LUBE-TECH

NO.11

DESIGNING AUTOMOTIVE ENGINE LUBRICATING SYSTEMS

1. Introduction

The primary purpose of the lubrication system is to lubricate sliding surfaces and reduce friction losses in the engine, whilst secondary issues are involved with heat transfer. It is recognised that the lubricating oil, which, after fuel, is the most variable engine 'part'of any internal combustion engine, and is probably the most open to abuse. The possibility of the use of inferior specification oils, and/or for periods exceeding the recommended drainage intervals, coupled with the possibility of excessive engine wear, has led to considerable emphasis being placed in lubrication system design and development.

The optimisation of engine lubrication systems traditionally involved much hardware testing, which was expensive and time-consuming.

Basic lubrication systems use a positive displacement oil pump feeding all bearings with full flow oil filtration, whilst more complex systems can include pressure relief valves, by-pass filtration, piston cooling, hydraulic lash adjusters and hydraulically activated cam phasing mechanisms. The main factors to be considered in the lubrication system design process are flow balancing and pump sizing (volumetric flow rate of oil) required for satisfactory operation. Items such as hydraulic lash adjusters and chain tensioners also require a minimum oil pressure at low engine speeds for adequate operation, which can have a significant affect on the oil pump size. Traditionally, the volumetric capacity of the oil pump (and pump speed) was determined according to the power rating, swept volume of the engine and design experience with a further allowance to account for the worn engine condition, but as engine lubrication systems became more complex a more analytical approach was required. A one dimensional fluid flow network can be generated to simulate the oil flow with specific models for galleries (pipes), filters, pressure controlled valves, bearings, piston cooling jets, orifices etc and further used to optimise the flow characteristics (oil velocity, volumetric flow rate and pressure) in the system.

2. Design of Lubrication Systems

2.1 GENERAL CONSIDERATIONS FOR THE LUBRICATION CIRCUIT

Typically oil velocities in excess of 3m/s in the pick-up pipe can result in cavitation reducing engine and oil pump life. However, at low temperatures the pressure in the lubrication system is high due to high oil viscosity and a majority of the oil is then re-circulated or directed back to sump via the pressure relief valve. To illustrate the difference in viscosity, for a typical 15W40 oil the kinematic viscosity at 120°C is 8.45 cSt increasing to 8546 cSt at -20°C.

Aeration is another of concern, which is caused by crankshaft churning (exacerbated by too high sump levels), oil break up (typically by a chain), high oil return velocities (from the cylinder head) and long suction lengths.

If the oil level is too low the pick-up pipe is not fully flooded under all conditions, again causing air bubbles to be mixed with the oil by the oil pump and circulated around the lubrication system. For drainage, if the oil velocity is in excess of 0.5m/s, air is mixed into the oil, which is the main reason for re-circulating oil from the pump relief valve rather than directing it straight back to sump at high speed.

Another important criterion to consider is the engine running temperature (or the average oil temperature in the sump), which ranges from 120°C-150°C for light duty automotive applications. If localised temperatures are too high, i.e. above 220°C for mineral hydrocarbon oils and above 300°C for synthetic oils, the oil is likely to carbonise into solid matter, which can accumulate in critical areas of the engine.

2.2 PUMP SPECIFICATION

Pump delivery requirements can be estimated using typical demand data for all component oil consumers, together with an allowance for worn conditions. As some of the bearings will wear, the clearances between the rotating shaft and bearing shell increase, thereby allowing more oil flow (although the minimum bearing flow requirement does not change) and therefore upsetting the flow balance in the circuit. Other parts of the engine which have not worn to the same degree will still require the same oil flow for satisfactory operation, hence the demand on the oil pump increases. However, for crankshaft bearings the flow requirement is based on temperature rise through the bearing and as the engine wears the oil temperature rise is reduced since increased clearances allow more oil flow.

The pump itself is most commonly a positive displacement oil pump of a gear or gerotor design and driven by the crankshaft directly or via a gear or chain. Current designs also have an integral relief valve (pressure actuated via a spring) to prevent very high oil pressures building up in the system, particularly at cold start.

2.3 RELIEF AND BY-PASS VALVES

Typically the oil pump relief valve would be set to open at

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4-5 bar with an additional control valve to regulate flow to piston cooling jets set at around 2 bar. Since the thermal loading on the pistons is generally a function of the engine load, flow through the piston cooling jets is not necessary at idle or low load. Hence the pressure control valve here has the opposite purpose, i.e. to allow oil to flow to the cooling jets at a minimum oil pressure, with the jet design and direction governing the flow to the underside of the pistons over the speed range.

