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Fluid-dynamic analysis of an in-line water piston pump

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Abstract

The paper focuses on the analysis of inline water piston pumps, by means of simulation of the fluid-dynamic behaviour and experimental characterization. These pumps are commonly used on the car wash systems, industrial washes systems and fire protection systems. They possess a robust architecture, made of a typical rod-crankshaft mechanism, which transforms the circular motion of the pump shaft in the reciprocating displacement of the piston. The pump analyzed in the paper, in particular, moves three pistons, shifted one another of 120°; each piston sucks from a tank and delivers water to a hydraulic line via automatic, spring loaded, poppet valves. The shaft, rod and other movable components are lubricated with mineral oil; the three ceramic pistons are isolated from this environment using opportune seals and work with water. This pump is robust and durable but suffers of some problems: the instantaneous pressure trend within each piston is ideally a square wave: the pressure is equal to the tank pressure during suction, while it is equal to the delivery pressure during the delivery phase. Instead, non –ideal behaviour of the poppet valves, leakages, fluid properties make the pressure trend more critical: during the pressure transient between the two high and low levels, pressure peaks and de-pressurization till aeration and cavitation occurrence may happen. These phenomena generate vibrations and noise and can damage the pump components. In order to study this, a lumped parameter fluid dynamic model of the pump has been realized, later compared with experimental results coming from the test rig to validate it. The model has been used to explore the dynamic pump behaviour in several operating conditions (various speed values and delivery pressure levels), calculating the flow irregularity, the pressure and forces instantaneous trends. The design characteristics of the poppet valves have been explored, in particular the spring characteristics, to discuss the trend of the previous variables.

Keywords: Water, In-line pump, 0D and 3D fluid dynamic simulations, volumetric efficiency

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1. Introduction

In line water pumps are relevant components of widely spread systems such as car wash systems, industrial wash systems and anti-fire systems. The kind of application in which they are used impose some issues: these components have to be robust and affordable; they are expected to operate in different conditions in terms of speed and pressure with good efficiencies, also considering the chance to have a suction port which is not pressurized; depending on the context, emitted noise and vibrations also can be serious issues.

In this work an in-line pump is analyzed, aiming at its fluid-dynamic behavior analysis. A 0D fluid dynamic model is firstly used to understand the dynamic behavior of the pump, monitoring the pressure transients, the flow rate and pressure at the ports, the timing in the displacement of the pumping elements and valves. From these characteristics the general behavior and possible critical concerns are highlighted. In the following, a 2D fluid-dynamic analysis is performed to better study the flow through the valves and inside the inlet/outlet manifold of the pump itself. These approaches are not new in the positive displacement pumps analysis: in particular, as shown in [1]-[4], positive displacement pumps of different architecture and applications are often modelled and simulated with a lumped parameter approach. The dynamic characteristics of the pump, such as the instantaneous flow, pressure at the ports, shaft torque are well depicted by this kind of approaches, which are able to highlight the influences of the design or geometry modifications on the pump behavior as well as the influence of the pump operating conditions. Moreover, the pump modelled can be integrated in the entire fluid power system to study the interaction between the different elements, with the objective of optimizing the system behavior or efficiency or to study noise and vibrations, as shown in [5]-[8]. However, not all the critical issues of a volumetric pump can be studied with such approach: when considering the delicate lubricating problems [9]-[11], cavitation, and flow velocity and pressure distributions within the pump, 2D or 3D fluid dynamic modelling is a more suitable approach; since all these aspects influence the pump efficiency, it is clear that these analyses are fundamental for the pump behaviour evaluation.

Despite the new potentialities offered by these different simulations approaches, experimental characterization is still considered an important step, either for validating a numerical model ([12]) or to study the robustness of different approaches for the efficiency determination ([13]), or, at last, to study complex phenomena as cavitation and aeration. While the modelling approaches used in this paper are consolidated in the positive displacement pumps used in mineral oil power systems, the kind of pump here studied is not so common to be found in literature. The fluid dynamic behavior of these pumps has been quite neglected in favor of robustness but nowadays, when efficiency and noise are very critical issues, it becomes important too.

