Journal Title
XX(X):1–7

DOI: 10.1177/ToBeAssigned
www.sagepub.com/
SAGE

The impact of running-in on the friction of an automotive Gasoline-engine and in particular on its piston assembly and valve-train

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Abstract

Generally, mating surfaces that are in tribological contact undergo a running-in process at the beginning of their operational lifetime. During this running-in phase the tribological operating condition change significantly leading ideally to long term operation with a minimum of continuous wear. While this process and its duration is rather well understood for single machine elements like journal bearings, it is the aim of this work to investigate the running-in behaviour of more complex systems like an internal combustion engine and its sub-assemblies. To gain insight into the influence and duration of this running-in phase a series of tests have been performed under realistic engine operating conditions. To be able to separate the running-in processes for the individual subsystems piston assembly, valve-train and journal bearings of the crank train, a large series of tests have been conducted for a conventional gasoline passenger car engine. The results show a strong influence of the running-in process on total engine friction, which can be attributed mostly to the direct acting valve-train and to a considerably lesser extent to the piston assembly.

Keywords

Running-in, friction, measurement, simulation, friction break down, automotive engine

Introduction

In the automotive and transportation sector, the efficiency of the powertrain and in particular of the employed internal combustion engine (ICE) is of great interest. Not only increasingly strict emission legislation, but also the customer demand due to increasing fuel costs motivate the continuous work to improve the efficiency by lowering the friction power losses. The general term friction refers commonly to the largest part of the lifetime of the tribological system, where the initial running-in phase has been finished and the parts operate with minimal wear. From a tribological viewpoint the running-in phase is central to obtain a long-term state with a minimum of continuous wear Blau (2006) and a better understanding of its influence on the tribological properties and on its duration would be beneficial. In addition, current trends in emission legislation put further focus on the friction losses over the entire lifetime by employing continuous monitoring systems.

While this process and its duration is rather well investigated for single machine elements like journal bearings Allmaier et al. (2015); Sander et al. (2015, 2017), the running-in behaviour of more complex systems like an internal combustion engine and its sub-assemblies are considerably more challenging. Although the major subsystems of the engine, namely the valve-train, the piston assembly and the journal bearings of the crank train, operate with the same lubricant and have been developed to have similar operating life time, the tribological conditions for these subsystems are distinctly different. It is commonly understood that the valve-train operates in significant mixed lubrication Dyson (1977); Schamel et al. (1997); Priest and Taylor (2000), the piston assembly in dominantly hydrodynamic lubrication with severe mixed lubrication only occurring for short periods Tian (2002); Allmaier et al. (2016) and the journal bearings in dominantly hydrodynamic lubrication conditions Allmaier et al. (2013); Sander et al. (2016). These different lubrication conditions certainly influence the running-in behaviour. The quantitative influence of running-in on engine friction as well as its duration are the scope of this work.

To gain insight into the influence and duration of this running-in phase a series of tests have been conducted. As it is known that the running-in depends on the operating conditions (e.g. Allmaier et al. (2015); Sander et al. (2017)), a broad range of operating conditions were covered to ensure that the running-in does not depend significantly on the chosen operating condition.

The friction power losses of the complete engine were obtained over a large range of operating conditions including low and full load operation as well as the full rotational speed

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range of the engine. As the temperature of the lubricant has a profound influence on the viscosity in the contact and, consequently, on the tribological conditions, the engine was operated with different lubricant temperatures ranging from 70-110°C. In addition, the friction power losses of the valve-train (and timing drive) were also obtained experimentally. As the journal bearings of the crank train (main and big end bearings) were already run-in, their friction losses did not change during the tests. Therefore, by subtracting the valve-train contribution from the full engine friction losses all remaining influence of the running-in process can be attributed to the piston assembly.

Testing

The engine under test

The passenger car engine put to test is a in-line 4 cylinder, turbo charged, spark ignition engine with a nominal volume displacement of 1.8 litres and a nominal power of 132 kW; more technical details of the engine are listed in Table 1. To be able to test the running-in behaviour of the base engine, all auxiliary devices like oil and water-pump, alternator etc. have been removed from the engine. To realize external temperature and pressure control of the media, several parts of the engine, e.g. the oil module and the oil pan, have been modified. The coolant and engine oil are supplied from an external supply unit which also controls 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 externally with an accuracy of about $\pm 0.5°$ C. The oil and coolant pressure was controlled externally with an accuracy of about $\pm$ 0.05 bar. The cylinder pressure is measured in all 4 cylinders using piezoelectric pressure sensors.

