Analyze engine friction in view of the new WLTC driving cycle

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With increasingly strict legislation and further the introduction of a new testing procedure WLTP, the highly detailed knowledge of engine friction becomes essential to obtain further fuel economy reductions. The new Worldwide Harmonized Light-Duty Vehicles Test Cycle WLTC driving cycle will not only test a larger range of engine operation conditions, but also brings fundamental changes to the way the powertrain is evaluated. In the following, a novel engine friction testing method is used together with a highly accurate simulation to analyze engine friction exemplary for the Renault Energy dCi 130 Diesel engine in great detail and discuss the results in view of the upcoming new testing procedure.

The differences between the NEDC and the WLTC driving cycle

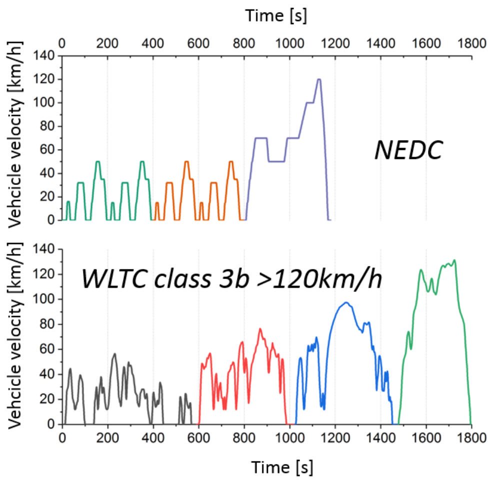

Figure 1: vehicle velocities tested in the NEDC and WLTP testing procedures (colors mark the individual phases of the tests).

	Unit	NEDC	WLTC class 3b >120km/h
Duration	s	1180	1800
Distance	km	11,02	23,26
Mean Velocity	km/h	33,61	46,51
maximum velocity	km/h	120,1	131,30
% Stop time	-	21,61	12,17
% Cruise time	-	33,62	11,17
% Acceleration time	-	26,02	39,33
% Deceleration time	-	18,81	37,17
Mean acceleration	m/s^2	0,48	0,44
Max acceleration	m/s^2	1,11	1,58
Mean deceleration	m/s^2	-0,66	-0,47
Max deceleration	m/s^2	-1,39	-1,49

Table 1: statistical comparison of the tested engine operation conditions for the NEDC and the WLTP testing schemes

Fig. 1 and Table 1 aid in the comparison of the two testing cycles. The WLTC driving cycle (for class 3b vehicles of more than 120km/h) is not only twice as long as the New European Driving Cycle NEDC, it features a higher mean velocity, a strongly reduced stopping time and only a minor amount of cruising time. Consequently, acceleration and deceleration have a much higher relevance. Due to the involved higher velocities and accelerations, different engine speeds and engine loads gain importance in comparison to the NEDC.

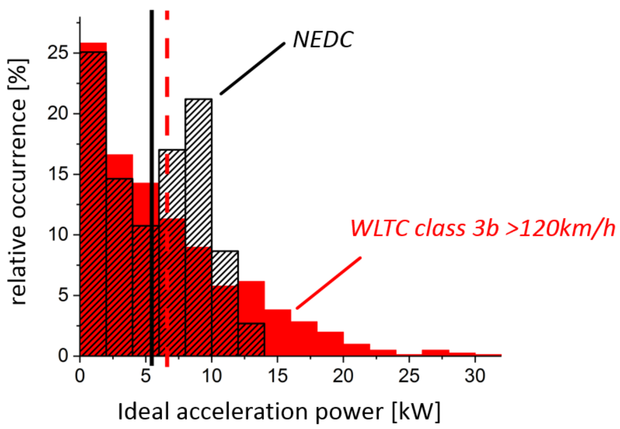

Figure 1b: occurrence in % of the ideal acceleration power required to reach the tested vehicle velocities in the NEDC and WLTP testing procedures, the averages are shown as vertical line in black for NEDC and in red for WLTC; stopping times have been excluded from this illustration.

