Measurement of the Crankshaft Seals Friction Losses in a Modern Passenger Car Diesel Engine

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Abstract

This paper presents results of experimental investigations on the friction losses of the crankshaft radial lip seals of a modern 4-cylinder diesel engine for passenger car applications. A two stage strip-test has been conducted on a motored engine test bed to obtain the friction torque of the radial lip seals. For the experimental investigations with the crankshaft seals removed from the engine, a special sealing apparatus has been designed and build. A wide range of tests have been performed covering the full speed range of the engine at lubricant temperatures of 70 °C, 90 °C and 110 °C. The results show a dependency on crankshaft speed and engine media supply temperature but also revealed the presence of constant plateaus of friction torque over engine speed.

1 Introduction

One of the crucial topics in engine development is to improve the engine efficiency to cut fuel consumption and subsequently reduce emissions. Very tight emission limits driven by the legislative and customers forces the automobile manufacturers to considerably lower the emissions of their vehicle fleet (95 g CO_2/km by 2021 (phased in from 2020)). For a diesel engine used in passenger cars this means a required fuel consumption of about 3.6 l/100 km [1]. Beside improvements of the thermodynamic efficiency (e.g. operating- and combustion-processes) and electrification, friction reduction represents a very efficient and economic possibility to improve fuel consumption. In addition to the largest contributors of the frictional losses in internal combustion engines (piston assembly, journal bearings and valvetrain) even small contributors like the radial lip seals need to be analysed and improved to reach the tight emission limits.

In automotive engines, the sealing of the crankshaft ends is generally ensured by radial lip seals which are installed at both ends of the engine crankshaft (flywheel side and FEAD[1] side). The main functions of the radial lip seals are to prevent that dust, dirt and water gets into the engine and to avoid engine oil leakage. Therefore, the crankshaft seals need to ensure static sealing between the outer side of the radial lip seal ring and the housing (e.g. crankcase) as well as dynamic and static sealing at the seal lip and shaft interface. These functions need to be guaranteed even under severe operating conditions like high oilsump temperatures above 130 °C, high additive low-viscosity engine oils, sooted and diluted engine oil and tumble motion in radial and axial direction of the crankshaft ends due to vibrations from the variying load of the reciprocating pistons. [2] However, modern radial lip seals for automotive applications are designed for lowest possible radial contact force to maximise durability and efficiency. The resulting radial force is caused by material expansion of the sealing body during the assembly and the spring

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[1] Front End Auxiliary Drive

force of the tension spring (so-called worm or garter spring) if used. In the automotive industry it is nowadays common to use radial lip seals without additional tension springs. For automotive applications, a common value for the radial contact force related to the shaft circumference is about 0.1 N/mm. [3] To ensure proper sealing capability with low radial contact forces during dynamic operation, the radial lip seals are manufactured with macroscopic recirculating structures (twist) to promote a hydrodynamic sealing assistance of these surface structures. The oil supply of the crankshaft radial lip seal rings is typically done solely due to splashed oil inside the crankcase. Nevertheless, in the contact zone between the surface of the rotating crankshaft and the seal lip, a lubricating and sealing oil film is formed. A lot of research has been performed in the past to understand and describe this sealing mechanism, which is a complex tribological interaction between radial lip seal, oil film and shaft. [4], [5], [6], [7], [8], [9] The turning of the crankshaft creates shear stress in the lubrication gap which generates frictional losses. Notably, only a minor number of publications discuss the friction losses caused by the radial shaft seals in modern engines. [10], [11], [12] To investigate the friction losses of the crankshaft radial lip seals a series of tests have been conducted. Therefore, measurements have been done with installed and with removed radial lip seals. To seal the engine crankcase during engine testing without the radial lip seals in place, a device has been designed and build which is able to prevent oil leakage from the engine without having contact with the crankshaft surface.

The friction power losses of the crankshaft radial lip seals were measured over the full rotational speed range of the engine. As the temperature of the lubricant has a considerable influence on the viscosity in the lubricated contacts and, consequently, on the tribological conditions [9], [12] in the seal lip-shaft interface, the engine was operated with different lubricant temperatures ranging from 70 °C to 110 °C.