The tested engine employs a direct acting valve-train with 2 overhead camshafts and 5 valves per cylinder, conventional aluminum pistons with coated piston skirts and coated piston rings in a cast iron crank case. Even brand new engines have already been operated by the manufacturer during the end of line testing, so an alternative approach has been used. A used engine was employed and all parts of the piston assembly and all relevant parts of the valve-train were replaced with new parts. However, the already run-in journal bearings of the crank train (main and big-end bearings) were kept and not replaced.

As the running-in behaviour of the journal bearings of the crank train was investigated in previous works Allmaier et al. (2015); Sander et al. (2015), the focus of this work is on the piston assembly and valve-train. Therefore, the already run-in journal bearings have been used and it was checked at the beginning and end of the tests that their contribution to the total engine friction did not change.

To lubricate the engine under test a standard 5W30 grade automotive lubricant was used and its basic rheological data are given in Table 2.

The dynamometer

For the measurements a pressurized motoring test-rig has been used. In contrast to fired engine operation, a pressurized air dynamometer supplies air under pressure to the engine

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

Volume displacement	1781 cm^3
Compression ratio	9.5:1
Bore	81 mm
Stroke	86.4 mm
Nominal torque	235 Nm
Nominal Power	132 kW
Cylinder diameter	88 mm
Conrod length	144 mm
Main bearing diameter	54 mm
Main bearing width	22 mm
Big-End bearing diameter	47.8 mm
Big-End bearing width	25 mm
Valve-train	DOHC
Timing drive	(dry) belt
Valve-train type	direct acting on bucket tappet
Valves	5 per cylinder

Table 2. Lubricant data

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

intake. In combination with the intrinsic compression of the engine realistic peak cylinder pressures of up to 200 bar can be realized in the combustion chamber Sander et al. (2013); Allmaier et al. (2016). This method has the major advantage of much higher measurement accuracy compared to fired engine tests. The main reason for the improved accuracy is the strongly reduced (by a factor of about 4) Indicated mean effective pressure (IMEP) in comparison to fired operation. Therefore, torque transducers with a much smaller measurement range and, consequently, much higher absolute accuracy can be used. In addition, in fired engine tests, the variations between individual cycles can be very large, in particular for Gasoline engines, where it is common to see more than 20% variation in peak cylinder pressure. The main drawback of pressurized motoring is the different thermal situation in the piston assembly. However, a direct comparison of friction measurements for the same engine in fired operation and pressurized motoring showed very similar results Allmaier et al. (2016).

The pressurized motoring technique uses the well known IMEP-method to determine the friction mean effective pressure (FMEP) by subtracting the brake mean effective pressure (BMEP),

$$\mathrm{FMEP} = \mathrm{IMEP} - \mathrm{BMEP} \tag{1}$$

with the minor difference being that the signs of the IMEP and BMEP are negative in comparison to fired engine operation. When using the IMEP-Method to investigate the friction losses of an engine it is essential to determine two quantities, the indicated power and the brake power of the engine with highest possible accuracy. The reason for this is that IMEP and BMEP are two large and very similarly sized quantities which are subtracted from each other to determine a rather small difference (FMEP). Therefore, any significant error in the determination of the BMEP and IMEP can easily lead to an measurement error of the same magnitude as the to-be-determined FMEP.

Practically, the IMEP is measured using piezoelectric sensors in the combustion chamber. However, the BMEP cannot be measured directly as pressure, but is commonly measured as brake torque. Following equation relates the mean effective pressure p_{me} to the corresponding torque T,

$$p_{me} = \frac{W}{V_D} = \frac{4\pi T}{V_D}, \tag{2}$$

where W refers to the work per cycle and V_D to the volume displacement of the investigated 4-stroke engine.

At the friction dynamometer test-rig a high accuracy torque measurement system (HBM T12) with built-in engine speed sensor is used for the brake torque and engine speed measurement. For the IMEP determination it is essential to determine the position of the top dead center (TDC) of the piston to great precision. To this task, a capacitive TDC-sensor (AVL OT428) was used to ensure a high accuracy TDC determination. Ideally, the TDC determination should be done for every operating condition as the engine load and even the oil temperature influences the exact position of the top dead center. A working compromise is to perform TDC determination for an important operating condition, which was defined as 2000 rpm. Figure 1 shows the capacitive TDC sensor installed in the opening for the spark plug of cylinder #1[*] during the TDC determination.

