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Development of a Sliding Vane Rotary Pump for Engine Cooling

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Abstract

The efficiency of a pump for engine cooling system in automotive sector can be very low (15%-20%) during the homologation cycle which is more oriented to medium and low engine loads. Actual pump technology makes reference always to centrifugal pumps, which suffer in terms of efficiency when the speed changes as well as when head and flow rate delivered. In order to reduce the power absorbed by the pump, a different type is needed.
A sliding vane rotary pump (SVRP) is a serious alternative having all the characteristics to fulfil the engine cooling circuit with high efficiency and reliability. In this work, a SVRP has been designed, built and tested for an existing engine cooling circuit: its performances were compared to the traditional (centrifugal) pump which today is mounted on that engine. The benefits over the homologation cycle in terms of mechanical energy and CO_2 saving have been emulated thanks to a comprehensive mathematical model.

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Keywords: engine cooling; sliding vane rotary pump

1. Introduction

As part of a policy aimed at reducing air pollution and optimizing fuel consumption, over the years, regulations governing the automotive industry have become increasingly stringent. In response to these needs, the research has therefore stepped up its efforts in the direction of several technological options which can participate to an overall energy consumption and emission reduction of the vehicle (Fig. 1).

In Fig. 1, "thermal management" is referenced as the plurality of the technologies which are referred to the engine cooling and vehicle thermal needs.

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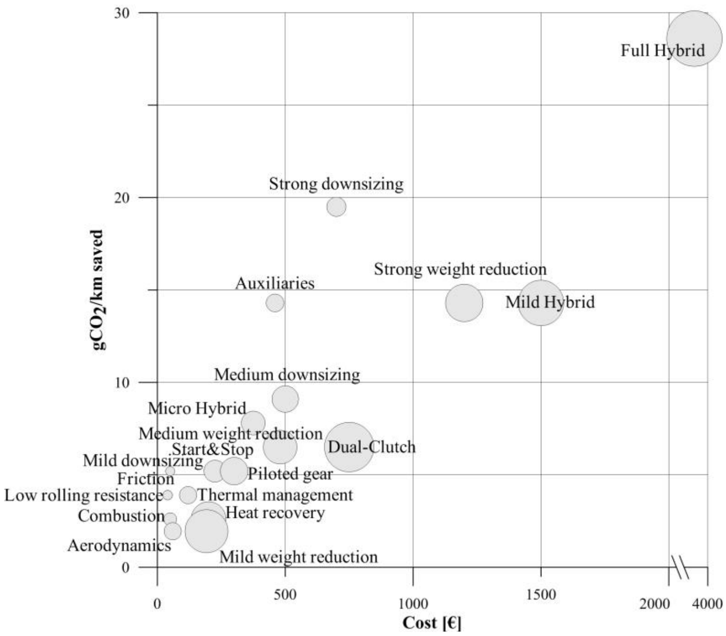

Fig. 1: vehicle cost increase expected and potential CO_2 saving for automotive technological options. Ball size represents specific costs [1]

Among the different options available, engine cooling system, which have remained almost unchanged in last decades, could produce significant fuel and CO_2 savings with a very low expected cost increase if revised. Fig. 1 shows that a cost of 25 €/gCO_2 is acceptable. In addition to the fundamental purpose of removing heat and maintaining the mechanical parts of the engine at a temperature compatible with the proper functioning and strength of materials, during last years other important functions have been added to the cooling systems. For example, exhaust gas recirculation and compressed air cooling as well as integrating important thermal needs of the vehicle (cabin heating and cooling) which progressively increase the overall complexity and specifications (radiator size, heated thermostat).

The fundamental component in the engine cooling system is the pump: it must fulfill the circuit requests in terms of coolant flow rate and pressure rise, which depends on fluid dynamic flow passages and layouts. Typically, a centrifugal pump driven by the engine is used in such system, with a significant low efficiency especially at off-design conditions, as they usually are in an internal combustion engine in the transportation sector due to its speed variations and to the head and flow rate requested by the engine.