In the following paragraphs the reader will find: the in-line pump description, the 0D model approach and the results on the dynamic analysis of the pump, finally the experimental-numerical comparison on the pump volumetric efficiency.

2. Pump description

The in line pump analyzed (Fig. 1) is equipped with three pistons, each one having two poppet valves to manage the suction and the delivery of each piston. The pistons perform an alternating displacement, induced by the rotation of the crank shaft of the pump, at which they are connected with a 120° angular phase shift one another. The pistons move in the pump body and head and are realized in ceramic. The isolated piston chamber within the pump head is alternatively filling and emptying with water; the ceramic piston is connected via a pin to the crank connecting rod. The bottom part of the pump, i.e. the body with the crankshaft and the piston rod are instead operating within lubricating oil. Opportune seals separate the piston chambers from the bottom part. The poppet valves open automatically thanks to the pressure difference between the piston chamber and the inlet/outlet ports. Going into detail, when the piston begins moving from the top dead center (TDC) to the bottom dead center (BDC), the volume in the chamber is increasing, the pressure decreases and, when it becomes lower than the suction port pressure plus the valve spring preload, the suction valve opens and the water can fill the chamber; on the other side, when the piston moves from the BDC to the TDC, the pressure in the chamber increases, the suction valves closes and , as soon as the pressure becomes higher than the delivery pressure plus the delivery valve spring force, the valve opens and the water is delivered.

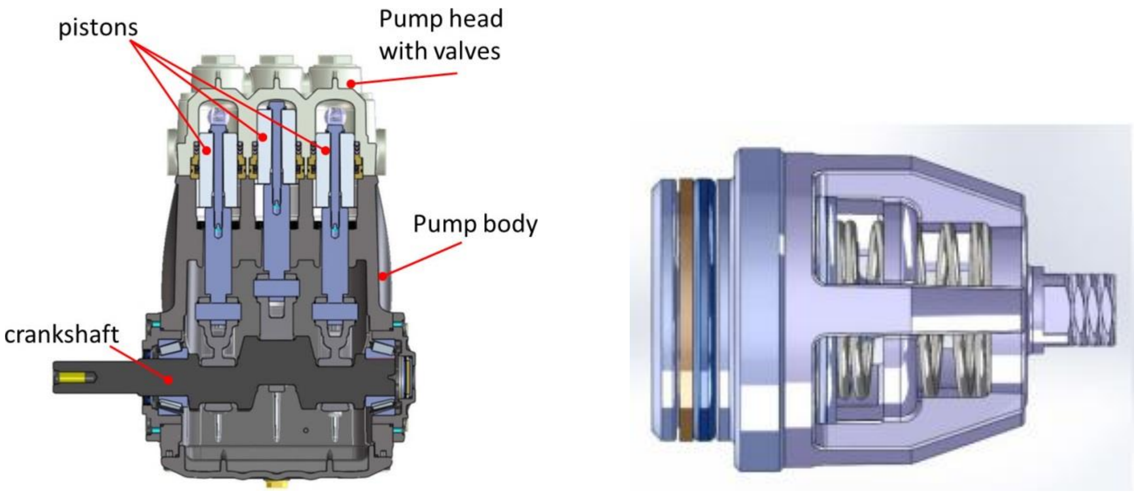

Fig. 1. In line water pump view (left) and poppet valve view (right)

It is obvious that a good phase shift between the piston movements and the valves opening and closing would guarantee smooth transients of pressure both at the TDC and BDC; on the contrary, when this does not happen, cavitation at the beginning of the suction phase and pressure peaks at the beginning of the delivery phase, make the pump operation more irregular, produce vibration and noise, reduce the pump efficiency. The 0D model proposed in this work has hence the aim to study this problem and to correlate the design parameters of the pump with the pressure transient within the piston chambers, and with the instantaneous trend of flow rate and shaft torque.