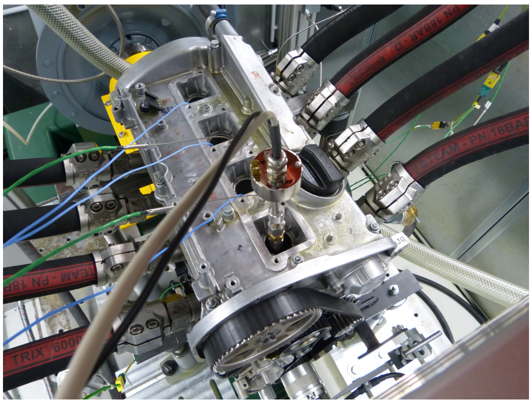

Figure 1. Mounted capacitive TDC sensor during TDC determination in cylinder #1

Test procedure

As shortly discussed in the introduction, to ensure the proper running-in of all engine components a wide range of engine operation conditions ranging from low to high engine speeds at low and high engine loads were performed for three different coolant/oil temperatures. For the tests the oil and coolant supply temperatures were kept identical; therefore, the term *media temperature* will be used for brevity in the following.

The engine operating conditions (speeds and loads) for the tests are listed in figure 2. These engine operating conditions were performed for 70/90/110 °C media temperature (in increasing order as listed). The increasing media temperatures are intended to promote a more gradual running-in process as mixed lubrication generally increases with increasing media temperature.

Full engine tests

The largest part of the experimental investigations comprises tests with the complete base engine. One set of measurements consists of all engine operating points (speeds and loads) shown as inset in figure 2. Such a test set was conducted for 3 different media temperatures 70°C, 90°C, 110°C (in this order) and then repeated starting again from 70°C media temperature. In total, 3 such tests were performed (denoted as test 1/2/3). After these 3 tests, additional measurements only for 3000 rpm engine speed were performed for all engine loads and for 70°C media temperature (denoted as test 4). Therefore, the tests 1/2/3/4 correspond to similarly spaced, increasing operating times of the engine. The total engine operating time was about 60 h.

Additional tests

Before and after the full engine measurements additional measurements were conducted to obtain the friction contributions from the valve- train with timing drive and the crankshaft main bearings with rotary shaft seals.

In *Stage 1* all four piston assemblies (pistons with con-rods) have been removed from the engine. As a replacement for the removed masses and to seal the oil supply holes in the crank shaft pin, master weights were mounted onto the crank shaft for this test. To summarize, in the stage 1 configuration the crankshaft supported by the main bearings, the rotary shaft seals, the timing belt and valve-train is motored. As the largest part of the measured friction torque in this engine configuration is attributed to the valve-train, this stage is called valve-train friction torque for simplicity in the following.

Stage 2 comprises the modifications from stage 1 and in addition the timing belt is removed from the engine under test. This allows the measurement of the crank train main bearings and rotary shaft seals. By subtracting the results from stage 1 with the results from stage 2 the friction contribution of the valve-train is obtained. In addition, this stage 2 test serves as a check at the beginning and end of the tests that the friction power losses of the main bearings do not change during the testing. Figure 3 shows the measurements of this stage 2 configuration at the beginning and at the end of the measurements and confirms that indeed the main bearings (and rotary shaft seals) do not change detectably.

Results and Discussion

This section discusses the results of the investigated running-in processes of the tested engine. Figure 4 shows the measured friction torques for the full engine after 4 different operating durations (as discussed in section test procedure). The results indicate that the chosen total running time of 60 h

[*]The used numbering of the cylinders starts from the side opposite of the clutch where most engines have the timing drive.