As can be seen in Fig. 1b, much more engine operating conditions are tested in the WLTC and higher power operating condition become important: 20% of the ideal acceleration power is higher than 10kW in WLTC in comparison to about 7% for NEDC. For this reason single CO_2 saving technologies like start-stop systems and improvements in engine heating up from cold-start have a reduced influence on the WLTC tested fuel consumption. Not only the impact of such technologies may potentially decrease. Due to this new testing procedure it might even happen that a technology which was beneficial for NEDC does not positively affect the WLTP results. This happened in a study conducted for different lubricants [2]. Such unexpected results can be explained generally by a detailed engine friction analysis.

New tools help in analyzing engine friction

Generally, in addition to trends like downsizing and -speeding, friction reduction offers still a significant potential to increase efficiency. As engines have already been optimized mechanically for a long time, it is often not possible to obtain significant improvements by optimizing only a single part. Therefore, it is commonly necessary to perform a number of small optimizations that yield in total a significant improvement [1]. The experimental assessment of such small optimizations in the smallish 100-200W range, however, is a challenge in itself due to the considerable measurement uncertainties of common measurement techniques. The method used in the following was presented in MTZ 10/2016 and is a valuable tool to investigate such small optimizations as it offers superior accuracy in comparison to other methods. Among the various possibilities to reduce mechanical friction, the usage of a low-viscosity lubricant in the engine is one of the most effective and most economic options. For this reason, continuously lower viscosity lubricants are being developed and offered on the market that promise to reduce engine friction while avoiding deleterious mixed lubrication and wear. Consequently, two low viscosity lubricants are investigated in the following after an introduction to the used testing and simulation tools.

Meet FRIDA

FRIDA – the short hand for Friction Dynamometer – is the test-rig utilized to measure the total engine friction under realistic operating conditions with improved accuracy. In the following the term friction relates to the power losses in dry or lubricated contacts and does not consider the power required to operate auxiliary devices (e.g. coolant/lubricant pumps etc.). FRIDA is basically a motoring test-rig that utilizes an external charging system to realize realistic peak cylinder pressures in the engine under test [3].

With the external charging system peak cylinder pressures of more than 200 bar can be realized. FRIDA utilizes an external charging system that supplies up to 200°C hot air to the engine inlet, which yields peak temperatures in the combustion chamber of about 800-1000°C, independent of the peak cylinder pressure.

The major advantage of the charged motoring method is the about 4-times improved measurement accuracy in comparison to other methods. The reason for the improved accuracy is the strong reduction of the IMEP for pressurized motoring, which reduces proportionally the measurement uncertainty at the same time. In numbers this means that for a Diesel engine with an output torque of +300 Nm in fired operation the motoring torque in charged motoring is only about -70 Nm. The friction power losses stay around 10 Nm, which gives a more than 4-fold advantage in measurement uncertainty.

The lack of combustion and the, consequently, different thermal situation in the combustion chamber represent therefore the most significant compromise of the charged motoring method. In a direct comparison of the results for the same engine from fired operation and from charged motoring it was found that the largest differences between the two methods are confined to part load conditions. For full load operation the results agreed within 0.5% accuracy (in relation to the nominal power of the engine) as well as trends were found to agree closely.

The simulation

For the calculation of the friction power losses of the main and big-end bearings of the crank train an accurate simulation method was developed at Virtual Vehicle. The method was validated with a large number of measurements and has been presented in detail in previous articles [3,4,5].

The simulation of the crank train journal bearings was realized using AVL Excite PowerUnit[1]. It considers both the elastic properties of the involved bodies (crank shaft, con rod etc.) as well as the microscopic geometry/roughnesses of the surfaces and the complex rheological properties of the lubricant. The simulation does not only allow to predict accurately the friction of the journal bearings, but also to determine if mixed lubrication is present and, if so, how severe it is. Due to the continuously increasing mechanical loads, this is a valuable property of the method [5].

