

Experimental and numerical analyses to improve the design of engine coolant pumps

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Abstract. In this paper, an experimentally based procedure is presented to re-orient the design point of the pump in order to minimize the energy absorbed during the homologation cycle or during any real driving one. During it, in fact, every benefit on the pump's efficiency is appreciated and produces fuel consumption and CO₂ reduction. The procedure takes the advantage from a dynamic test bench for coolant pump, realized and engineered at University of L'Aquila. It has been linked to a model-based methodology, which evaluates, according to a specified vehicle's mission profile, the speed and load variation of the engine propelling the vehicle, and, therefore, the pump speed. The knowledge of the engine cooling circuit for closed and fully opened thermostat allows the calculation of the flow rates and pressure delivered in each time instant of the drive cycle. The speed-flow rate-pressure delivered pump profile has been reproduced on the bench, and all the relevant quantities have been measured: an exact evaluation of the scatter of the efficiency of the pump, the instantaneous power and the overall energy absorbed have been obtained. Results show how the pump efficiency is far from its Best Efficiency Point. This conclusion invited the Authors to reorient the design pump considering an operating condition, which has a greater occurrence among all the operating points characteristic of a drive cycle. Four pumps have been designed following this approach, showing a sensible reduction of the energy absorbed: this represents a key point also for pump electrification.

1 Introduction

In recent years, the research and development on Internal Combustion Engines (ICE) in the on-the-road transportation sector is pushed by the necessity to reduce their environmental impact. This is mainly related to emissions produced by fuel combustion. Considering the share of fossil fuels on the transportation of people and freights and other aspects referred to the life of the fossil fuel economy, it is well recognized that the expected technological life of ICE will be still long and any effort to improve vehicle efficiency will be certainly welcomed.

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In fact, global on road transportation is responsible for about 26.5% of the CO₂ emissions [1], since fossil fuels-based propulsive architectures actually cover more than 99% of the vehicle's market share [2]. Commercial transportation plays the most important role in this scenario: it is responsible of more than 30 million barrels per day (over a total quantity that is close to 100 million barrels per day) of oil equivalent, due to heavy loads carried over long distances. Heavy commercial and long hauling vehicles consumed about 55 % of the fuel for trucking in 2015, also if they made up only the 15 % of the fleet [3]. Hence, international commitments have proposed several regulations to impose a reduction of CO₂ emissions from vehicles and always more stringent limits and huge fees have been introduced to achieve these goals [4, 5]. Moreover, "Diesel gate" scandal and recent attention gained by electric vehicles have slowed down the interest on the development of thermal engine-based vehicles, but all the perspectives still give to ICE the role of the principal propulsion system by 2040, in particular related to the freights transportation and long haul coaching, driven by increases in population and GDP [6, 7]. In this regard, electrification and hybridization of propulsion gained great attention: they are able to realize near zero local emissions vehicles, in particular in urban contexts. In an optimistic scenario, fleet electrification alone leads to an energy consumption reduction of about 40% by 2050 compared to 2015 [8]. Actually, this value does not consider a life cycle evaluation of impacts, which is surely lower, according to well-to-tank analysis, taking into account different energy vectors [9].

From this point of view, technologies that can be more easily introduced in the market are those that have low ratio between costs and benefits expected. Thermal management of the engine and the vehicles is one very attractive area of interest, since it covers the optimization of the conventional thermal engine and also hybrid and pure electric powertrains [10, 11]. Reported benefits associated with thermal management options for passenger car and heavy duty vehicles are estimated to be 5-7% and further 3-4% can be obtained by the integration of vehicle thermal needs [12, 13]. A strong opportunity is related to the exploitation of wasted energy in ICE. The most studied application is the exhaust heat conversion into mechanical or electrical power through a thermodynamic cycle, usually ORC one [14, 15], but more than one limiting aspects should be overcome [16, 17]. Direct heat recovery seems a technology more easily to be implemented, with similar benefits [18, 19]. Also the coolant energy can be exploited for low grade thermal recovery [20, 21] and this integration can have additional benefits also in electrified powertrains and hybrid vehicles [22, 23]. Other opportunities to integrated thermal management are referred to the cooling of the intake air of the engine, by means of water charge air cooler or chiller-based system, in order to increase engine volumetric efficiency [24, 25]. Air conditioning unit can be also integrated into the cooling systems, particularly considering a liquid cooled condenser [26, 27]. Moreover, multi temperature radiators and improved fan will lead to a reduction of heat exchanger sizes, frontal area occupation and reduction of drag coefficient of the vehicle [28].

