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## Fuel economy benefits of a new engine cooling pump based on sliding vane technology with variable eccentricity

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### Abstract

During a homologation cycle, engine cooling pump operates at off-design conditions that are characterized by low efficiency and, in turn, impacts on the vehicle fuel economy and emissions performances. Furthermore, the conventional centrifugal technology struggles with the need of implementing thermal management strategies. In order to address these issues, the current paper presents the development of an innovative engine cooling pump for automotive applications that is based on the sliding vane technology. The experimental performances of the novel pump were compared with the ones of a conventional centrifugal device through a test bench that reproduced the real operating conditions of a passenger car engine cooling circuit. Characteristic curves and efficiency maps were further used as input data for a comprehensive model that takes into account heat transfer and hydraulic phenomena of a real engine cooling circuit. The simulation platform was eventually used to assess the energy benefits of the sliding vane pump over a centrifugal machine along the Worldwide harmonized Light vehicles Test Cycle (WLTC) with respect to a fixed geometrical configuration and an optimized one achieved through a control on the eccentricity between stator and rotor that tuned up the flow rate on the cooling demand. A mechanical power reduction of 12% was estimated for a sliding vane pump mechanically linked to the engine crankshaft while the benefits reached 22% if the eccentricity actuation was considered. In terms of CO<sub>2</sub> reduction, the benefits related to the introduction of the sliding vane pump are of about 0.5 g/km.

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**Keywords:** engine cooling, automotive, sliding vane rotary pump, WLTC, WLTP

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### 1. Introduction

Cooling circuit assumes a fundamental relevance in internal combustion engines since it allows to keep the materials of the combustion chamber within suitable operating temperature ranges. The magnitude of the thermal power acquired by the coolant medium and further dissipated at the radiator strictly depends on the engine operating conditions. Therefore, current engine circuits layouts are based on a belt coupling between engine shaft and coolant pump, which is usually a centrifugal device designed at operating conditions close to the maximum engine mechanical power. However, the ones that occur during a homologation cycle are far away from the design point. Hence, the design pump efficiency of 45-50% tends to drop to 15-20% because of losses due to incidence, secondary flows, volute passage, etc. On the other hand, rotary displacement pumps do not suffer of such abrupt efficiency reduction.

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	duration	maximum speed	stop phases	gear shift
WLTC	1800 s - 23 km	131 km/h	23.7	imposed at a given time
NEDC	1180 s - 11 km	120 km/h	12.6	adapted to the vehicle

Table 1: Main differences between WLTC and NEDC

The recent development of thermal management approaches to increase the engine efficiency and to reduce emissions struggles with the mechanical connection because of the difficulty to implement optimization control strategies [1] such as Lyapunov-based non-linear control techniques [2]. Hence, a transition towards electrified vehicle auxiliaries is currently under development. In this framework, the linear relationship between pump revolution speed and coolant flow rate that characterizes displacement machines might forecast a future employment of these technologies in automotive contexts.

One of the major advantages of an electrified engine cooling system is the decoupling between engine and pump that currently constrains the coolant mass flow rate. In particular, after the engine cold start, the surplus of coolant mass flow rate with respect to the engine cooling demand increases the warm up period: literature studies state that this phase is responsible of 2/3 of the overall harmful emissions during a New European Driving Cycle (NEDC) homologation cycle [3,4]. Hence, a faster warm up would participate in a positive way to the CO<sub>2</sub> emissions reduction and to the fulfillment of environmental emission regulations, thanks to greater engine mechanical efficiency due to friction reduction. This could be achieved decreasing the pump speed but ensuring a minimal coolant mass flow rate to prevent local thermo-mechanical stresses [5,6]. An efficient cooling on demand under all operating conditions could be further achieved tuning the coolant mass flow rate through the use of electric components such as pump and intelligent thermostat valve [7,8]. In [9–11] several topologies of electrified engine cooling systems were studied, and it was demonstrated that a 1-3% of fuel consumption decrease might be achieved through lower parasitic losses of the coolant pump and a decrease of coolant temperature fluctuations.