The virtual strip-down

By combining simulation and testing, a full *virtual* strip-down of engine friction becomes possible for all operating conditions of interest including full load. In the conventional strip-down method the engine is motored without a realistic peak cylinder pressure and by removing engine components their contribution to total engine friction is estimated. The main compromise with this approach is that the engine is not operated under load and that the removal of certain subassemblies (like the valve train) affects also the friction contribution of other subassemblies (like e.g. the piston assembly), so their contributions are not truly independent from each other. All these drawbacks are not present for the virtual strip-down.

By using pressurized motoring, realistic mechanical loads can be realized in the engine. With the accurate simulation of the crank train journal bearings, a *virtual* strip-down can be performed in 3 steps: first the total engine friction under load is measured. In addition, a separate measurement of the valve-train and timing drive friction is conducted. Finally, with the accurate simulation of the crank train journal bearing friction, the friction contributions of the piston assembly, crank train journal bearings and valve-train with timing drive can be separated for any operating condition of interest including full load operation.

The Energy dCi 130 engine from Renault

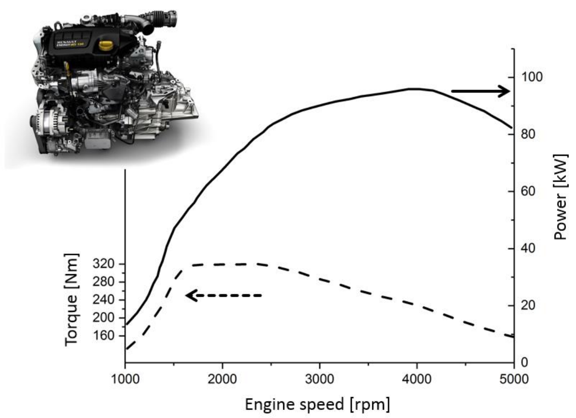

Figure 2: torque and power of Renault's Energy dCi 130 engine

[1] AVL List GmbH, Advanced Simulation Technology, Hans-List-Platz 1, Graz, Austria 8020, www.avl.com

To investigate the friction benefit of low viscosity lubricants a well-established engine was chosen which is in operation in a wide range of vehicles – Renault's Energy dCi 130 inline 4-cylinder engine. As shown in Fig. 2, the engine realizes 130hp from 1.6l volume displacement and it features carefully-developed Stop & Start system combined with deceleration/regenerative braking. It uses cold-loop, low-pressure exhaust gas recirculation (EGR), where Renault was the first manufacturer to introduce this technology in Europe. Further, thermal management technology, a variable displacement oil pump, variable swirl technology and a multi-injection system designed to optimize particulate filter regeneration was realized. In terms of friction, the choice of a shorter stroke allowed the dimensions of the rotating parts (crank assembly) to be optimized for less friction. The calibration of the piston rings and the surface treatment of the bearings also contributed to reduce friction.

Friction analysis for two low viscosity lubricants

The influences of two different low viscosity lubricants on engine friction are in the focus of the present investigation. The two lubricants will be called Oil A and Oil B in the following, where Oil B has about 25% lower viscosity than Oil B. Oil A features also an equivalent reduction in the so called high temperature, high shear (HTHS) viscosity. Despite the apparent large difference of 25 % and more in the lubricant viscosities of low viscosity oils, the corresponding effect on engine friction is much smaller and highly accurate test-rigs are required to measure the benefit of the individual lubricants experimentally.

The following Fig. 3 gives a first global perspective by looking at the total engine friction for the two lubricants Oil A and Oil B for two different oil sump temperatures (70°C and 90°C) under engine load. The lower viscous oil A shows a global benefit in comparison to oil B. As expected, the differences between the two lubricants are much smaller than the large differences in their viscosities. For both lubricant temperatures and over the entire range of engine operating conditions studied the effect on engine friction from Oil A is less than 1 Nm and ranges from 0.3-0.6 Nm. However, from a total engine friction measurement it is not possible to draw conclusions whether specific engine subassemblies like the piston assembly might show a larger benefit than others. Therefore, a *virtual strip-down* is performed in the following to gain a more detailed understanding of the engine friction.

