

Valvetrain Friction - Modeling, Analysis and Measurement of a High Performance Engine Valvetrain System

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ABSTRACT

Engine efficiency is one of the key aspects to reduce CO₂ emissions. Lamborghini S.p.A. has focused attention on the engine friction modeling, analysis and measurement to understand and control the phenomena. To reduce friction it is necessary to improve understanding of the behavior of the engine components and to pay attention to detail at every tribological contact. The valvetrain can make a significant contribution to whole engine friction especially at low engine speed and this is particularly true for a high speed sports car engine. Direct acting valvetrains are often used for this type of engine to minimise the moved mass and so enable high speed operation. However the sliding contact between the cam and tappet results in higher friction loss than the roller finger follower valvetrain used on many modern passenger car engines. In addition, the high maximum engine speed demands a large valve spring force to maintain contact between cam and tappet. The large spring force can lead to increased valvetrain friction at low engine speed when the inertia force is low. Thus the development of calculation methods to quantify friction of direct acting valvetrains and support the design of components is important.

This paper describes the use of advanced mathematical models to quantify power loss at cam/tappet contact, tappet/bore contact and camshaft bearings.

The mathematical models are sufficiently detailed to capture the major influencing factors while being quick enough to use to enable engine designers to make decisions in the required time frame. This paper compares calculated and measured

friction data for the valvetrain of a high speed passenger car engine as tested on a motored cylinder head test rig. The system friction was measured and calculated across the operating speed range with different oil supply temperature. The effect of diamond like carbon (DLC) coatings on the tappets was quantified by measurement and analysis.

INTRODUCTION

Valvetrain friction makes a significant contribution to the whole engine friction loss, especially at low engine speed. The total amount depends on the valvetrain type, engine architecture, engine speed and lubricant temperature among many other factors. In [Reference 1](#) a graph, derived from motored strip test data, shows the valvetrain for a modern spark ignition car engine contributing ~35% to total friction at 1000 rpm and ~10% at 6000 rpm. This type of test typically includes the timing drive as part of the valvetrain group. On an engine for a high performance sports car the friction of the valvetrain at low engine speeds is relatively high (compared with valvetrains for normal passenger car engines) for the following reasons. This kind of engine typically uses a direct acting valvetrain because it is very hard to make a roller type valvetrain light and stiff enough to enable very high engine speed (>8000 rpm) without significantly compromising the cam profile design. Direct acting valvetrains have inherently higher friction than roller types due to the sliding contact between the cam and tappet and this is also indicated in another graph in [Reference 1](#). In addition, sports car engines typically have a high bore/stroke ratio (>1.2) and large valve diameter to give high performance. The large valve head diameter leads to high valve mass which, combined with the high maximum engine

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speed and large valve lift, leads to the requirement for high valve spring forces to ensure acceptable high speed dynamic behaviour. At low engine speed the high cam/tappet contact force leads to high friction loss.

At the cam/tappet contact the power loss is a function of contact load, sliding velocity and friction coefficient. The contact load is a function of spring force, inertia force and vibration. The sliding velocity is determined by the cam profile, the geometry of the mechanism and the engine speed. The friction coefficient is a complex variable dependent on contact force, sliding speed, lubricant film thickness, lubricant viscosity, lubricant temperature, surface texture, materials combination etc. The lubrication regime at the cam/tappet contact varies rapidly during the cycle from hydrodynamic on the cam flanks at high engine speed to boundary lubrication on the nose at low speed.

At the camshaft bearings the power loss is a function of shaft speed, valvetrain load, bearing diameter, bearing length, oil viscosity and bearing clearance. The lubrication regime at the cam bearing is predominantly hydrodynamic but may involve some asperity contact friction at low engine speed with hot oil.

To enable optimisation of the valvetrain system and component design it is necessary to be able to predict the various contributions to valvetrain friction at the design stage so that friction level at low speed (and effect on fuel consumption) can be weighed against dynamic performance (and effect on engine power) at high speed.