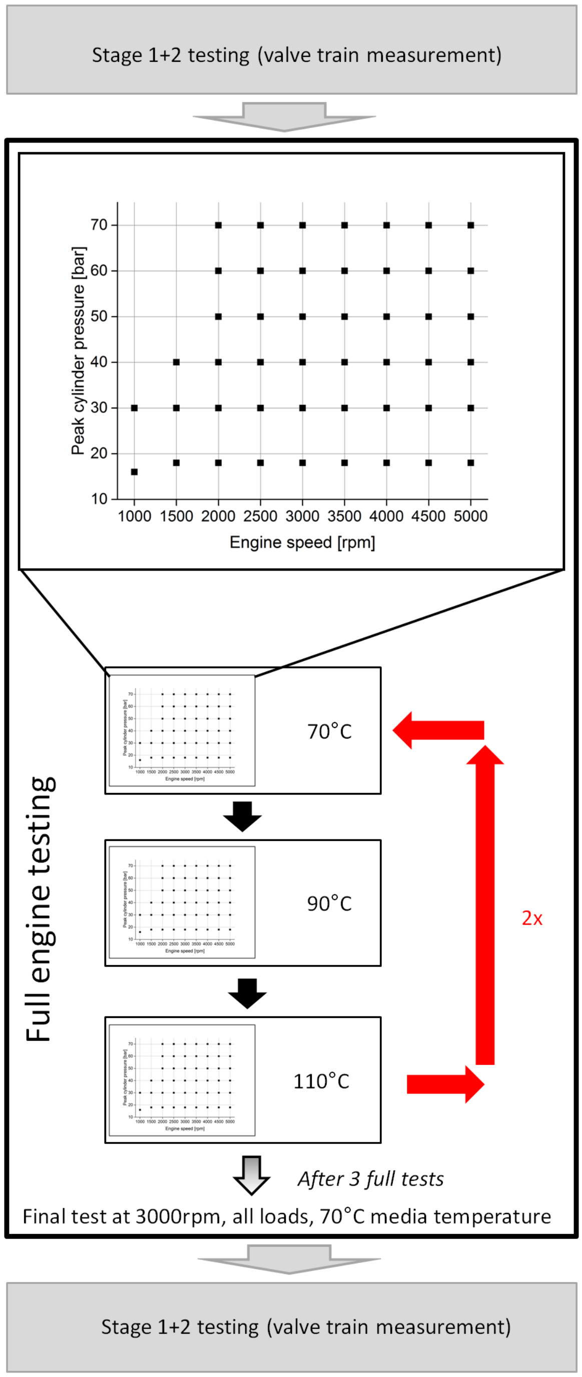

Figure 2. Flow chart of the tested engine operating conditions. The tests covered the full range of operating conditions of the engine (in terms of speed and load), were performed for 3 different media temperatures (70°C, 90°C and 110°C) and conducted 3 times in total. At the end, a final test at 3000 rpm was performed for all loads ranging from motoring to full load and 70°C media temperature. At the beginning and at the end measurements for the stage 1+2 configuration were conducted to obtain the friction torque of the valve-train (see testing section).

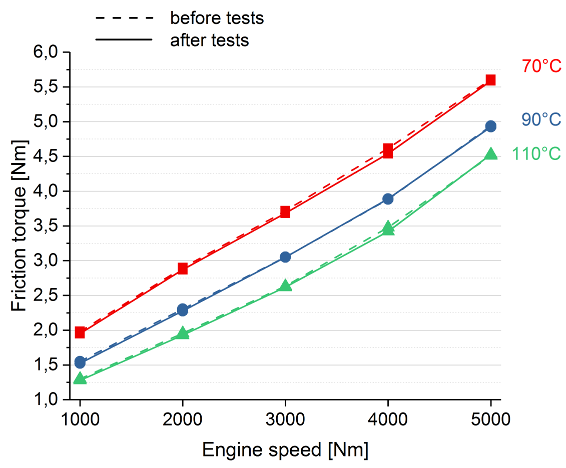

Figure 3. Comparison of the measured friction torque for the stage 2 engine configuration (motoring only the main bearings of the crank train and rotary shaft seals, see text) before and after the full engine test program.

is sufficiently long so that the running-in process is almost completely finished and a stable friction torque is achieved for most operating points. Already the 2nd test indicates that most of the running-in process is finished and only small changes in the friction torque occur in the later tests. The running-in process causes a significant change in full engine friction torque as is found from the results. For e.g. the full load (70 bar peak cylinder pressure) operating point at 70°C media temperature, a total change due to running-in of 0.82 Nm is observed, which is equivalent to about 6% of the initial friction torque of 14.62 Nm. Also the change in friction torque is strongly regressive with the most part of the change occurring very quickly. Between the 1st and the 2nd test already 0.57 Nm change occurred, which is 70 % of the total change. Between the 2nd and 3rd test only a change of 0.16 Nm is observed, which corresponds to about 20 %. The remaining 10 % change are seen between the 3rd and 4th test with an absolute change in friction torque of 0.09 Nm. For the other operating conditions similar behaviour is observed.

