21

New achievements with “old” technology

Rolling bearings in automotive engines

Peter Solfrank

Addressing the requirements for modern drive trains for individual mobility in this paper would be like bringing sand to the beach. There is no doubt that the optimization of thermodynamics, i.e. the conversion of thermal into mechanical energy, represents an essential aspect in this context. However, over the past few years, increased consideration has been given to the significance of the purely mechanical efficiency of engines. Obviously, this aspect represents an important potential market trend for companies such as the Schaeffler Group with a traditional focus on rolling bearings.

Rolling bearings in engine design

Even in the very early years of the reciprocating piston engine, the design and installation of rolling bearings in these systems played an important role [1]. They were used not only in exotic applications but in a wide range of applications until the fifties and sixties of the last century: Examples of crankshaft bearings include the Porsche 936 or Auto Union 1000 S de Luxe Coupé 1962-1963. In motorcycle engines, crankshaft rolling bearing supports continue to play an important role. Since the proliferation of roller finger followers in valve trains, rolling bearings can also be regarded as a generally widespread technology in modern internal combustion engines in passenger vehicles. Accessory drives include a large number of bearings and have traditionally been a rolling bearing domain, as are water pump and alternator bearings.

Benefits of rolling bearings compared to plain bearings

Rolling bearings not only allow a precisely guided relative motion of two components. In most cases, the friction losses generated during this motion also represent the absolute minimum of what can be considered reasonably achievable from an engineering standpoint. In addition, the lubricant requirements for rolling bearings are often so low that in applications within the engine these requirements can be satisfied by the oil mist that is present nearly everywhere. This may potentially lead to the elimination of the need for pressure oil supply, resulting in considerable design and manufacturing-related savings. The reduced oil pump capacity also represents a reduction of parasitic losses and thus an efficiency increase for the engine.

Finally, the rolling contact of the rolling bearing's component surfaces results in both low friction and extremely low wear with minimal lubricant availability. This makes rolling bearings especially suitable for applications requiring wear-free motion without the availability of pressure oil supply. This aspect of almost dry-running capability with lubricant starvation is even more relevant for engine applications since there can be considerable loads on the bearings even when the engine is at a standstill. Examples are the bearing positions on crankshafts and camshafts which are subject to traction drive transmission loads. In cases where plain bearings depend on pressure oil supply, these bearings are extremely susceptible to wear. The system-related benefits of rolling bearings in these types of operation situations obviously make these suitable for both engines with a stop-and-start system and for hybrid applications. Both applications have a drastically increased number of starts compared to previous requirements.

Even for cold starts, rolling bearings clearly have benefits as demonstrated by a low breakaway torque and good dry-running capability with lubricant starvation and minimal wear.

Crankshaft

Serving as the dominant rotational joints in the engine, the main and conrod bearings of the crankshaft would seem to be an ideal occasion to replace the common plain bearings by rolling bearings. The implementation of this idea on a laboratory scale is shown in [2] with the impact regarding the potential fuel savings being quantified as 5.4 % in the NEDC.

These crankshaft bearings are subject to very high and highly dynamic loads. Bearing loading is characterized by forces in varying directions combined with the highly dynamic tilting. Both loads as well as tilting effects to some degree are influenced by the stiffness and damping properties of the bearings. This needs to be considered for every single point of the wide range of engine operation. All of this makes the mechanical system a challenge with regard to rating life calculations that are required as a starting point for the development of a reasonable bearing design. In addition, during the course of such a development, acoustic considerations must be addressed in an early stage, which places additional challenges on the overall system development (comp. [3]). Generally speaking, such a bearing system can only be developed as a combined system of shaft and housing, and also manufacturing and installation methods must be considered from the very beginning.

Compared to other rolling bearing applications conrod bearings are exposed to very special loading conditions: they not only allow for the relative rotational motion of shaft and surrounding structure but at the same time are subject to a guided motion which generates significant centrifugal forces upon the bearing components themselves. Finally, the cranktrain kinematics leads to a very dynamic fluctuation of the rotational relative motion between the crank pin and the conrod. This kinematic cyclic irregularity is superimposed by the fluctuation of the crankshaft's rotational speed caused by the irregular combustion process. The consequence of all of these effects is that the potential for reducing friction by rolling bearings in this position is not nearly as great as for the main bearings. Moreover, the technical challenges are relatively high as a result of the special loading and motion conditions described above.

When looking for a good compromise between minimized friction and reasonable engineering effort, a step-by-step process should be considered which begins with designing the crankshaft main bearings in the form of rolling bearings and continues to use the conrod bearings as plain bearings. This would gain most of the potential friction reduction while also limiting technical risks. However, the pressure oil supply to the conrod bearings must be ensured. Depending on the available design space, this may be carried out as a separate pressure oil feed into the crankshaft, or one of the main bearings remains a plain bearing to meet pressure oil supply requirements to the crankshaft in the conventional way.