This paper describes research work performed by Lamborghini and Ricardo to evaluate the potential of software for calculation of the losses in a direct acting valvetrain. Measurements of mean camshaft drive torque were made on a motored cylinder head under controlled conditions. The effect on friction loss of oil supply temperature, oil supply pressure and tappet coating were measured across the operating speed range and some measurements were also taken with camshaft bearings present but valvetrains removed in an attempt to separate the losses due to camshaft bearings. The paper presents a record of the calculation input data used to obtain a good match to the measured data.

MEASUREMENT EQUIPMENT

A motored cylinder head from a V12 engine was used for this investigation (see [Figures 1](#) and [2](#)). The intake camshaft only was driven by an electric motor connected to the end of the camshaft. The exhaust camshaft was not present. The camshaft operated 12 intake valves via hydraulic tappets. The drive torque was measured by the electric motor. The losses in the electric motor were measured by operating it with no

connection to the camshaft and these losses were subtracted from all subsequent measurements of drive torque.

The rig was also used to take measurements of valve motion using a laser system. This was used to verify the physical dynamic behavior of the valvetrain and to check the fidelity of the dynamic analysis and is not the subject of this paper.

Measurements of camshaft drive torque were made across the normal operating speed range for the following builds.

- Whole valvetrain with standard steel tappets
- Whole valvetrain with steel tappets with diamond-like carbon (DLC) coating
- Camshaft only with no tappets or valves present

In each case the measurements were made with oil supply temperature controlled to 50°C, 70°C, 90°C and 110°C and oil supply pressure controlled to 3 bar.

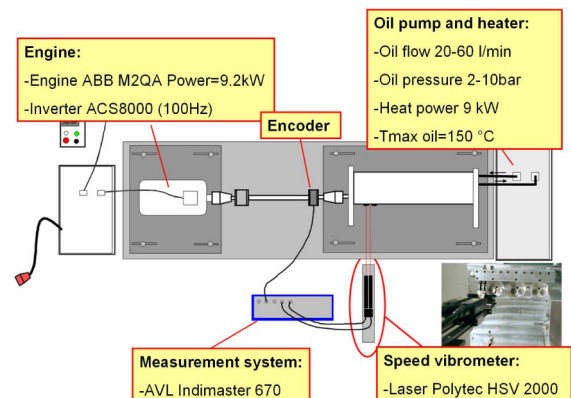


Figure 1. Test rig and measurement system



Figure 2. Test rig photo

MATHEMATICAL MODELS

A dynamic model of a single valvetrain was modelled using Ricardo Software VALDYN as shown in [Figure 3](#).

This valvetrain was represented as a series of lumped mass/inertia nodes connected by stiffness and damping values. The

cam node was suspended on a stiffness element representing camshaft bending stiffness and camshaft support stiffness. The tappet top stiffness was modelled as a function of eccentricity of the cam/tappet contact. The hydraulic tappet was modelled as two mass nodes connected by a special element to account for the action of the high pressure chamber, the expansion spring and the check valve. The tappet was connected to the valve using a lash stiffness element representing the stiffness of the valve stem between the tip and the centre of mass. The valve and spring retainer were modelled as a single mass node. The valve node was connected to ground by another lash stiffness element representing the valve head and seat stiffness. The valvetrain had a double spring pack and each spring was modelled using 4 mass nodes per coil connected by stiffnesses.

