ENERGY SAVING IN THE HYDRAULIC CIRCUIT FOR AGRICULTURAL TRACTORS: FOCUS ON THE POWER SUPPLY GROUP.

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Abstract: This work aims to analyze a complete hydraulic system of a medium size agricultural tractor, in order to perform an energy dissipation analysis and to suggest possibly alternative configurations and solutions. The fuel consumption and energy dissipation in off road vehicles have in fact become a key feature, given the great attention devoted to the need of reducing pollutant emissions, in order to satisfy the future emission limits. In this work the focus is on the fluid power supply group and the standard architecture of this unit has been compared with an alternative solution, called variable pump margin strategy and compared on the basis of the power consumption to perform the same duty cycle, showing that a relevant percentage of energy may be saved with simple modifications in the hydraulic power generator group.

Keywords: Agricultural Tractor, Simulation, Power Saving, Duty Cycle

1. Introduction

This paper is intended to describe part of the work done as a research program whose final aim is to test and verify the effectiveness of some high efficiency hydraulic architectures in agriculture tractors. Hydraulic circuit in mid-size tractors has different key roles: feeding the auxiliary utilities, the hitch valve, the trailer brake, the steering, the driveline and PTO clutches, the lubrication line and the pilot lines to control the electro-hydraulic valves. Traditional configuration of this circuit involves the use of separate pumps to feed the auxiliary utilities line, the hitch valve and the steering on one side and lubrication on another side and finally the clutches and pilot lines.

The first step of the research is the analysis and the evaluation of the energy dissipation of the auxiliary utilities of a standard circuit, which is a typical load sensing multi-actuators system. The load sensing concept has been introduced in the second half of the last century and it is still used in hydraulic applications because it has proved to be more efficient than the hydraulic power system which entrusted power management to dissipative components (as the traditional pressure relief valve). At the same time, hydraulic load sensing systems are robust and reliable (see for instance [1], [2], [3] and [4]). Nevertheless, the need to limit pollutant emissions of agricultural machines has become more and more pressing in the last years. Many researchers and manufacturers have focused their efforts on reducing the energy consumption in the system without compromising its functionality and its performance. From this point of view, the combined use of simulation tools and experimental testing represents the most promising way to develop alternative solutions characterized by lower energy consumption (see for example works from [5] to [11], which proposes different solutions spanning from independent metering valves to displacement controlled actuators).

In this kind of system, possibilities to achieve more energy saving lay on the fact that high dissipative distributors are used to manage the flow to the actuators and to maintain control of multiple loads eventually working together and that the fluid power group must work under a fixed pressure margin that is higher than the maximum working pressure generally of about 2-3 MPa; this margin of pressure has to be sufficient to overcome all the pressure losses at the maximum flow rate that the flow come across from the pump to the actuators.

Possibility to reduce power losses lay hence both on the power generation side and in the power utilisation side. In this paper the focus is more on the power generation group improvement.
It's noteworthy to underline that hydraulic circuit in a tractor can serve a wide range of actuators, depending on the equipment connected at the moment on the tractor, for example a seeder, or a loader or a harrow. The actuators can be both motors and cylinders and standardized duty cycles for these kind of circuit do not exist at the moment. This requires more efforts in the analysis to be devoted at the experimental measurement of the main hydraulic variables during a working cycle of the tractor with different equipment. On the basis of previous analysis ([12], [13]), here it was chosen to adopt a standard duty cycle created analysing the whole life and average use of a mid-size power tractor. The alternative solutions evaluated and compared with the standard architecture are built through the adoption of electronically controlled variable displacement pumps. In the future, the logical following step of this work will be to select the best solutions from the simulation results and to realize them on the vehicle, experimentally testing on vehicle the ability of new solutions of maintaining the vehicle performances beside the introduction of energy saving.

2 The Standard Load Sensing Hydraulic System and its Modelling

Given the complexity of the hydraulic circuit of the agricultural tractor, it is convenient to use a top-down approach that divides the entire circuit in different operational sections.

The hydraulic circuit is composed by a fluid power supply group, and a complex whole of hydraulic actuators (high and low pressure users) and control valves.

The power supply unit represents the heart of the tractor. This is composed of two pumps (a charge pump of the gerotor type and an axial piston pump) that generate the operating flow rate required by the users. The power supply unit is a typical load sensing open-circuit group that involves the use of local pressure and flow compensators. Advantages and critical aspects of these kind of systems have been thoughtfully analyzed in literature, for example in [14] and [15].When speaking of the energy dissipation of these systems, the critical aspect is that they work at the pump delivery with an higher pressure level than the one required by the load. These pressure difference (pump margin) is regulated in order to be able to guarantee the maximum flow rate request at the actuators and it strictly depends on the losses that flow encounters in the passage from the pump delivery to the user. In the standard hydraulic pump, this pressure difference is fixed, even if it could be reduced when the flow request is not at its maximum level, thus reducing dissipation.