Many modern engines are also designed to include a by-pass valve on the full flow filter, set to open at approximately 1-1.2 bar, with the main purpose being to protect the filter and prevent oil starvation in the event of a blocked filter.

2.4 FULL FLOW AND BY-PASS FILTRATION

Light duty diesel and gasoline engines tend to have a single full flow screen filter which can typically remove in excess of 90% of particles greater than 30μm (depending on manufacturer) with lower removal rates for smaller particles. In addition some modern passenger car diesel engines also have fine element by-pass filtration to capture the smaller particles.

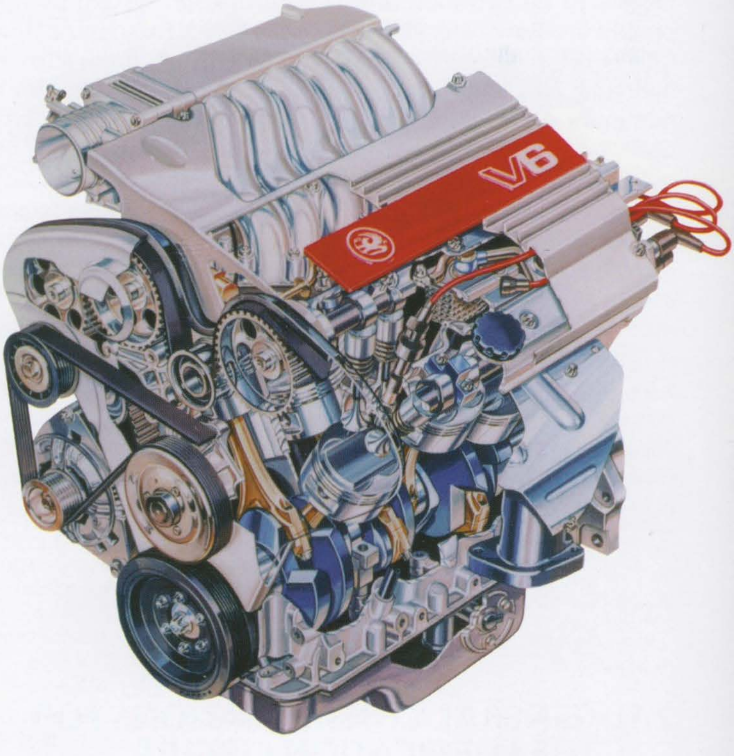
Different types of filters also affect the oil flow and pumping since they all exhibit different flow characteristics. Typically 10-15% of the oil pump delivery is reserved for screen by-pass filtration and 5-10% for centrifugal by-pass filters. Centrifuges are particularly effective for 'heavy' particles, due to the method of operation - oil flows through rotating nozzles into a cylindrical chamber where the particles denser than the lubricating fluid gather at the outer surface. However relatively high oil pressure is required for effective filtration since the important parameter is the angular velocity of the rotating nozzles which is governed by the oil velocity.

2.5 OIL DRAINAGE INTERVALS

Safe oil drainage intervals are based on the degree of oil degradation, traditionally determined by engine and laboratory testing. Diesel engines characteristically generate particulates (e.g. soot) which contaminate the oil and generally have low particle sizes, e.g. 1-2μm, which will pass through normal full flow filters. The only effective way of removing these soot particles, which is necessary in order to reduce engine wear and extend drain intervals, is to use a centrifugal filter. Even here, since soot is of a similar density to oil (approximately 1.1 times the density of oil), removal is still difficult, requiring more oil to be by-pass filtered at high centrifuge speeds. In practice, oil drain periods depend on the amount of 'dirt' being deposited into the oil and upon additive depletion which is engine and application specific. The drainage period ultimately depends on the application, type of engine (indirect or direct injection), oil consumption, emissions control, oil specification and even oil additives. The benefit for gasoline engines is much greater than for diesels since relatively small amounts of particulates (soot) are produced and deposited into the oil, therefore a longer oil drain period can be sustained. Studies indicate that the oil drain period for gasoline engines could be increased considerably from 10,000km recommended for many gasoline engines in production by use of an optimised filtration system and superior quality synthetic oil, up to 30,000+km.