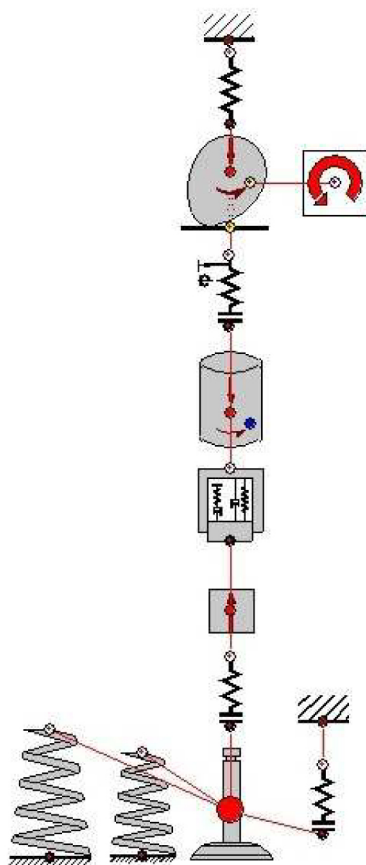


Figure 3. Valvetrain model

The cam profile was designed to meet the many conflicting requirements for engine breathing, acceptable durability, high speed dynamics etc. The spring pack was designed to maintain contact between cam and follower at high engine speed. Figures 4 and 5 show the calculated dynamic valve lift and dynamic valve acceleration and against crank angle at camshaft speed of 4000 rpm. These graphs demonstrate the good dynamic response even at high engine speed.

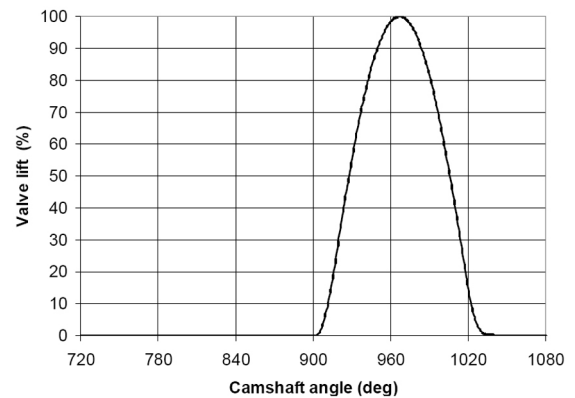


Figure 4. Calculated dynamic valve lift at 4000 rpm camshaft speed

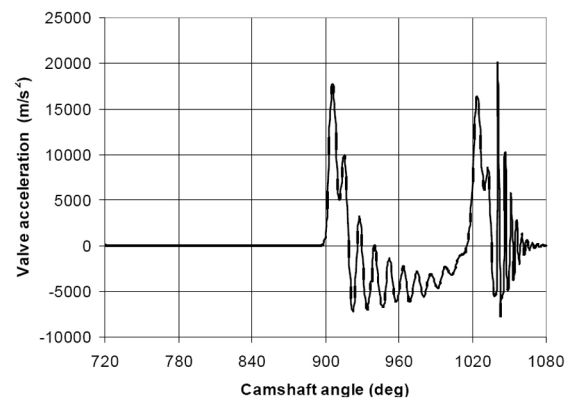


Figure 5. Calculated dynamic valve acceleration at 4000 rpm camshaft speed

The calculation model has the capability to make calculations of friction loss at the cam/tappet contact based on the methods described in [References 2 and 3](#) but dynamic forces were used instead of pseudostatic forces used in those references. There is not enough space in this paper for a full treatment of this calculation process which is very involved and the following description is intended as a rough guide only. At each time step the program calculated the cam/tappet contact force and the dimensions of the contact ellipse based on Hertzian contact theory. The minimum oil film thickness at the cam/tappet contact was calculated using isothermal elastohydrodynamic theory ([Reference 2](#)) and compared to the combined surface roughness. If asperity contact was predicted to occur then the Greenwood and Tripp algorithm (see [Reference 4](#)) was used to determine the asperity contact friction loss using a user-specified friction coefficient. In addition the energy required to shear the thin oil film in the Hertzian contact was also calculated accounting for the non-Newtonian behavior of the oil film under the influence of high pressure between the contacting surfaces. Thus the total friction due to oil shear effects and asperity contact effects was determined at each point in the simulation and summed to give the mean total friction torque at each speed. In

addition the losses at the tappet/bore due to tappet translation and tappet rotation were calculated using the methods described in [Reference 3](#). The cam/tappet friction and tappet/bore friction algorithms require the special program inputs shown in [Appendix 1](#) in addition to the other data required to calculate dynamic forces and motions at each contact.

