

# Experimental Procedure for Measuring the Energy Consumption of IC Engine Lubricating Pumps during a NEDC Driving Cycle

**Massimo Rundo**

Politecnico di Torino

**Raffaele Squarcini**

Pierburg Pump Technology Italy S.p.A.

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## ABSTRACT

The paper presents an experimental procedure for comparing different families of IC Engine lubricating pumps in terms of total consumed energy in a NEDC driving cycle.

Measures are performed on a test rig able to reproduce the oil temperature profile, the lubrication circuit permeability and its variation during the engine warm-up.

The pump under test is driven by a variable speed electric motor supplying the engine velocity profile of the driving cycle. The load on the pump is generated by means of a variable restrictor controlled in a closed loop by a proper combination of speed, temperature, flow rate and pressure signals in order to replicate the typical permeability of the lubricating circuit.

The method has been applied to a 15 cc/rev crankshaft mounted gerotor pump for a medium displacement Diesel engine; the flow-pressure characteristics of the lubricating circuit at different temperatures and the oil heating rate measured on the engine have been supplied to the test rig control system.

The reference pump has been contrasted with a crankshaft mounted variable displacement vane pump with absolute pressure limiter device and an off-axis external gear pump with the same displacement and pressure setting.

Experimental tests bring to evidence the advantage of the variable displacement pump during short trips with cold start, while the external gear pump prevails at the end of NEDC cycle where highest engine speeds and oil temperatures are reached.

## INTRODUCTION

Due to increasingly stringent regulations for CO<sub>2</sub> emissions reduction, the lowering of consumed power from ancillary

equipment, e.g. the lubrication pump, is becoming of great relevance.

In the last few years a steady development was observed of new types of variable flow pumps, variable displacement vane pumps [11], [4], [16], variable timing with fixed displacement [18] and hybrid solutions [12]; beside these architectures that allow continuous control of delivered flow rate, also discrete flow control methods have been developed [2], [15]. The purpose of controlling flow rate, regardless of how this is achieved, is the removal of the relief valve that generates an energy waste.

Flow control can also provide additional advantages if coupled with pressure control in the lubrication network.

Proposed solutions consider open loop pressure controls as function of temperature [17] or speed [15], as well as closed loop controls through an additional high dynamics pressure transducer providing the feedback signal [1]. This last strategy can also be used for controlling the main gallery pressure by regulating the speed of an electric pump complementing a standard low capacity fixed displacement unit [9], [14].

The selection criterion amidst these viable solutions must account for diverse aspects such as cost, reliability, space and energy saving.

In the automotive industry a procedure based on a standard driving cycle is firmly established while assessing attainable savings from optimized solutions or from new engine components. For example, in [5] and [13] engines are tested with different ancillaries to quantify the relative saving in fuel consumption in a NEDC cycle.

This type of investigation requires engine tests on a dynamometric rig, or even on the entire vehicle. Often, the energy saving obtained by a proposed component is of the

same order of magnitude as the fuel consumption measurement accuracy of the engine test bench, thus impairing the comparison with respect to the baseline. The alternative is to focus on the component under scrutiny tested on a specific rig reproducing the operating conditions encountered during the driving cycle.

Dealing with a lubrication pump this last procedure is rather complicated to effect: one basic issue is the reproduction of a load history representative of the engine operating conditions. In fact, in the test rig the load, i.e. the lubrication network, must be simulated through a restrictor that exhibits a turbulent characteristic, experimental measurements indicate instead a quasi-linear flow-pressure characteristic due to prevailing zones with laminar flow regime. Furthermore, being the cycle originated at ambient temperature, the test must be run following as closely as feasible a law representing the oil temperature during the warm-up phase.

Consequently, owing to these difficulties, the procedure commonly followed by component manufacturers consists in evaluating the required power for some combinations of temperature, speed and pressure only; however, correlation of these data to the vehicle fuel consumption is yet another issue.

Aim of the present work is to contrast diverse types of pumps using as a screening factor the energy consumed during a NEDC cycle. A sufficiently fast and accurate experimental procedure deployed in a specific lubricating pump test rig will also be presented.

## TEST RIG

**GENERAL CHARACTERISTICS** - Fig. 1 shows the rig located in the Fluid Power Research Laboratory of the Politecnico di Torino. The rig, conforming to ISO standards (ISO 4409) and in class A precision, allows attainment of characteristics and of steady-state and dynamic performance, of low power (max 15 kW) pumps and motors. More specifically it is tuned for testing ICE lubricating pumps (crankshaft mounted and off-axis architectures) in the fluid temperature range 5÷140 °C.



Figure 1: view of the test rig

The rig, confined in a sound-proof cell is remotely controlled from an external position situated behind a window with impact resistant glasses (Fig. 2). A National Instruments real time system and a graphical user interface with a synoptic display capability [7] are implemented.