2 Testing

The crankshaft seals put to test are the original radial shaft seals belonging to a modern in-line 4-cylinder, turbo charged, passenger car diesel engine with a nominal volume displacement of 2 litres and a nominal power of 135 kW; more technical details of the engine are listed in table 1.

Table 1: Technical data of the in-line 4-cylinder turbocharged engine under test:

Volume displacement	1995 cm^3
Compression ratio	16.5:1
Bore	84 mm
Stroke	90 mm
Nominal torque	380 Nm
Nominal Power	135 kW
Maximum Speed	4600 rpm
Cylinder diameter	91 mm
Conrod length	138 mm
Main bearing diameter	55 mm
Main bearing width	25 mm
Big-End bearing diameter	50 mm
Big-End bearing width	24 mm
Valve-train	DOHC
Timing drive	chain
Valve-train type	roller-type cam follower
Valves	4 per cylinder

Figure 1 shows the flywheel side and the FEAD side of the crankshaft with mounted radial lip seals whereas the basic geometry and elastomer material is listed in table 2 respectively.

The crankshaft seal ring on the FEAD side is shown in detail in figure 2. The seal ring construction on the flywheel side is identical.

The crankshaft seal construction consists of a metal housing with an o-ring seal on the outer face of the seal ring. To seal the crankcase on the crankshaft surface, the seal ring consists of two lips. The sealing lip is preventing oil leakage from the oil side (crankcase) and the dust lip is preventing that dirt and water gets inside the crankcase.

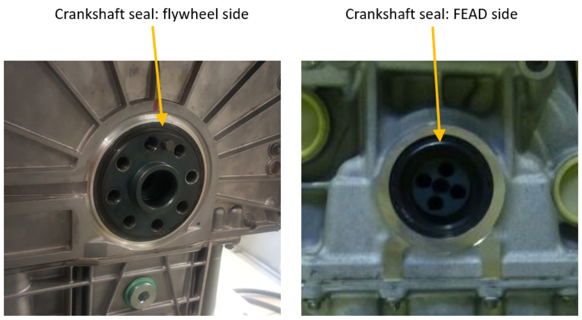

Figure 1: Mounted crankshaft radial lip seals

Table 2: Technical data of the crankshaft radial lip seals:

Mounting position	Flywheel side	FEAD side
Material	FPM[1]	FPM
Outer Diameter	104 mm	68 mm
Hight	9 mm	8 mm
Inner Diameter	90 mm	55 mm
Twist type	left-hand	right-hand

[1] Fluorinated rubber

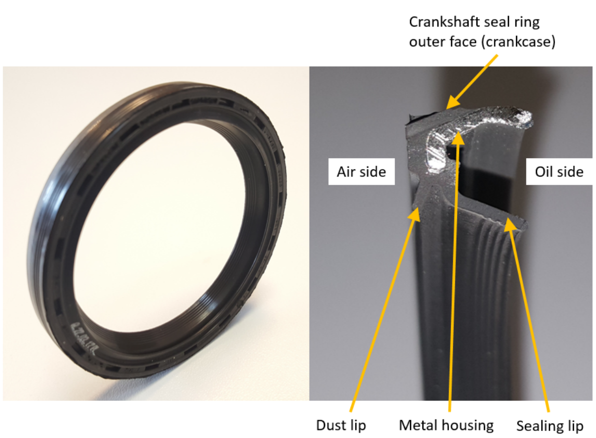

Figure 2: FEAD side crankshaft seal ring: Seal ring (left); Sectional view (right)

2.1 Engine preparation and set up

To separate the frictional losses of the radial lip seals from the other engine parts and components, all auxiliary devices like oil- and water-pump, alternator, balancing shafts, vacuum pump, etc. have been removed from the engine or are deactived. Also, conrods and pistons as well as the valve train timing drive (upper timing drive) have been removed. At the crankshaft, balancing weights have been installed on the crank pins to compensate the missing rotational mass forces of the conrods and to prevent excessive oil leakage from the oil supply bores. The lower timing drive was mounted during the experimental investigations but the high pressure fuel pump which is normally driven by the lower timing drive was deactivated. To deactivate the high pressure fuel pump, a ball bearing was mounted between the fuel pump drive shaft and chain sprocket.
To prevent oil leakage from the engine crankcase during the measurements without mounted crankshaft seals, a convenient sealing at both ends of the engine is necessary. On the FEAD side, the sealing of the crankcase can be realised by using a simple plug because the crankshaft shoulder is not touching the seal lip when unmounting the belt pulley (see figure 3).