3. 0D modelling

The 0D fluid dynamic model proposed considers the single piston chamber as a control volume, within which continuity equation is applied and solved:

$$\frac{dp}{dt} = \frac{V}{B} \cdot \left(Q_{in} - Q_{out} - \frac{dV}{dt} \right) \tag{1}$$

In Equation (1), B is the fluid Bulk Modulus, p is the pressure within the pumping chamber, V the volume of the chamber; moreover, Q_{in} is the flow rate entering the volume, Q_{out} is the flow rate getting out of the control volume towards the delivery. The flow rate across the valves is calculated according to Equation (2).

$$Q_{in/out} = C_d \cdot A \cdot \sqrt{\frac{2|\Delta p|}{\rho}} \cdot sign(\Delta p) \tag{2}$$

Where A is the flow area through the valves, C_d id the discharge coefficient (variable as a function of the flow Reynolds number), Δp is the pressure drop across the valve, ρ the fluid density.
The poppet valves are represented as movable elements, whose dynamic equilibrium is described by equation (3):

$$\begin{gathered} m \cdot \ddot{x} = \sum_{i=1}^{n} F_i + F_m + F_{flow} \\ F_m = F_0 + k \cdot \Delta x, \;\; F_{flow} = \rho \cdot Q \cdot v_{flow} \cdot \cos\theta_{flow} \end{gathered} \tag{3}$$

Where m is the poppet mass, F_m is the spring force (k is the spring stiffness, Δx the spring compression, F_0 the spring preload), F_{flow} is the flow force on the poppet, expressed as function of the flow rate Q, fluid density ρ, fluid velocity v_{flow}, and angle of the fluid velocity θ_{flow} with respect to the poppet axis; finally F_i are the friction forces contributions (static, dynamic and viscous) and pressure forces acting on the poppet.

A proper definition of the pump geometry (piston chambers volume and flow areas as a function of the angular position of the shaft) is needed as input data in the model.

The volume of the chamber is a function of the piston displacement as shown in Fig. 2 left side, while Fig. 2 right side the poppet valve flow area is plotted as a function of the poppet displacement; the two valves have the same geometry but the spring stiffness and preload are different. The model is realized in AMESim environment [14] as shown in Fig. 3.

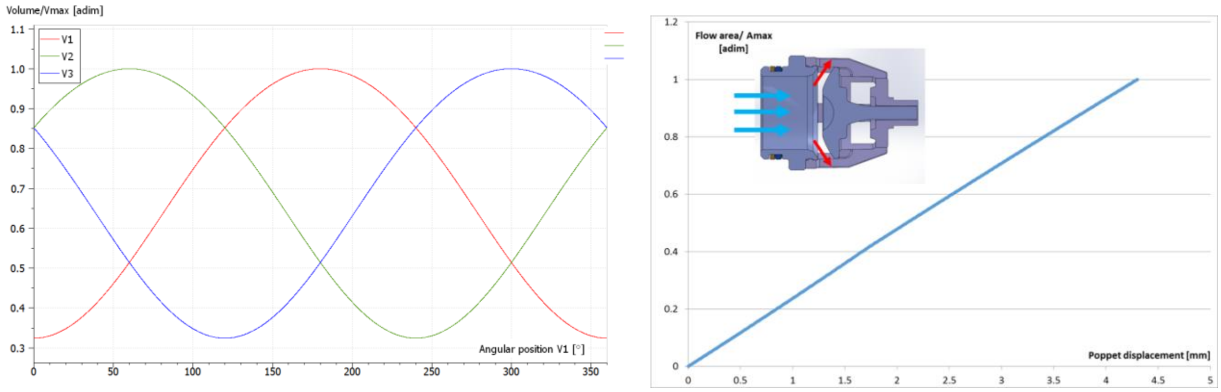

Fig. 2. Piston chambers volume transients as function of the reference shaft angular position (piston V1 at the BDC); poppet valve flow area as function of the poppet displacement