In recent years, cooling fluid circulation control was the main aspect: electrical water pumps or electromagnetically or mechanically switchable water pumps have been proposed [2, 3, 4] and put on the market. New pump arrangements were proposed including devices to stop the flow or to reduce it according to controlled variables [5, 6, 7]. All of these were mainly directed to the reduction of the engine warm up time, this phase being responsible of 60% of the overall harmful emissions during a homologation cycle [8]. A faster warm up, also, participates in a positive way to the CO_2 emission reduction, thanks to the benefits on the mechanical efficiency due to friction reduction. A system view of all the thermal needs on board (engine & vehicle) is expressed by other cooling technologies which introduce multiple cooling circuits (operating in parallel) at different temperature levels [9, 10, 11], rearranging engine thermal needs with those of the vehicles [12, 13]. Other studies are focused on the benefits of phase changing coolant to seriously decrease the flow rate delivered [14] or insuring different cooling temperature levels between head and engine block [15, 16, 17]. All these technologies are mature but, for cost reasons, they are struggling to reach a wide application: some of them were also conceived to improve traditional engine performances and component size.

It is important to remark that all these technological proposals never regarded the pump itself. Actual pumps are of a centrifugal type, often characterized by very rough designs. Inlet and outlet sections, as well as impeller and volute passages are far from being optimized from a fluid dynamic point of view: geometrical constraints, usual pump mounting situations, cost reduction issues and carelessness on auxiliary systems have prevented the reaching of high pump efficiencies and also the best operating condition for critical components inside the pump (sealing systems and bearing).

Moreover, current centrifugal pumps are designed at engine operating conditions close to the maximum mechanical power. A maximum cooling fluid temperature increase is fixed (≈ 15 °C) when it crosses the engine and, once the thermal power to be removed is known, the flow rate is defined. Usual values are of the magnitude of 100-200 l/min in a medium size passenger car. Only today on new or revised engines, thanks to specific studies, overall pump efficiency at best efficiency point is close to 40%-50%: several engines are still characterized by pump geometries so unchangeable to have very low efficiencies. Even in the case of optimized pumps, during the most part of the homologation cycles (with the thermostat or a similar component closing the branch toward the radiator - i.e. engine cold -), the efficiency of the pump is very low (15% - 20%). Hydraulic efficiency drops down mainly for fluid-dynamic reasons but also for the power absorbed by the bearing and conventional sealing systems which, at reduced flow and head, has the same amplitude of the hydraulic power.

Pumps which do not suffer such efficiency reduction are of a rotary volumetric type. Among those, sliding vane rotary pumps (SVRP) have additional advantages in terms of geometrical shaping (aspect ratio), weight, reliability and constructional features. Fluid dynamic losses are mainly related to the filling and emptying process which could be easily limited by an optimum ports design and, intrinsically, they are not influenced by the pump speed. Moreover, volumetric efficiency (fluid leakages among vanes) at lower speed can be kept high with some additional devices which help the blades to immediately reach their final position (on the stator surfaces or when re-entering inside the slots). Friction, instead, could be under control thanks to a suitable tip blade shape and thanks to arrangements which reduce relative speed among tip blade and stator. Rotary vane pumps, finally, could be easily managed in order to reduce flow rate (without changing rotational speed) modifying the eccentricity between stator and rotor.

In this paper, the Author discusses the benefits related to the introduction, in an engine cooling circuit of a Citroen C3 car, of a SVRP. This novel pump was designed in order to fulfill the previous cooling requests of the original engine and, then, prototyped. A wide experimental campaign done on this pump gave a full comparison with a traditional centrifugal pump which originally equipped the engine, demonstrating the higher efficiency. The fluid dynamic characteristics of this new pump were introduced in a comprehensive mathematical model of the engine (and of the correspondent vehicle) which was extensively experimentally tested and validated [19, 20] on that car and on the engine propelling it. The model has already demonstrated to be suitable as a virtual platform for engine cooling optimization. The mechanical energy consumption was calculated for the two pumps when the vehicle equipping that engine runs the European homologation cycle (NEDC). The benefits in terms of mechanical energy absorbed, fuel and CO_2 saving are discussed.