On the other hand, although efficiency of electric water pumps is higher than that of conventional mechanical devices, electric energy is provided by the alternator whose efficiency can be less than 60% [12]. For this reason, several studies placed their attention to the improvement and the replacement of the mechanical water pump in conventional [13] and heavy-duty vehicles [14], whereas the effect of fuel saving is higher. In [15] the effect of two mechanical pump technologies was evaluated on the engine cooling system: a planetary gear pump, which could reduce the revolution speed up to 65% through a pulley, and an on/off centrifugal pump which could completely stop the rotation of the impeller. The performances of the two devices were assessed through warm up tests (target coolant temperature of 80 °C) and along the NEDC. The warm up test of the planetary gear pump was 7.3% faster than a conventional water pump, while the one of the on/off centrifugal pump was 24.7% faster. For the NEDC test, the result was an improvement rate of fuel consumption up to 1.4% and 4% respectively.

The current paper addresses the issues of a high efficiency engine cooling pump through the development of an innovative device based on the sliding vane technology. The reference homologation cycle is the Worldwide harmonized Light vehicles Test Cycle (WLTC), whose differences with respect to the NEDC are summarized in Table 1. It turns out that the contribution to the total cycle emissions due to the cold start phase is about half than that in NEDC. A performance comparison between the Sliding Vane Rotary Pump (SVRP) and a conventional centrifugal device was accomplished through an experimental test bench that is able to reproduce the operating conditions of a passenger car engine cooling circuit. The measurement data were further used to calibrate a comprehensive engine cooling circuit model which allowed to predict the fuel consumption benefits over the WLTC. The analysis carried out in this paper firstly considers a direct coupling between SVRP and engine crankshaft. However, since the SVRP was not able to constantly restore the same coolant flow delivered by the centrifugal pump, a sliding vane pump with variable eccentricity was also evaluated. The eccentricity variation might be accomplished using different technological solutions like electric or oleodynamic actuators as well as the employment of a friction wheel.

## 2. Experimental setup and measurements

In order to compare the performances of a conventional centrifugal pump and the ones of a novel sliding vane water pump prototype with fixed eccentricity (Figure 1a), the experimental test bench schematized in Figure 1b was built and instrumented.

The water circuit is composed of storage and expansion tanks for the water-glycol mixture and a series of valves whose combination was able to simulate the head losses that typically occur in the cooling circuit of an engine installed

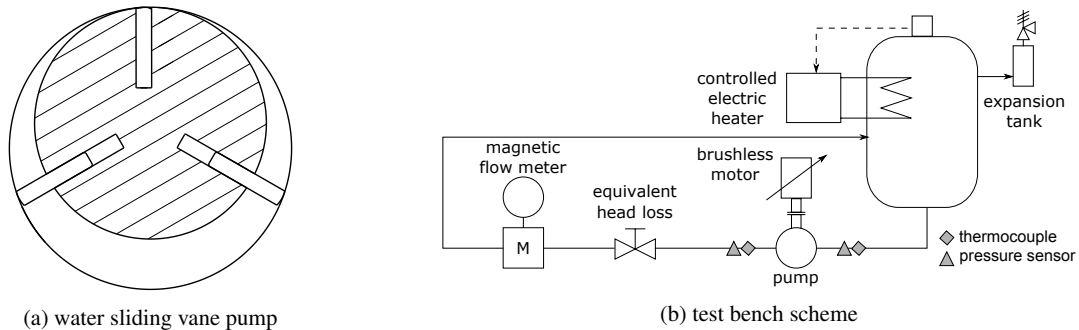


Fig. 1: Experimental setup

on passenger cars. Among them, the cyan and red curves in Figures 2a and 3a represent the head losses that occur at open and closed thermostat respectively. A controlled electric heater in the storage tank further allowed to set fluid temperatures typical of real applications ( $90^{\circ}\text{C}$ ). The pump revolution speed was varied using a brushless electric motor which also provided the indirect torque measurement from the electrical current. The pressure rise given by the pump was monitored through pressure sensors across the device. In the same location, T-type thermocouples were also installed. The pump flow rate was eventually measured using a magnetic flow meter. Tests were carried out at several steady operating points represented with circles in the performance maps of Figures 2a and 3a both for the centrifugal and sliding vane pumps respectively.

