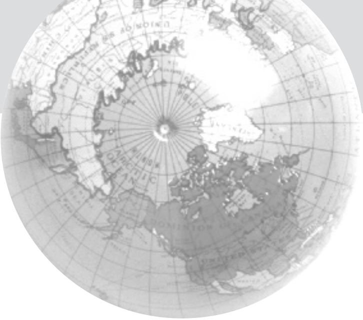
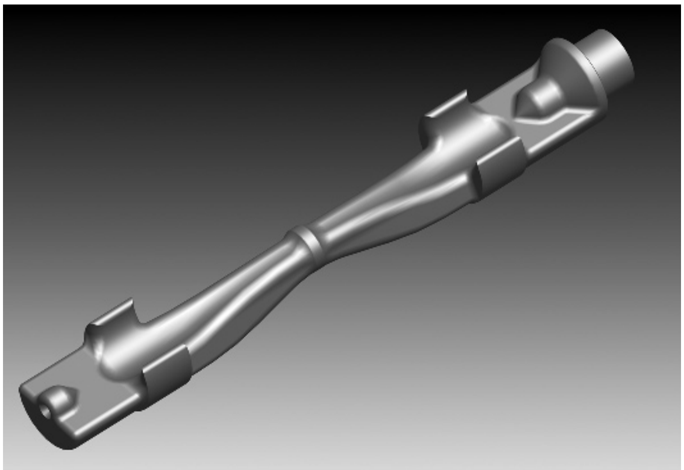


You will find the figures mentioned in this article in the German issue of MTZ 06|2007 beginning on page 460.

Partiallagerwelle –
Entwicklung einer Leichtbau-Ausgleichwelle

Partial Bearing Shaft
Development of a Light-weight Balancer Shaft



The oscillating movement of crank mechanism in piston engines produces free mass forces that are compensated by means of balancer shafts with specific unbalance. Based on an expedient arrangement of unbalanced masses and a new design of the bearing points, Herzog Intertec GmbH has developed the partial bearing principle, which provides considerable weight reduction as compared to the conventional balancer shaft geometry.

Authors:
Ewald Herzog, Raphael Herzog and Patricia Will

1 Introduction

Balancer shafts in piston engines are not designed for main tasks such as crankshafts, pistons or connecting rods. They are integrated into the design only if the external effect produced by free mass forces could not be borne by the overall system. Balancer shafts only serve to compensate the undesired side effects of the crank mechanism. They involve, however, a deterioration of the engine efficiency. For this reason, eliminating the necessity of balancer shafts is a main target in engine design. Especially with four-cylinder in-line engines, balancer shafts are indispensable to compensate the second-order oscillating mass forces. Optimising the balancer shaft in terms of manufacturing costs and own weight is thus continuously gaining in importance.

As the four-cylinder in-line engine is the most frequently used drive concept in the passenger car sector, this engine type alone already offers a huge saving potential. In addition, increasing requirements in terms of driving comfort are also to be met by medium-sized and compact cars, with engine smoothness being of decisive importance.

2 Function

2.1 Analysis

The main component of shafts used to compensate mass forces is the unbalance body of semicircular cross-section that extends almost over the entire length. In general, one or two loose bearings and one fixed bearing are provided. In order to subsequently mount the shaft in the housing, the unbalance body must not exceed the finished bearing cross-section. The bearings are often connected by means of a reinforcing bar on the opposite unbalance side. Without this reinforcing bar, the admissible bending moment depending on the centrifugal force and the resulting stress would be exceeded in operating condition. In addition, the deflection causes a misalignment of the bearings. The fixed bearing side often comprises a connection to the drive unit. The drive is ensured by the crankshaft either via integration into the chain drive or directly by cylindrical gear pairs.

2.2 Rating

The component rating is aimed at producing a centrifugal force by a static unbalance of minimum mass. When calculating the static unbalance, the relation shows that at constant unbalance and decreasing total mass, the eccentricity of the centre of mass has to be increased. As the available space is always restricted, the engineering limits for this constructional measure are rapidly reached.

To analyse an additional option for mass reduction, the balancer shaft is theoretically subdivided into two segments in longitudinal direction. While the unbalance side is located below the axis of rotation, the opposite unbalance side is arranged above the axis, **Figure 1**.

The eccentricity of the component's centre of mass is determined by the mass difference between both sides. This implies that for component weight reduction, constructional measures are required to reduce the opposite unbalance side.

The shaft rotation produces a centrifugal force. Statically speaking, the resulting force vector is always symmetrically arranged in the unbalance cross-section, as shown in **Figure 2**. This means that only this section of the shaft bearing surface is subjected to load. Considering the bearing cross-section, the maximum bearing clearance is produced on the opposite unbalance side.

3 Partial Bearing Geometry

Based on the conclusion that there is no bearing load acting on the opposite unbalance side, the bearing surface can be reduced on this side to save mass. One possible design of the bearing cross-section considering manufacturing aspects is shown in **Figure 3**. In this case, a semilunar bearing cross-section is obtained. The remaining running path can be subdivided into two zones: the load zone, where the force is transmitted by rotation and the supporting zone ensuring centric orientation even at standstill.

The potential offered by this innovative design is shown in **Figure 4**. As the original cross-sectional area is clearly reduced, a corresponding bearing cylinder mass saving is obtained.

In addition, the centre of mass is displaced from the axis of rotation towards the load zone. After having been changed to the partial bearing geometry, the original rotationally symmetrical bearing cylinders thus contribute to the total unbalance of the balancer shaft. The eccentricity of the component's centre of mass increases, while the component weight is reduced at the same time. This produces a positive side effect. The mass saving on the opposite unbalance side alone increases the eccentricity of the centre of mass proportionally more than the total mass is reduced. This results in a higher static unbalance. As this increase is actually not required, the mass on the side opposite the unbalance can be reduced until the original static unbalance is achieved again. In addition, an optimised mass arrangement on the unbalance side substantially reduces the degree of deflection. Reinforcing elements on the opposite unbalance side are thus no longer necessary, which has a positive effect on the overall weight balance.

4 Summary

Based on an innovative bearing point geometry, the new balancer shaft design according to the partial bearing principle ensures a decisive component weight reduction. With the present model example, the own weight of 1731 g per shaft of conventional design can be reduced to 1124 g when using the partial bearing principle. The required static unbalance and the resulting centrifugal force remain unchanged. Thanks to the lower mass inertia of the partial bearing shaft, the drive power required for the balancing system can be reduced, which in turn produces a positive effect on the efficiency of the entire engine.

As compared to the production of conventional balancer shafts, the manufacturing process is only slightly changed. Material savings will rapidly more than compensate the expenditure involved in the necessary adaptation of individual production steps.

In the test bench series comprising a 1000 hours endurance test carried out with cyclic speed groups, the partial bearing shaft has already proven its efficiency in practice.

By the way the patent is pending. ■