Camshaft

The number of camshaft bearing positions is so high in most modern engines that the total friction losses and thus the potential reduction would appear to be significant. The loads, however, compared to the crankshaft are less extreme so that the friction heat that is to be expected is fairly low in relation to the heat conductivity of the ambient components of the camshaft and cylinder head structure. Under these circumstances, pressure oil supply often is not required any more, as in many cases it serves more for cooling purposes than for lubrication (for which a very small amount of oil is sufficient).

Friction calculations for various operating conditions of camshaft bearings as well as experimental results for the driving torque of cylinder heads lead to the conclusion that, in lower speed ranges, the plain bearings operate in mixed friction conditions. For medium to high speeds, on the other hand, the surfaces are separated by the oil film, resulting in hydrodynamic conditions, so that friction losses here can be reduced considerably.

Classic camshaft design

One-piece camshafts with small bearing diameters require the bearings (cage and outer ring) to be split. The bearing's separation line must be set in a relatively low-load area in accordance with the preferred load direction to avoid a significant load capability reduction. However, over the past few years more and more built camshafts have come into use. This concept allows for unsplit bearings to be used, either by mounting bearing inner rings onto the shaft or by using the camshaft base surface as a direct raceway for the rolling bearing, provided the appropriate material and processing methods are used.

Tests on friction reduction using camshaft rolling bearings have shown that a great amount of losses generated by mixed friction in the plain bearing can be avoided in the lower speed range.

For higher speeds, on the other hand, potential reductions are fairly low. Figure 1 shows a summary of measurements taken on four engines (4-cylinder DOHC, measurements on one camshaft each).

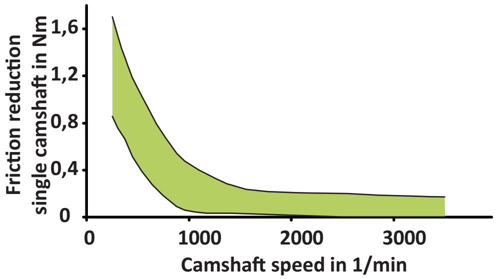

Figure 1 Scatter band of friction reduction by rolling vs. plain bearings for classic camshafts

Based on these results it seems recommendable to perform specific measurements individually for each potential application, and to thoroughly consider the cost benefit ratio keeping in mind all the secondary beneficial effects like elimination of pressure oil supply both from a design/manufacturing point of view as well as in terms of the reduced amount of oil in the cylinder head.

Tunnel camshafts

Despite the frictional disadvantages of such designs, there are still reasons for using camshafts with bearing diameters large enough to allow the camshaft to be installed into the housing in axial direction. The reduced number of bearing supports compensates only partly for the system-related frictional disadvantage of this principle, so that using rolling bearings to reduce friction losses could be an advantageous solution. The one-piece design with a thin-section inner and outer ring possible in this case can be used with hardly any changes to the overall camshaft concept.

Measurement results for this type of system show a significantly lower mixed-friction area on the plain bearing side. The advantage of rolling bearings with respect to friction is fairly consistently in the order of 0.2 to 0.5 Nm (Figure 2) over the full speed range. The greater potential savings in combination with the relatively moderate engineering effort in implementing the design and manufacturing technology thus lead to an extremely favorable cost-benefit ratio in the case of tunnel camshafts.