Figure 2a shows the characteristic curves of the centrifugal pump at different revolution speeds (from 500 rpm to 7500 rpm) retrieved using a quadratic fit of the experimental data. On the other hand, a linear fit was used to draw the characteristic curves of the sliding vane pump from 100 rpm to 1500 rpm. From the comparison of the performance maps, the operating curves of the sliding vane pump tend to be vertical lines. Hence, the device is able to provide an almost constant flow rate for a given revolution speed along the whole pressure rise range. The slope of the lines tends to decrease (i.e. becomes more negative) at lower revolution speeds. Therefore, flow rate decreases if pressure rise increases. This effect is due to a worsening of the volumetric efficiency. Indeed, at low revolution speed the sealing action due to the centrifugal force is less intense and leads to leakage paths across blade tips that result in an internal flow recirculation inside the pump. Furthermore, since pressure difference across the pump is the driving force for all leakage phenomena, higher pressure rises tend to increase them. Anyway, it should be pointed out that minimal recirculation is a common pump design procedure to prevent that the occurrence of overpressure might lead to mechanical damages of the machine.

Figures 2b and 3b additionally provide the experimental variation of the pump efficiency, defined as the ratio of the hydraulic power over the mechanical power. Although the maximum values for the pump efficiency are both close to 40%, the distribution along the performance maps is highly different. Centrifugal pump presents low efficiency at small flow rates and this trend is independent from pressure rise. On the other hand, sliding vane pump shows remarkable efficiencies even at small flow rates. For instance, at  $0.5\text{ m}^3/\text{s}$  and 1 bar, centrifugal pump operates with an efficiency of 20% while sliding vane works with an efficiency of around 35%. Conversely, at high flow rates (e.g.  $2\text{ m}^3/\text{s}$  and 1 bar) centrifugal pump operates close to the design point with an efficiency greater than 40%, while the sliding vane pump is lower than 35%. Maximum efficiencies occurs between  $2\text{ m}^3/\text{s}$  and  $2.5\text{ m}^3/\text{s}$  for the centrifugal pump while the correspondent maximum values for the sliding vane machine are in a flow rate range between  $1\text{ m}^3/\text{s}$  and  $2\text{ m}^3/\text{s}$ .

### 3. Engine cooling model

Behavior and performances of the engine cooling circuit and components were predicted thanks to a mathematical model which describes thermal and fluid dynamics phenomena that occur in the cooling circuit [16,17]. This engine cooling model consists of different sub-models that interact each other (Figure 4a): thermal model of the engine, hydraulic model of the circuit, thermodynamic model of each circuit component and aerodynamics of the frontal end of the vehicle. An extensive description of the model has been already presented by the Authors in previous papers and its validity verified thanks to an experimental campaign performed on a real diesel engine (1.4L DV4 TED) mounted on a Citroen C3 passenger car [18,19].

