20

The missing link

From a simple component to a sophisticated chain drive system

Bolko Schuseil
Steffen Lehmann
Stefan Wehmeyer, IFT Clausthal Zellerfeld
Christopher Lehne, IFT Clausthal Zellerfeld

Introduction

Over 90 years ago, chains were for the first time used for linking the crankshaft and camshaft of internal combustion engines. The bush chains used in this application were based on a patent of Swiss inventor Hans Renold from 1880. Since then, the chain drive has become a highly complex system the development of which requires an in-depth understanding of systems due to its interaction with other engine components. Starting from the component "hydraulic chain tensioner", the Schaeffler Group has taken on this challenge with state-of-the-art development methods. It offers customers not only all components of a chain drive system but also the relevant systems engineering.

Figure 1 First hydraulic chain tensioner

Components and function

Chain tensioners

INA developed the first chain tensioners based on a customer request in 1982. These elements were introduced to the market in a six-cylinder engine in 1984.

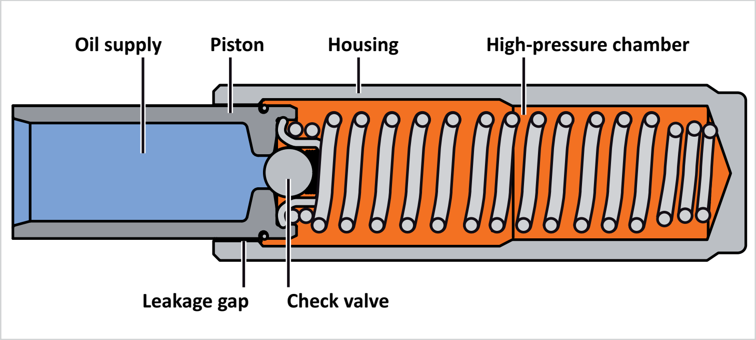


Figure 2 Structure of a hydraulic chain tensioner

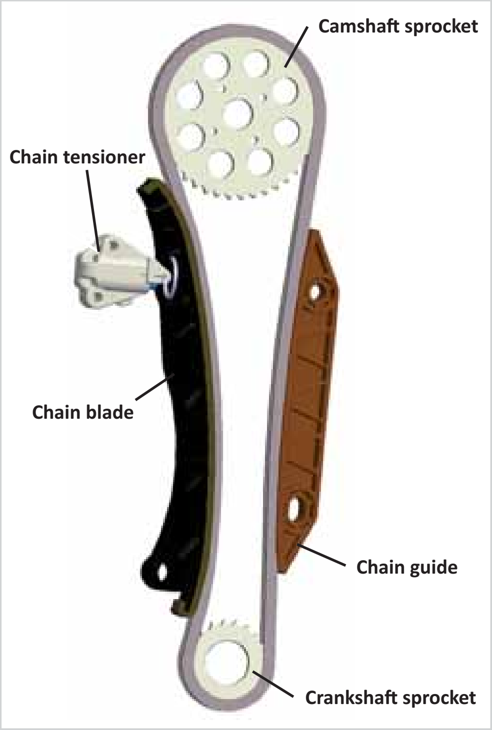


Figure 3 Structure of a chain drive system

The structure and function of the tensioning elements were derived from hydraulic valve lash adjustment elements (HVA). They work according to the principle of leakage gap damping that acts proportional to the speed.

A piston that can be moved longitudinally is positioned in a cylindrical housing so that a high-pressure chamber (red) is created that is linked with the engine oil circuit (blue) via a check valve. The extremely narrow gap (leakage gap) between the piston and the housing seals the high-pressure chamber almost completely against external influences. The rotational irregularity of the crankshaft and/or camshaft leads to chain oscillations that apply dynamic loads to the tensioner. The oil is forced out of the high-pressure chamber through the leakage gap, and oscillation is damped dependent on its speed and the size of the leakage gap.

Chain tensioner elements have been developed further to meet the requirements of specific engines. The described functional principle, however, has remained unchanged to this day. Tensioners are usually located on the slack side, i.e. the driven side, of a chain drive system as shown in Figure 3.