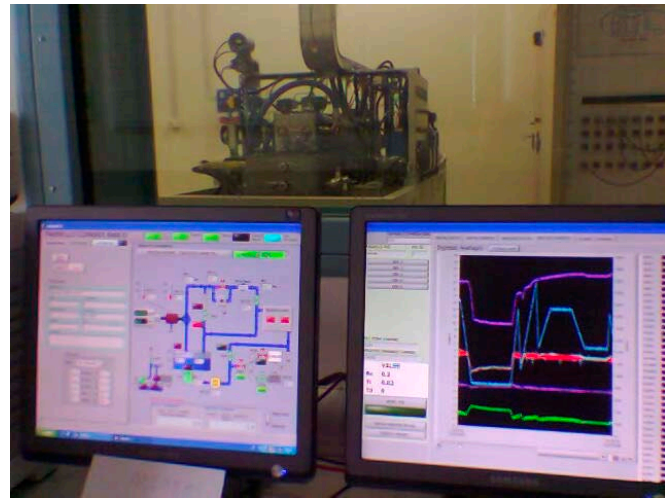


Figure 2: view of the test rig from the operator control unit

**PUMPS TEST MODE** - Fig. 3 shows the hydraulic scheme in the open circuit test mode; main components are:

- M1: variable speed electric drive.
- TF: torque meter HBM T10F with range 0÷50 Nm.
- E1, E2: pneumatic actuated segmented ball valves FISHER RF-V150.
- P1: pressure transducer<sup>1</sup> GS XPM5, with range 0÷20 bar and linearity error ± 0.25% F.S.
- FM: mass flow meter MICROMOTION CMF100M.
- T1, T2, T3, T4: PT100 thermoresistances.
- EH: 5 kW electric heating elements.
- CH: chiller.
- SC: water-oil heat exchanger.
- PU: screw pump for the conditioning circuit.

The pump under test sucks fluid from an insulated 30 liters tank that can be displaced along the three axes.

Tank temperature is obtained from the average of readings of differently located thermoresistances T1 e T2. Torque measured by the torque-meter is reduced by the quota relative to the bearings of the interface block: this contribution is evaluated through an expression derived from experimental measurements without the pump and at varying speed being temperature given from T4. The rig utilizes a synthetic base oil Shell Helix Ultra AG 5W30.

**LOAD HISTORY ON PUMP** - Starting from experimental data collected on 1.9 L Diesel engine, it was ascertained that within the following ranges for pump delivery pressure 1.5÷8 bar, fluid temperature 20÷120 °C and speed 750÷4500 rpm, the characteristic equation of the lubrication circuit can be approximated with the expression (1):

$$Q = a(T) \cdot p^{b(T)} - c(T, n) \quad (1)$$

1. this transducer is not part of the measurement equipment of the rig but rather it is fitted directly or nearby the component under test

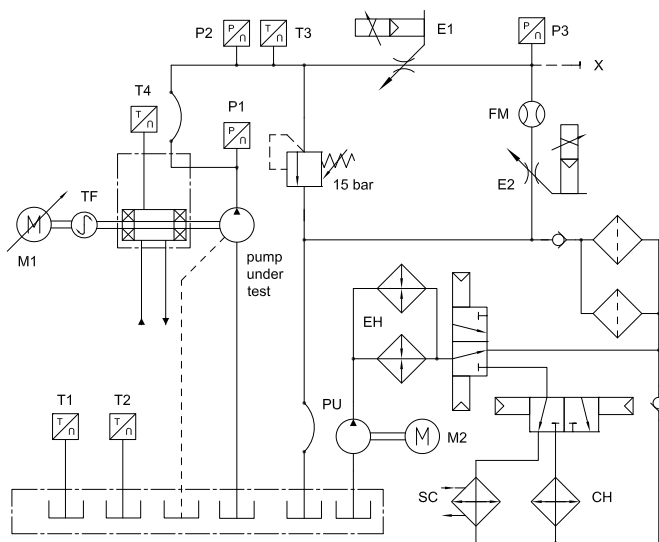


Figure 3: layout of the hydraulic circuit (simplified)

being  $Q$  the net volumetric flow rate,  $p$  the pump delivery pressure,  $T$  the oil temperature and  $n$  the angular speed. This expression also accounts for the pressure drop in the fluid conditioning unit (filter and heat exchanger). The exponent  $b$  is between 0.9 and 0.7; the higher values are relative to low oil temperatures that determine prevalent laminar flow regimes.

The term  $c$  accounts for the fact that permeability of the lubricating circuit is also an increasing function of engine speed for various reasons among which:

- the flow rate through bearings is a function of the eccentricity ratio that, in turn, for a given load varies with speed;
- the oil temperature within bearings, at equal main gallery temperature, rises with speed (up to some tens °C [19]) owing to viscous dissipation: this implies a local viscosity reduction within the bearing and an increase in flow rate;
- the oil feed pressure of the conrod bearings, at equal main gallery pressure, rises with speed due to the effect of centrifugal force on the oil within the crankshaft drillings [20]: this effect entails an increase in flow rate through the bearing.