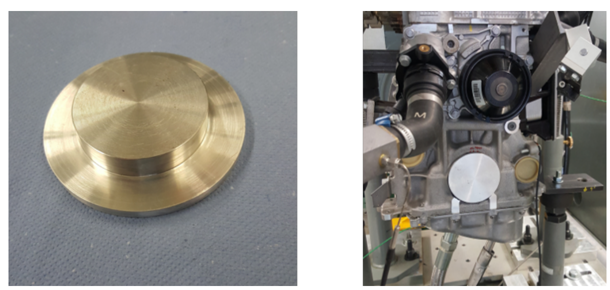

Figure 3: Sealing plug to prevent oil leakage on the FEAD side of the engine

Proper sealing of the engine on the flywheel side without mounted crankshaft seal demands a special apparatus to prevent oil leakage from the resulting sealing gap (see figure 4).

For this purpose, a custom sealing case has been designed and realised. The sealing case consists of

Figure 4: Flywheel side of the engine without installed crankshaft seal

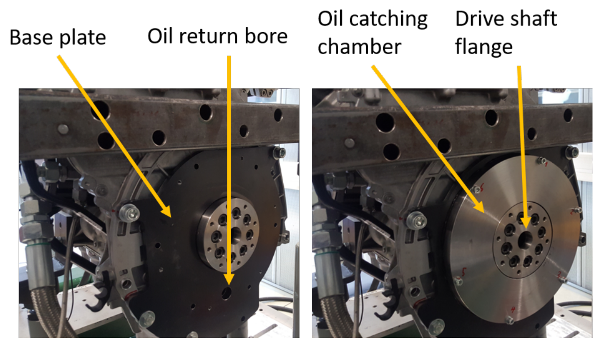

Figure 5: Realised crankshaft sealing system on the flywheel side

a base plate and an oil catching chamber. The base plate is mounted on the engine crankcase and oil pan using the gearbox flange fixation and is sealed with standard automotive flange sealant. The oil catching chamber captures the oil which is flying off the rotating crankshaft and drive shaft flange when operating the engine. In addition, a bore was realised to allow the collected oil to flow back to the oil sump. The oil return bore prevents a filling of the oil catching chamber and undesired effects due to oil churning of the rotating shaft during the tests. The oil catching chamber is mounted and aligned on the base plate and also sealed with standard automotive flange sealant on the front end. The catching chamber is precisely manufactured on the inner bore to prevent touching the crankshaft or drive shaft flange and to avoid oil leakage. Therefore, the bore clearance is within 0.1 mm to the crankshaft/drive shaft flange surface and is precisely aligned during the mounting procedure. The designed sealing system worked very well and no oil leakage was detected during the conducted measurements. Following figure 5 shows the realised sealing system on the flywheel side.

The drive shaft flange connects the engine crankshaft to the torque measurement system and dynamometer with a test bed drive shaft. For the torque and engine speed measurement, a high accuracy torque measurement system (HBM T12) with built-in engine speed sensor is used. The connected engine crankshaft to the dynamometer is shown in figure 6.