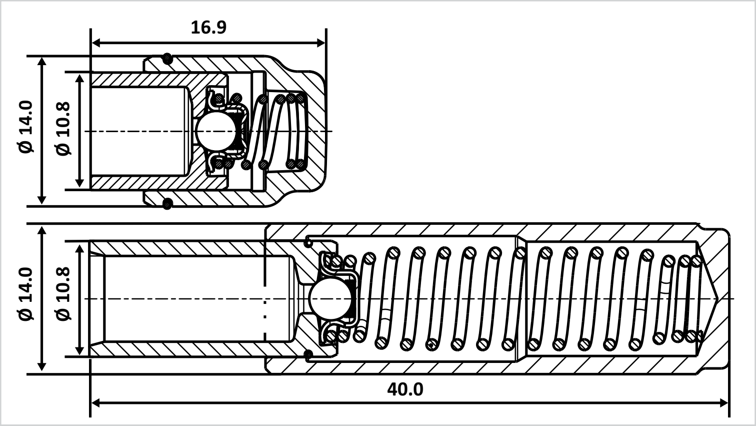


Figure 4 Comparison of the size of a chain tensioner and hydraulic valve lash adjustment element

Figure 4 compares the size of a hydraulic valve lash adjustment element and chain tensioning element. The significantly larger stroke of the tensioning element is required for compensating chain wear over the engine life and the tolerances of the chain drive system. The two elements also differ in terms of the size of the leakage gap and thus their rigidity.

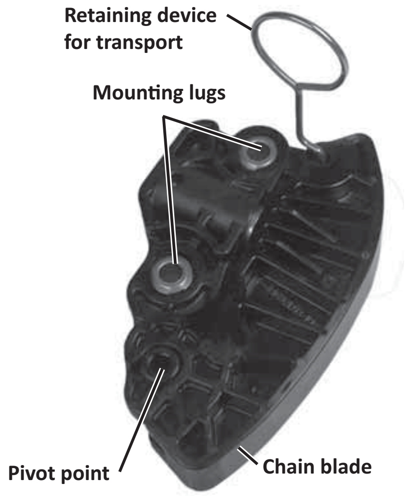


Figure 5 All-plastic tensioning element

The customer's design envelope requirements and the dynamic behavior of the chain drive system that is measured or calculated during development form the basis for the design of tensioning elements. Valid standards and the objective of ensuring the required function at lowest possible manufacturing costs are taken into account early in the design phase. The tensioners shown in the two figures below were developed for current V8 engines. Figure 5 shows an all-plastic element.

In another engine, high dynamic demands required a concept with a pressure control valve. This valve is integrated in the high-pressure chamber of the tensioner (Figure 6, right ball) and serves to compensate high force peaks.

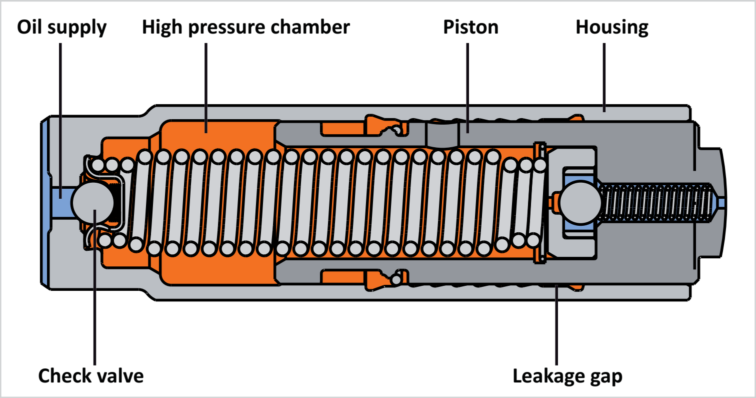


Figure 6 Hydraulic tensioner with a pressure control valve

Chain guides and blades

In contrast to belts that have unguided sections, timing chains are always supported by guides and blades in the free strands. These elements prevent skewing of the chain when it enters the sprockets, but also chain oscillations and therefore high loads on the chain joints. The friction on these guides and blades can be minimized by using suitable materials and geometries.

Cast metal elements have been largely replaced by injection molded parts made of polyamides for cost and weight reasons. A large variety of base materials, which often contain reinforcing fibers, are used in these elements. The direct contact surface to the chain, however, is usually not reinforced. Plastic supports have an improved internal damping behavior compared to sheet metal supports and therefore reduce the loads acting on the chain.