For the engine under consideration, making use of L/min for the flow rate, bar for pressure, °C for temperature and rpm for speed, the following expressions have been obtained for  $a$ ,  $b$  and  $c$ :

$$a = 0.00018 \cdot T^2 + 0.0486 \cdot T + 1.23 \quad (2)$$

$$b = -4.8 \times 10^{-7} \cdot T^2 - 8.09 \times 10^{-4} \cdot T + 0.91 \quad (3)$$

$$c = (0.03 \cdot T + 0.74) \cdot \left(1.2 - \frac{n}{4250}\right) \quad (4)$$

With these coefficients the following figures have been obtained that reproduce the experimentally measured delivery pressure with an error of  $\pm 0.2$  bar, at equal flow rates.

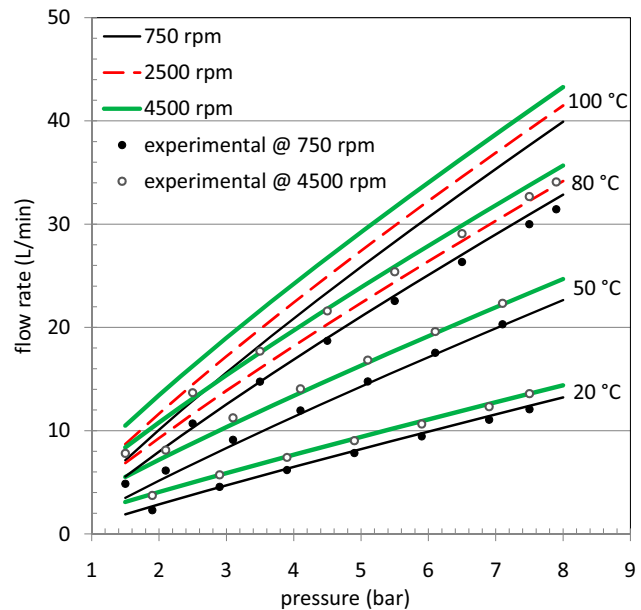


Figure 4: lubricating circuit characteristics

To replicate the curves in the test rig a variable CAL is introduced in the control program [7] defined by (5):

$$CAL = \frac{a(T_{out}) \cdot p_d^{b(T_{out})} - c(T_{out}, n)}{Q_d} \quad (5)$$

where  $Q_d$  is the net volumetric flow rate of the pump detected by FM,  $T_{out}$  is the delivery temperature measured by T3,  $p_d$  is the delivery pressure measured by P1 and  $n$  the pump speed.

If CAL equals 1 then the link among flow rate, pressure, speed and temperature follows expression (1).

The program that permits handling of the rig includes 10 PID to control proportionally actuated components based on feedback signals from a rig transducer or from an external source or from a variable evaluated as function of other signals. Taking advantage of this situation it is possible to implement a closed loop control of valve E2 using the variable CAL as feedback signal and enforcing the value 1 as set point. By so doing valve E2 originates a flow passage that reproduces, for each temperature and velocity condition, the resistance generated by the lubrication circuit.

In case it were desired to separate the contribution of the conditioning group from the lubricating circuit it is possible to control both valves E1 e E2. This procedure is useful when remotely piloted pumps (the relief valve in case of fixed displacement pumps or the absolute pressure limiter in case of variable displacement units) are tested; in such a situation the pilot signal is taken at port X (Fig. 3) located between E1 and E2.

**VELOCITY PROFILE** - For the engine velocity profile during the driving cycle use has been made of a model available in AMESim® [3] where data relative to tires, gear and differential ratios were provided as inputs; choosing as “mission profile” the mixed cycle NEDC, plots of the urban cycle ECE 15 (also known as UDC) and extra-urban EUDC.

shown in Fig. 5 and Fig. 6 have been obtained. Simulated plots were thereafter approximated with linear traces removing some peaks and then assigned as set point for the electric motor M1. As an alternative it is also possible to start from engine speed data collected during the cycle.