As can be seen from figure 4, the measured decreasing trends appear not to be entirely perfect. To put this into perspective, the smallest observed differences in the friction torques are of the magnitude of 0.04-0.1 Nm, which corresponds to less than about 0.7% of the maximum measured motoring (BMEP) torque of about 60 Nm and corresponds to even only 0.04% when related to the maximum nominal torque of the engine (235 Nm, see table 1). It was found that the cleanest results with the smallest fluctuations where obtained for 70°C media temperature, so these will be investigated in particular in the following.

Several interesting points can be observed from figure 4: notably the influence of running-in is about as large as the temperature difference between 70°C and 90°C media temperature and its impact on engine friction. In numbers, the running-in process caused a 0.6-0.8 Nm reduction in the full engine friction torque for 70 bar peak cylinder pressure (for both 70°C and 90°C media temperature), whereas the 20°C difference in media temperature caused a change of about 1.2 Nm in friction torque. Also the influence of the engine load (peak cylinder pressure) is of the same magnitude with about 0.6-0.8 Nm change in friction torque between 30 bar and 70 bar peak cylinder pressure.

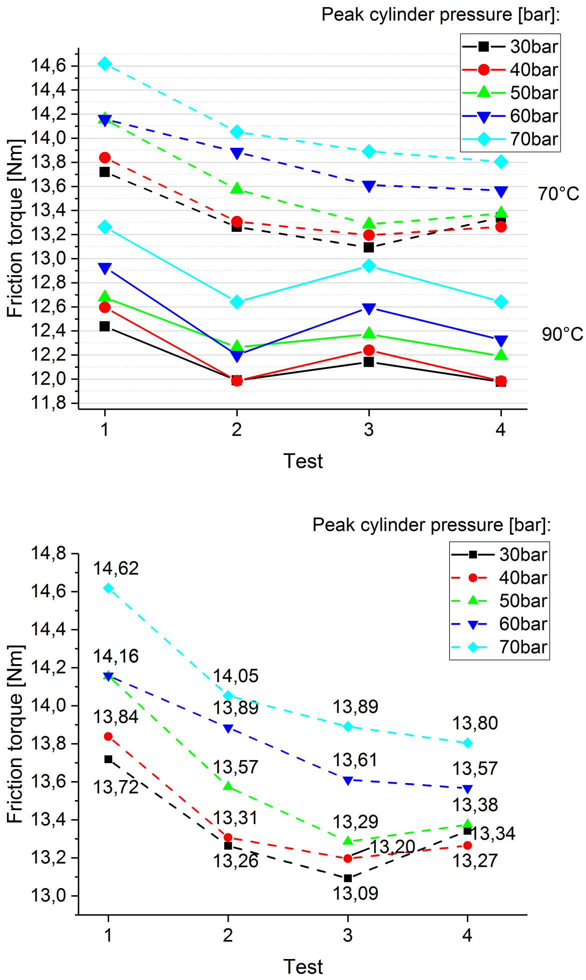

Figure 4. Change of full engine friction torque with the number of tests shown for a rotational speed of 3000rpm: the top plot shows the results for 70°C and 90°C media temperatures, the bottom plot shows the 70°C results in more detail.

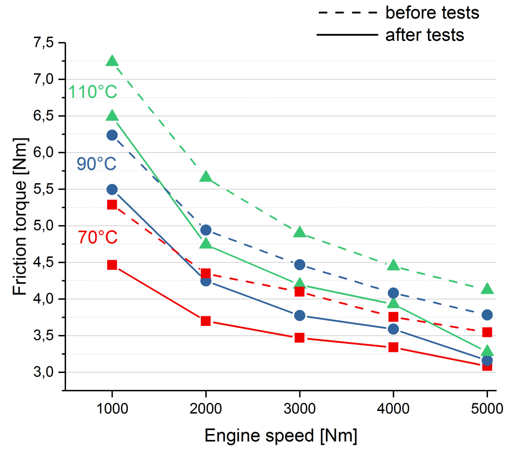

Figure 5. Comparison of the friction torque of the valvetrain before and after the running-in tests