Focusing ICE thermal management, it directly concerns the thermal behavior of metal walls of cylinders and engine head [29], optimizing temperature to improve combustion [30], saving fuel and reduce emissions [31, 32], avoiding after-boiling of the cooling fluid and mitigating knock [33, 34]. Lubrication circuit has been also a main topic of investigation, due to its strict interactions with the engine coolant [35] and the fuel consumption benefits related to faster oil warm up [36], which implies lower oil viscosity at higher temperature and, so, low engine friction: split sump [37, 38], heat-to-oil [39], thermal storage and variable flow lubrication [40, 41] are the principal technologies available at the moment.

Unconventional components have been proposed, like multi-way and smart thermostats [42] or the use of nanofluids [43] to improve engine thermal management effectiveness,

assuring a more flexible control of the engine temperature in order to shorten warm up time. From this point of view, the most important contribution for a fast warm up time is due to coolant pumps and to their actuation. Positive displacement pump has been proposed in order to have high efficiency at low loads [44] and they can benefit of an additional degree of freedom if they have a variable displacement option, leading to an optimization of the coolant flow rates during a daily trip [45]. Unconventional actuators, decoupled from the engine crankshaft, are under strong attention. This is the case, for instance, of electrical water pumps where a speed-controlled electrical motor fixes the flow rate of the pump, sharing the benefit related to the increasing hybridization of the vehicles [46, 47].

In reality, conventional pumps are mechanically linked with the engine shaft, via pulleys and belt and their design is typically done fixing a specified flow rate, which is needed to cool the engine at its maximum mechanical power. Knowing these engine conditions, the speed of the pump is known (by the ratio of the diameters of the pulleys) and also the pump head, once flow rate is fixed and the permeability of the cooling circuit, too. Considering that the condition of maximum engine power is far from typical operating conditions of the engine during usual operation, the pump operates very frequently as overdesigned. This results in average low efficiency of the pump during common operations (i.e. real driving conditions or homologation cycle), far from the best efficiency point (BEP) of the pump (i.e. design point). For a medium and low engine loads which cover in real cases a great part of the operating engine conditions, efficiencies of the pump lower than 0.3 have been measured [44] while at BEP, usually chosen at maximum engine power, higher performances are obtained (efficiencies in the range of 0.5-0.6 are typical).

Hence, the Authors are convinced to re-orient the design pump considering an operating condition at BEP, which has a greater occurrence frequency among all the operating points characteristic of a real driving. In order to be close to a real operation, the WLTC has been chosen. If a vehicle is specified, the instantaneous propulsion power can be calculated and, therefore, the sequence of engine speed and loads. The mechanical connection between engine shaft and pump allows the calculation of the pump's speed. Having defined the hydraulic characteristics of engine cooling circuit (with thermostat branch opened and closed), the instantaneous flow rates and pressure delivered have been calculated, as well as the instantaneous power requested to drive the pump once the efficiency is known. The working conditions of the pump have been reproduced on a dynamic test bench specifically developed to understand the energy absorbed by the pump, when different BEP was chosen.

In this paper, a new design criterion for the pump's design is proposed, searching an operating point for the design of the pump (so, a condition in which it exploits its best efficiency) which guarantees the minimum of the energy absorbed during a driving cycle. The availability of a dynamic test bench capable to operate a pump as it is driven in real engines, allowed to verify which design point (DP) minimizes the energy absorbed during a WLTC. In the test bench, pump speed has been reproduced thanks to suitable actuators, as well as flow rates and head delivered. The minimization of the energy requested is of particular interest when the optimization applies to an electric water pump [48]. The application of the criterion produces an energy saving close to 15%, which is a very interesting result within the WLTC procedure as reference cycle for CO₂ measurement.