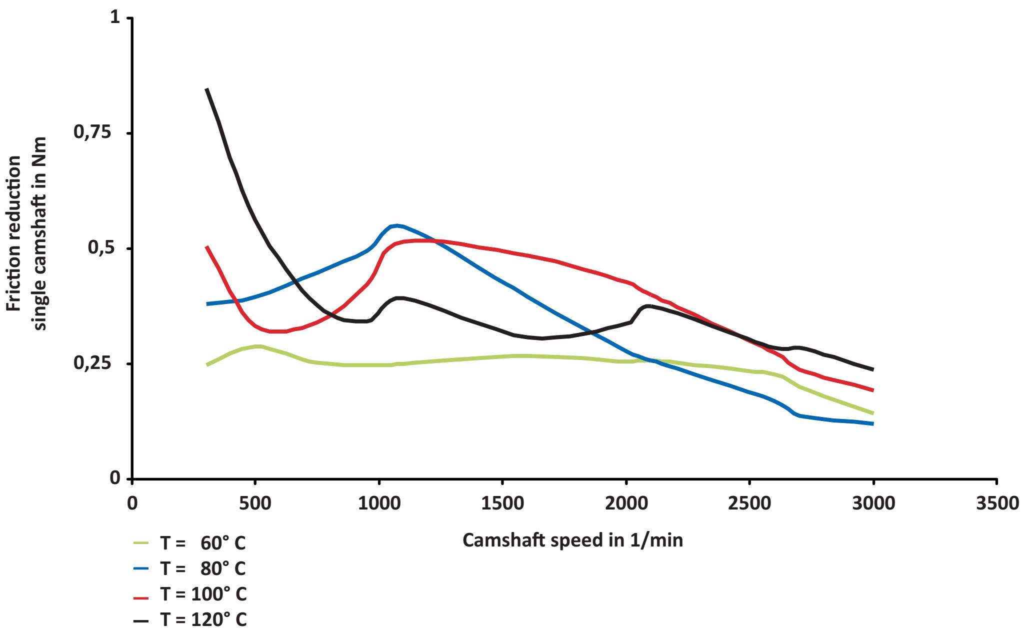

Figure 2 Friction reduction by rolling vs. plain bearings for tunnel camshafts

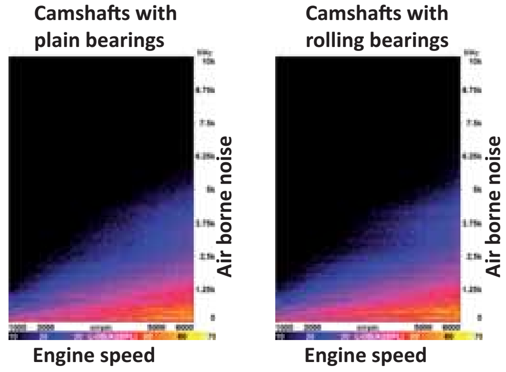

Figure 3 Comparison of airborne noise vs. engine speed
Left: camshaft with plain bearings, right: camshaft with rolling bearings

Acoustics

Compared to plain bearings the oil film between metallic surfaces in a rolling baring is significantly lower which is an indication for different acoustic behavior. In a representative application the customer relevant differences have been investigated. Purposely a 4-cylinder gasoline engine (1.4 liter displacement) in a compact car was selected for this investigation because the relatively low amount of acoustic insulation in the respective vehicle should make potentially unfavorable rolling bearing effects particularly evident. A direct comparison of acoustic measurements (dummy head microphone on the passenger side) in various driving situations has led to differences between both systems that are below the limit of perceptibility in almost all cases (Figure 3).

Besides friction reduction, the bearings' dry-running and starting characteristics play an important role, especially for camshaft bearings. It is the drive-side bearings in particular that are not only exposed to forces from the valve trains but also to additional loads from the tooth belt or chain drive. These additional forces are considerable even at engine standstill. In applications with stop-and-start systems or in hybrid applications, the bearings are subject to frequent stop and restarting under such considerable loads. Since these boundary conditions do not allow a lubricating oil film with load capacity to establish, plain bearings tend to show increased wear in such applications. In contrast, the rolling contact design provides the needle roller bearings or ball bearings with significantly better running characteristics for lubricant starvation, loads and low speeds so that they are particularly suitable for the drive-side camshaft bearing positions.

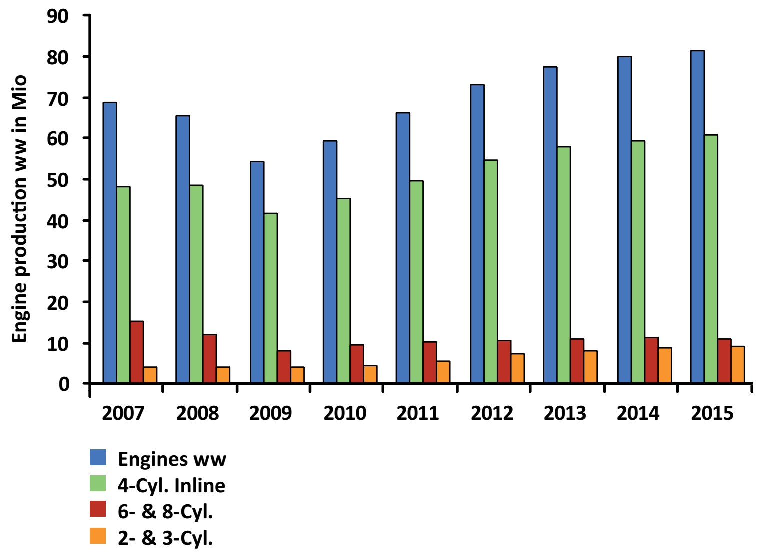

Figure 4 World market for passenger vehicle engines

Balance shafts

Some special aspects of the increasing use of balance shafts in passenger vehicle engines and particularly the benefits that rolling bearings offer for these applications are discussed below. The outlook for these components is demonstrated by taking a look at market trends.

Market trends

After many years of discussion the concept of downsizing engines to improve their efficiency has reached the market with more than just a few striking examples. This trend can also be observed when investigating the overall figures of the worldwide market and, especially, the shift between the various engine concept shares within the market. Even though vehicle weight has not decreased significantly on the European market, a shift can be seen in favor of smaller displacements and a smaller number of cylinders.