2. SVRP design and testing

In this work, an innovative water pump for the cooling circuit of an internal combustion engine has been proposed. The idea was conceived by the fact that traditional (centrifugal) cooling pump has fluid dynamic losses dependent from revolution speed, flow rate and head. A volumetric pump, instead, acts like a device which tends to impose the flow rate ("flow rate generator"): pressure delivered is fixed by the downstream circuit which reacts with a given backpressure if a flow rate is forced inside it. The choice fell on Sliding Vane Rotary Pump, that has higher efficiency, high flexibility and a very good manageability and reliability. So, a SVRP with 3 vanes has been designed having specified a design point in terms of flow rate (100 l/min) at a given pump speed (1000 RPM). The prototype has been built by the Ing. Enea Mattei S.p.A. company. In order to improve pump reliability insuring absence of leakage and noise, a new mechanical seal supplied by the Meccanotecnica Umbra Group S.p.A. company has been used to further reduce, the mechanical power absorbed with respect to a conventional seal.

The performances of the SVRP have been studied on a test bench which reproduced pressure drops seen by the pump and fluid temperature. Pressure drops of the circuit are produced by a flush valve and an electric heater is used

to simulate the effect of the engine on the cooling circuit. Flow rate is measured thanks to a magnetic Foxboro IMT25 931HA flow meter. Revolution speed and torque of the pump were measured and controlled by a brushless electric motor which gave the revolution speed and torque measurements. A proper mixture of water and glycol at 90°C was used in order to reproduce the real cooling fluid.
Considering the accuracy of the sensors used, overall uncertainty for the efficiency was close to 2%.
Fig. 2 shows the results of the experimental campaign: dashed curves represent the characteristic curves of the SVRP for different revolution speed (from 50 rpm to 1400 rpm). It reports also the pressure losses of the circuit of a Citroen C3 car and of the engine propelling it (1.4 DV4 TED engine) when the thermostat was opened or closed: on this car the effectiveness of the new pump was evaluated. The deviation of the characteristic curves of the pump from vertical lines corresponds to the difference between ideal and real behavior: for a given revolution speed, when pressure increases flow recirculation happens inside the vane's pump and real flow rate delivered decreases.

In Fig. 2 the comparison between the two pumps on the characteristic curves of the cooling circuit with the thermostat opened or closed are reported: a set of operating points have been reproduced on the hydraulic test bench on the pressure-flow rate curves of the engine (and vehicle). Data on Fig. 2 refer to: (a) SVRP and engine speed; (b) pressure and flow rate delivered by the two pumps; (c) fluid dynamics and mechanical power of the two pumps; (d) volumetric efficiency of the SVRP and overall pump efficiency. The benefits are evident for that which concerns the SVRP. It is important to remark that the ratio between the two revolution speeds (engine and SVRP) remains almost unchanged: this means that in real conditions a fixed gear can be used.