2 Experimental set-up

The evaluation of the performances of a cooling pump has been done on an experimental test bench properly designed to perform a dynamic cycle [48]. In fact, the test bench developed at the University of L'Aquila comprises a variable speed actuation with a high response time (<1 s), which is able to reproduce the speed variation of a pump when it runs a dynamic cycle, like the one realized in engine cooling applications, during a

common daily trip or during a homologation cycle. Conventionally, in fact, the pump is mechanically actuated by the engine shaft; this means that the revolution speed of the pump is continuously changed proportionally to the engine one, which on turn depends on the vehicle velocity. These high-speed variations can be reproduced in the test bench, if the pump tested is both mechanically actuated or actuated by an electric motor. In this way, the test bench is able to test not only conventional engine cooling pump (mechanically actuated), but also innovative pump for hybrid vehicle (24/48 V_{DC}) or for the cooling of other auxiliaries (batteries, electrical equipment, etc.). When the pump is driven following a homologation cycle, the instantaneous power is measured by a torque meter (Fig. 1) and, by integration during time, its energy absorbed during the cycle. The CO₂ emissions accountable to the cooling pump can be evaluated. Its eventual optimization based on a new choice for the BEP can be oriented to minimize CO₂ emission.

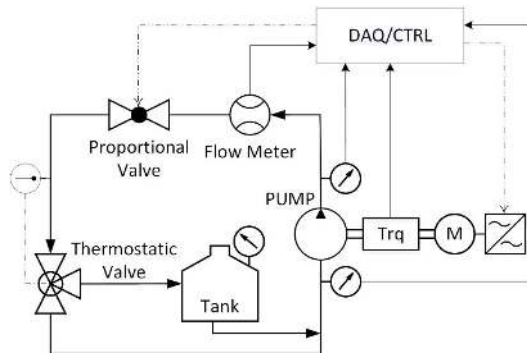


Fig. 1. Experimental layout of the dynamic test bench for pump testing

The test bench is also composed by a proportional pneumatic valve, which can set in order to impose a specific pressure drop on the circuit, imposing variable pressure head on the pump. Moreover, a three-way valve, thermally actuated, is able to represent the thermostat valve on a conventional cooling circuit and a pressurized tank gives the right hydraulic head to the pump and the possibility to increase it through static pressure increase. Finally, magnetic flow meter and pressure transducers consent to calculate the hydraulic power, while torque meter and electric power meter allows the determination of the efficiency chain (Fig. 1 and Fig. 2).

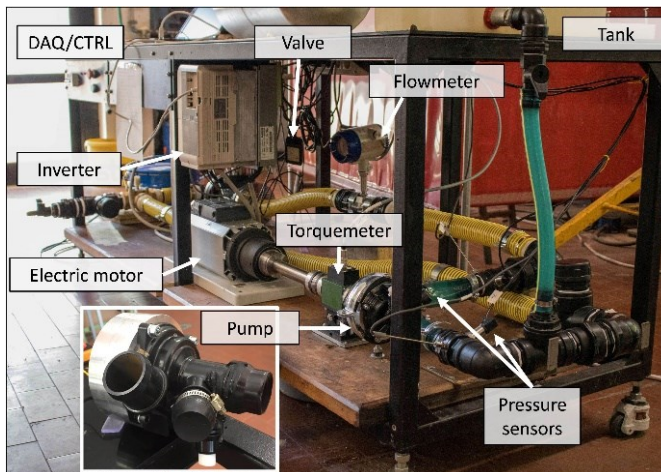


Fig. 2. Picture of the dynamic test bench and the pump tested

3 Pump modeling and validation

A zero-dimensional model of a coolant pump has been developed in order to evaluate the effects related to different designs of the pump when it is operated following a sequence of working points. Each different design corresponds to a different working point of the pump ($\Delta p @ RPM @ Flow\ rate$) where the best efficiency is requested, removing the design criterion which would fix the maximum efficiency of the pump when the engine delivers its maximum mechanical power. Once the design has been done, characteristic curves (pressure delivered and efficiency versus speed and flow rate) of the pump are known either because they are measured or because they are theoretically predicted. The hydraulic performance of the cooling circuit accounting for a closed and an opened thermostat, which modifies the permeability of the equivalent circuit seen by the pump. Once the revolution speed of the pump is known (either because mechanically driven by the engine or because electrically actuated), the head and flow rate delivered by the pump can be easily evaluated and, therefore, the hydraulic power given to the fluid as well as the mechanical power absorbed on the pump shaft. If the pump follows a sequence of operating points that derive from a given driving cycle of the engine (vehicle), the overall mechanical energy absorbed can be calculated (integrating during time the instantaneous power requested by the pump). If the sequence represents a homologation cycle, this would be particularly meaningful: the energy requested by the pump can be compared with the one needed to propel the vehicle.