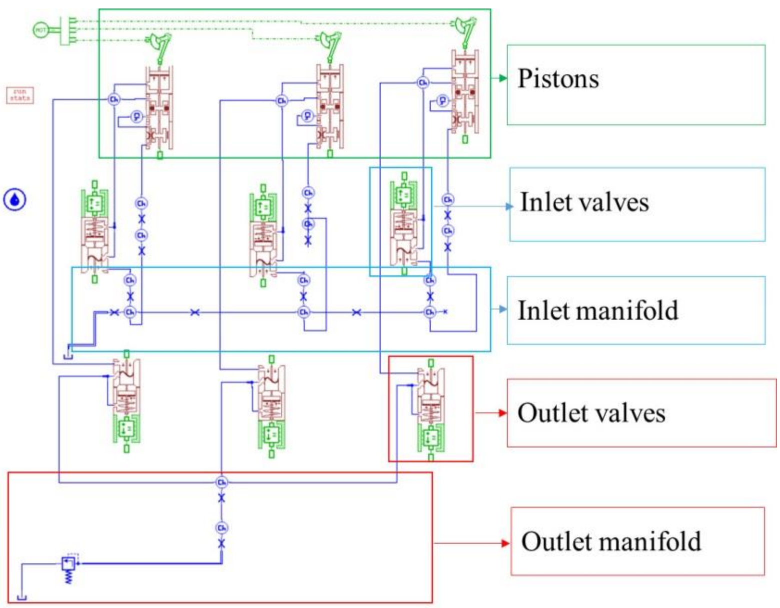

Fig. 3. In line pump model in AMESim

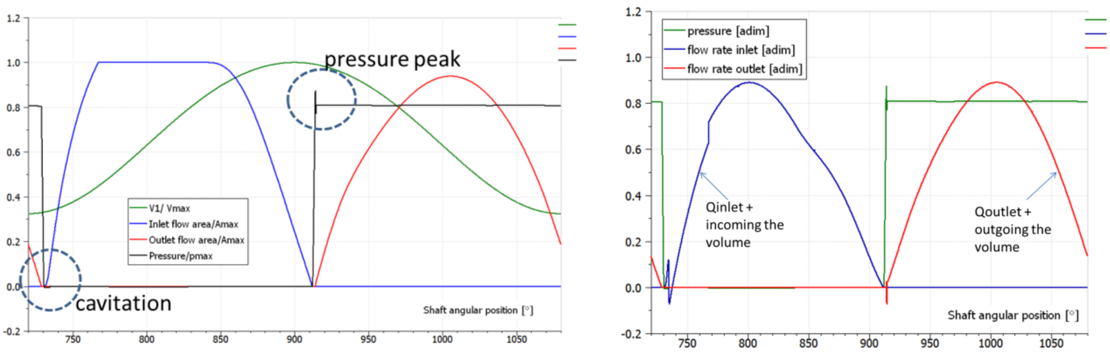

Fig. 4. Instantaneous characteristics of a single piston chamber: volume, inlet and outlet flow areas, pressure transients and flow rates.

Considering a tank at atmospheric pressure and a regulated orifice to pressurize the delivery, the resultant pressure transient in volume V_1 is shown in Fig. 4, where the flow rates incoming and outgoing the chamber are both represented with positive sign to make the understanding easier. Two critical phases are highlighted within the circles: the first one, which is a depressurization in the piston chamber, occurs when the piston overcomes the TDC and it is due to the fact that a small depressurization in the piston chamber is needed to open the inlet valve; the second one is a pressure peak and occurs a little after the BDC, when the inlet valve has already closed but the outlet valve is not open yet, because pressure in the chamber has to rise enough to overcome the valve spring preload. This behavior also causes some instability in the flow rate incoming and outgoing the volume: for example nearby the BDC, the pressure in the chamber does not rise until the inlet valve closes, even if the chamber volume has started decreasing, as soon as the piston has overcame the BDC. Successively, the rapid pressurization opens the outlet valve, but soon after, when the pressure decreases again as a consequence of the flow outgoing the chamber, the flow rate changes sign for a small angular interval and enters the volume.

