

Application of a Split Lubrication Gasoline Engine to the Screening and Understanding of Friction Modifier Behaviour

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ABSTRACT

A series of viscous and surface friction modifier additives has been studied in a modified SI engine with separable valve train lubrication. From the results, it has been possible to classify the hydrodynamic or boundary lubrication nature of the observed effects for a series of lubricant additives. It is shown that the frictional benefit of a given additive depends not only on the engine operating condition but also on the engine components on which it is acting. For some additives a fuel economy benefit can switch to a disbenefit as operating regime changes and a different aspect of the additive properties becomes important. Such observations are rationalised in the context of conventional lubrication theory.

INTRODUCTION

With increasing global challenge on CO₂ emissions, there is enhanced focus on the opportunities presented by fuel and lubricant additives in delivering automotive fuel economy. One such class of additives are those designed to reduce friction and can be formulated directly into the lubricant or have the potential to be delivered directly to the piston assembly via the fuel under normal operating conditions [1, 2]. The identification of such options is greatly assisted by improved experimental tools that can not only screen additive candidates but can also build understanding of their mechanism and location of action.

LUBRICATION REGIMES

Friction in an engine arises from two main components. One involves surface contact and the other arises from viscous loss in liquid lubricating films. The relative contributions of the two sources of energy loss vary according to the lubricant properties, the temperature, the surface roughness and the relative speed of motion of the two surfaces. This non-linear behaviour is often summarised in terms of a Stribeck

diagram depicted in Figure 1. The horizontal axis can be envisaged as representing lubricating film thickness. On the left side of the diagram, the boundary regime, lubricating films are much thinner than the surface roughness and there is significant surface contact. On the right side, the hydrodynamic regime, the surfaces are well separated by thick liquid films and friction is dominated by viscous or hydrodynamic loss. In between, there is the so-called mixed regime where lubricating film thickness is comparable to surface roughness and friction has contributions from both mechanisms.

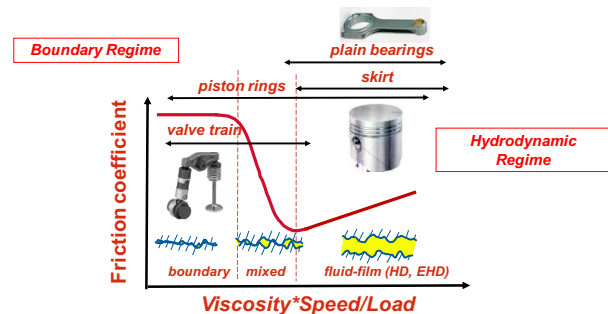


Figure 1. Lubrication Regimes: stylised Stribeck Diagram

In practical terms this means that different engine components can operate under different lubrication regimes. As illustrated in the figure, the sliding cam-tappet interface operates in the boundary regime whereas plain bearings, such as those associated with crankshaft, typically operate in the hydrodynamic regime. In reality the contribution of these different regimes can change as temperature and relative surface motion changes with engine operating condition.

Figure 1 can be used to illustrate the two key means of reducing friction in the formulation of fuel economy lubricants. One approach is to reduce viscosity which, from the figure, can be seen to be only effective for engine components operating on the right side of the frictional minimum in the Stribeck diagram. The alternative approach, the use of surface friction

modifier materials, would be expected to be efficient for components operating on the left side of the Stribeck minimum under boundary or mixed lubrication regimes. Friction modifier additives operate in engine regimes where there is surface to surface contact and these are expected to be mainly the valve train and piston assembly.

To explore the interplay between viscometric and surface friction effects and the impact of the two main sites of boundary friction under true operating conditions, a modified engine was prepared. This not only allowed detailed study of the impact of friction modification but also allowed separation of the piston assembly friction modifier response from that of the valve train.

EXPERIMENTAL

The key experimental tool for the work to be described was a highly-instrumented Ford 2.0l Zetec-R engine modified with a special split lubrication system to allow isolation of the cylinder head lubrication.

ENGINE

Engine	Ford Zetec 2.0l I4 PFI Normally Aspirated
Valves	DOHC direct acting cam on tappet
ECU	Motec aftermarket unit

SPLIT LUBRICATION DESIGN

The design criteria for splitting the valve train from the engine's main lubrication system were:

- Maintain lubricant pressure, temperature and flows to original design values
- Facilitate simple switching between whole-engine and separated valve train configurations
- Eliminate possibility of crossover between separated lubricant systems
- Minimise changes in mechanical loading on the engine when switching configurations
- Maintain constant sump volume when switching configurations

These requirements were met by three key design features:

- Interception of the main oil gallery feeding the Zetec head
- Interception of head oil drains and conveyance of oil to a manifold external to the engine

- Use of a separate electric lubricant pump and tracking pressure regulator to feed the valve train and ensure that valve train feed pressure is matched to oil gallery pressure in the cylinder block.