A specific pump was designed, built and tested on the dynamic bench reproducing the sequence of the homologation cycle WLTC class 3. From the engine speed, the speed of the pump was calculated knowing the ratio of diameters of the two pulleys. Once the hydraulic performance of cooling circuit was known, for each pump speed the equilibrium between the characteristic curve of the pump and that of the circuit allows the calculation of the flow rate delivered and pump head. The sequence of the operating points in terms of speed were the set points on the dynamic test bench, so reproducing real pump operation.

Pump speed was changed by means of a brushless-type synchronous electric motor; flow rate is adjusted actuating a proportional valve. The derivation of the sequence of the operating points of the pump followed the procedure described in [49]. Fig. 3 shows the quality of the speed control of the test bench: set points and measured speed values are shown for the sequence of pump operating points reproducing a WLTC homologation cycle.

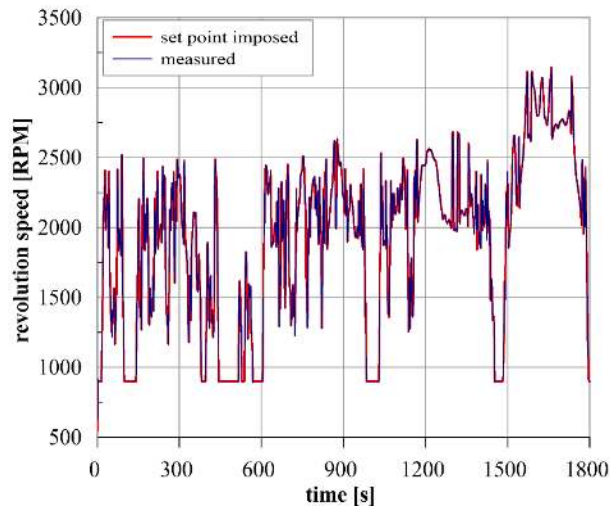


Fig. 3: comparison between imposed revolution speed and measured one on the pump test bench during WLTC

The resulted flow rate delivered by the pump is shown in Fig. 4, where the experimental one is compared to that calculated by the zero-dimensional model. Experimental flow rates are a little bit smoother with respect to the one calculated by the model. However, the model catches with good accuracy the experimental data and the error is limited.

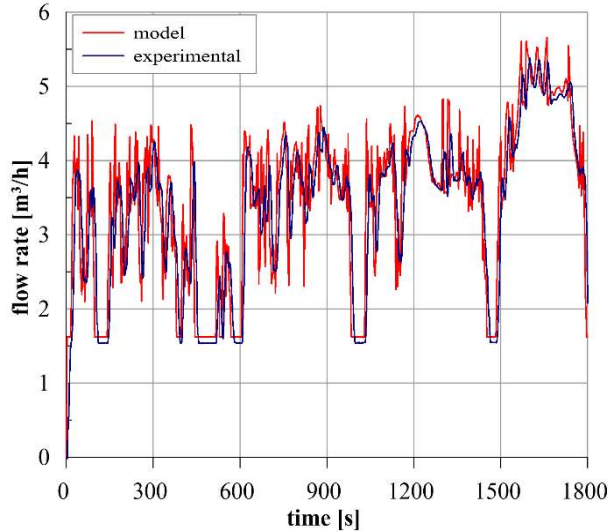


Fig. 4: comparison between experimental and model data of flow rate during WLTC

An interesting comparison is reported when the mechanical power absorbed by the pump (Fig. 5) is compared with the modelled data. The matching is very satisfactory: a root mean square error is 4.6 % of the maximum value.

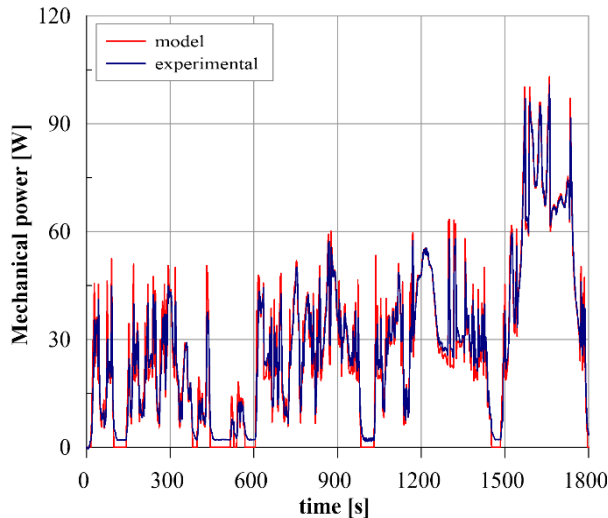


Fig. 5: comparison between experimental and model data of mechanical power during WLTC

Moreover, the cumulated mechanical energy has been evaluated by integrating during time both experimental and model data, and represented in Fig. 6. The final energy consumption of the tested pump on the bench is about 52 kJ, and the error of the value calculated by the model is under 0.3%.