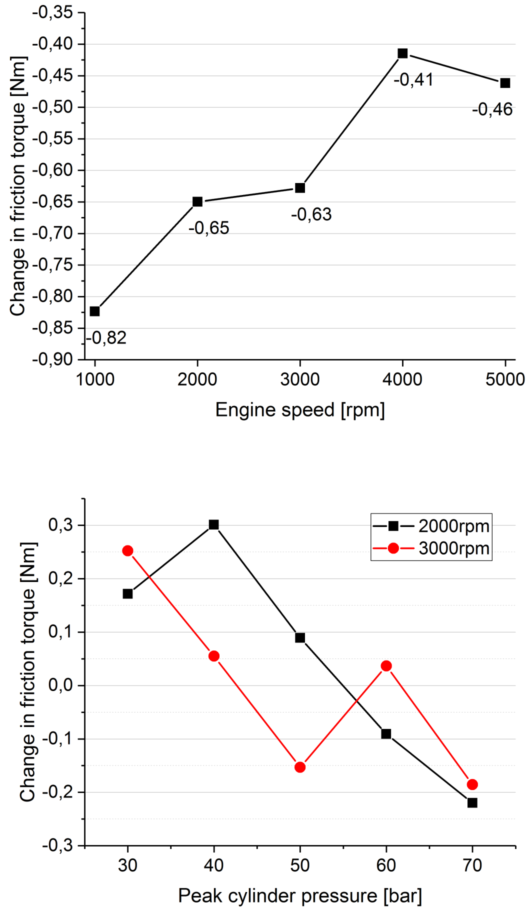

Figure 6. Difference of the friction torque of the valvetrain before and after the running-in tests for 70°C media temperature; the bottom plot shows the change in friction torque for the piston assembly for 3000rpm engine speed before and after 3 full testing maps. For comparison, also the results for 2000 rpm are shown before and after 2 testing maps (all for 70°C media temperature).

While the results for the full engine tests provide a global perspective, it is interesting to investigate the influence of running-in and its effect on the sub-assemblies valve-train and piston assembly (the journal bearings of the crank train do not change during the tests as discussed previously in section additional tests and shown in figure 3).

Running-in results for the valve-train: in figure 5 the friction torques of the valve-train before and after the tests are shown for different media temperatures and figure 6 shows the corresponding change in friction torque for 70°C media temperature.

As it is to be expected from lubrication theory, the influence of running-in directly correlates with the amount of mixed lubrication which is strongest for low engine speeds. This is supported by the shown plots (figure 5 and figure 6) as the influence of running-in reduces with increasing engine speed. From the results it becomes apparent that for the investigated direct acting valve-train always significant mixed lubrication is present; even at the highest engine speed a change due to running-in can be observed for all three media temperatures.

Running-in results for the piston assembly: Figure 6 shows the change in friction torque for the piston assembly by subtracting the friction torque of the valve-train from the full engine friction torque before and after the entire testing-procedure. The results are shown for an engine speed of 2000 and 3000 rpm, which are the engine speeds where most mixed lubrication is expected.

While the results for both engine speeds show very similar trends, it is necessary to point out that these two engine speeds correspond to different engine running times. Only the engine speed of 3000 rpm was tested at the end of the three test sets (see section full engine tests), therefore the results for the 2000 rpm engine speed were taken from the last of the three tests and correspond to a shorter engine running time. Consequently, the 2000 rpm engine results in this figure shall only serve to confirm the 3000 rpm results.

Apparently the modifications to the engine and/or to the test-rig caused a measurement offset as the change in friction torque obtained for the piston assembly is positive for low peak cylinder pressures. This increase would mean that the running-in process causes an increase in friction of the piston assembly for low peak cylinder pressures, which - even if it might be physically possible - appears unlikely. Also the obtained results are on the border of the measurement accuracy of the test-rig, therefore, the focus is only on the observed trend in the following.

Neglecting this change in absolute torque offset, a trend can be identified for the piston assembly. With increasing peak cylinder pressure a stronger effect of running-in on the friction torque can be seen for the piston assembly. This is consistent with the understanding of the occurrence of mixed lubrication in the piston assembly where increasing peak cylinder pressure leads to increased mixed lubrication. Therefore, running-in optimizes the mating of the surfaces in contact and this improvement is expected to gain influence with increasing peak cylinder pressure.