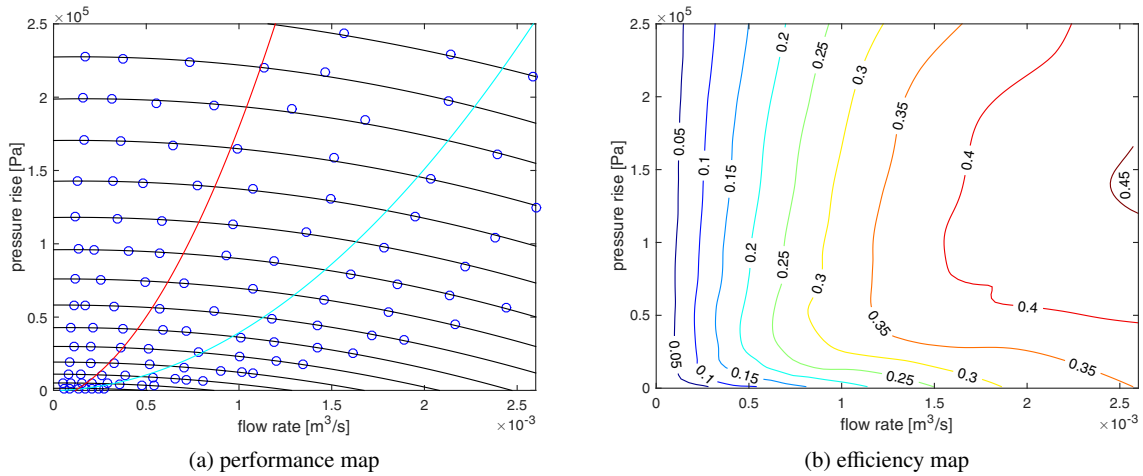


Fig. 2: Experimental results - centrifugal pump

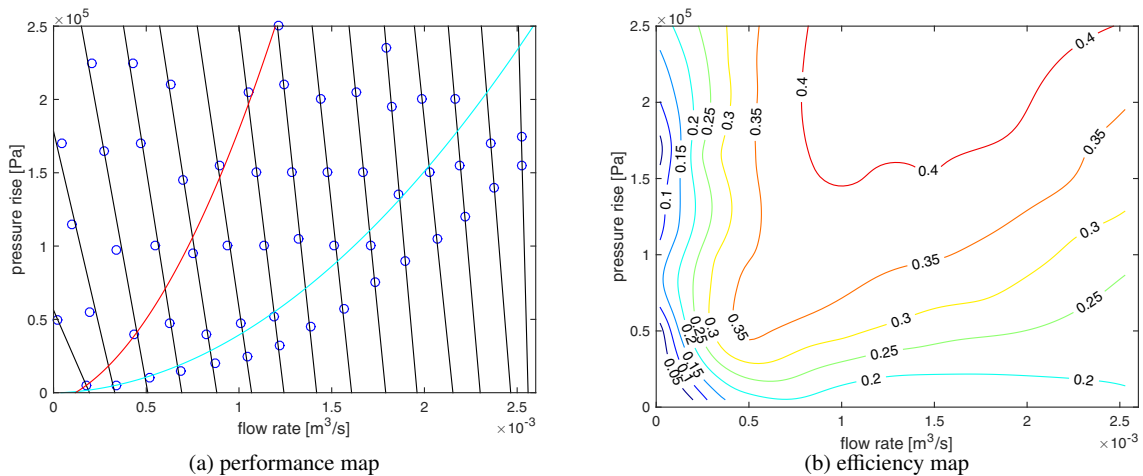
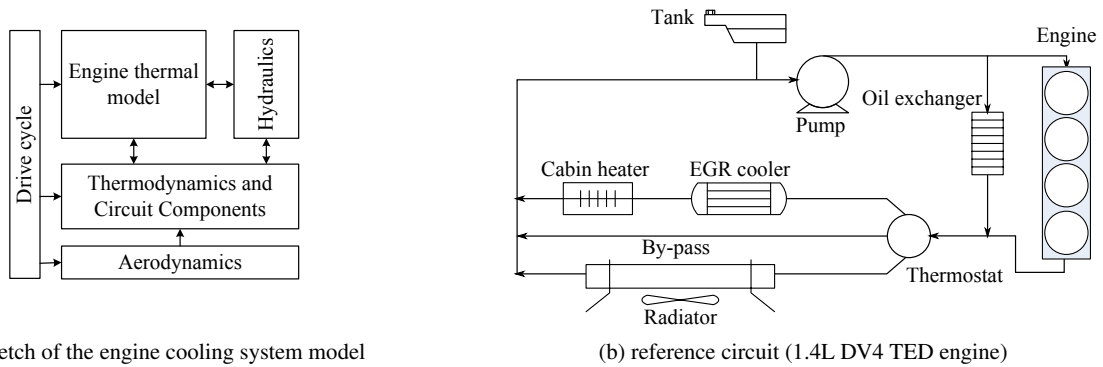


Fig. 3: Experimental results - sliding vane pump

### 3.1. Engine thermal model

First purpose of the cooling circuit is to remove heat from the engine in order to keep it at a regime temperature. Therefore, engine behavior is modeled accurately: combustion process is evaluated according to a zero dimensional Wiebe's two parameters model. Intake manifold conditions are known as a function of engine speed and torque. Hence, instantaneous heat transfer between gases and walls is calculated according to Woschni correlation and a modified Heichelberg approach, which considers the variation of heat exchange surfaces due to piston motion. This resulted in a mean heat transfer coefficient and an equivalent gas temperature, allowing a direct prediction of the heat exchanged by the burning gases through the cylinders walls. The engine is represented through an electrical analogy: each cylinder and each correspondent engine head's portion are represented by equivalent metallic masses having thermal capacity property. Each mass represents a portion of metallic cylinder. These masses are linked each other by equivalent thermal resistances. They represent conduction through walls, convective heat transfer between gases and walls and that one between walls and cooling fluid. Therefore, the thermal model of the engine is a wire of thermal masses (representing metallic component) linked by thermal resistances (representing heat transfer). In this regards, the specification of the coolant heat transfer coefficient requires the solution of the model of the hydraulic circuit. Once the fluid passage inside the engine block and head is specified (as well as the cooling fluid repartition among



(a) Sketch of the engine cooling system model

(b) reference circuit (1.4L DV4 TED engine)

Fig. 4: Engine cooling model

different engine parts), the coolant temperature inside the engine can be calculated and, so, thermal power reversed through the coolant is evaluated.