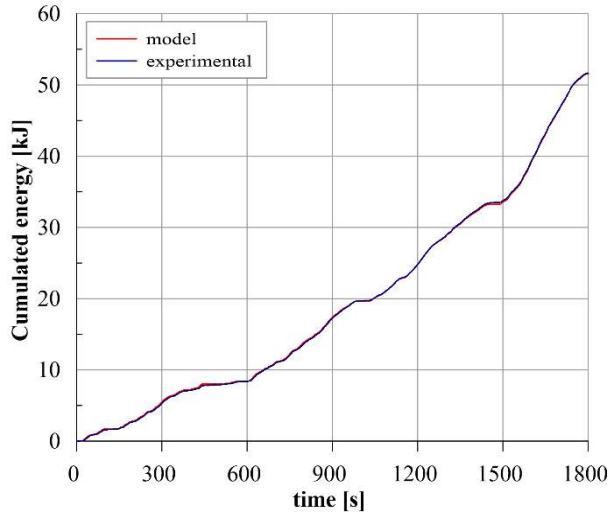


Fig. 6: comparison between experimental and model data of cumulated mechanical energy during WLTC

4 Pump design

In order to evaluate the influence of the choice of design point of the pump on the energy required during a homologation cycle, four pumps have been designed modifying the couple (*RPM@flow rate*) at which the best efficiency of the pump will be guaranteed. The intention is to re-orient the choice from the present criterion (maximum engine power) to the one that minimizes the energy absorbed by the pump over the WLTC.

The new design point of the centrifugal pumps has been chosen by applying the dimensionless theory for turbomachines. The main advantage of this approach is that only one characteristic curve is used for all possible operating speeds of the tested pump using typical dimensionless numbers. Flowrate@RPM@ Δp @efficiency were known from the experimental results of the tested pump. The efficiency was, by definition, the highest value of all of those reached by the pump. The reference dimensionless numbers, in terms of flow rate coefficient (φ , eq.1), manometric coefficient (ψ , eq.2), specific speed (n_s , eq.3) have been calculated once the impeller outer diameter D is considered. As it is known, all the pumps that have the same dimensionless numbers at design points are in similarity to each other and will have the same maximum hydraulic efficiency.

To generate a new pump in similarity with the other ones (keeping the same value of the maximum efficiency), the efficiency curve has been shifted over the map and a different operating point has been chosen as design one. Thus, the BEP has been moved graphically along the same rotational speed curve, by maintaining constant all the quantities shown in eq. 4, where i and j indicate two different design points.

$$\varphi = \frac{Q}{nD^3} \tag{eq.1}$$

$$\psi = \frac{gH}{n^2 D^2} \tag{eq.2}$$

$$n_s = n \frac{\sqrt{Q}}{\sqrt[4]{gH^3}} \tag{eq.3}$$

$$\begin{cases} \left(\frac{gH}{n^2 D^2}\right)_i = \left(\frac{gH}{n^2 D^2}\right)_j \\ \left(\frac{q}{n D^3}\right)_i = \left(\frac{q}{n D^3}\right)_j \\ n_i = n_j \\ \eta_{i,max} = \eta_{j,max} \end{cases} \quad (\text{eq.4})$$

In that way, a new impeller geometry and volute of a different pump have been generated, keeping the same maximum efficiency value (Best Efficiency), but at a different operating point (P). This procedure has been repeated for four design points, which have the same rotational speed of the original pump ($n_i=n_j$), but different combinations of flow rate and pressure head. Thus, four points have been placed on the characteristic curves of the circuit, according to four opening degree of the thermostat branch (DP1, DP2, DP3 and DP4, green points of Fig. 7). This choice was done in order to find suitable operating conditions of the pump to which it would correspond a maximum frequency of occurrence among the operating points of a WLTC. In that way, each new pump has its own performance map, with different characteristic curves, in terms of flow rate vs. pressure head and efficiency. All of them operate in fluid dynamic similitude and have the same maximum efficiency value at BEP.