In contrast to metal supports, plastic supports must be designed in increasingly large dimensions particularly in the direction of the force application. This is due to the high temperature dependency of the strength of polyamides. However, small engine design envelopes often require compromises that would not be possible without knowledge about the interactions between the systems. Figure 8 shows a plastic chain blade in direct comparison with a steel chain blade. A suitable design can achieve both a weight and price ratio of a factor of 1:5.

Sprockets

Chain sprockets link the chain to the crankshaft and the camshaft by geometrical locking. Their design, especially the connection with the shafts, is determined by the technical requirements of the specific application. The tooth geometry of roller and bush chains is generally based on international standards. Toothed chains, which have been developed with the objective of reducing noise and friction, can have very different tooth profiles. These profiles are often specially designed for a chain and therefore not interchangeable, even if different chains have an identical pitch. This must be taken into account, especially when using toothed chains in engines with variable camshaft phasing units, as the teeth are often an inherent part of the phasing unit and therefore part of expensive tools. Figure 9 shows a selection of sprockets from current production. In addition to flat or dished fine-blanked sprockets, there are sprockets made of sintered materials and machined sprockets.

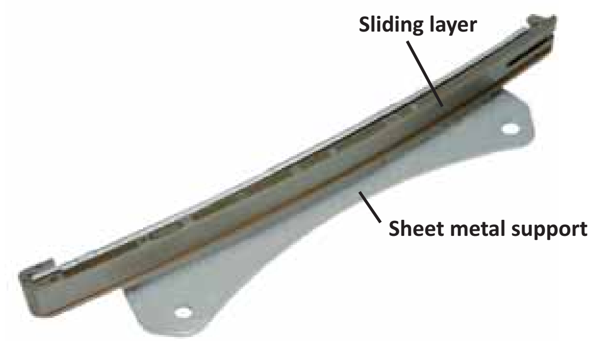


Figure 7 Chain guide with a formed sheet metal part as support

Figure 8 Chain blades in different designs

Figure 9 Chain sprockets in different designs

Figure 10 Chain sprocket with rubber lining on the flank

The bush chain sprocket in Figure 10 is a special variant with a vulcanized rubber lining on the flank. The rubber lining minimizes the noise generated by the chain when it enters the sprocket as the link plates run on the rubber shoulder before the bush comes into contact with the tooth root.

Chain

INA worked as a systems developer for chain drive systems and as a component supplier for chain tensioners for almost 20 years until fall 2005. INA supplied systems in cooperation with external partners, who were often also competitors. Nearly at the same time as the development of a toothed transmission chain by LuK, the INA TechCenter in Troy, USA started to design a toothed chain for primary drives in 2005. This 3/8” toothed chain went into volume production in a V8 engine in the USA in spring 2007. INA acquired the chain manufacturing plant of Renold in Calais, France in fall 2006 and has since then been able to offer its own comprehensive range of chains. This marked INA’s leap from a systems developer to a systems manufacturer.

INA transferred its manufacturing technologies to the chain production processes. For example, all chain pins are manufactured using high-precision grinding processes from needle roller production. The acquired product range could thus be significantly improved. Figure 11 shows the improvement in the wear behavior of 8 mm bush chains.

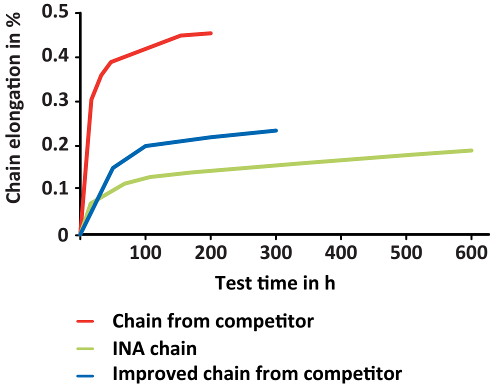


Figure 11 Improvement in wear behavior by using optimized manufacturing technology

Comparison ⅜" toothed chain	INA	Competitor	Difference/%
Width/mm	10.5	13.2	-20
Weight/kg/m	0.5	0.67	-26