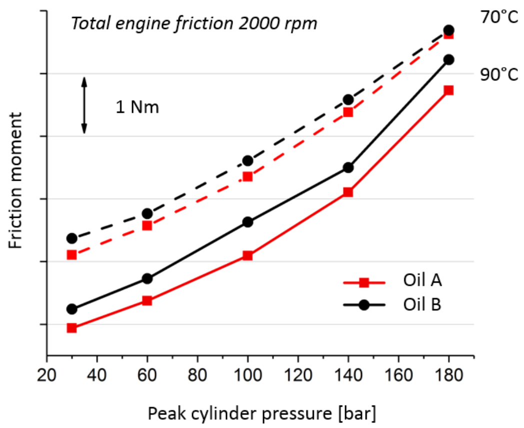

Figure 3: Total base engine friction over engine load for a rotational speed of 2000rpm for both oils at 70°C and 90°C oil temperature.

Valve train with timing drive and rotary shaft seals

From the measurement of the engine with pistons and conrods removed, the friction of the valve train, timing drive and rotary shaft seals can be obtained. The friction power losses of the main bearings have been subtracted using the accurate simulation as discussed previously. Due to the reduced rotation speed of the valve train in comparison to the engine speed and due to the high contact loads, the valve train is commonly the engine subsystem with the largest amount of mixed lubrication. A reduced viscosity lubricant may negatively promote mixed lubrication consequently; this, however, is not the case for the investigated engine.

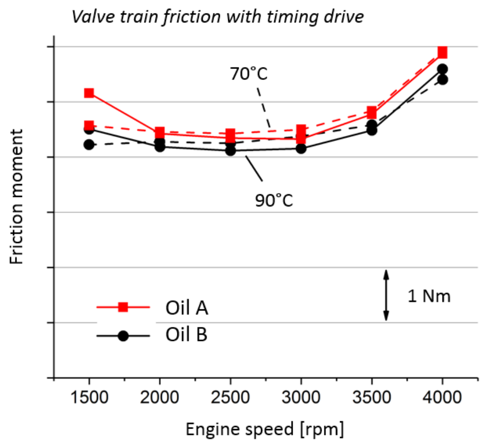

Figure 4: friction of the valve train/timing drive combination for both oils at 70°C and 90°C oil temperature.

From Fig. 4 it is evident that the friction power losses of the valve train and timing drive are only slightly affected by the two tested lubricants. For both oils, a slight increase in friction at very low engine speeds can be seen, which is more pronounced for the lower viscosity Oil A. Otherwise the plots for the two oils have the same trends but are only shifted to a slightly higher friction moment for Oil A (by about 0.2 Nm).

Piston assembly

The piston assembly is the largest single contributor to engine friction and is, therefore, of primary interest. Plot 5 shows the obtained piston assembly friction for a representative engine speed of 2000 rpm.

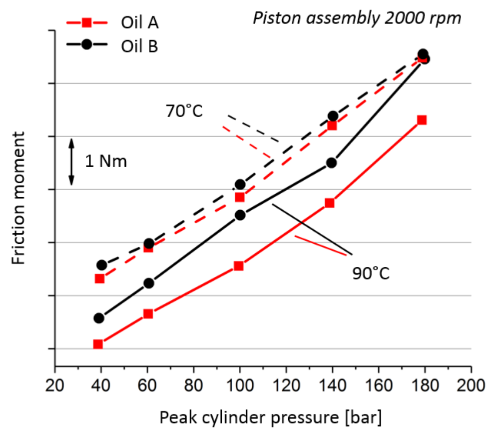

Figure 5: friction of the piston assembly at 2000rpm engine speed for both oils at 70°C and 90°C oil temperature.