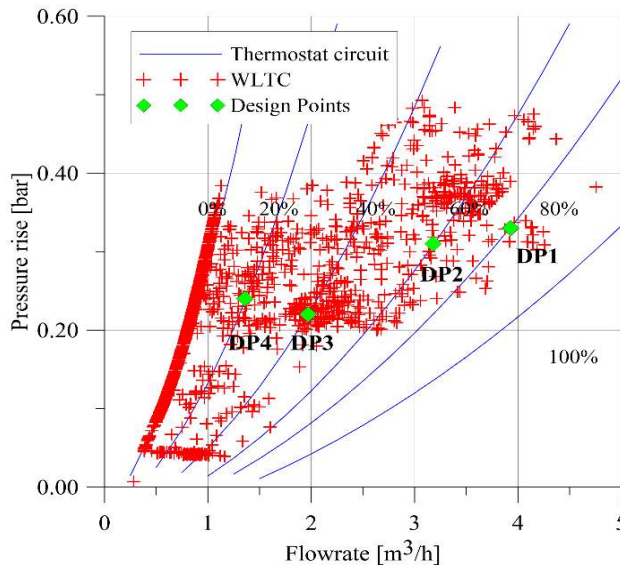


Fig. 7: operating points of a WLTC mission profile and the different four design points of the pump (DP1, DP2, DP3 and DP4).

5 Results and discussion

By using the developed model, the performances of the four different pumps have been evaluated, in order to assess the pump that minimizes the overall energy absorbed on the cycle. The (flowrate@ Δp) of the cooling circuit considered the opening of the branch toward the thermostat when the temperature reaches the right value (80-85 °C, Fig. 8): the opening is progressive, according to the coolant temperature. For each thermostat opening degree, a different characteristic curve of the circuit has been considered, so closely

reproducing the working points of the pump as it was mounted in a real engine during a WLTC driving cycle. All the operating points of the pump have been shown (red crosses, Fig. 7) and it is evident how a great part of the points stays on a line which represents the branch of the thermostat closed (engine warm up).

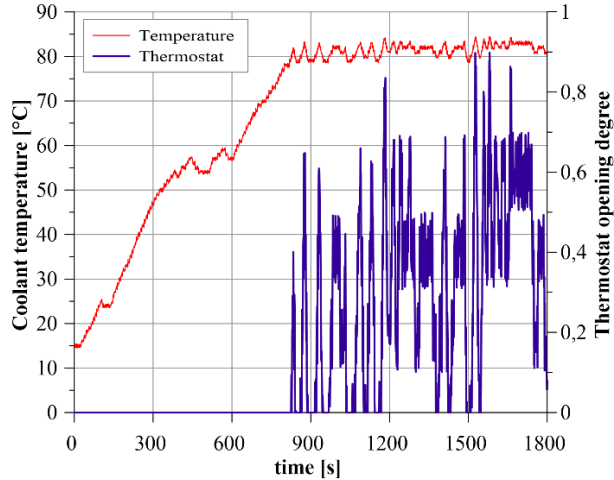


Fig. 8: Coolant temperature variation considered during WLTC and consequent thermostat opening degree

In Fig. 9, the flow rate of the four different pumps as a function of the revolution speed is reported. It can be observed how the pumps designed in less permeable circuits (DP4), characterized by a higher pressure drop on flowrate ratio, need a higher revolution speed for a fixed flowrate to be delivered. This issue is further outlined in Fig. 10, where the speed profile in the four different cases during the WLTC is reported. In fact, moving from pump #DP1 to pump #DP4, size is smaller and, so, it should spin at higher revolution speed to deliver a specific flow rate. In fact, the flow rate delivered by the pump is fixed by the degree of cooling required by the engine which remains the same for the four pumps; so, for a defined operating point of the engine, the four pumps must deliver the same flow rate. A different speed of rotation for the four pumps will readjust the flow rate needed.