Figure 12 Comparison of an INA chain and a chain from a competitor

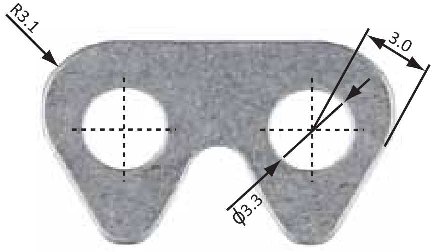


Figure 13 Wall thickness ratios of a chain link

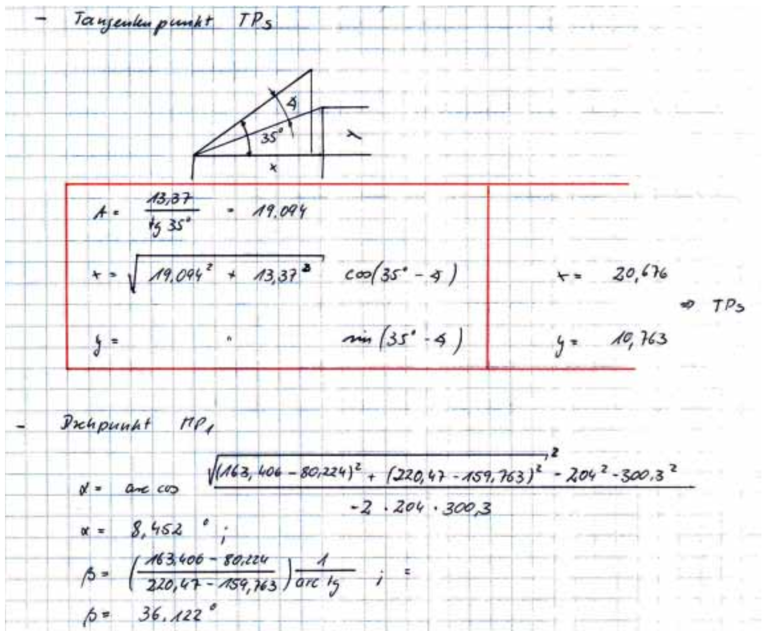
The INA toothed chain has fine-blanked functional links. Figure 12 compares the design of a 3/8" toothed chain with a current product from a competitor.

The table shows that the INA toothed chain is characterized by a very narrow design. This was achieved by using a set of only five link plates of different thicknesses. The intermediate plate is pressed onto the pin. Figure 12 shows the high clean cut proportion on the functional surfaces in comparison with the competitor's chain. The resulting excellent joint surface is supported by the press fit of the intermediate link which reduces deflection of the pin under load. An extremely low chain wear was achieved with these measures. None of the engines that are currently in volume production or in the testing phase showed wear elongations larger than 0.15 % over long running times. This is significantly below the values achieved by the competitors' chains. The wall thicknesses are larger than the usual limits (Figure 13) and can only be reliably achieved through comprehensive tool optimization.

All INA toothed chains have an identical basic design. Adaptations for different future market sectors (e.g. motorcycle applications) are being developed.

System

Figure 14 shows a copy of a tensioner stroke calculation from 1987.



- Tangentenpunkt TPs

$$x = \frac{13{,}37}{tg\,35°} = 19{,}094$$

$$x = \sqrt{19{,}094^2 + 13{,}37^2}\;\cos(35° - φ) \qquad x = 20{,}676$$

$$y = \quad " \quad \sin(35° - φ) \qquad y = 10{,}763$$

⇒ TPs

- Drehpunkt MP1

$$α = \arccos \frac{\sqrt{(163{,}406 - 80{,}224)^2 + (220{,}47 - 159{,}763)^2}^{\,2} - 204^2 - 300{,}3^2}{-2 \cdot 204 \cdot 300{,}3}$$

$$α = 8{,}452° \;;$$

$$β = \left(\frac{163{,}406 - 80{,}224}{220{,}47 - 159{,}763}\right) \frac{1}{arc\,tg} \;; \;=$$

$$β = 36{,}122°$$

Figure 14 Tensioning element calculation from 1987

Back then, calculating the stroke of the chain tensioner and the curve of the catenary using a pocket calculator and trigonometric formulas was a time-consuming task. This basic design activity was further developed using the first simple CAE calculation programs and has meanwhile become a complex systems analysis where the calculation of the tensioner stroke requires comparably little outlay. Often, a chain drive system can only be designed by taking into account the function and influences of adjacent systems/components such as balancer shafts (AGW), variable camshaft phasing units or valve train systems. The integration of stop-start functionalities in engine operation and the requirement for further reductions in friction will place new demands on chain drive systems. INA can draw on the experience of the different sectors of its Engine Systems division in solving these tasks, which is regarded as a significant competitive advantage by the customer.