It is interesting to note that the piston assembly closely resembles the behavior of the two investigated lubricants. Both oils A and B belong to the same cold temperature SAE viscosity class which yields almost identical piston assembly friction at 70°C oil temperature. For 90°C oil temperature, the two oils yield significantly different friction levels which reflect their different hot temperature SAE viscosity grades. Notably, oil A gives a significant reduction in friction over the entire engine load range studied.

Main and big end bearings of the crank train

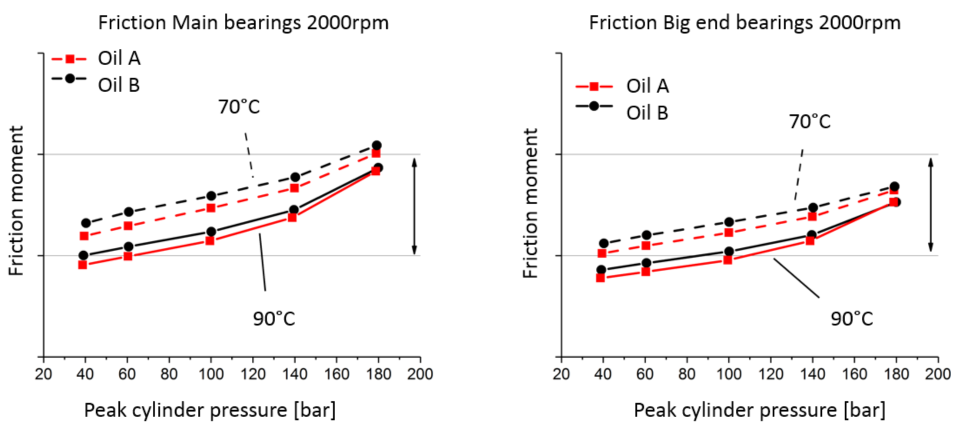

Figure 6: friction of the main bearings (left) and big end bearings (right) at 2000rpm engine speed for both oils at 70°C and 90°C oil temperature.

The journal bearings of the crank train are considered to represent the 2nd largest contributor to engine friction following the piston assembly. The friction power losses of the big end and main bearings obtained from the accurate simulation method are shown in Fig. 6. For the journal bearings the differences in friction between oil A and B are very small. The reason for this is the fact that lower viscous oils yield lower bearing temperatures which reduce the effectiveness of the lower viscous oil.

Comparison of the friction contributions of the individual subsystems

Now that all major sources of base engine friction have been analyzed, it is worthwhile to obtain a more global perspective by comparing these sources to each other.

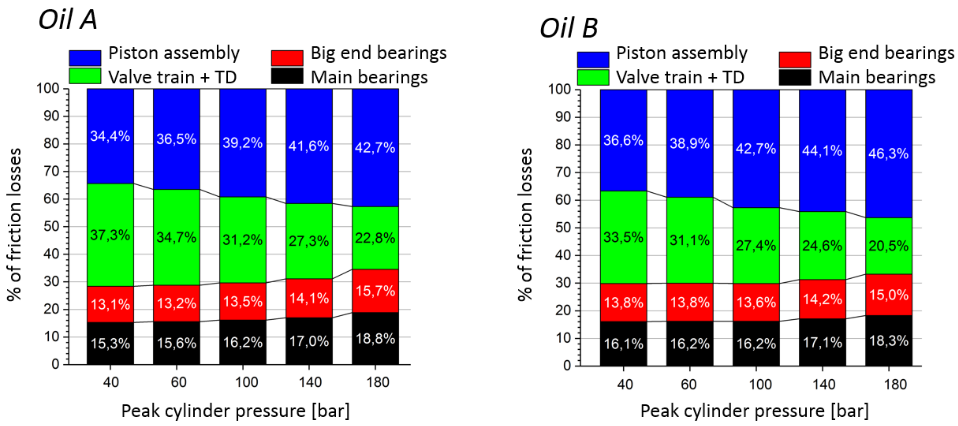

Figure 7: relative contributions of piston assembly, valve train with timing drive and crank train journal bearings at 2000rpm engine speed for both oils at 90°C oil temperature (oil A on left, oil B on right).