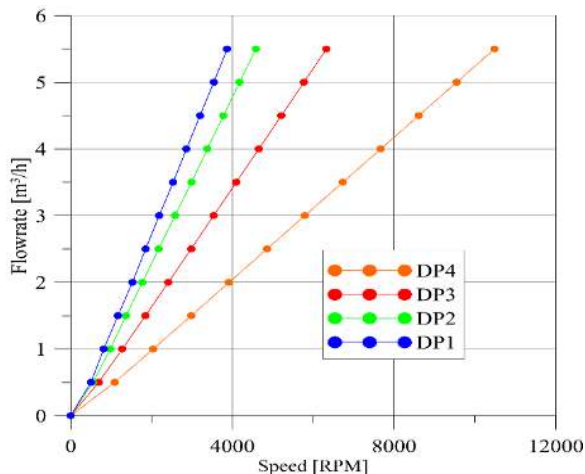


Fig. 9: flow rate delivered by the pumps and revolution speed variations

In order to compare the four pumps when they operate to cool the engine during a WLTC, the zero-dimensional model, which fixes the equilibrium between the cooling circuit and each pump, has been used. It evaluates the instantaneous mechanical power, redefining the speed of each pump in order to guarantee the same flow rates toward the circuit and, therefore, the same pressure rise. This produced, for instance, when pump design #DP4 is considered, the maximum speed as high as 10 kRPM (Fig. 10). This would be possible if an electric actuation is considered.

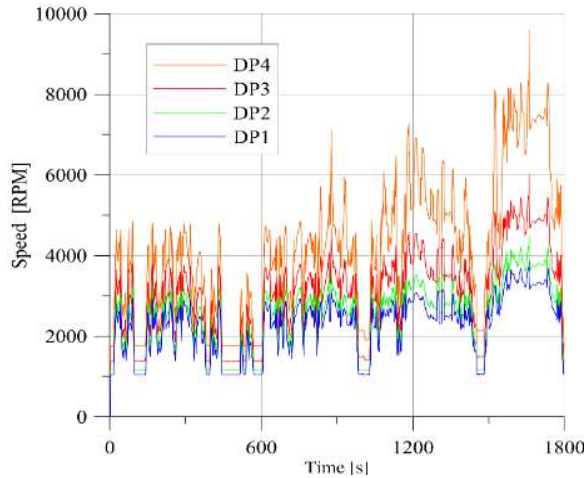


Fig. 10: revolution speed of the four pumps designed during WLTC

Fig. 11 and Fig. 12 show the results obtained in terms of instantaneous mechanical power requested and cumulated energy absorbed by the four pumps, when a homologation WLTC is run. Pump #DP4 shows lower power absorbed in the first part of the cycle, when the branch toward the thermostat is fully closed. However, in this part of the cycle the flowrate requested by the circuit is still low and, consequently, the mechanical power. When the WLTC proceeds and the temperature of the circuit gradually opens the branch of the circuit toward the thermostat, the flow rates requested by the engine increases and the bigger pump (#DP1) operates in a region in which efficiency values are greater; the smaller pump (#DP4) is certainly able to deliver the requested flow rate but this requires a higher revolution speed and the operation of this pump in a region of lower efficiency.

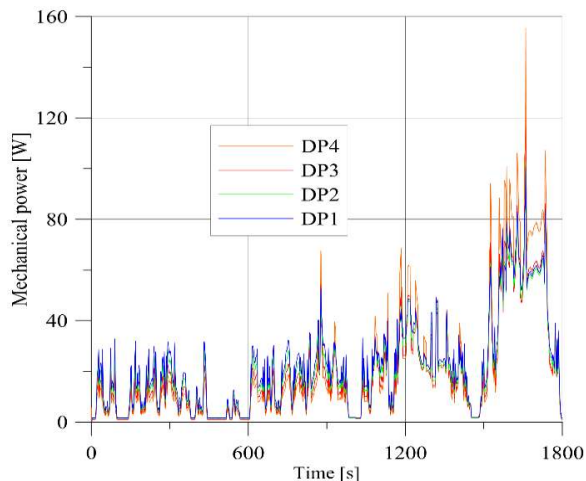


Fig. 11: mechanical power absorbed by the pumps during WLTC

Looking at the cumulated mechanical energy shown in Fig. 12, the best performances are obtained by pump #DP3. In fact, during the last part of the WLTC, thermostat opens, determining an increase of the flowrate that has to be delivered by the pump. The hydraulic power is much higher and so better efficiency of DP3 with respect to DP4 in the last part of the cycle plays a greater role. However, the greater efficiency at high loads does not compensate the behavior at lower load. Pump #DP3 is the best choice; its final energy consumed is about 34.1 kJ; a maximum energy saving is about 15% compared with pump #DP1.