After 2008 and 2009, when worldwide engine production dropped by 5 % and even a dramatic 17 % respectively, the total market is expected to recover quickly at rates of approximately three times 10 %. The 4-cylinder in-line engines

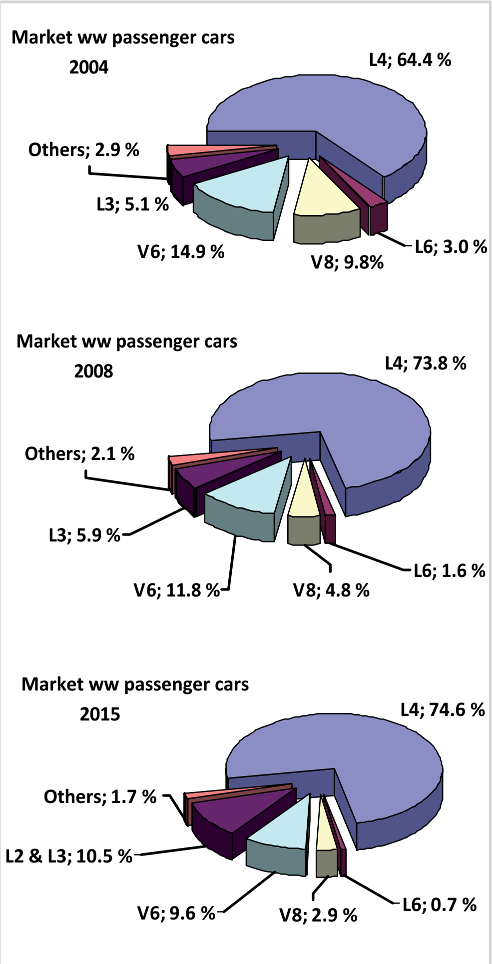

Figure 5 Market shares of various engine designs in 2004, 2008 and 2015 (L: in-line engine, V: V engine)

were not affected as much by the sharp decline on the global market in these years, so that their market share increased significantly. In contrast, 6 and 8-cylinder engines decreased considerably above average by 20 % and more than 30 %, and a recovery cannot compensate for this plunge, neither in absolute numbers nor with regard to market share. For 2012 to 2015 they are expected to reach no more than about 70 % of the 2007 production output. The outlook is entirely different for 2 and 3-cylinder engines. They will remain unaffected by the general market downturn and will increase annually by 10 to almost 30 % from 2010 to 2013, doubling their market share to more than 10 % by 2015.

Market shares for 4-cylinder engines increased from 65 % in 2004 to around 75 % in 2008, a value that is expected to remain stable until 2015. The remaining quarter is expected to experience a continuous shift from 6 and 8-cylinder engines to 2 and 3-cylinder engines. The number of 6 and 8-cylinder engines replaced by 4-cylinder engines corresponds fairly accurately to the volume of 2 and 3-cylinder engines that have replaced 4-cylinder engines.

Advances in engineering obviously support the development reflected in the market figures. Displacement-specific performance is expected to increase permanently from 44.9 kW per liter in 2004 to approximately 53.8 kW per liter in 2015. The average displacement, however, was down to 2.1 liters in 2008 from 2.4 liters in 2004 and is expected to stay fairly constant at nearly 2 liters until 2015. Taken together, both effects ensure that the global average engine performance will remain at around 105 kW, with the exception of the economic crisis from 2008 to 2010. As a result of the downturn in the production of large-volume engines, this inevitably is only possible with an increase of average performance in the 4-cylinder segment from 80 kW in 2004 to 92 kW in 2015.

This performance increase is a requirement when using this engine concept in higher-quality vehicles, allowing the driving dynamics expectations in this segment to be met and simultaneously reducing vehicle-specific consumption by using engine operating points with higher specific loads and thus better efficiency.

Market forecast for mass balancing systems

End customers who used to drive vehicles with 6-cylinder or even 8-cylinder engines will increasingly be offered 4-cylinder drives. In this process, the demands this group of customers places on engine vibration and noise behavior are not likely to decrease. For 4-cylinder engines, this means a significant increase in requirements for noise and vibration behavior.

The shift in market shares from 4-cylinder to 3-cylinder engines, on the other hand, is taking place in a market segment in which customer expectations have not necessarily made a balancing system for 4-cylinder engines appear necessary. Altogether, this makes the forecast of considerable market growth for balance shafts in four cylinder engines nothing but a logical consequence. This is obvious from the market shift shown in Figure 4 and is additionally supported by increasing rather than decreasing requirements for engine vibration and noise behavior from the end consumer's point of view, which is true for the classic large markets in Japan, the U.S. and Europe.

Mass balancing for 4-cylinder engines

Technical goals and mechanical principle

Inherant to its design, the piston engine has somewhat irregular component motion. This is particularly the case for its conrods and pistons, which results in free inertial forces and/or moments in most engine applications. For 4-cylinder engines, in particular, it is the second-order inertial forces that are not balanced within the engine, unlike other lower-frequency contributions. Unless adequate countermeasures are taken inside the engine, increasingly with engine speed these inertial forces will tend to cause vibrations in the entire engine structure. In 4-cylinder engines, these vibrations occur in a vertical direction, i.e. in the direction of the cylinder axis. Without countermeasures inside the engine, secondary measures must be taken to prevent vibrations from being transmitted to the vehicle structure and from being acoustically perceptible by the driver and passengers. Besides these effects, which are purely subjectively perceived as annoying, the increased engine vibration levels often make it necessary to reinforce the structures to mount adjacent structures, such as accessories. Overall, this may even be a worse solution with regard to weight and engineering effort than using balance shafts to counteract the vibration excitations in the engine.