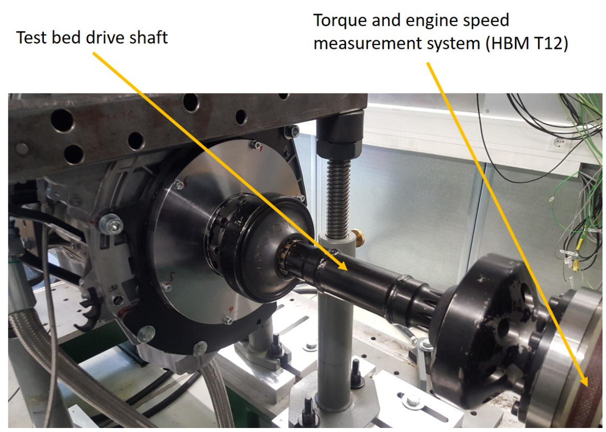

Figure 6: Crankshaft sealing system with connected crankshaft to the dynamometer

To realise reproduceable thermal boundary conditions during the measurements, the engine media (engine oil and coolant) are supplied from external supply units. This enables a accurate control of the supply temperatures and pressures, which are measured directly at the coolant circuit and oil gallery of the engine, respectively. The oil and coolant supply temperature was controlled with an accuracy of ±0.1 °C. The oil and coolant pressure was controlled with an accuracy of ± 0.05 bar. A standard 5W30 grade automotive engine oil was used to lubricate the engine. The basic rheological properties of the used lubricant is given in table 3.

Table 3: Basic rheological properties of the lubricant

SAE[2] class	5W30
Density at 15 °C	853 kg/m^3
Dynamic viscosity at 40 °C	59.88 mPas
Dynamic viscosity at 100 °C	9.98 mPas
HTHTS[3] viscosity	3.57 mPas

2.2 Testing Procedure

To determine the friction losses of the crankshaft radial lip seals, a strip-measurement campaign was realised. Figure 7 shows the process steps during the experimental investigation.

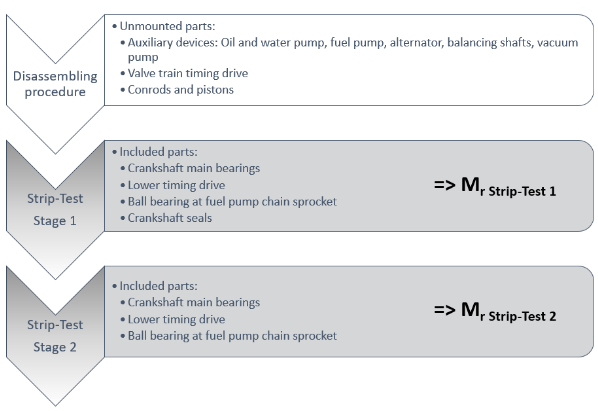

Figure 7: Measurement campaign process to determine the frictional losses of the crankshaft seal rings

The testing procedure consist of a two stage strip-measurement campaign. In the first stage, the drag torque ($M_{\mathrm{r\ Strip\text{-}Test\ 1}}$) of the crankshaft main-bearings, the lower timing drive, the ball bearing of the fuel pump chain sprocket and the crankshaft sealings (FEAD side and flywheel side) was measured. The measurements were done within a crankshaft speed range of n=1000 rpm to n=4000 rpm using speed steps of Δn=500 rpm. In addition, the tests are conducted at three different engine media supply temperatures (70 °C, 90 °C and 110 °C) to investigate the influence of the engine media supply temperature on the frictional losses. In the second stage, the drag torque ($M_{\mathrm{r\ Strip\text{-}Test\ 2}}$) of the engine setup from strip-test stage 1 without the crankshaft seals is measured at the same engine speeds and media supply temperatures. The resulting friction torque ($M_{\mathrm{r\ Crankshaft\ seals}}$) is calculated by subtraction of the measured drag torques of strip-test stage 1 and strip-test stage 2.

$$M_{\mathrm{r\ Crankshaft\ seals}} = M_{\mathrm{r\ Strip\text{-}Test\ 1}} - M_{\mathrm{r\ Strip\text{-}Test\ 2}} \tag{1}$$

To minimize variations in torque measurement due to different lubricant supply temperatures, specific focus has been spend on the lubricant supply temperatures when running the strip-tests. Following figure 8 shows the difference of the oil supply temperature between strip-test stage 1 and strip-test stage 2 over the entire speed range for a lubricant supply temperature of 90 °C.