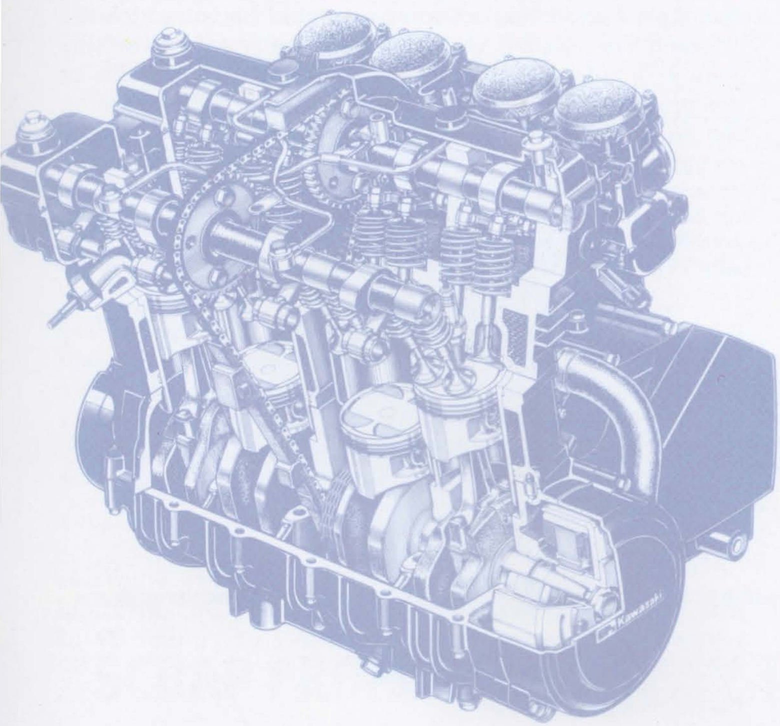
Note that for long oil drain periods the engine would still require oil to be 'topped up' in between each oil drain to account for the oil consumed. High oil consumption can be beneficial since topping-up would replace depleted oil additives. For example, Cummins have investigated a system which includes an oil reservoir for continual automatic 'top up'to extend drainage periods up to 500,000km.

There is also now a movement towards more intelligent oil servicing, with a view to the use of on-board monitors, which assess the degree of oil degradation. The four main parameters used to determine oil degradation are contaminants (soot, water, fuel, metallic debris), total acid (or base) number, oxidation and viscosity. Of these, proposals have been made to measure oil degradation by measuring the electrical properties of the oil. There are already many vehicles in use which use a simpler system of service indicators, by assessing the frequency of cold starts, and of the duration of high load operation, high oil temperature operation and length of running, as a guide towards servicing or oil change.



2.6 BEARINGS

Oil pressure and flow should be sufficient so as to give acceptable minimum film thicknesses in bearings as the oil temperature rise. To achieve this, ideally the flow should be continuous throughout the engine cycle to all bearings. Often, the flow is discontinuous since the connecting rod big end bearing is fed from the main bearing which has a 180° circumferential groove via a single drilling in the crankshaft. A better alternative would be to have an additional drilling passing through the centre of the main journal allowing oil to flow under pressure to the big end throughout the engine cycle. In fact, a minimum 180° groove with a cross drilled main journal is recommended for bearings with a feed to the crankpin for continuous oil flow characteristics although applications exist with bearing grooves less than 180° with satisfactory performance. Note that for a main bearing with a full 360° groove the oil flow to the big end is



continuous for both designs making the cross drilling in the main journal redundant.

Pressure/flow relations have been developed for journal bearings with and without an oil groove, with special consideration to 180° and complete 360° groove.

Furthermore, flow rates depend upon flow induced by the hydrodynamic pumping action and also by the 'squeeze' action resulting from the eccentricity of the journal, in addition to the flow resulting from external feed pressure. Crankshaft bearings are the most difficult to predict due to the changeable bearing loads from cylinder pressure and counterweight design. Other bearings in the engine with relatively low bearing loads are more easily predictable. Equations are used to determine the pressure flow behaviour for individual bearings in an engine as an average over an engine cycle, and also at different engine speeds. This allows improved prediction of oil demand from the lubrication system and hence a more optimised design solution.

3. Analysis Techniques for Lubrication Systems

Whilst fluid flows in engines are three dimensional the lubrication system can be simplified and treated as a series of one dimensional passages, and the method of solution adopted can be simplified to steady state and isothermal with oil viscosity and density calculated for different thermal conditions of the engine.

Traditionally engine designers have used oil pressure in specific parts of the engine to gauge acceptable lubrication with the majority of the lubrication system considered during the engine development phase. However for most critical components satisfactory performance is governed by the volumetric oil flow rate and flow balance in the system. The engine speed can be simulated by considering the oil pump delivery (taking into account volumetric efficiency), and, since positive displacement pumps are most commonly used in engine applications, oil pump delivery is theoretically proportional to engine speed. In practice volumetric efficiency varies slightly with pump speed, oil pressure and the type of oil pump. Flow characteristics over the speed range would be given by results for steady state isothermal solutions at a number of engine speeds or more accurately for the equivalent oil pump delivery.