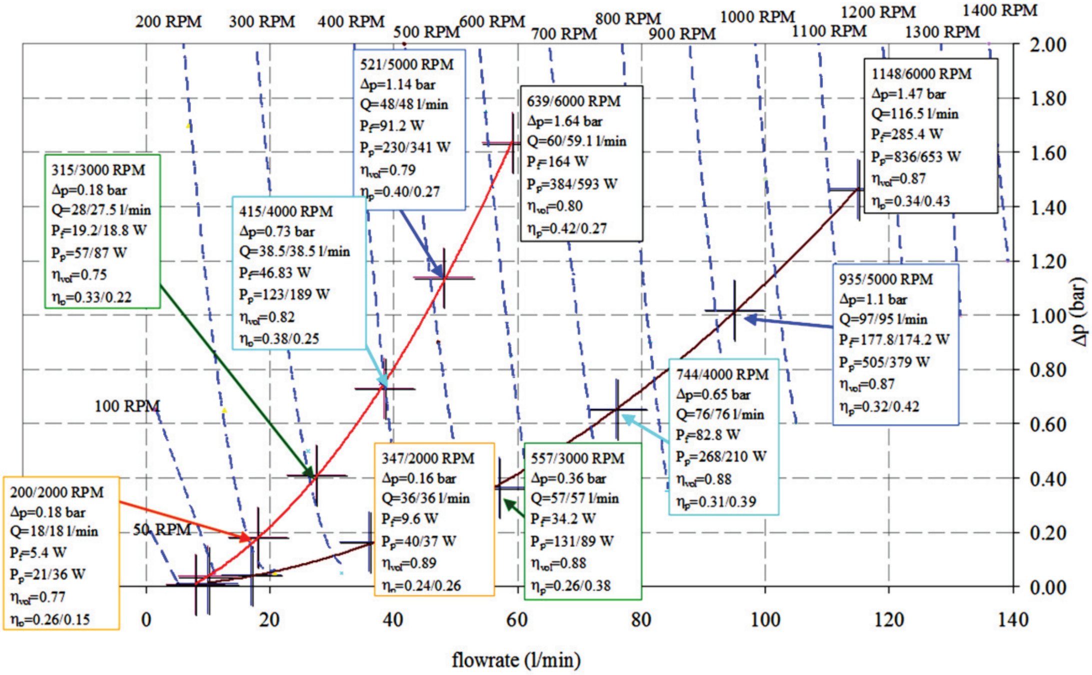

Fig. 2: experimental working point of the cooling circuit with open (brown line) and closed thermostat (red line): comparison between SVRP and centrifugal pump

Fig. 3 shows the efficiency (η_P) of the both pump in the cases of open and closed thermostat. It is evident the real higher efficiency of the SVRP on a circuit with closed thermostat, while when the thermostat is open the efficiencies are similar: when the thermostat is closed (the most part of the homologation cycle) the SVRP efficiency is about 10-15% higher than the centrifugal one.

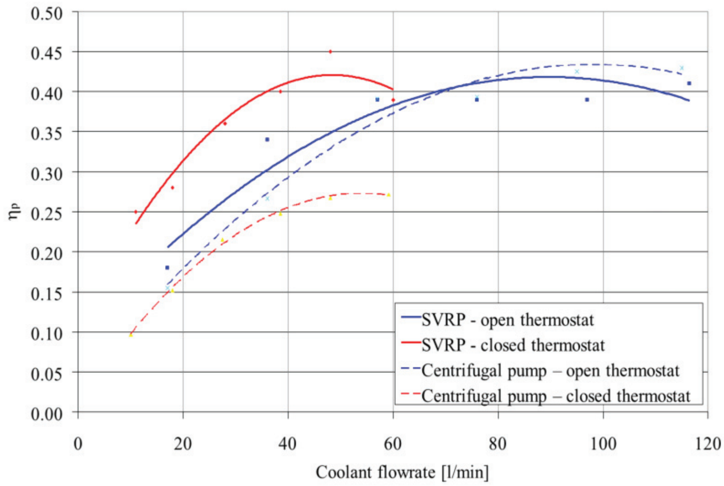

Fig. 3: comparison between pump efficiencies on tested working points

3. Energy and CO_2 emission saving

Substitution of the actual pump with a new type SVRP, the greater efficiency and lower absorbed mechanical energy will necessarily result in lowering of the CO_2 emissions. In order to evaluate the avoided emissions, the NEDC has been chosen as reference mission profile and the mechanical power requested by the two pumps has been simulated thanks to a comprehensive mathematical model which describes the engine behavior. The model describes engine processes (intake, combustion, exhaust, etc.) and predict with high accuracy the mechanical performances (torque) of the engine and the thermal transients of the engine block and head, as well as main operating characteristics of the components usually present on the cooling circuit [18, 19, 20].

The engine model was specified (and validated) on a 1.4 DV4 TED engine propelling a Citroen C3 car. A deep experimental validation was done concerning mechanical performances and thermal power exchanged by the engine toward the lubricant and cooling circuits [20]. Thanks to its accuracy, the overall model has been used as virtual platform to compare the two pumps in terms of energy absorbed: the model, in fact, represents cooling flow passages inside engine block and head as well as all the branches on which heat exchanger and components are mounted (cabin, EGR and oil heat exchangers, radiator, thermostat, etc.) in terms of concentrated and distributed pressure losses. A deeper description is available in the cited literature.