The temperature variations between the strip-test stages could be maintained within 0.1 °C. The measurement uncertainty at n=1000 rpm and an engine media supply temperature of 90 °C is ±u=0.07 Nm which includes the sensitivity tolerance, linearity deviation including hysteresis, temperature influence, relative standard deviation of the repeatability and parasitic loads (axial force, lateral force and bending moment). At n=4000 rpm and an engine media supply temperature of 90 °C the calculated measurement uncertainty is ±u=0.09 Nm. The very stable testing boundary conditions in combination with the highly accurate torque and speed measurement system are enabling experimental investigations with high reproducibility of the results.

[2]SAE: society of automotive engineers

[3]HTHS: high temperature high shear rate viscosity (defined as the dynamic viscosity of the lubricant measured at 150 °C and at a shear rate of $10^6 s^{-1}$

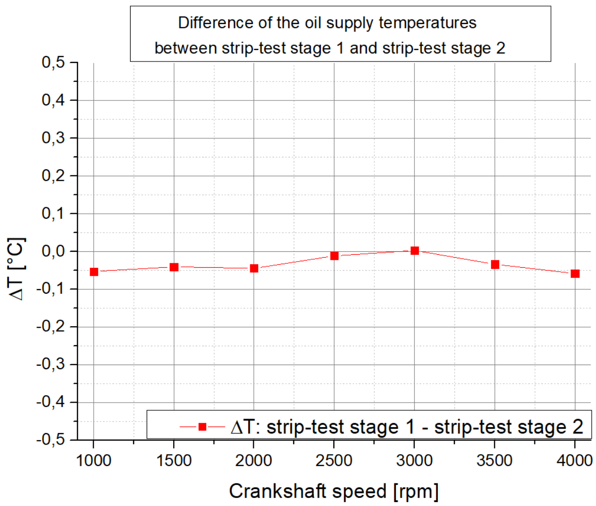

Figure 8: Variation of oil supply temperatures

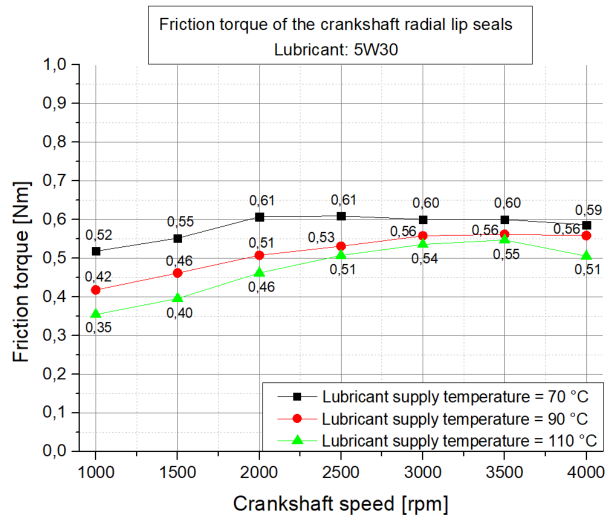

Figure 9: Resulting friction torque of the crankshaft seals (flywheel side and FEAD side) using equation 1

3 Results and Discussion

In this section, the obtained friction losses of the crankshaft seals from the used strip-measurement procedure are discussed. Therefore, the influence of the engine speed and the lubricant supply temperature on the friction losses is investigated. The strip-test measurements are carried out for 21 different measurement points for every strip-test stage (42 measurement points in total) and the resulting friction torque of the crankshaft seals using equation 1 is shown in figure 9.

Several interesting points can be observed from figure 9: The friction torque shows a dependency on the engine speed for lower engine speeds and enters a stable plateau at specific values. At an engine media supply temperature of 70 °C, the friction torque increases from 0.52 Nm at n=1000 rpm to 0.61 Nm at n=2000 rpm which is an increase of 0.09 Nm (17 %). For engine speeds above n=2000 rpm the friction torque reaches a constant plateau and remains constant at about 0.6 Nm. Further, it is interesting to note that for a lubricant supply temperature of 90 °C the friction torque increases from 0.42 Nm at n=1000 rpm to 0.56 Nm at n=3000 rpm which accounts to an increase of 0.14 Nm or 33 %. For a lubricant supply temperature of 110 °C the friction torque increases from 0.35 Nm at n=1000 rpm to 0.54 Nm at n=3000 rpm which accounts to an increase of 0.19 Nm (54 %).