Although simple in concept, the implementation was not without significant challenges, particularly in the interception of the head drain channels within the very limited and obstructed space available.

A functional schematic of the split lubrication system is shown in Figure 2.

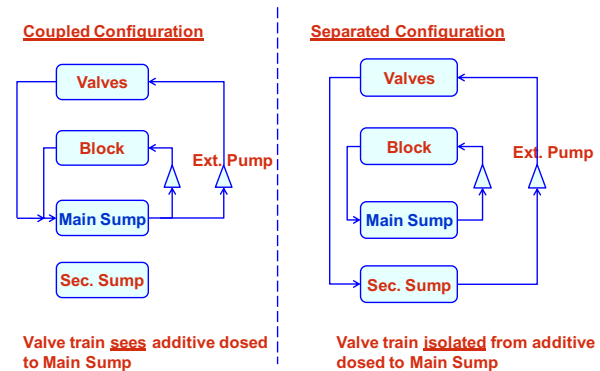


Figure 2 Schematic of Zetec Split Lubrication System

The design allows separate probing of the friction modifier response of:

- the whole engine (coupled configuration);
- the whole engine *except* the valve train (separated configuration);
- the valvetrain alone (not used in this work);

The integrity of the split lubrication design was confirmed by addition of an elemental marker to a lubricant to show the absence of crossover in the separated configuration.

Prior to conducting the main experimental programme, the modified engine underwent 100 hours break-in utilising the stages listed in Table 1.

EXPERIMENTAL PROCEDURE

The methodology adopted was to measure the impact of additive addition to the crankcase of the Zetec system whilst the engine was running. From experience this has been found to give a greater precision in determining friction modifier responses than the alternative approach of stopping the engine and changing lubricant.

Each experiment followed the scheme shown in Figure 3 in which all measurements are made over 17 evaluation cycles through each of four steady-state engine conditions (Table 1). These were chosen to represent core and extreme operating

Lubricant Flush	Lubricant Stabilisation	N Evaluation Cycles	Additive Addition	P Evaluation Cycles
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Content of Evaluation Cycle

Stabilise Stg.1	Measure Stg.1	Stabilise Stg.2	Measure Stg.2	Stabilise Stg.3	Measure Stg.3	Stabilise Stg.4	Measure Stg.4
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Figure 3. Generic Experimental Scheme. N=17, P=17 was employed. The stabilisation phase is typically 3 minutes and the measurement phase is 1 minute.

points for the European NEDC emissions drivecycle for a medium-sized passenger car. The table also shows accompanying weighting factors representing the effective proportion of time spent at each of these four key points derived from a coarse residency time analysis of the speed & torque points visited during a hot NEDC¹.

The measurement phase was preceded by three lubricant flushes and an oil stabilisation (degreening) period of 16 hours during which the engine was cycled through the same four load stages to achieve stability in BSFC & FMEP response.

Table 1 Key Experimental Operating Conditions

Stage	Speed (rpm)	Brake Torque (Nm)	Oil Gallery (°C)	Coolant (°C)	Hot NEDC Weight (approx)
1	1500	40	93	95	0.67
2	1500	80	93	95	0.24
3	3000	40	93	95	0.01
4	3000	80	93	95	0.08

The additive addition phase involved 'on the fly' crankcase dosing (main sump) in which a small sample of lubricant was withdrawn and used to prepare an additive concentrate which was re-introduced to the crankcase whilst the engine was operating at Stage 1. Following this, the engine was returned to its four-stage cycling sequence. A lubricant sample was taken from the crankcase for viscometric evaluation at the end of test.

The main observables for all experiments were:

- gravimetric fuel flow (to determine BSFC)
- cylinder pressure (for determination of mechanical friction via FMEP)

¹ This, of course, neglects the impact of the lower temperature phases of a conventional cold-start NEDC but nevertheless provides indicative real-world fuel-economy improvement.

Critical controls were:

- engine speed/brake torque
- cell, oil and coolant temperatures

Figure 4 shows typical output for the engine total mechanical friction in terms of Friction Mean Effective Pressure (FMEP) response at Stage 1 resulting from the sequence depicted in Figure 3 with additive addition after 17 cycles. FMEP was calculated from cylinder pressure measurements (FMEP=IMEP-BMEP). The sensitivity of the approach is apparent in the significant shift in FMEP mean caused by the additive.

The response to the additive was obtained by comparing the mean signal of interest over all 17 cycles before the additive addition with that of the post-additive cycles.