### 3.2. Hydraulic model

It gives the flow rates in each branch of the cooling circuit, according to the equilibrium between pressure rise given by the pump and pressure losses inside pipes and inside main components (heat exchangers, thermostat, etc.). The reference engine cooling system is the one of a 1.4L DV4 TED engine, where the model has been fully validated, and it is shown in Figure 4b.

The pump is described by its characteristic curves (pressure rise VS flow rate) experimentally evaluated at different revolution speeds, mechanically linked through a gear to the engine crankshaft. The hydraulic model of the circuit is a one-dimensional one: a series of equivalent branches (operating in series or in parallel) is used to represent the engine cooling circuit. Length, diameter, material and roughness are the main properties of each duct and pressure losses through the pipes are evaluated by solving the mass and momentum conservation equations. EGR cooler, cabin heater, radiator and all other components along the circuit are properly represented by concentrated pressure losses.

### 3.3. Thermodynamics and circuit components

The cooling circuit is needed to cool engine and fulfill the needs of other engine and vehicle components (EGR cooler, oil exchanger, charge air cooler, cabin heater, air conditioning exchanger, radiator). The dynamic thermal behavior of the cooling fluid is represented considering the energy conservation equation. Thermal exchanges inside each heat exchanger are described by specific balances between coolant and other fluids involved (oil, exhaust gas recirculated, engine charging air, air cabin heater, air conditioning fluids, external air) considering calibrated values of heat transfer coefficient. The radiator has an additional complexity related to the air flow rate that crosses it. Therefore, it requires the solution of an aerodynamic model in order to evaluate such mass flow rate. The model considers two contributions: one is the ram contribution of the environmental air (linked to the vehicle speed and solved in the aerodynamic model of the radiator), the other is the air flow rate imposed by the fan when it is turned on.

## 4. Results and discussion

In order to assess the benefits related to a SVRP installed on a real engine cooling system, the characteristic curves of the pumps, resulted from the experimental campaign, have been transposed in the engine cooling model. In this way, it was possible to evaluate the behavior of the SVRP compared with a traditional centrifugal pump, when a WLTC was performed by the reference engine and vehicle (1.4L DV4 TED on a Citroen C3). First of all, considering a mechanical coupling of the SVRP, a proper fixed gear ratio has been considered between engine crankshaft and pump in order to fulfill the engine heat removal at its maximum working point: this value is equal to 1.158 for the centrifugal pump and to 0.250 for the SVRP.

Figures 5.a and 5.c show coolant flow rate and pressure drop of both pumps considered. It is evident, how the SVRP produces higher flow rates (and, so, pressure drops along the circuit) with respect to the centrifugal pump. Therefore, an optimized actuation of the SVRP can be introduced varying the pump geometry with realtime or slightly delayed approaches in order to restore the coolant flow rate imposed by the centrifugal pump. This actuation might be accomplished modifying the eccentricity of the SVRP, i.e. the distance between rotor and stator centers whose maximum

value, achieved when the rotor becomes tangent to the stator, is equal to the radii difference. The dependency between eccentricity and flow rate is twofold: on one hand, eccentricity influences the magnitude of the maximum suction volume displayed in Figure 1a since it reduces, by definition, the gap between inner stator and outer rotor surfaces. On the other hand, when eccentricity decreases, the recirculation volume that builds up across the exhaust and intake ports increases, provided that the angular range between the above mentioned ports is equal to the vane width to ensure a proper operation of the device. These two volumes assume the same extent at zero eccentricity and, in turn, lead to a condition of zero flow rate. From a physical point of view this situation corresponds to a continuous rotation of the fluid inside the vanes without filling and discharging processes.