In an optimized pump, the inlet valve would open a little earlier, nearby the BDC of the piston, thus reducing the angular interval in which cavitation occurs; moreover, the inlet valve would close a little earlier, nearby the TDC, allowing the pressurization of the chamber and, consequently the opening of the outlet valve. The behavior described strongly depends on the pump speed, so the designer has to guarantee a regular behavior in the whole range of the pump operation adjusting the spring choice and eventually modifying the valve geometry.

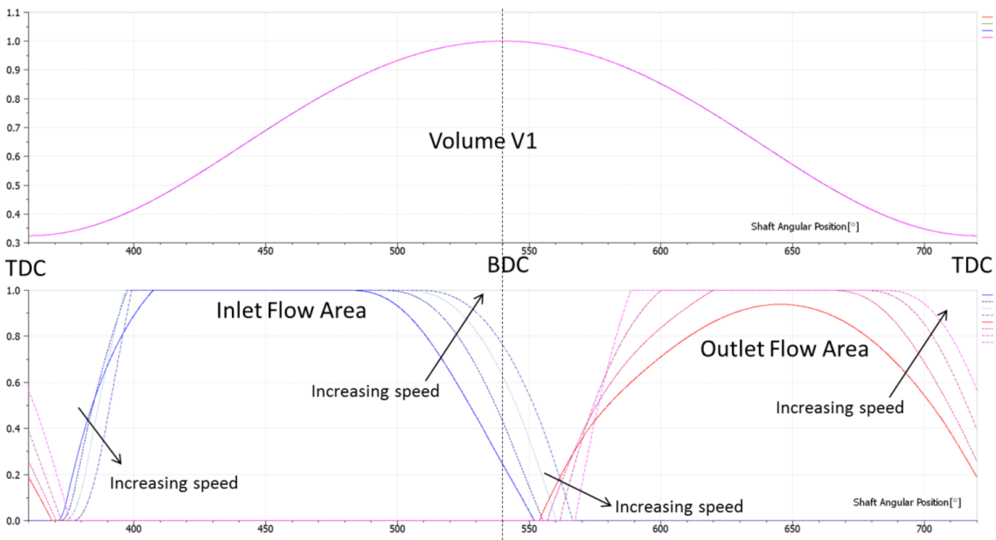

Fig. 5. Instantaneous characteristics of a single piston chamber: volume, inlet and outlet flow areas for different speed values (950-1450 rpm).

In the following figure, the behavior of the pump is analyzed for increasing pump speed values. The opening and closing of the automatic poppet valves strongly change according to the speed: the inlet valve closing and the outlet valve opening are more and more delayed with respect to the TDC, hence also the pressure transients is shifted forward, the pressure peaks are higher as a consequence of the stronger volume reduction caused by the piston displacement and also the permanence of the pressure under the atmospheric value increases, a sign that indicates a possible cavitation occurrence.