4. Engine cooling system simulation over NEDC

The characterization of the cooling pumps (both centrifugal and SVRP) was used within the virtual model of the engine cooling system in order to evaluate the benefits related to the use of a SVRP during an homologation cycle. Starting from cold conditions, the model simulates engine warm up and main characteristics of the cooling circuit: in particular the power instantaneously absorbed by the pump and, so, the energy absorbed during a cycle.

When the NEDC is specified, the mechanical power requested by the vehicle can be calculated according to a representation of the vehicle based on a single equivalent mass on which the equilibrium of the acting forces is applied and propulsion power is calculated. Then, engine speed is known also considering the tire radius and the engine-shaft gear ratio. At that engine speed would correspond a mechanical power previously calculated, hence the torque that the engine has to deliver. From engine data (speed of revolution and torque), the boost pressure is known as well as air/fuel ratio: the overall engine mathematical model can be run and all the quantities calculated (cylinder and head temperatures, flow repartition among branches, mechanical losses, thermal power exchanged toward cooling circuit, etc…).

In this paper a focus has been done on the pump performance and, in particular, on the sequence of pressure delivered, flow rates and efficiency. Both pumps (the centrifugal and the sliding vane one) have been considered as mechanically linked to the engine, but with a different gear ratio in order to keep the same volumetric flow rate at design condition: in the original equipment it is equal to 1.39, when equipped with a SVRP it is 0.25.

Fig. 4 shows the coolant flow rate in both cases during a NEDC cycle: performances on high load are maintained, but at lower loads the SVRP gives a slightly higher flow rate (+2.5%). This value, however, is not very significant and, in fact, does not affect the thermal behavior of the coolant and the engine (warm up): therefore, the two pumps can be considered perfectly equivalent in terms of engine and vehicle functions. Fig. 5 shows the pressure difference delivered by the two pumps, which corresponds to the characteristic of the cooling circuit with closed thermostat till to 800 s, and for the remaining part to that of its specific opening degree.