Considering that the exceeding coolant flow rate was known from Figures 5.a, the target eccentricity value to use as set point in the control strategy was calculated using the geometrical model for sliding vane machines developed by the Authors in [20]. In particular, since the employment of a variable flow rate pump allows to reduce the engine warm up time, the eccentricity control system is only related to the cooling fluid temperature. Indeed, considering that maximum cooling temperature remains constant, no specific requirements or links are required neither to the thermostat valve nor to the cooling fan control.

Figures 5.b shows the eccentricity variation, with respect to the maximum geometrical value of the sliding vane pump, correctly adapted in order to have the same flow rate of the centrifugal pump. As it seems very difficult to realize a realtime actuation of the eccentricity, a moving average is proposed in Figures 5.b with 5s and 10s delays. In this way, the actuation is smoother and can be more easily achieved, without producing significant variations on SVRP mechanical power and energy saved.

Thanks to such actuation, in Figures 5.a and 5.c, centrifugal pump line (in black) and optimized SVRP line (in green) are perfectly overlapped. In this way, previous engine cooling conditions realized by the centrifugal pump, are ensured and extra flow rates and pressure drops of the mechanical SVRP (which brings to higher hydraulic powers) are avoided. Moreover, this actuation gives an extra degree of freedom in engine thermal management strategies: it can be used to reduce (or make zero) the coolant flow rate during the engine cold start phase to shorten the warm-up time and, in heavy-duty applications, it can be employed to adjust the flow rate according to the engine load during long-haul so leading to fuel saving and emissions reduction.

Since hydraulic power depends on coolant flow rates and pressure drops realized along the circuit, mechanical SVRP has almost always higher hydraulic powers during a WLTP driving cycle. Nevertheless, SVRP has significantly higher efficiency in both configurations (mechanical and optimized one, Figure 5.f). Indeed, during a WLTP cycle, SVRP has a mean efficiency 6-7% higher than the centrifugal one. Only in the region of high engine working points (last 400s of the driving cycle) efficiencies are comparable. Hence, mechanical power requested by the pump can be easily calculated by knowing hydraulic power and pump efficiency: SVRP has always a lower mechanical power with respect to the actual centrifugal pump and this is more noticeable with the eccentricity actuation (Figure 5.d). A mean reduction of mechanical power is accounted for 12% with a SVRP mechanically linked to the engine crankshaft and reached 22% if an eccentricity actuation is considered.

Figure 5.e eventually shows the mechanical energy saved during a WLTP cycle thanks to the introduction of the SVRP. Standard SVRP configuration and the three strategies of the eccentricity actuation have been compared to the centrifugal pump. With a mechanical SVRP an energy reduction of about 5% with respect to the centrifugal one is achieved, but this saving can reach 22% if an optimized SVRP, with an eccentricity actuation, is considered.

## 5. Conclusions

This research work investigated through experimental and numerical approaches the employment potential of sliding vane devices in automotive systems, namely for engine cooling applications. A test campaign on an equivalent passenger car engine cooling test bench showed greater efficiencies for a fixed eccentricity SVRP with respect to a conventional centrifugal pump, especially at low flow rates.

The experimental maps of both devices were further included in a comprehensive model of an engine cooling system to assess the performances of both pumps along a WLTC. Even though a proper gear ratio between engine crankshaft and pump shaft was considered to fulfill the request of the maximum engine working point, simulations showed that fixed eccentricity SVRP had higher coolant flow rates and pressure drops with respect to the centrifugal pump, leading to a higher overall hydraulic power. To overcome this issue a variable flow rate SVRP was proposed, whereas flow rate change was accomplished modifying the pump geometry through a change of the eccentricity between stator and rotor. This innovative design solution led to significant improvements of the pump efficiency: if

