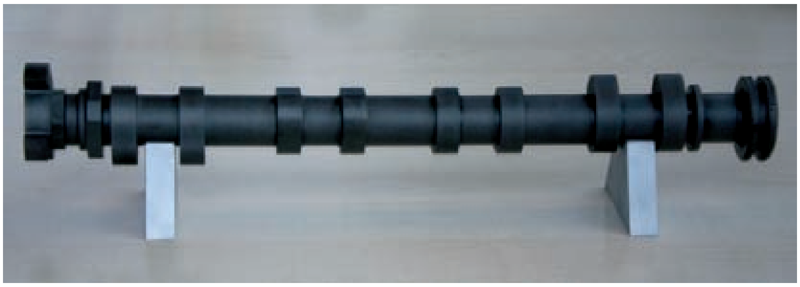

Figure 2: DLC-coated exhaust camshaft

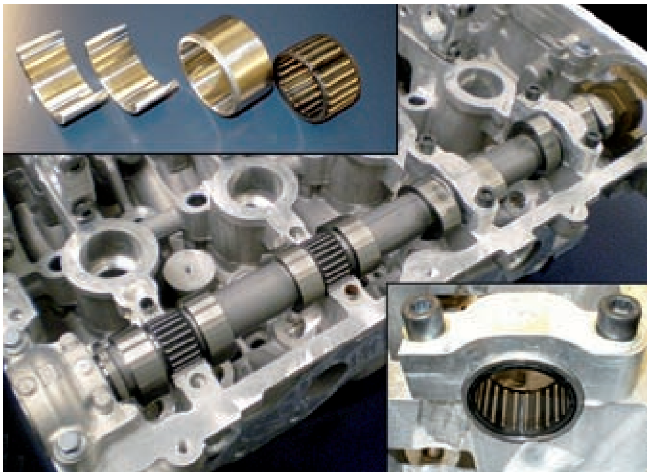

Figure 3: Cylinder head with a roller bearing supported exhaust camshaft

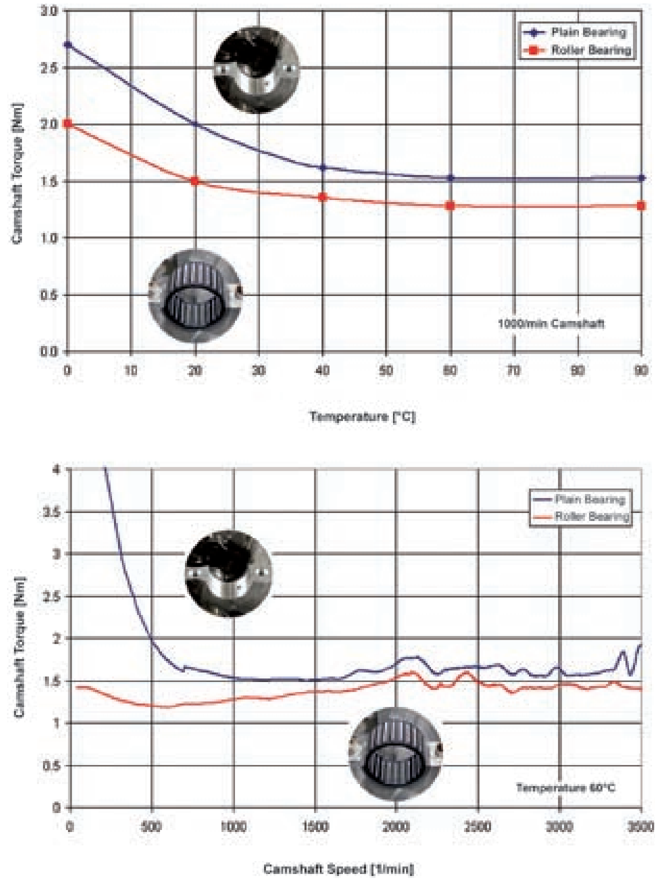

Figure 4: Comparison of camshaft torque for plain and roller bearing

3 Examination of the Complete System

The following examinations are based on a modified standard 16 valve DOHC cylinder head from a 1.6 l four-cylinder spark ignition engine, **Figure 3**. Static engine speeds and sweeps at constant oil temperatures between 0 °C and 90 °C were examined. **Figure 4** shows the dynamic driving torques of a camshaft in the relevant speed range. It becomes obvious that the effect of mixed friction is eliminated at a low engine speed and thus the driving torque is clearly reduced. The constant level of drive torque is reached at a higher engine speed where improvements reduce. The static measured points clearly show an important transition around 25 % on the field of lower temperature. This decreases with rising temperature as expected, and results in the characteristics of the plain bearing performing better due to decreasing oil viscosity. However, the roller bearing requires significantly lower oil volume than the plain bearing, thus the oil pump design can be optimised leading to a power consumption requirement of around 5 % to 8 %. This can be achieved with normal pumps through a reduction of the rotor width. This design optimisation can be simply achieved and requires no further measurements. With regards to the lower temperatures, the DLC-coated camshaft also leads to friction loss, that also increases with higher temperatures.