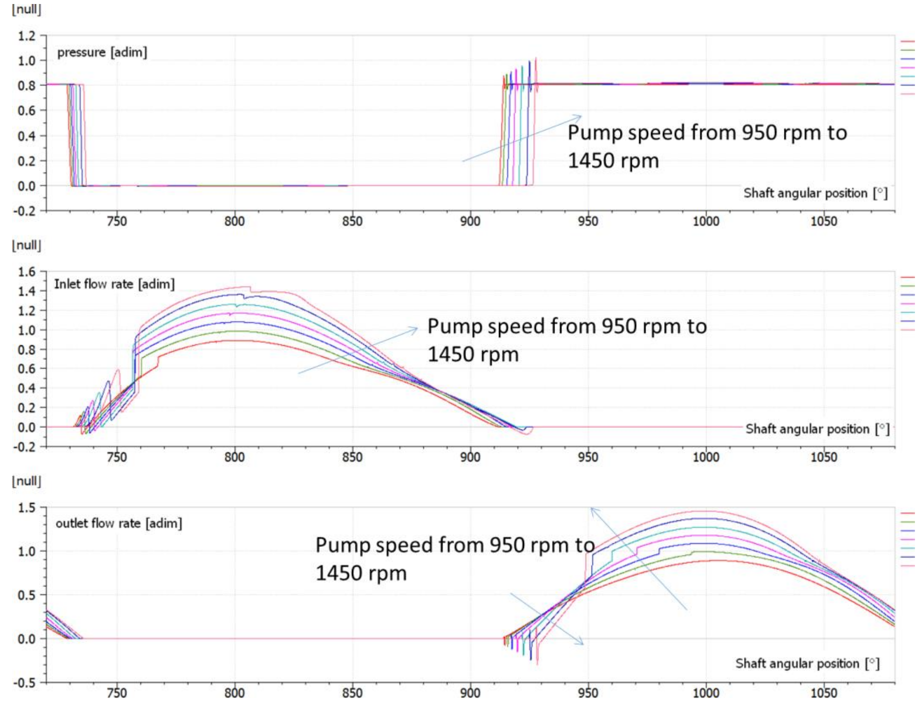

Fig. 6. Results on the pressure transient within V1 and flow rates for increasing shaft speed values.

Obviously, the spring characteristics influence the behavior described and this model can be used to optimize the spring's choice. At the same time, it is important to guarantee a good volumetric efficiency and the valves behavior also plays a role in this. Hence, the pump optimization needs to be developed controlling the pressure peaks and cavitation issues but also the volumetric efficiency. It is clear that a lumped parameter model can only depict the qualitative trend of the pump efficiency, which depends on parameters that are unknown a priori, and has to be tuned by comparison with experimental data.

To do this, the pump was tested on a test rig in different operating conditions, both changing the pressure at the inlet to simulate the suction from a tank or from a pressurized water net, and changing the pump speed and delivery pressure.

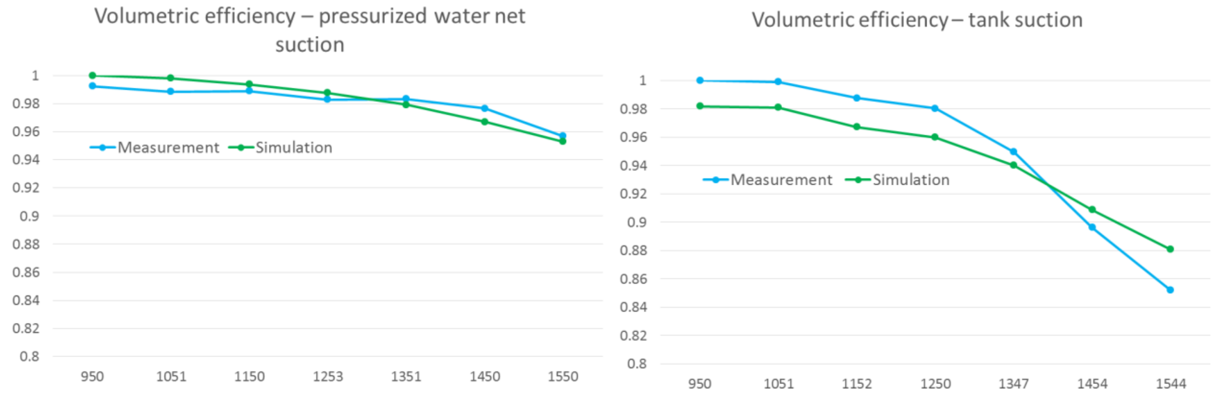

Fig. 7. Numerical and measured “normalized” volumetric efficiency as function of the pump rotational speed.