In many positions, including the engine, damping is used to reduce vibration effects to an acceptable level, irrespective of whether these vibrations are perceptible as unpleasant vibrations of the chassis or interior surfaces, as acoustic disturbances or increased component loads in individual positions of the entire vehicle system. The principle of such damping measures is based on dissipation, resulting in energy losses. In contrast to this principle, the balancing of the vibration exciting forces by means of balance shafts, similar to mechanical compensation systems, represents a measure that does not have a negative effect on the system's energy balance. Only the friction in the balance shaft drive and the friction in the bearing positions may still cause energy losses.

Design

When balancing second-order inertial forces in the direction of the cylinder axis, a system consisting of two eccentric shafts running in opposite directions can be considered a standard. These balance shafts necessarily rotate with double the crankshaft speed and are synchronized to allow the horizontal forces to cancel each other at any time while the vertical forces, i.e. the forces applied in the direction of the cylinder axis, add up. Synchronization in relation to crankshaft rotation ensures that the second-order free inertial forces of the crankshaft drive are balanced.

The design includes two shafts with at least two bearing supports each operating at double the crankshaft speed und transmitting considerable forces to the cylinder crankcase. Consequently, seeing this in the context of efforts to reduce fuel consumption of the engine one easily notices that there are significant unavoidable losses even with friction-optimized plain bearing designs. As measurements of various applications have shown, however, these losses can be reduced considerably by using rolling bearings. Moreover, needle roller or cylindrical roller bearings offer significant additional potential for optimization with respect to mass and mass moment of inertia of the system, which is described below.

Design types

When putting mass balancing designs into practice in 4-cylinder engines, there are two very different design types. What may be considered the classic design type represents a combination of both balance shafts in a module, i.e. in a separate housing usually installed in the oil pan. Over the past few years, however, integrated solutions that involve installing long and slender balance shafts in the cylinder crankcase, have been increasingly used. This type of design has a number of benefits in terms of crankcase rigidity, engine weight and costs [4]. These benefits appear quite logical when considering that the modules arranged in the oil sump require a separate housing, which often is also required to evacuate the balance shafts against the oil level in the sump. The balance shaft housing when using the classic design, i.e. plain bearing supports, must be connected to the engine pressure oil supply to ensure the functional operation of the plain bearings. All this requires considerable development, manufacturing and assembly efforts.

Development of balance shafts with rolling bearing supports

Special care is required when designing rolling bearings for balance shafts, particularly in 4-cylinder engine designs, due to high load and speed requirements and an environment characterized by design spaces restrictions and the permanent pressure to optimize with respect to function and costs. The initial design of such rolling bearing supports is performed using classic methods. In a next step additional analytical methods are used to take the effects of shaft deformation on fatigue life into consideration [5]. Even at this stage, the development is a non-trivial optimization task that involves minimizing bearing friction by varying the bearing diameter as well as the number, diameter and length of the rolling elements while also maximizing fatigue life with the same variation parameters. Here, shaft tilting definitely plays an important role due to the fact that designs incorporating line contact are the ones primarily used. This requires the balance shaft shape to be included in the optimization strategy wherever possible; although the requirements upon their function impose additional optimization criteria (i.e. amount of unbalance and functional location of unbalance forces possibly in combination with geometrical

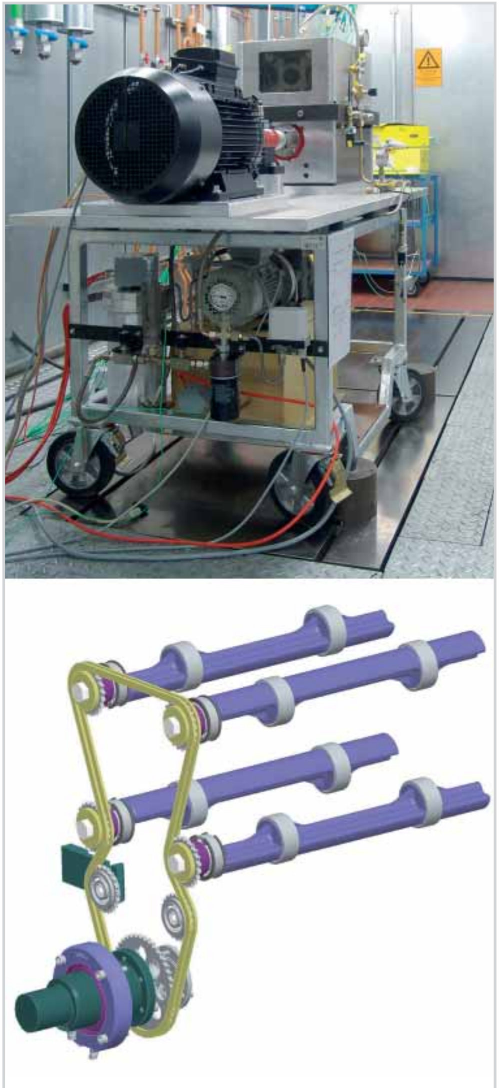

Figure 6 Test stand design and test stand with drive, oil supply and measuring system

restrictions from the relative motion of the cranktrain components).