Further, it is interesting to note that the impact of running-in is considerably smaller for the piston assembly than for the valve- train: the friction torque of the piston assembly changes by about 0.3 to 0.5 Nm, whereas the friction torque of the valve-train changes by about 0.4 to 0.8 Nm in comparison. A possible explanation for these results might be that the valve-train does not employ coatings neither on the cams nor on the bucket tappets. In contrast, both the piston rings and piston skirts are coated which commonly reduces wear. In addition, the valve-train employs the direct acting principle, where the cam directly acts on the tappet without any (roller) finger follower. This working principle leads to a highly loaded sliding contact Dyson (1977); Colgan and Bell (1989); Katoh and Yasuda (1994); Schamel et al. (1997); Masuda et al. (1997); Priest and Taylor (2000); Lawes et al. (2010) which in combination with the unusually high number of 5 valves per cylinder is the likely explanation for the observed magnitude of the valve-train friction power losses.

Conclusions

The influence of running-in on the friction torque of a 4 cylinder Gasoline engine was investigated with a series of tests for different engine speeds, loads and media temperatures using a conventional 5W30 lubricant. With the exception of the journal bearings of the crank train, the engine can be considered to be like-new at the beginning of the tests. By removing the pistons and con-rods before and after the measurement campaign, the change in friction torque of the valve- train with timing drive can be assessed. As the friction torque of the full engine and the valve-train is measured and the journal bearings of the crank train did not change during the tests, the influence of running-in for the piston assembly can be derived in addition.

The results for the full engine indicate that the running-in process is almost finished after the investigated operating time of about 60 h. After this total 60 h operating time, the friction torque decreased by about 6 % from its initial value. The largest change in friction torque (about 70 % of the entire change) occurs already between the 1st and 2nd test, which corresponds to an operating time of about 15 h. The change in friction torque due to running-in observed between the 2nd and 3rd test is about 20 % and between the 3rd and 4th test is about 10 % of the entire change in friction torque due to running-in.

From the results it is further found that the direct acting valve- train with 5 valves per cylinder is strongest affected by the running-in process. The friction torque of the valve-train changes by a maximum of about 0.8 Nm which represents about 18% of the total valve-train friction torque (refering to the same operating point). As expected, the influence of running-in is strongest for low engine speeds where the sliding speeds are smallest and mixed lubrication is strongest.

In contrast, the influence of running-in on the piston assembly is minor. While the piston assembly contributes considerably more friction torque than the valve-train[†] a maximum change in friction torque due to running-in of merely about 0.4 Nm is observed. However, an increasing influence of running-in with increasing peak cylinder pressure could be identified from the results.

A possible explanation for this unexpected result is unusually high number of 5 valves per cylinder and, in particular, the lack of coatings in the valve-train (neither on cam or tappet) Lawes et al. (2010); Gangopadhyay et al. (2004). While in the piston assembly both the piston ring faces as well as the piston skirt are coated and are, therefore, more resistant to wear.

Finally, it is interesting to compare the results to the running-in behaviour of the crank train journal bearings. In a previous work Allmaier et al. (2015), the running-in behaviour of journal bearings with similar size and operating conditions were investigated by employing a combination of simulation and experiment. It was found that the running-in process of journal bearings is finished very quickly; the most significant part of running-in was observed in the first 20 minutes of testing on the journal bearing test-rig. Also in terms of influence on friction, the magnitude of the friction torque changed by about 10% in the first 20 minutes of operation and was already insignificant for longer operating times. Summarizing, it is concluded that for the investigated engine the running-in process affects most strongly the

[†] While the absolute contribution of the piston assembly to the measured friction torque cannot be accessed using this approach, it is estimated to be at about 6 Nm for 3000 rpm engine speed, full load and 70°C media temperature.

valve-train where the observed change in friction torque is largest with about 18%. By considering the results from the previous work, it appears that the journal bearings of the crank train are also significantly affected by the running-in process with a roughly 10% change in friction torque. Finally, the piston assembly shows the smallest change for the investigated engine.

Future work could address the running-in process of a Diesel-engine as the considerably higher peak cylinder pressures of up to 200 bar might involve a stronger running-in behaviour for the piston assembly.

Acknowledgement

The authors would like to acknowledge the financial support of the ”COMET - Competence Centres for Excellent Technologies Programme” of the Austrian Federal Ministry for Transport, Innovation and Technology (bmvit), the Austrian Federal Ministry of Science, Research and Economy (bmwfw), the Austrian Research Promotion Agency (FFG), the Province of Styria and the Styrian Business Promotion Agency (SFG).

Furthermore, we acknowledge the partial financial support of the Austrian Science Fund (FWF): P27806-N30.

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