Optimising the lubrication system is a complex task, having to consider flow balances, optimum flow requirements and specifications for relief valves simultaneously leading to an iterative process. This is further complicated for systems with oil coolers and thermostatically controlled by-pass valves.

Once a model has been analysed results for interrogation against acceptable guidelines are:

i. Volumetric flow and velocity of oil through piston cooling jets per cylinder or average flow rate per engine cycle for pistons directly fed through the connecting rod.

ii. Average flow through major bearings per engine cycle.

iii. Flow through pressure relief valves.

iv. Oil velocity in pick-up pipe.

v. Oil pressure at critical components, hydraulic lash adjusters for instance.

vi. Hydrodynamic pumping requirement (parasitic losses due to oil pumping only).

In some cases the lubrication system is required to provide a pressure (with little or no flow) for specific functions (e.g. hydraulic lash adjusters). Here, the system may be designed by using fluid power techniques and may be analysed

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separately with knowledge of the pressure distribution with engine speed from the flow analysis.

4. Case Study - Assessment of the Lubrication Circuit of a Four Cylinder Passenger Car Diesel Engine

Consider a light duty four cylinder HSDI engine for passenger car applications; the main features for consideration in the lubrication system are balancer shafts, piston cooling jets, single full flow filter, gear oil pump with integral relief valve and feeds to a turbocharger and belt tensioner. The information required to make the lubrication system flow analysis effective is:

i. Oil pump characteristic - this would be in the form of pressure flow curves at different speeds for an oil pump with integral relief valve, or volumetric efficiencies plus pressure flow curves for the relief valve in the case of a separate oil pump and relief valve design.
ii. Typical pressure flow curves for filters (both full flow and by-pass) are usually sufficient.
iii. Pressure flow curves for the piston cooling jets or jet geometry plus pressure flow curves for the check valve.
iv. Flow characteristics for the engine bearings taking into consideration load, clearance and geometry.
v. Dimensions of pick-up pipe and machined and cast-in galleries.
vi. Oil specification and average sump temperatures for a range of engine running conditions.

The results of the analysis showed expected pressure levels in the main gallery, 4.5 bar at rated speed, but with excessive oil flow through the piston cooling jets based on the engine power curve and assuming 9 litres/hour per kW as the target flow. This excessive flow has also resulted in high jet velocities providing excessive piston cooling even at idle speed conditions not to mention part load through the mid speed range.

The cause of this excessive flow is an oversized oil pump, with a theoretical output of 26 litres/min per 1000rpm and a relatively low volumetric efficiency at high engine speeds, approximately 75% giving an output of 86 litres/min at rated speed.

The easiest and cheapest solution would be to reduce the opening pressure on the relief valve since this invariably involves simply replacing a spring, although the benefits of this are minimal. The more expensive route of redesigning the oil pump for reduced capacity would be benefited by smaller packaging requirement for the oil pump and lower hydrodynamic pumping losses.

5. CONCLUSIONS

Design guidelines for the entire lubrication system are available with the potential for further optimisation through analysis. In addition optimised practical solutions would give minimum friction losses whilst still achieving satisfactory performance. A further technical benefit is the investigation of more complex lubrication systems which may not be considered otherwise.

This article is an abbreviated version of one which was prepared by Abid Mian of Ricardo Consulting Engineers, and which was presented in full in Issue 2 of Volume 51 of Industrial Lubrication & Tribology.

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NEW SURFACE COATING PROCESS DEVELOPED TO REDUCE COMPONENT FAILURE IN THE AUTOMOTIVE INDUSTRY

A new surface coating technology, developed by German company Dr. Tiliwich GmbH, is set to reduce the risk of failure of electronic or optical components used in the automotive industry.

The surface infecting fluids (SIF) coat helps prevent surface contamination of components. This happens when substances such as brake fluids, gear lubricants, and fuel residues, are deposited across a component during use. if this causes failure, vehicle owners are often forced to pay for expensive repair work and, in a worse case scenario, face the risk of vehicle failure.

The SIF coat stops fluids from spreading and residual contaminants from sticking. The application process is simple. The parts are dipped in or sprayed with 3M™ Fluorad™ coating, a primer and an anti-flocking additive. They are then dried at 80°C for two hours, a "burn-in" process that gives the coating non-stick properties without impairing the component's performance.

Typical applications of the new coating include steel axles in electromotive actuators, metal-coated surfaces, optically active elements and electromechanical sensors. These components, which are increasingly being used in the automotive industry, are getting ever smaller while having to withstand higher specific loads and upper limit thermal stresses and therefore need superior protection against SIF's.