Additional impacts on bearing life include oil quality with respect to viscosity, contamination and additives as well as the actual peak temperature reached during operation. These specific requirements are supplemented by installation designs often specified due to the engine structure and assembly concept and often require special bearing or cage design types: The balance shafts discussed above, which call for relatively long, thin shafts integrated in the cylinder crankcase, must almost always be mounted in an axial direction.

For cost reasons, experimental verification of bearing designs with regard to a number of design goals is often initially performed on a component test stand. To facilitate the operation of an adequate test rig, it should be configured in such a way that it includes the crankshaft drive with the piston group and the balance shaft drive to avoid unbalanced mass forces. Alternatively, a system can be configured with two housings with two balance shafts each that can then be synchronized to balance all the inertial forces within the system (Figure 6).

Verification targets of testrigs as the ones mentioned include the following:

- Speed suitability of the bearing design
- Bearing fatigue life affected by realistic shaft and housing deformation
- Temperature conditions at high speeds under nearly realistic lubrication conditions
- Investigations of the impact of oil contamination

If a reasonable technical design of a plain bearing variant is available, a direct comparison with this bearing design, in terms of the required driving torque and achievable friction reduction can also be made. The advantages of rolling bearing supports are generally in a range of approx. 50 %. Depending on the design, absolute values for an operating temperature of approx. 100 °C in the upper speed range can be up to 1.5 kW of friction reduction (Figure 7).

Pressure oil supply directly into the bearing leads to friction values that are in the same range as plain bearings. Therefore, lubricant supply exclusively using oil mist is essential when aiming at friction reduction. Oil mist supply has proven to be sufficient in many cases, especially since reduced bearing friction results in reduced heat generation within the bearing. Often, heat conduction into the ambient components is sufficient to limit peak temperatures in the bearing to an acceptable level.

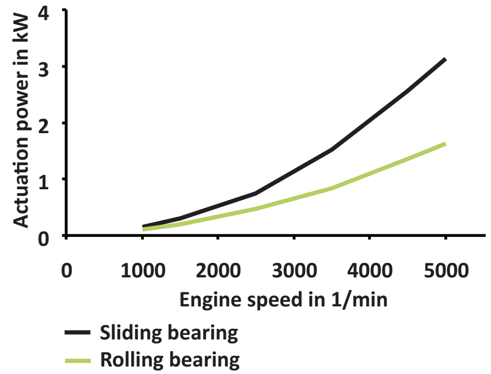

Figure 7 Comparison of driving torques for balance shafts with rolling bearing supports and plain bearing supports in a 4-cylinder engine

Eliminating the necessity of pressure oil supply reduces the effort in design and manufacturing of the balance shaft housing to a considerable level. Moreover, the elimination of the pressure oil volume flow that would otherwise be required represents a considerable reduction of parasitic losses in the engine. Finally, reducing friction in the bearing positions also reduces the amount of heat transferred into the engine oil so that oil coolant requirements also decrease. Overall, various automobile manufacturers have agreed that these effects result in fuel savings in the range of 1 to 2 % [6].

Additional potential for optimization

The elimination of pressure oil supply described above in combination with a direct raceway on the shaft permits additional optimization that can be justified as follows.

Since the force action of balance shafts is generated by their unbalance, the force is circumferential with the shaft, i.e. its effective direction with respect to the shaft geometry does not change. For the rolling bearing this means that only the rolling elements that in a given situation are on the side of the unbalance mass will transmit forces from the shaft to the housing (Figure 8). The rolling bearings in the area opposite to the unbalance mass do not transfer forces between shaft and outer raceway. They remain in contact with the outer race due to their centrifugal forces, if the shaft rotates fast enough. This actually eliminates the need for a shaft raceway in that section. A force supporting function for this area may only be necessary when the engine is turned off and incidentally stops in a rotational position with the unbalance mass directing upwards. However, in this case the forces generated are so low that a significantly reduced raceway width suffices. In many applications, maintaining a certain raceway width is also advisable for acoustic and rolling bearing technology reasons.

Bearing width reduction on one side removes mass in the area opposite to the unbalance, thus increasing overall shaft unbalance. Therefore, mass is also removed on the unbalance side to restore the nominal unbalance. This compensation, however, is done in a position as close to the axis of rotation as possible resulting in the amount of mass removed on this side being considerably higher than the mass remove on the opposite side. Overall, this results in significant potential for mass reductions compared to the original design. In typical 4-cylinder engine applications, this amounts to 20 to 40 % of the original mass and up to 1 kg of overall weight.

The balance shaft mass reduction inherently goes along with a reduction of the mass moment of inertia. As a consequence, the driving torque for acceleration and deceleration is lower in the relevant partial drive train which, on the one hand, facilitates strength design and, on the other hand, dynamically leads to reduced acoustic excitations.