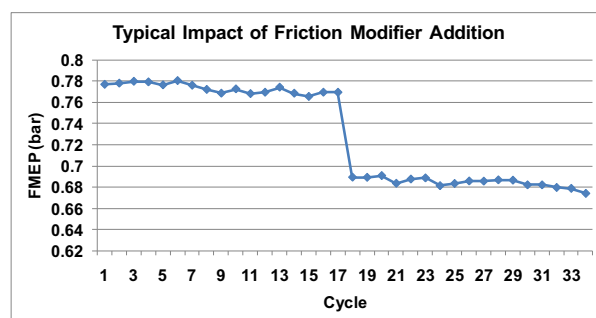


Figure 4. Typical output showing steady-state FMEP response to additive addition to the crankcase

ADDITIVE EVALUATION

A series of organic, surface-active additives known to show friction modification behaviour in engines and exhibiting a range of molecular weights were chosen. In addition, a base fluid component, with a low molecular weight was selected in order to provide viscosity reduction (dilution) but not surface modification. These materials are listed in

Table 2 together with dosages that were found to give a friction response which did not increase with further dosage (except in case of E).

The additives were evaluated in two experimental sets grouped by molecular weight, as indicated in the table. Additive B was present in both sets to provide a reference point. In each case, the chosen lubricant was a friction-modifier free 10W-40 Shell Helix Plus drawn from two batches and the fuel was sourced from a common batch of unleaded RON 95 gasoline.

The experiments were designed with two key factors of additive type and engine configuration and run as a randomised, full factorial design with duplicate tests. A null experiment, or dummy additive addition, was included as a control level within the additive factor.

Table 2. Additive Set

ID	Mol. Wt.	Dose (mass %)	Set	Description
A	Medium (ca.700)	1%	1	Nonionic surfactant
B	Low (ca. 360)	2%	1 & 2	Nonionic surfactant
C	High (ca. 1000)	2%	2	Nonionic surfactant
D	High (ca. 1000)	2%	2	Nonionic surfactant
E	Low (282)	10%	1	2 Cst. Fluid

RESULTS

Each experimental matrix was analysed as a complete set by fitting a linear model using ANOVA techniques. The resultant mean responses were then ratioed to those for the control level (null) to give percentage reduction in BSFC and FMEP. The fitted model was refined to include atmospheric pressure, fuel temperature and manifold air temperature as covariates to compensate for the significant influence of these variables. An illustration of the impact of these covariates is detailed in Appendix A.

MEASURED ENGINE QUANTITIES

The main experimental results will be presented in the form of fractional changes in measured quantities and so Table 3 is provided for baseline context.

Table 3. Engine Operating Characteristics (prior to additive addition)

Stage	BSFC (g/kWh)	IMEP (bar)	BMEP (bar)	FMEP (bar)	PMEP (bar)
1	373	3.260	2.513	0.747	0.579
2	282	5.865	5.027	0.838	0.380
3	360	3.555	2.513	1.042	0.585
4	274	6.131	5.027	1.104	0.422

BSFC RESPONSE – LOW MOLECULAR WEIGHT ADDITIVES

The BSFC results for the matrix are shown with respect to the whole engine (with valve train) and valve-train excluded (without valve train) configurations in Figure 5.

Focusing initially on the response of 1% A and 2% B we see similar gross features:

- Similar magnitude of response
- Response is greater at low load and low speed
- Valve train involvement enhances response, particularly at low speed (approximately by a factor of 3)

The latter point is in line with previously published observations involving motored engine frictional work [2].

The significantly different response to a pure viscosity reduction caused by E is noteworthy. Without the valve train, BSFC benefit is essentially independent of speed and load. With the valve train, the response is small/negative at low speed and small/positive at high speed.

Using the Hot NEDC weighting factors of Table 1 to combine the observed BSFC improvements (with valve train in Figure 5), it is possible to approximate the expected equivalent vehicle fuel economy improvement as ca 1.47% for a *hot* cycle.

FMEP RESPONSE – LOW MOLECULAR WEIGHT ADDITIVES

Engine-averaged FMEP response provides a more direct means of observing mechanical friction. For this reason the responses are larger when compared to those in BSFC terms. Consequently it provides a more sensitive means of exploring additive response.

In approximate terms, an upper value for the BSFC reduction at a given engine operating condition can be estimated as :

$$\text{BSFC reduction} =$$

$$\text{FMEP reduction} * \text{FMEP}/(\text{IMEP} + \text{PMEP})$$

Comparing Figure 5 & Figure 6 it can be seen that BSFC reduction tends to be below this estimate. This is due to confounding factors such as increase in pumping losses (throttle closure) and change in spark timing resulting from a change in engine operating point when running with fixed brake output. In reality this is the situation which holds in real-world fuel economy evaluation in drive cycles such as the NEDC.

In general, the observed speed/load and configurational dependence seen in Figure 6 show the same directional behaviour as that of BSFC i.e. the responses in separated and coupled configurations appear to decrease with speed, as does their ratio. In detail, however, the relative ranking of additive effects in each of the BSFC and FMEP cases is not identical. The reason for this is not clear, although the signal-to-noise of the FMEP data is expected to be superior.