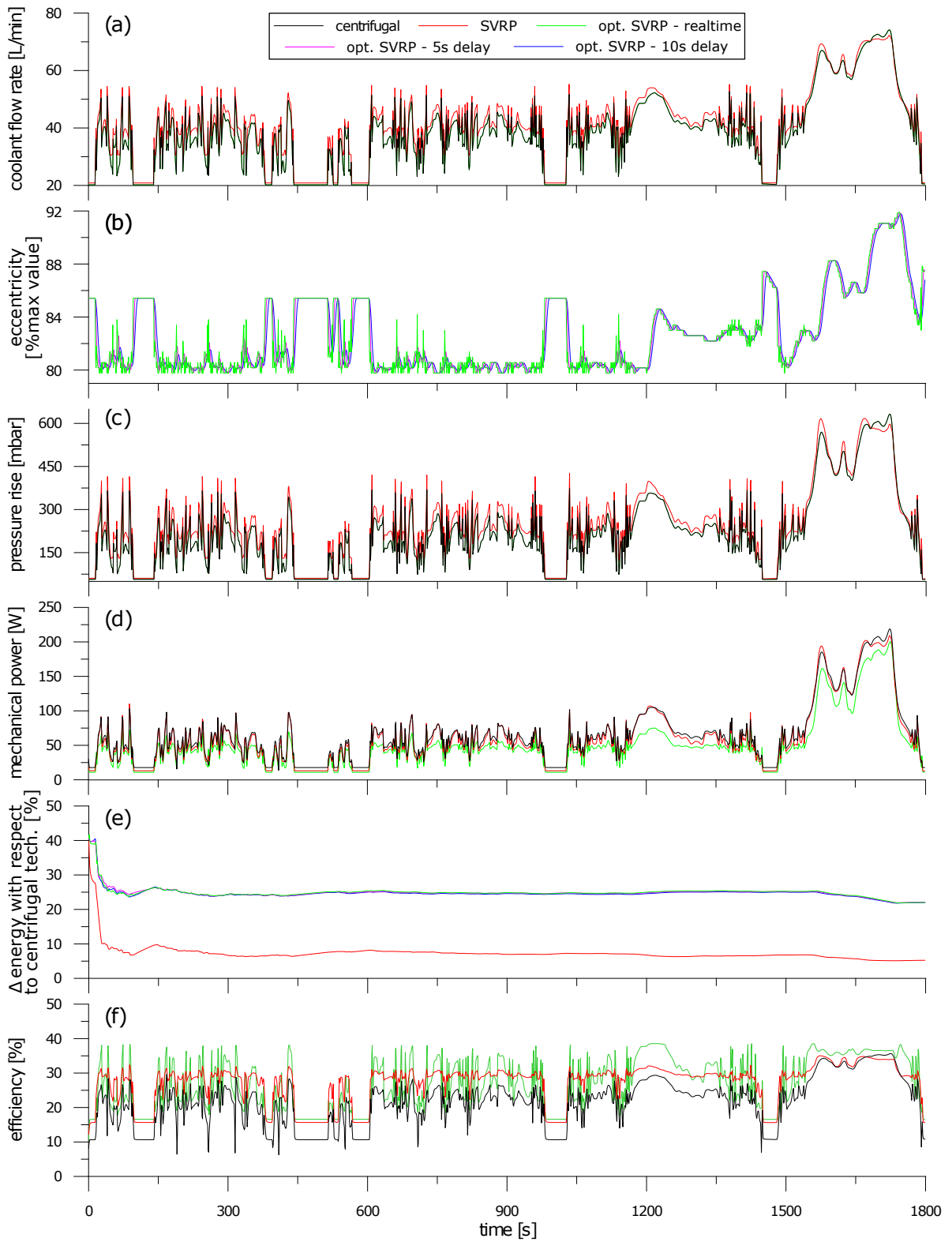


Fig. 5: Operating conditions and performances of centrifugal and sliding vane automotive pumps over a WLTC

mechanically driven SVRP resulted in an efficiency 6-7% higher than the centrifugal pump one, the one of eccentricity actuated SVRP is even greater. Definitely, if this optimized actuation is considered, the mechanical energy requested by a SVRP over a WLTC can be about 22% lower than the one requested by a centrifugal pump. This value, however, is strictly dependent on the working conditions: higher benefits are related to lower engine loads and, therefore, higher energy saving is achieved in low powerful drive cycles. Considering the reference passenger car of this study (Citroen C3) and the reference engine (1.4L DV2-TED) the CO<sub>2</sub> reduction related to the introduction of the SVRP is of about 0.5 g/km, which still is a remarkable value in the automotive sector.

As concerns a prospective employment of sliding vane pumps in the automotive sector, SVRPs should not lead to drawbacks in terms of overall vehicle weight since they will be made of thermoplastic materials. Furthermore, for a given delivery flow rate, a proper selection of pump aspect ratio (length/diameter) and operating revolution speeds will allow to properly address space requirements issues and to reduce friction losses between pump blades and stator such that benefits on the overall reliability of the device will be achieved.

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