It is worthwhile pointing out that one of the essential advantages of this principle lies in the fact that oil mist from the engine compartment can now easily access the rolling bearings due to the partial elimination of the raceway, ensuring sufficient lubricant supply if the oil mist density is adequate. A good example for implementing the ideas discussed here is the OM651 Daimler 4-cylinder diesel engine.

System optimization

Unless specific design measures are taken, rolling bearings – similar to plain bearings – can only withstand a very limited amount of tilting without suffering rating life losses. This makes system design a highly complex optimization task that can include opposite tendencies and boundary conditions that must be met:

- The shape of the balance shafts must be optimized in terms of its bending behavior during operation to minimize bearing tilting. In the process, the required unbalance und

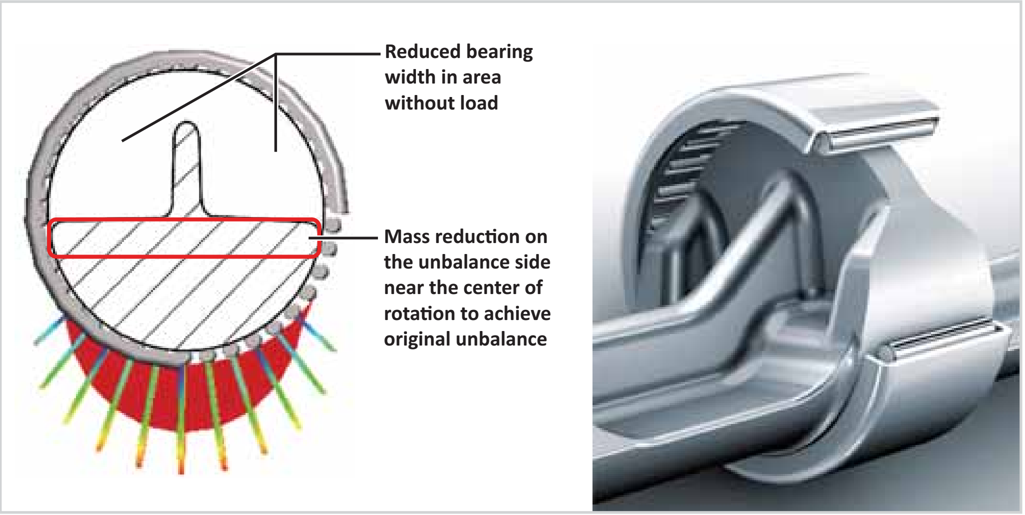

Figure 8 Optimized balance shaft bearing design

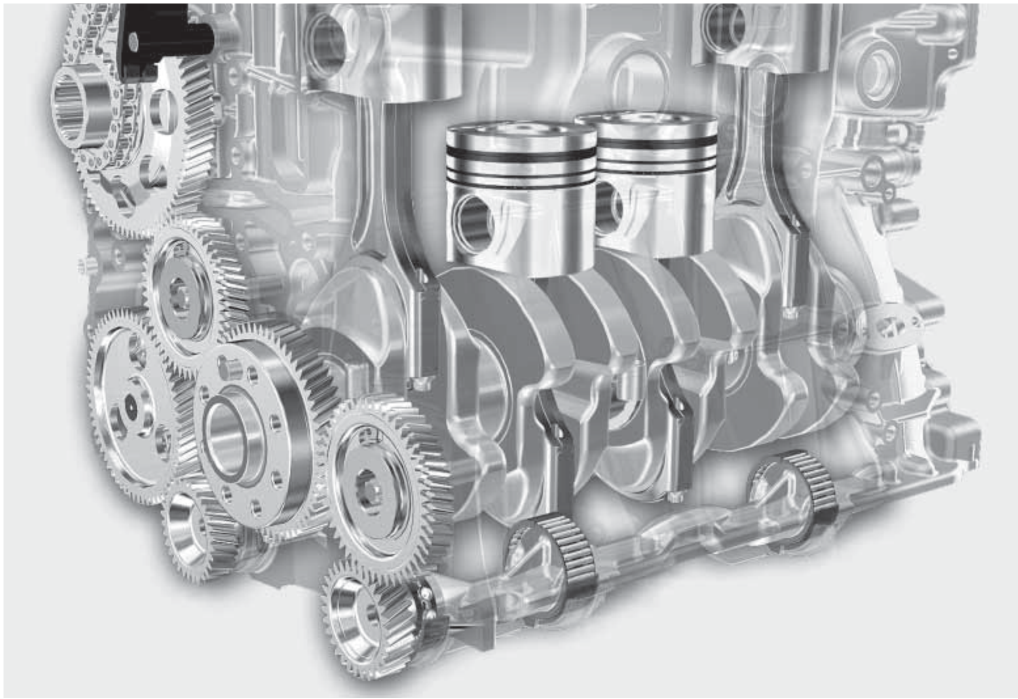

Figure 9 Optimized balance shafts in the Daimler OM651 engine

the effective position of the unbalance forces in the longitudinal direction of the engine (engine center for 4-cylinder engines) must be maintained.