Figure 15 Chain sprocket with integrated vibration damper

An example of the described system influence is the chain sprocket from a balancer shaft drive of a four-cylinder engine (Figure 15).

The 7 mm chain used in the turbocharged version of an engine was overloaded due to increased irregularities of the balancer shafts compared to the normally-aspirated version. Durability of the chain drive was achieved by integrating a damper in the crankshaft gear and thus significantly reducing the load peaks in the drive. The diagram in Figure 16 compares the torsional vibrations measured on the balancer shaft with and without an integrated damper in the gear.

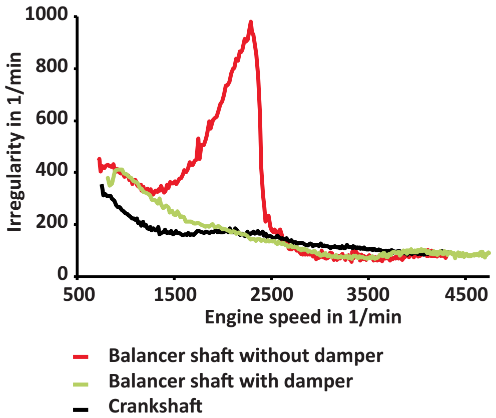


Figure 16 Influence of a torsional vibration damper in the balancer shaft drive

Similar effects were achieved for the balancer shaft of a three-cylinder diesel engine with a tensioner-guide application called "Motion Guide" (Figure 17).

Figure 17 "Motion Guide" tensioning element for the balancer shaft

In this case, the systems simulation recommended the use of a reinforced chain, which moved the resonance speed range towards higher speeds (Figure 18). The diagram shows the envelopes (maximum/minimum values) and the mean values of the chain forces over the speed range as a function of the chain used.

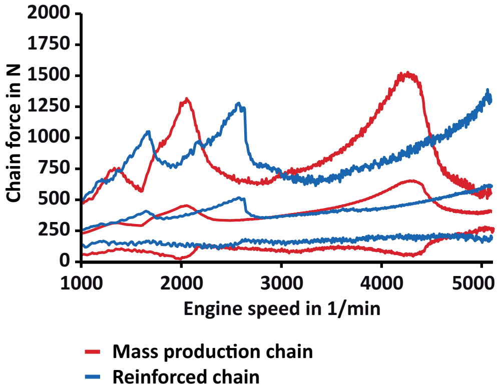


Figure 18 Influence of the chain rigidity on the drive resonance

Some engines today have out-of-round chain sprockets for reducing the effects of torsional vibration on the chain drive. Here, the irregularity caused by the engine is compensated by an out-of-round chain sprocket. In an ideal case, this reduces the system loads by up to 20 %. The design of such a sprocket (Figure 19 shows a tri-hexagonal sprocket for a three-cylinder engine in comparison with a round sprocket) depends on the dominant engine orders and requires a defined wrap angle on the chain sprocket. The design is always matched to a critical speed range.

Compared to round sprockets, this can sometimes lead to increased loads on the drive in defined speed ranges, especially in applications for variable camshaft phasing systems. Here, simulations provide comprehensive and useful information for the extremely complex tests.

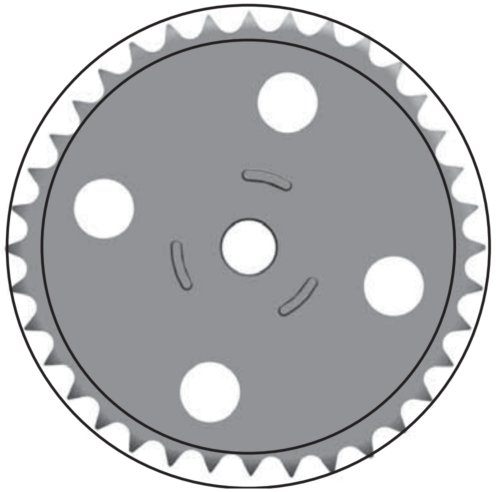


Figure 19 Out-of-round chain sprocket for a three-cylinder engine with round envelopes