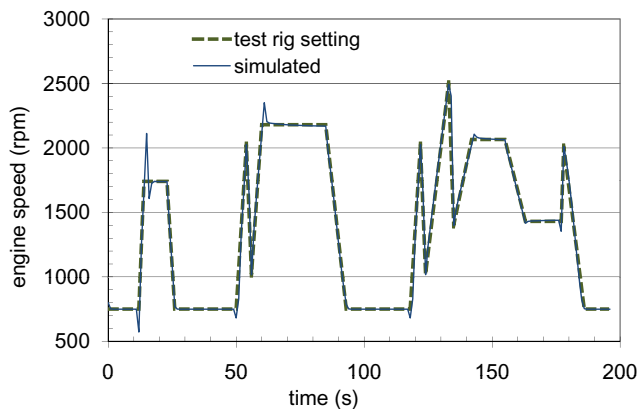


Figure 5: engine speed during the ECE 15 driving cycle

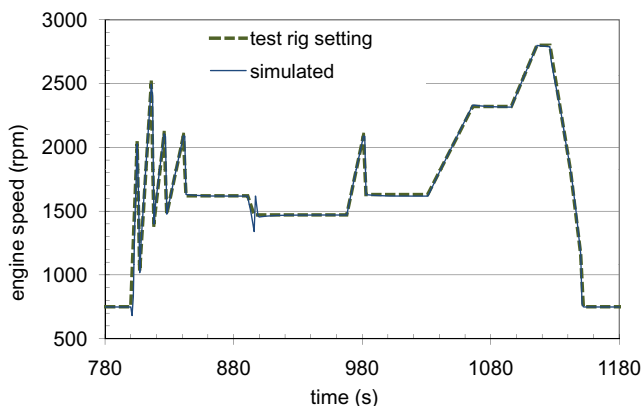


Figure 6: engine speed during the EUDC driving cycle

**TEMPERATURE PROFILE** - The law of temperature variation during the driving cycle is extremely relevant, as testified by experimental results presented hereafter.

The rig allows reproduction, acting on the power of resistances EH, of an experimental oil temperature profile measured in the engine. A present constraint of the rig is represented by the temperature rise speed that, despite the reduced tank capacity and the pipes insulation, can at most reach 4.5 °C/min, while the highest gradients detected in the engine and also confirmed by other researchers [5] reach 8-9 °C/min. This constraint can be overcome by installing most powerful heating elements or by slower execution of the cycle, in the present case by doubling the time span: this provision also allows attainment of more gradual velocity ramps (at most 275 rpm/s) with less problems bound with control of valve E2. In the data processing phase, it is then sufficient to divide by two the energy consumed; it was checked with tests that this artifice has negligible influences on the final result.

For tests reproducibility, it is necessary that the initial oil temperature be always the same (20 °C) regardless of ini-

tial ambient conditions; furthermore experimental measurements on the engine led to the observation that the temperature gradient in the first minute of the cycle is extremely low, about 2.5 °C/min, while in the second already reaches 8 °C/min.

To meet these requisites the following automatic start-up procedure was devised:

- motor M1 is brought to minimum speed (e.g. 750 rpm) correspondent to time 0 of the ECE cycle;
- tank temperature is set a few degrees<sup>1</sup> below the initial cycle temperature, 16 °C for the case of cycle start at 20 °C;
- closed loop control of valve E2 is put into action according to permeability curves of the circuit: this way pressure rises and stabilizes;
- the chiller is excluded and oil warms up for dissipative effects only;
- when the temperature sensed by T3 reaches 20 °C the test proper is started.

Fig. 7 shows parameters of interest in the start up phase of the test.

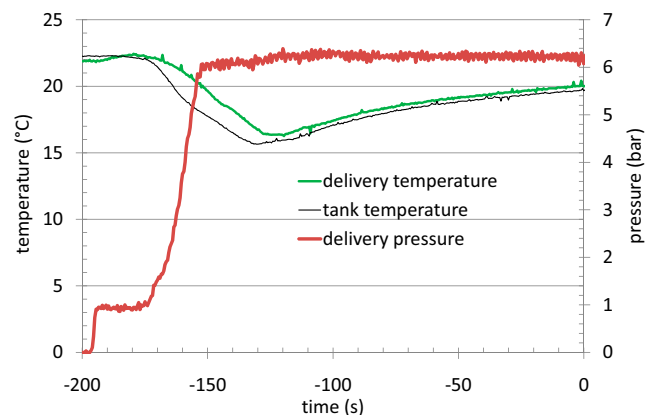


Figure 7: temperatures and delivery pressure during the starting procedure

Once the test is started, an appropriate command profile is given to heaters EH so to reproduce the measured temperature in the engine. Fig. 8 shows the temperature attained in the rig in comparison with that measured in the engine during a NEDC cycle: the maximum error is of 2° C.

## METHOD VALIDATION

**REFERENCE UNIT** - Tests aimed at calibrating and tuning the rig have been carried out using a crankshaft mounted gerotor lubrication pump shown in Fig. 9; significant data are:

- Displacement = 15,3 cc/rev
- Outer diameter of the external gear = 80 mm
- Gears thickness = 11 mm
- Pressure setting of relief valve = 5.5 bar

1. it was verified that the temperature of 16 °C represents the best compromise to arrive at 20 °C in a relatively short time and with a low gradient.