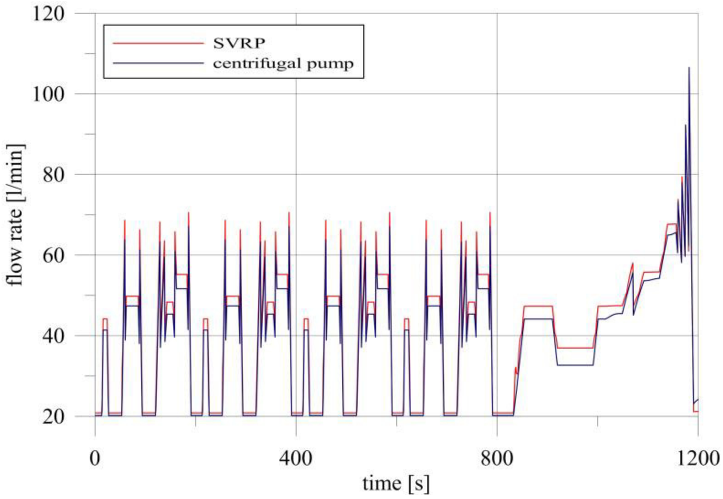

Fig. 4: coolant flow rates of the two pumps on a NEDC cycle

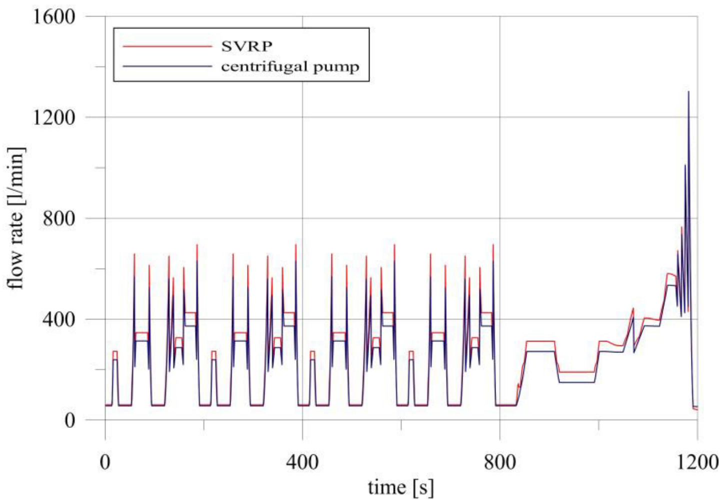

Fig. 5: pressure rise of the two pumps on a NEDC cycle

The data concerning flow rate and pressure rise during time allow the calculation of the mechanical power absorbed by the SVRP and by the original pump: this calculation requires the knowledge of the pump efficiency which is dependent on the two variables head and flow rate: Fig. 6 and Fig. 7 report the working points of the pumps when a NEDC is run. It is evident how the SVRP operates at higher efficiency with respect to the centrifugal pump. In terms of mean values, the centrifugal pump has an efficiency of 0.20, while a SVRP reaches a value of 0.28 (about +40%).

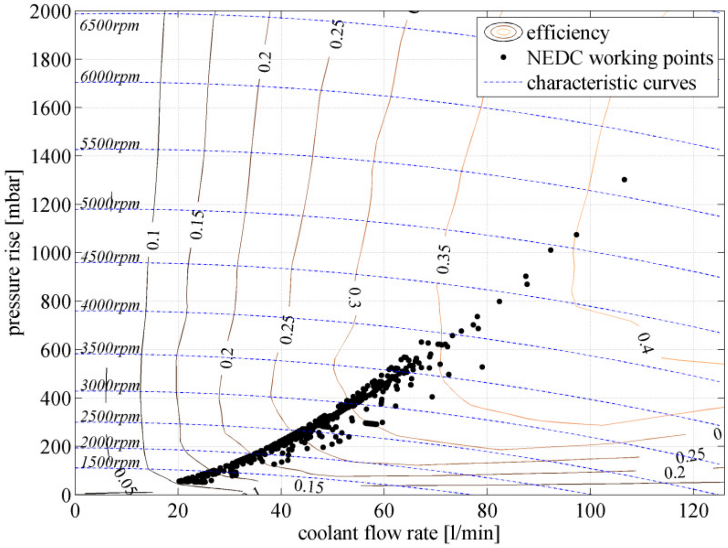

Fig. 6: efficiency map of a centrifugal pump and its working points on a NEDC cycle

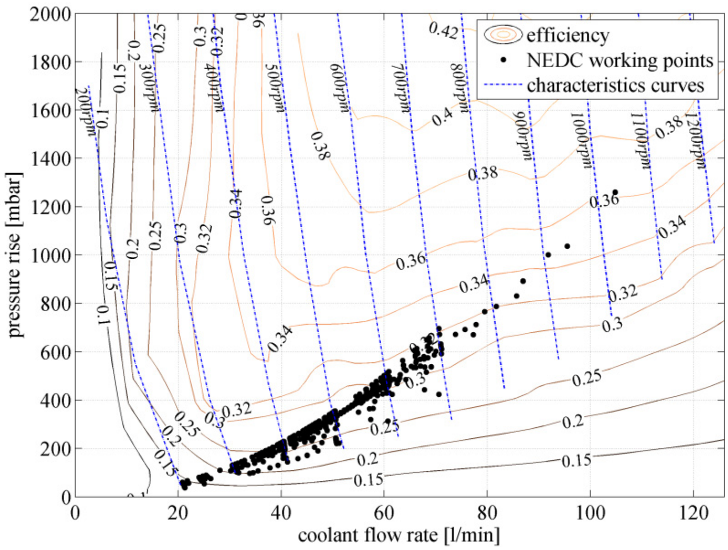

Fig. 7: efficiency map of the SVRP and its working points on a NEDC cycle

The improvement of the efficiency is clearly shown in Fig. 8, where it is evident the higher efficiency of the SVRP at low and medium loads: during the extra-urban part of the NEDC cycle, the centrifugal pumps increases its efficiency (speed, head and flow rate which get closer to the operating conditions to the design point).