It is also found that at higher lubricant supply temperatures of 90 °C and 110 °C the specific value of engine speed - where the stable friction torque plateau begins - is shifted to higher engine speeds above n=3000 rpm.

When analysing the friction torque for different lubricant supply temperatures, the lowest friction torque occurs at the highest lubricant supply temperature of 110 °C over the entire speed range. The highest friction torque, on the other hand, occurs at the lowest lubricant supply temperature of 70 °C. It is interesting to note that the largest differences occur at low engine speeds. At an engine speed of n=1000 rpm, the friction torque reduces from 0.52 Nm at an oil supply temperature of 70 °C to 0.35 Nm at an oil supply temperature of 110 °C which represents a reduction of 0.17 Nm (33 %). Furthermore, the friction reduction due to higher engine media supply temperatures is not following a linear trend. Despite that the

temperature difference between the measurements at different engine media supply temperatures is always ΔT=20 °C, the reduction of the friction torque between 70 °C and 90 °C is 0.1 Nm and between 90 °C and 110 °C it is 0.07 Nm at n=1000 rpm.
In addition, the advantages in the frictional losses at 110 °C supply temperatures is decreasing with incrasing engine speeds. Whereas the difference between the friction torque at 90 °C and 110 °C at an engine speed of n=1000 rpm is 0.07 Nm, the difference decreases to 0.01 Nm at n=3500 rpm. Also, the difference between the friction torque at 70 °C and 90 °C at an engine speed of n=1000 rpm is 0.1 Nm, and decreases with increasing engine speed to 0.04 Nm at n=3000 rpm.
A potential explanation for the frictional behavior of the radial lip seal is found in [12] who identified a similar trend. With increasing temperature, the radial forces of the sealing lip decreases because of changes in the rubber material properties. Additionally, the amount of oil in the lubricated contact impacts the lubrication gap and consequently the friction torque. The oil amount is dependent on crankshaft speed, due to recirculation effects in the lubricated contact of the sealing lip. Therefore, at low engine speeds, the reduced radial force in combination with the reduced lubricant viscosity at higher temperatures explains the varying friction torque at different temperatures. At high engine speeds, the uniform results may be caused by the reduced amount of oil in the lubricated contact. In addition, an increase of temperature with increasing crankshaft speed can be expected which results in a further reduction of the radial force of the sealing lip. Finally, the resulting friction power losses of the crankshaft radial lip seals at different temperatures are analysed. In figure 10 the resulting friction power losses are plotted over engine speed for different engine media supply temperatures.

As expected, the friction power losses show a strong increase over the crankshaft speed and is ranging from 37 W to 250 W depending on engine speed and lubricant supply temperature. The results show, that significant energy saving is possible beside large contributors like the crankshaft journal bearings [13] also at apparently small contributors of frictional losses. With an assumption of about 10 different

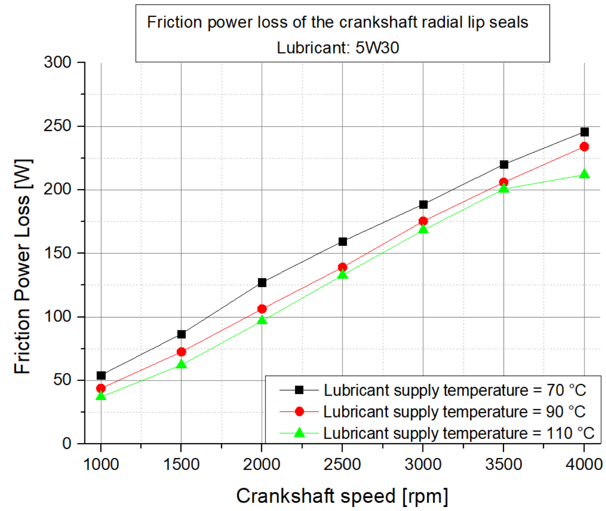

Figure 10: Resulting friction power loss of the crankshaft seals (flywheel side and FEAD side)

lip seals mounted in a conventional passenger car to seal rotating shafts, the amount of frictional losses of these machine elements is significant [14]. For comparison the friction power losses of the crankshaft journal bearings of a passenger car engine in motored conditions at an engine speed of n=4000 rpm and an oil supply temperature of 90 °C are in the magnitude of about 2 kW.