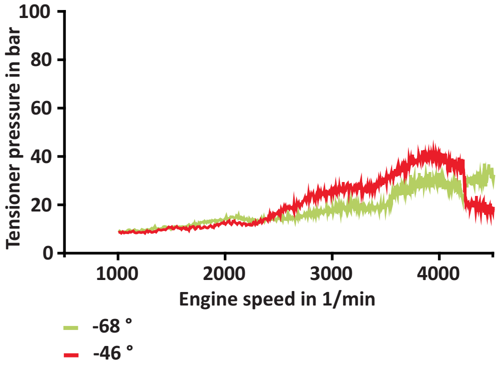


Figure 20 Influence of the orientation of the diesel injection pump on the tensioner load

Another example of cross-system expertise in designing chain drive systems is the connection of diesel injection pumps with chain drives. The force development in the chain drive can be strongly influenced by changing the angular position of the pump. Here, the drive torque of the pump superimposes the drive torque of the camshaft. Detailed knowledge of the pump and its application on the engine is therefore useful for optimizing the chain drive. Figure 20 shows the corresponding curves of the pressure in the tensioner's high-pressure chamber and thus the chain load.

Extremely high chain forces that occurred in a four-cylinder high-speed engine at speeds over 10 000 1/min could only be explained after investigating in more detail the backlash of the intermediate gear used between the crankshaft and the primary chain sprocket. The results can be taken from Figure 21, which shows the maximum chain forces during startup as a function of the tooth backlash.

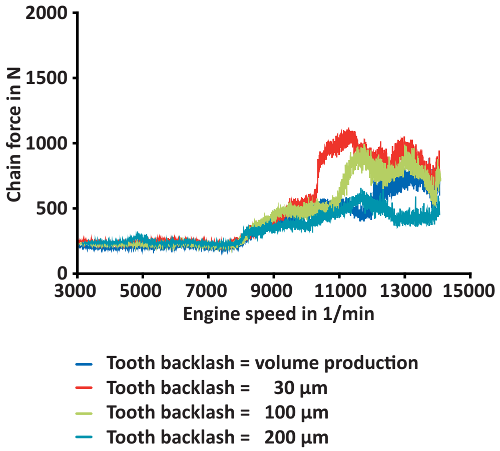


Figure 21 Influence of the tooth backlash of an intermediate gear on the chain forces

The diagram shows that an increased tooth backlash significantly reduces the chain forces in the primary drive. Thus, the intermediate gear has the effect of a vibration damper.

Many of the above described developments were tested with measurement technology that was specially developed for chain drive dynamics. During almost 20 years, INA has gained expertise in this field thus often being one step ahead of the state of the art. INA also uses an increasing number of simulations that can significantly shorten variant testing if the models have been calibrated with tests. Figure 22 shows an example of the high correlation between the test results and the calculations for the balancer shaft damper described in Figure 16.

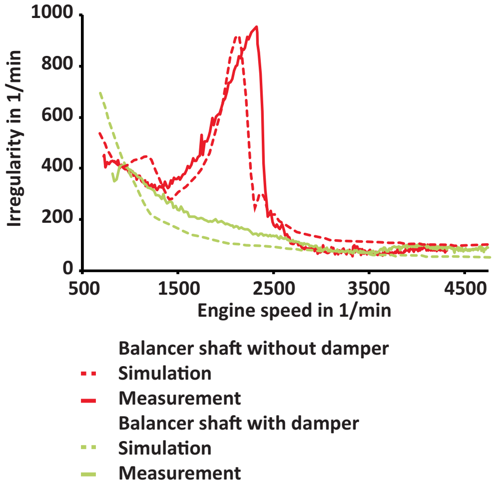


Figure 22 Correlation between the measurement results and calculations for the balancer shaft system in Figure 16

INA now uses simulation tools that can combine the systems valve train, variable camshaft phasing unit and chain drive, and, if required, balancer shaft damper or dual mass flywheel. Precise input data of the affected systems is required to achieve reliable results in the simulations. This is another field where INA Engine Systems can draw on the wide range of expertise of its sectors and make use of synergies.

Summary

Starting with the replica of a shock absorber 25 years ago, INA has developed a wide range of products that covers all elements of the chain drive system. The design of components for chain tensioning elements has become a complex task that requires an in-depth understanding of systems. Cost-effective solutions can only be developed within the available timeframe and budget by using special simulation tools and measurement technology. Profound knowledge about the systems that interact with the chain drive, such as variable camshaft phasing unit, balancer shafts, valve train and injection pumps, is essential for an optimum design of the entire system. INA is optimally positioned for this task due to the close cooperation between the different sectors.