Fig. 7 shows a direct comparison of the individual friction losses of the subsystems piston assembly, big end bearings, main bearings and valve-train/timing drive for both oils, A and B, for 2000rpm engine speed and 90°C oil temperature. While for both lubricants the results are similar, they show interesting details. For idle operation, the combination of valve train and timing drive dominates the friction power losses for Oil A and generates more friction than even the piston assembly. This, however, is not the case for Oil B where the piston assembly is for all studied cases the largest friction source. Comparing the results for Oil A and B, it becomes evident that for full load operation with the highest peak cylinder pressure the piston assembly contribution is almost 4% higher for Oil B (46.3%) than for Oil A (42.7%). Therefore, for the studied lower viscous Oil A, the relative contribution of the piston assembly is reduced and optimizations for this subassembly become less effective.

How the testing cycle affects the friction evaluation

So far, the detailed friction evaluation revealed that lower viscosity lubricant not simply lowers the friction level of the engine, but it can e.g. also shift a bit the relative contributions of the piston assembly and the crank train journal bearings.

As the new WLTP testing procedure evaluates higher engine speeds and loads, it is of particular interest to analyze how this different testing relates to engine friction. Fig. 8 shows the friction losses for the piston assembly for two different engine speeds and different engine loads.

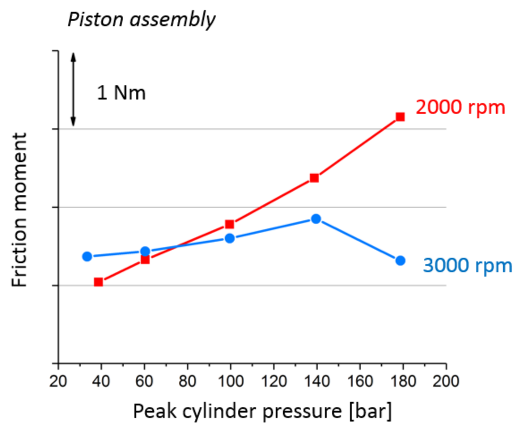

Figure 8: friction of the piston assembly at different engine speeds for oil A at 90°C oil temperature.

As can be seen from this figure, the engine load affects fundamentally the friction of the piston assembly and how it changes with increasing load. While for 2000rpm engine speed the friction of the piston assembly is continuously increasing with engine load, the piston assembly becomes almost independent from engine load for 3000rpm engine speed. A possible explanation for this observation is that for higher engine speeds the inertia forces dominate the friction behavior; the gas forces – that increase with the peak cylinder pressure - counteract these forces and yield the observed trend.

The WLTP test procedure has a higher average velocity and tests higher engine loads, so such effects directly affect the test result. For the NEDC test with its very large cruising time (34% of the total time) the friction at low engine loads had an important influence on the evaluated fuel consumption. As can be seen in the Fig. 8, at low engine loads (less than 80bar peak cylinder pressure) the friction of the piston assembly is indeed lowest for 2000rpm. For the WLTP testing procedure higher engine loads gain importance which might require a reevaluation of established engine friction optimization technologies targeted at low speed/low load operation.

Summary

Engine friction has been in the focus of engine development for many decades and modern engines are commonly already on a highly optimized level. To yield further significant fuel economy savings by friction reduction requires the use of sophisticated testing and simulation techniques. Many small separate optimizations need to be combined to obtain a considerable benefit. Detailed knowledge is required on how these single optimizations interact with each other and behave with engine speed and load as otherwise unintended side effects may be possible. For this task, a new testing technique and simulation method have been combined to investigate in detail the friction for Renault's Energy dCi 130 engine. With the new WLTP testing procedure being introduced, the relevant engine operating points have shifted and require the reevaluation of established friction reduction technologies.

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