A priority valve is placed after the hydraulic power supply unit; this valve has the task of ensuring the proper operation of the three main users of the agricultural tractor: steering system, trailer brake and agricultural equipment whose actuators are controlled with auxiliary distributors.

The priority valve, consisting of two sections that work in parallel, sets limits of operating pressure and in emergency conditions guarantee a level of flow and pressure to the users with a specific priority.

In connection with the priority valve are the following utilities: steering, trailer brake valve and auxiliary distributor, with their respective lines of load pressure sensing . Depending on the load detected, the valves in the priority block are used to adjust the flow rate and the pressure coming from the power supply unit, ensuring a priority order in the functions of power steering, trailer brake and auxiliary utilities.

The trailer brake valve has the task of managing the pressure signal generated by the brake pedals in the cab and the pressure to be provided for the parking brake of the machine.

The auxiliary actuators may work together hence local pressure compensators are introduced on each section of the auxiliary distributor to guarantee the correct flow sharing. These elements, because of the geometry complexity introduced on the distributor section and of the multiple oil passages needed to perform the control and to guarantee the flow, are critical for the dissipation introduced ([16]).

To complete the hydraulic circuit of the agricultural tractor, the main accessory components are the filter and the heat exchanger.

The hydraulic circuit of the tractor in its standard configuration is represented in Figure 1, by means of a functional representation in which the blocks described before are recognizable. These blocks contain the mathematical models that define the components behaviour, developed in LMS Imagine.Lab AMESim and AMESet, following a lumped parameters approach (Figure 1).

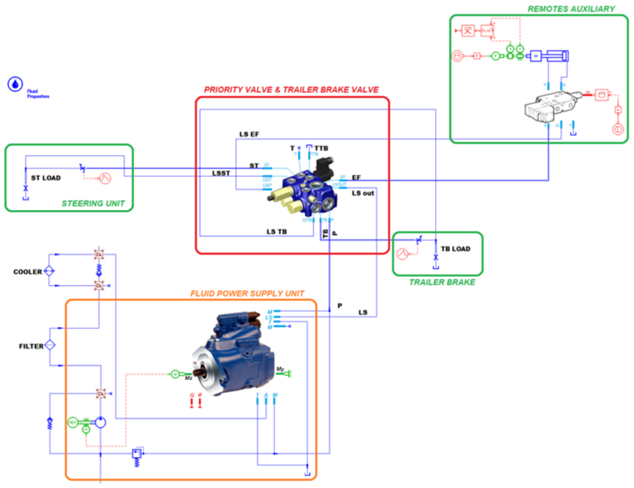

***Figure 1**: Standard Hydraulic Circuit of a medium sized tractor*

The lumped parameters model is divided in three main sections. The first one (orange block in the Figure 1 named FLUID POWER SUPPLY UNIT), includes the detailed model of the axial piston pump of the tractor, together with the flow and pressure compensators, and the model of the charge pump. The instantaneous pressure at the axial piston pump delivery and suction, the flow rate and the torque transients are typical output variables of this block.

The second section (red block), named Priority & Trailer Brake Valve, includes the detailed model of the priority valve and a “functional model” for the trailer brake valve. These models ensure the correct pressure settings and flows-sharing towards the users.

The green blocks contain the hydraulic models of the main users of the agricultural tractor. The steering user and trailer user are modelled by means of variable orifices, suitably controlled and calibrated.

A detailed hydraulic model is instead realized for the auxiliary distributors, since they have shown from previous analysis ([12]) to be quite dissipative elements.

The hydraulic model involve also the filter and lubrication section not shown in Figure 1.

Focussing on the fluid power group, in the following a detailed description is given of the main pump, the variable displacement swashplate axial pistons pump. The hydraulic model realized in LMS Imagine. Lab AMESim environment is an evolution of the model presented by the authors in [17], consisting mainly of three main sections:

-rotating group of axial piston pump, including the pistons moving within the cylinder block, the valve plate, the swashplate

-actuator that perform the adjustment of the displacement of the machine

-flow and pressure compensators: it implements the modelling of the 3 way valves that contribute in the determination of the control pressure within the actuator chamber.

The model allows the determination of the pressure transients in the piston chambers, the instantaneous forces exerted between the movable parts of the pump, the torques at the swashplate and at the shaft and the instantaneous flow. The basic module is represented by the piston chamber which is a typical control volume that communicates with high and low pressure ports through variable orifices, opportunely controlled in order to replicate the opening and closing of the flow areas due to the cylinder blocks and valve plate interposition.