The effect of the spring optimisation and the keyless retainer connection are also now examined. To consider the interaction between spring and retainer we consider the single valve train. For example: If the keyless retainer has a mass, approximately 5 g lower than the

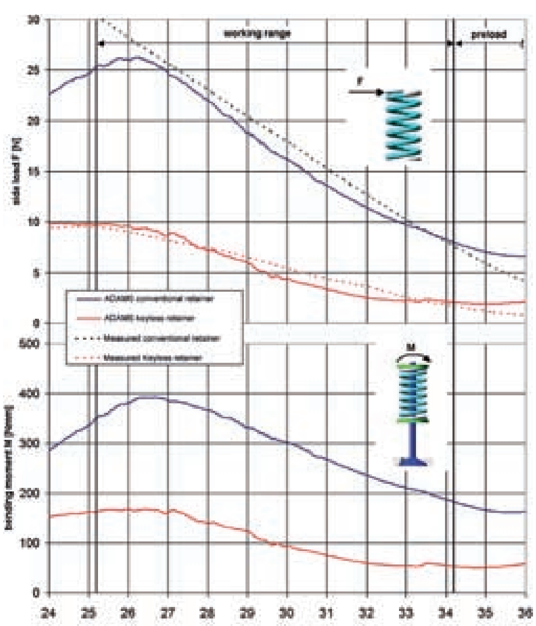

Figure 5: Reduction of side load bending moment in the valve stem guiding by using the Keyless Retainer

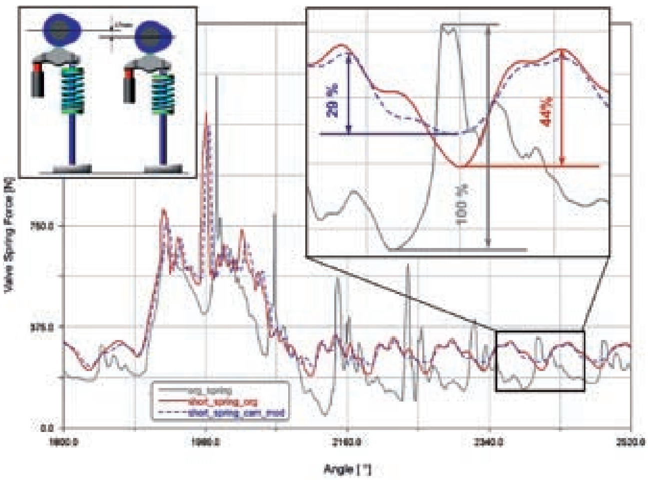

Figure 6: Improvement of valvetrain dynamic by optimisation of valve spring and closing ramp

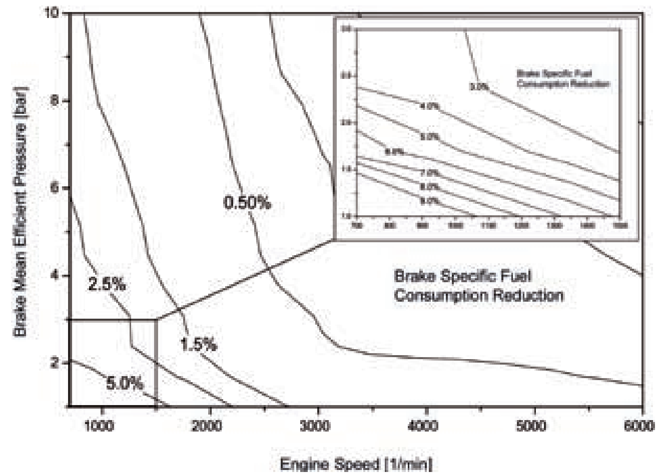

Figure 7: Engine characteristic of fuel consumption reduction by valvetrain optimisation

standard solution, then the maximum spring power (F_2) can be decreased by about 20 N at a constant total engine speed [3]. Thus, the friction in the valve train will be decreased. The additional effective degrees of freedom at the free spring end can be considered by a modification of the ADAMS/Engine spring model. In **Figure 5**, the spring side load at the retainer is shown over the total applied stroke and is shown in comparison to the keyed retainer design. The forces were measured, as per the simulation, with a dedicated instrument in support of the model, showing excellent correlation. The bottom of the figure highlights any significant bending moments generated by the spring side load. Side loads and bending moments resulting from the roller-type cam follower are not shown in the figure and they are retained unchanged. The side load and the bending moment are similar to each other and are proportional to the friction on the valve stem. The data from the keyless retainer solution shows a marked improvement in comparison to the standard keyed design.

As mentioned above, the spring optimisation achieves a reduction of both dynamic mass and installation length. This result was considered in the multi-body-simulation. An excerpt of the results is shown in **Figure 6**. A spring at a high engine speed with a high natural frequency, equal stiffness and minor mass has a positive effect on the valve train dynamic. The force amplitudes are reduced and the closing ramp of the cam influences the seating velocity of the valve. Because of that scenario the coordination of the closing ramp on the valve spring seems to be effective.

In this stage of optimisation only the ramp form is influenced, whereas time, height and length of the ramp are retained unchanged. This leads to another reduction of the force amplitude in the base-circle phase [1, 4]. For friction in the valve train, this optimisation is not relevant.

4 Summary

After considering several aspects of the complete system design, our aim was to calculate possible savings in a driving cycle. In **Figure 7** the engine characteristic is summarised. It describes the percentage of specific fuel consumption savings according to engine speed and the mean effective pressure. As mentioned above, the savings in the lower speed range and load range are the highest. Different scenarios were also examined through the use of a cycle simulation tool, which is able to examine fuel consumption in the New European Driving Cycle (NEDC) according to vehicle and engine characteristics. It is clear that there was a saving between 1.5 % to 2 % for a compact car with a 1.6 l engine in NEDC. This corresponds to a carbon-dioxide reduction of between 2 g and 3 g per km. It is important to realize when considering this reduction that on a vehicle with an emission level of 130 g CO_2/km it has enormous value. Engines with other valve train topologies are reaching higher values.

In any such study it is important to consider the commercial practicalities of implementing such innovations. As an overview, the additional costs generated by such optimisations equate to about 2.50 € per gram of CO_2. In contrast to the discussed extra duty in the European Union there is a ratio of 1:10 regarding the cost-efficiency. It is therefore clear that the optimisations and innovations explained herein are key steps in the right direction of achieving target emission values and thus the inevitable extra duties involved.

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