The camshaft bearing friction was calculated using Ricardo Software ENGDYN. Initial calculations were made using the following assumptions.

- each bearing was assumed identical
- each bearing was initially loaded only by a rotating force due to the out-of-balance mass of the two adjacent cam lobes (so no valvetrain actuation forces)
- camshaft journal and housing were assumed to be rigid
- camshaft journal and housing were assumed to be cylindrical shape
- finite volume hydrodynamic bearing solver was used
- Greenwood and Tripp algorithm (see [Reference 4](#)) was applied if mixed lubrication regime was predicted (if predicted oil film thickness is similar to combined surface roughness)

The bearings were also analysed with increased loading due to the action of the two adjacent valvetrains as calculated by VALDYN at each speed. Some of the program input data for this analysis was difficult to obtain with certainty and the following strategy was used to identify the finally chosen values during this study.

- Validate the valvetrain dynamic model by comparison of predicted valve motion with measurements (not reported here)
- Choose values for those special parameters required by analysis models (using data from component drawings and oil specification data where possible)
- Calculate camshaft bearing friction loss with no valvetrain actuation loads and compare with measured data for camshafts only
- Calculate camshaft bearing friction loss with valvetrain loads included and subtract these values from the total measured camshaft friction torque with valvetrains present
- Calculate friction due to cam/tappet and tappet/bore contacts and adjust asperity contact friction coefficient to obtain an acceptable match to the whole valvetrain measured data
- Test the response of the model to variations in asperity contact friction coefficient and oil supply temperature and compare with measured data (see later sections)

To obtain the best match between calculation and measurement it was found necessary to introduce a variable

coefficient of asperity contact friction and the effect of this is shown in the section comparing measured and calculated results. In addition different values of this friction coefficient were used to match the measured data for standard steel tappets and DLC tappets.

RESULTS

MEASURED RESULTS

The first build involved measurement of camshaft drive torque with standard steel tappets. The results are shown in [Figure 6](#). The y-axis shows normalised drive torque where 100% is taken as the highest measured value (at 350 rpm camshaft speed with oil supply temperature of 110°C). This same scale was used for all measurement graphs. The graph shows the classic response of a direct acting valvetrain as shown by many other papers in the literature (for example see [References 3,5,6](#) and [7](#)). Friction torque reduced with increasing camshaft speed. Increasing the oil supply temperature gave increased friction at low speed and reduced friction at high speed but the effect was quite small. No significant change in behavior at very high engine speed was observed - friction torque continued to decrease.

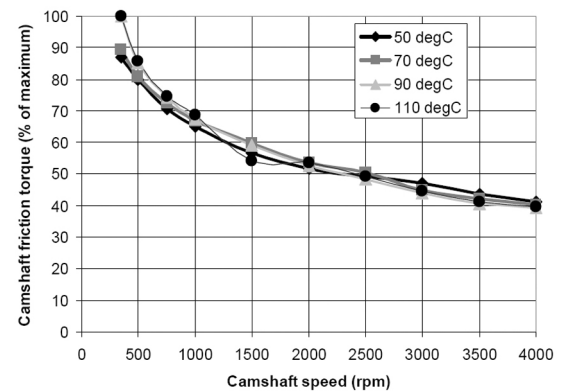


Figure 6. Measured camshaft drive torque with standard steel tappets

In the next build the steel tappets were replaced by tappets coated with diamond-like carbon. The results are shown in [Figure 7](#). The coated tappets gave a lower friction level across the speed range compared with the steel tappets and a slightly greater response to varying oil supply temperature.