Looking at Fig. 7, where a “normalized” volumetric efficiency is shown (i.e the ratio between the volumetric efficiency and the maximum value obtained during the tests), a first consideration can be immediately derived from experimental data: the volumetric efficiency decreases with the speed, in an opposite way respect the traditional positive displacement pump (swashplate axial piston pumps, external gear pumps, vane pumps..). In fact, in the case of this in line pump, what affects most the volumetric efficiency are not the leakages (almost null in the part of the pump that operates with water because of the presence of seals on the pistons) but rather the cavitation occurrence at the pump suction and the flowing back of water in the piston chamber from the delivery or at the suction from the chamber. Reducing the risk of cavitation, i.e. using a pressurized suction, heavily improves the volumetric efficiency; the numerical-experimental results comparison is very good when the pressurized suction is considered, while there’s more distance (maximum 2.5%) when the tank is considered This is probably due to the more relevant role played by cavitation occurrence in determining the efficiency in the case of the tank suction; moreover, it has to be considered that cavitation is modelled here with a fluid equivalent approach ([14]) , an effective method when the dynamic behavior of systems are under study (an ”easy” way to consider the change in the fluid compressibility for example), but too much rough to well depict this complex phenomenon. Once ascertained that the model is able to depict the qualitative trend of volumetric efficiency, some modifications in the spring characteristics are proposed. For example in the figure below, springs with higher stiffness have been used at the suction: in particular the spring stiffness at the suction in the actual pump is about the half of the spring stiffness at the delivery, while in the simulation with “higher stiffness spring” the suction valves spring stiffness is taken equal to the delivery valves spring stiffness. Fig. 8 shows that the general improvement that can be obtained when the pump works with a pressurized suction, is maintained only at low speed with a tank suction, while at high speed the higher stiffness spring at the suction valves worsens the cavitation occurrence.

4. Conclusions

In this paper an in-line piston water pump has been analyzed with the aim of evaluating the fluid-dynamic behaviour and the critical issues. This pump is robust and durable but suffers of some problems: the instantaneous pressure, ideally a square wave, presents some points in which cavitation and pressure peaks over the delivery pressure value occurs. As shown in the analysis this depends on the poppet valve opening and closing and on the pump speed; in particular increasing the pump speed the pressure peaks are higher and the permanence into cavitation condition is longer too. Using the model, modifying the poppet valve characteristic and spring it is possible to improve these critical issues, always keeping attention to the volumetric efficiency.

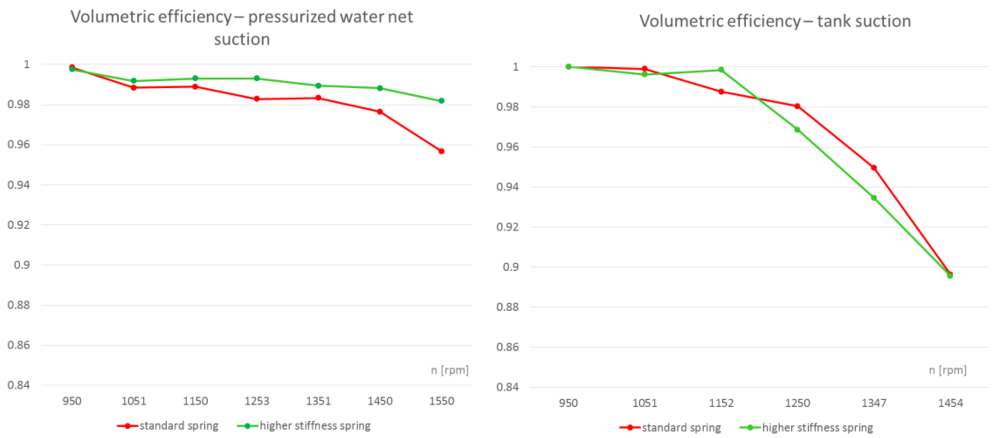

Fig. 8. "Normalized " volumetric efficiency as function of the pump rotational speed for different spring settings of the automatic valves.

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