4 Conclusions

The friction losses of the crankshaft seals of a modern 4- cylinder diesel engine for passenger car applications have been investigated with a two stage strip-test procedure. The measurements contained tests with mounted and unmounted crankshaft seals. For the tests with unmounted crankshaft seals, a proper sealing system at both ends of the engine was designed and realised to prevent oil leakage during engine operation. A series of tests for engine speeds ranging from n=1000 rpm to n=4000 rpm and engine media supply temperatures of 70 °C, 90 °C and 110 °C have been conducted using a standard 5W30 grade automotive lubricant.

It was found that the highest crankshaft seal friction losses occur at the lowest lubricant supply temperature. In contrast, the lowest frictional losses appear at the highest lubricant supply temperature over the entire engine speed range. This is caused by a decreasing radial force of the crankshaft seal rings with increasing temperature in combination with the reduced lubricant viscosity at higher temperatures.
Furthermore, the results showed that the friction torque enters a stable plateau at specific engine speeds. The transitional crankshaft speed shifts from 2000 rpm at a lubricant supply temperature of 70 °C to an engine speed of n=3000 rpm at 90 °C and 110 °C. A potential explanation for this behaviour can be the reduced amount of oil in the lubricated contact due to the recirculation effects of the sealing lip resulting in lower friction losses. This in combination with a speed dependent temperature increase, a further decrease of the radial force of the sealing lip reduces the friction torque resulting in stable plateaus of friction torque.
Finally, it was found that the friction power losses of the crankshaft seals are ranging from 37 W to 250 W depending on engine speed and lubricant supply temperature. When considering that about 10 lip shaft seals are mounted in a conventional passenger car, improvements to minimize the friction losses can make a significant contribution even at apparently small contributors to meet future emission limits.

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6 Appendix

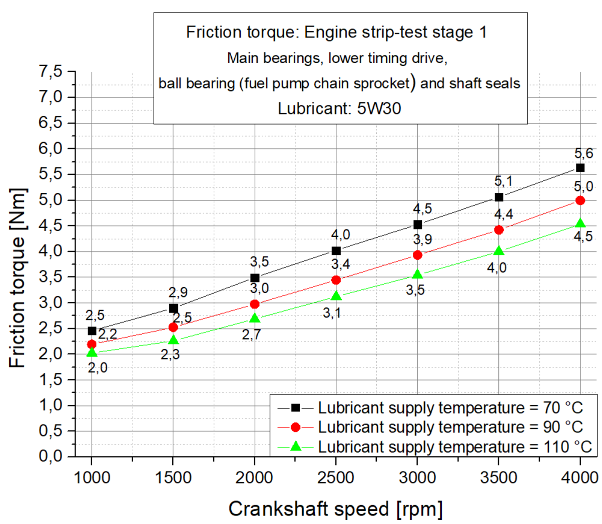

Figure 11: Measured friction torque for engine strip-test stage 1

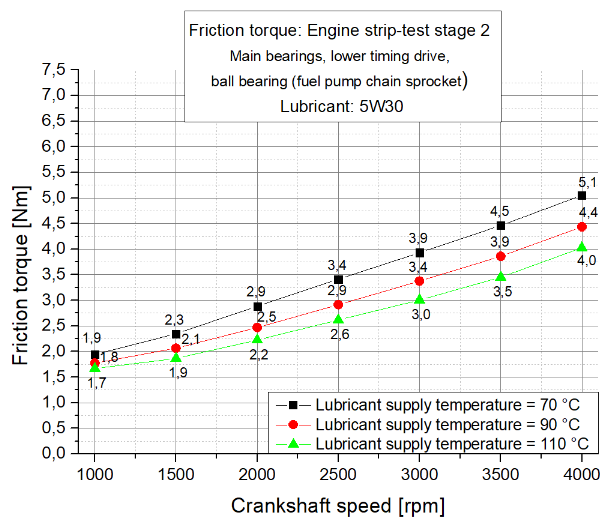

Figure 12: Measured friction torque for engine strip-test stage 2