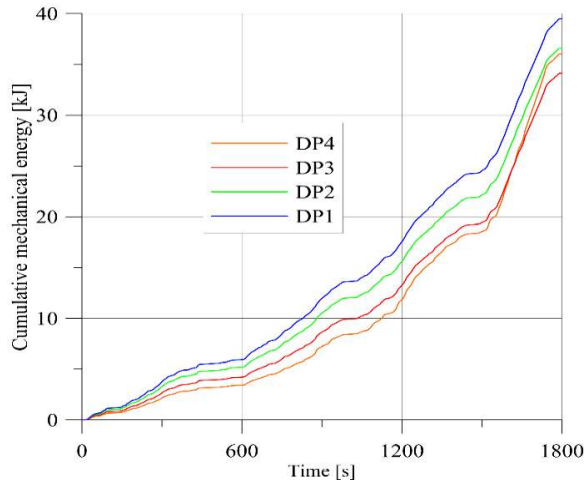


Fig. 12: cumulated energy absorbed by the four pumps during WLTC

A more precise evaluation should be done considering the energy absorbed by the auxiliaries (seal and bearing), which at lower flow rates (and pressure delivered) seriously reduces the overall pump speed.

A quantitative explanation of the greater performance of pump #DP3 is demonstrated in Table 1, which shows the repartition of the energy in the four different phases of the WLTC (low, medium, high, extra-high). Pump #DP4 has the best efficiency in I and II phases (lower energy absorbed), while pump #DP2 has best efficiency in IV phase. It is in accord with the design point chosen for each pump (Fig. 7): pump #DP4 has been designed in a working point near to the fully closed thermostat curve, so it has better performance in the early phases of the driving cycle (cold phases). On the other hand, pumps #DP1 and #DP2 have been designed in the opened thermostat region, so they have better performance in hotter engine phases. The best compromise is obtained by pump #DP3.

Table 1: energy consumption of the pumps during WLTC cycle

	time start [s]	time end [s]	Hydraulic energy [kJ]	Mechanical energy [kJ]			
				DP1	DP2	DP3	DP4
<i>WLTC I phase</i>	0	589	1.9	5.9	5.2	4.2	3.4
<i>WLTC II phase</i>	590	1022	2.8	7.7	6.9	5.8	5.1
<i>WLTC III phase</i>	1023	1477	5.2	10.6	9.9	9.3	9.9
<i>WLTC IV phase</i>	1478	1800	8.5	15.2	14.7	14.9	17.7
<i>WLTC total</i>	0	1800	18.4	39.5	36.6	34.1	36.0

6 Conclusions

Today, the design point of a pump for cooling ICE considers an operating point of the engine in which it delivers maximum mechanical power. This orients the design toward the highest flow rate, which is needed to remove the maximum thermal energy from the ICE. This corresponds to a maximum value of the pressure to be delivered. Pump speed follows the mechanical connection between engine shaft and pulley of the pump; at the design point of the pump, a high value of revolution speed is requested. So, the best efficiency point of the pump is obtained at high revolution speed as well as high flow rate and pressure delivered.

During real driving conditions, the ICE is often far away from its maximum power, so the pump operates most frequently at off design conditions and it operates as over-designed. Being the efficiency of the pump extremely sensible to the flowrate and revolution speed, typical real operating efficiencies are low. This situation also applies when the pump operates during a homologation cycle (of the engine): an eventual reduction of the energy absorbed by the pump would be strongly welcomed, producing a fuel saving and, therefore, a CO₂ emission reduction.

The paper demonstrates how a new criterion that defines the design point of a pump more oriented to an average working condition of the pump during real driving (as it happens during a WLTC) saves about 15% of mechanical energy absorbed by the pump. Starting from a conventional design of an existing pump experimentally tested, four pumps have been redesigned using the dimensionless theory and similarity laws. The conventional pump has been tested on a dynamic test bench in which it has been run reproducing the operating condition which happened when a homologation WLTC is considered. During this testing, most important relevant variables have been measured and, particularly, the instantaneous mechanical power requested. A mathematical model has been successfully validated; it gave the time variation of the set points (to the dynamic test bench) in terms of flow rate delivered and pump speed in order to reproduce real pump behavior during the homologation (or any other real driving conditions). The four pumps redesigned, inputting with their characteristics curves the mathematical model, gave the quantitative result before mentioned. This result tends to be greater when the mechanical efficiency of the pump is accounted; in fact, at lower flow rates and pressure delivered, the weight of the sealing system and of the bearing in terms of mechanical power absorbed is dominant with respect to the hydraulic request. The energy saved appears to be particularly interesting for electrical actuation of the pump.

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