- The potential mentioned above for optimized shaft design in the bearing area should be utilized as much as possible.
  - The effective unbalance radius must be as large as possible to minimize mass.
  - By contrast, reducing the driving torque in acceleration/deceleration and acoustic excitations requires an effective unbalance radius that is as small as possible.
- Loss must be minimized by using adequate bearing design (raceway diameter, rolling element diameter and bearing width); load distribution and bearing tilting also play an important role
- At the same time, the necessary bearing life must be ensured using the same parameters.
  - Larger raceway diameters increase rating life as well as bearing friction torque.
  - Larger bearing width leads to both increased bearing rating life and higher friction torque. However, the unfavorable impact of bearing tilting is reinforced and the design space remaining to generate the required unbalance is reduced.
  - Larger rolling element diameters increase bearing rating life but also require more radial design space.

Finally, specified assembly concepts must be considered that further limit design options. Depending on the focus of all requirements, optimal results may vary significantly. In some cases, reducing the shaft raceway diameter to a value smaller than that of the unbalance envelope dimension has proven to be a good idea. For axial mounting, this would then require a bearing outer ring that is purely cylindrical and without boards. Such boards are not required with this design since the axial guidance of the rolling bearing cage is ensured by contact surfaces on the shaft. These contact surfaces, however, are interrupted in a circumferential direction so that the design of this type of system requires great care when coordinating the cage design and bending behavior as well as the load distribution of the shaft.

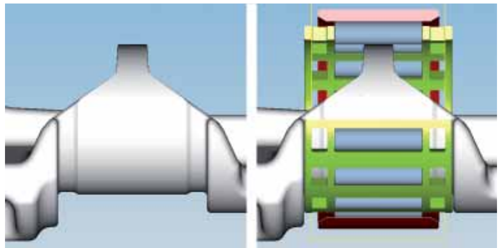

Figure 10 Possible design of a needle roller bearing for friction optimization

This is only possible as part of a very thorough overall system optimization, as shown above.

Development environment

Schaeffler has coordinated and combined the steps to be taken in this type of optimization in a self-contained toolbox. The following steps all build up on one another, from the initial idea to a sophisticated balance shaft and bearing layout, and have been designed with the greatest possible consistency.

- The application specific requirements in terms of unbalance and bearing positions easily lead to a basic idea of bearing sizes necessary.
- A preliminary shaft design study is prepared using the same customer specifications such as unbalance, unbalance effective position and drive concept. In this process, it has been helpful to learn that, for an initial approximation, the cross sections of balance shafts can be assembled from a very limited number of geometric primitives. The geometric design can be described using an easy-to-edit Excel chart that evaluates unbalance, unbalance effects, mass and mass moment of inertia.
- The Excel template includes an interface to Schaeffler's in-house rolling bearing design software, Bearinx, which is used to perform the following calculations:
  - The balance shaft's deflection is determined based on its approximate flexural rigidity while taking unbalance-related distributed loads, which are also defined by geometry, into consideration.
  - Calculating the shaft deflection not only provides an initial approximation of load distribution across bearing positions. Since models of designed rolling bearings can already be integrated in this phase a detailed rolling bearing calculation including a rating life calculation with consideration of tilting effects is performed automatically without requiring additional user input. Moreover, the effect of bearing rigidity on balance shaft deformation is already included in the results.
  - The automated interface between the list of the geometric shaft description and the shaft and rolling bearing calculation program allows simple, quick and thus highly efficient optimization with respect to the parameters stated: rating life, tilting, mass, mass moment of inertia and design space.
- As soon as a satisfactory design suggestion has been achieved, a CAD interface is used to transfer the geometry from a list of geometric drafts to a CAD design concept so that a design engineer can further process the geometry.
- The CAD system has an integrated balance shaft development environment that provides significant support to designers.
  - It contains an automated evaluation of unbalance, unbalance distribution, mass and mass moment of inertia.
  - The entire record can be transferred via another interface to the rolling bearing calculation software at any time to be able to review and correct the shaft and bearing designs during this phase.
  - The balance shaft development environment also includes a standardized finite element analysis (FEA) to allow an exact calculation under varying speed and load conditions during the end phase of the design process.
  - One of the tasks in this phase is geometric optimization in terms of manufacturing processes to maintain the required geometric tolerances and to ensure a cost-efficient shaft production process.

Summary

Due to their low friction, robustness and reliability, rolling bearing supports are well established as the standard design in piston engine valve trains and continue to conquer additional applications. Their specific characteristics, such as running capability at lubrication starvation, minimal losses and load capacity even at low speeds and design flexibility, make them particularly suitable for the various requirements of future applications. In addition to camshafts and balance shafts, examples for such applications include gear drives and chain wheels where, due to low bearing internal clearance, double-row angular contact ball bearings may even have benefits in terms of acoustics compared to plain bearings.

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