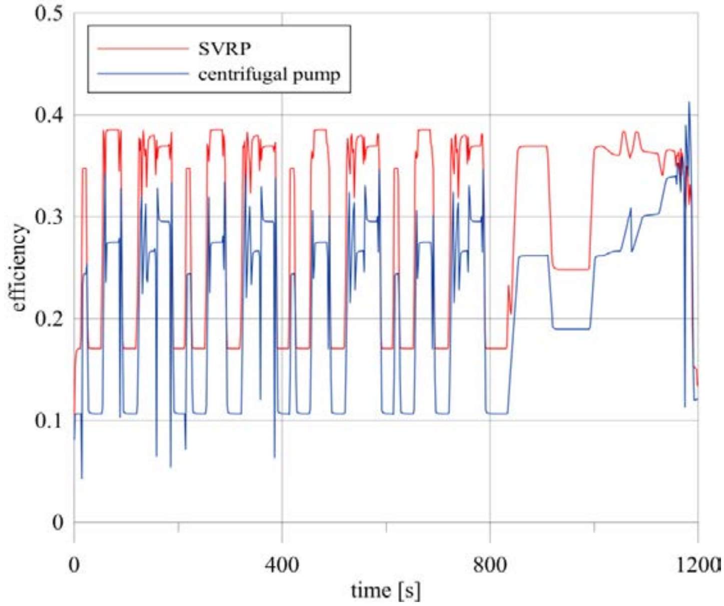

Fig. 8: Efficiency of both pumps on a NEDC

The overall mechanical energy required by the pump can be easily calculated by knowing the instantaneous mechanical power absorbed: if the pumps are compared as they are (accounting for the extra flow rate produced by the SVRP), mechanical energy reduction of about 10 kJ (-15 %) is in favor of the SVRP based on the previous energy absorbed. This saving represents 0.5 % of the overall mechanical energy for propulsion. A preliminary environmental improvement due to the SVRP can be done simply referring the same percentage to the fuel saved and CO_2 emission avoided: in terms of CO_2, the datum shows a decrease of about 0.5 gCO_2/km which is a datum worthy of attention.

5. Conclusions

The pump which guarantees the cooling fluid circulation inside internal combustion engines has been always of a centrifugal type: other pump types have never been considered as technological alternative. Unfortunately, these pumps have efficiencies strongly dependent on speed and flow rates and head delivered. When the engine propels a vehicle during a homologation cycle, its mean operating conditions are in a low speed and load region. This results in a very low pump efficiency since the pump is operating far from its design conditions set at maximum engine power (high load and speed). This produces an amount of mechanical energy absorbed by the engine which could be reduced adopting a new pump type whose efficiency is less dependent from operating conditions.

A new pump for cooling application has been presented: it is of a sliding vane rotary type and its efficiency is not so dependent on pump speed and operating conditions. The pump has been designed in order to replace the actual pump working on 1.4 DV4 TED Engine on a Citroen C3 car, built and tested. The pump characteristic curves have been introduced in in a comprehensive mathematical model which reproduces real 1.4 DV4 TED engine cooling circuit and vehicle operating conditions. A NEDC has been simulated and the working point of the pump predicted: on the same engine and vehicle, the original equipment was known (characteristic curves of the original centrifugal pump).

The operating working points of the cited engine propelling the specified vehicle have been calculated, as well as all the thermal and hydraulic properties of the cooling fluid; the engine warm up phase has been predicted as well. For both type of pump, centrifugal and SVRP, the instantaneous mechanical power and energy absorbed during a NEDC have been predicted and the results compared each other. Mean efficiency of the original pump was 20 % while the SVRP reached a value of 28 %. When the difference is compared with the mechanical energy requested by the vehicle, the mechanical energy saving is almost 0.5 %; a preliminary estimation of the CO_2 saved ranks at 0.5 g/km: in this datum, a slight extra-flow produced by the SVRP with respect to the original equipment was accounted for. The datum is worth of attention: in a technological environment which will bring the CO_2 emissions toward more severe targets (95 gCO_2/km by 2021 for the class of vehicle considered in this paper) and the introduction of a 95 Euro fine for each gram of CO_2 which exceeds these targets, the SVRP seems to be a serious alternative.

Acknowledgements

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