For each control volume continuity equation can be written under the hypothesis of an incompressible fluid and considering isothermal flow:

$$\frac{dp_i}{dt} = \frac{B}{V} \cdot \left(\sum_k Q_k - \frac{dV_i}{dt} \right), \quad i = 1 \ldots N$$
$$\frac{dp_i}{d\theta} = \frac{B}{V \cdot \omega} \cdot \left(\sum_k Q_k - \omega \cdot \frac{dV_i}{d\theta} \right) \qquad \omega = \frac{d\theta}{dt} \tag{1}$$

In equation (1), B is the Bulk Modulus of the fluid, Q_k represents the generic flow coming in or out the piston chamber.The piston chamber volume is defined as a function of the angular position of the machine shaft θ and of the swashplate β, as in equation (2) where V_0 is the dead volume of the chamber, R_{cp} the cylinder block radius, β the swashplate angle, D_p piston diameter, θ_i the angular position of the piston and ω the angular speed.

$$V = V_0 + R_{cp} \cdot (1 - \cos\theta_i) \cdot \tan\beta \cdot \frac{\pi}{4} D_p^2$$
$$\frac{dV}{d\theta} = \frac{R_{cp}}{\omega} \cdot \frac{\pi}{4} \cdot D_p^2 \left(\sin\theta_i \cdot \tan\beta \cdot \omega + \frac{1 - \cos\theta_i}{\cos^2(\beta)} \cdot \dot{\beta} \right) \tag{2}$$

The flow through the valve plate ports is determined using the equation of flow through an orifice having geometrical passage equal to A:

$$Q = C_d \cdot A \cdot \sqrt{\frac{2 \cdot |\Delta p|}{\rho}} \cdot sign(\Delta p) \tag{3}$$

In equation (3) C_d represents the discharge coefficient, variable as a function of the Reynolds number as described in equation (4), where D_h hydraulic diameter of the orifice, υ kinematic viscosity e ρ the fluid density; R_{ec} the value assumed by the Reynolds number where the transition between laminar and turbulent flow happens; $C\infty$ is the value of the discharge coefficient in fully turbulent flow condition across the orifice.

$$C_d = C_\infty \cdot \tanh(2 \cdot \mathrm{Re}/\mathrm{Re}_c), \qquad \mathrm{Re} = \frac{D_h}{\nu} \cdot \sqrt{\frac{2 \cdot |\Delta p|}{\rho}} \tag{4}$$

Particular care has also been devoted to the description of the leakages through the lubricating gaps whose geometry (height) is constant and has been defined on the basis of experimental and numerical comparison of data regarding the average flow, swashplate torque and shaft torque for different operating conditions. Equation (5) is used to calculate the leakages across the slipper bearings-swashplate gap; R_{ext} and R_{int} represents the external and internal slipper radii, μ the fluid absolute viscosity, p_{int} and p_{ext} the pressure values at the external and internal radius of the slipper.

$$q = \frac{\pi \cdot h^3}{6 \cdot \mu} \cdot \frac{(p_{\mathrm{int}} - p_{ext})}{\ln\left(\frac{R_{ext}}{R_{\mathrm{int}}}\right)} \tag{5}$$

A similar expression is used for the leakage across the piston ball joint-slipper gap, considering axially-simmetric geometry and again constant gap height.

Equation (6) describe the leakage across the cylinder block-valve plate gap, where leakage may flow both to drain and to the adjacent pistons chambers.

Equivalent geometric dimensions have been defined to describe the gap, the length l and the width b_{cy}, and again a constant heigth h_{cy} (distance between the cylinder block and the valve plate) has been considered.

$$q = \frac{b_{cy} \cdot h_{cy}^{\ 3}}{12 \cdot \mu \cdot l} \cdot (p - p_D) \tag{6}$$

Finally, equation (7) expresses the flow across the gap between the pistons and the cylinder block, considering a zero eccentricity piston position. In equation (7) h_{pi} is as usual the gap heigth, l_e the length of the gap and $\dot{s}_P$ the piston speed:

$$q = \frac{\pi \cdot D_p \cdot h_{pi}^{\ 3}}{12 \cdot \mu \cdot le} \cdot (p - p_D) \pm \frac{\pi \cdot D_p}{2} \cdot h_{pi} \cdot \dot{s}_P \tag{7}$$

The piston-slipper assembly force, due to the chamber pressure, inertia, viscous friction, determines one of the contribution of the torques acting on the swashplate; the swashplate, modelled as a rotary inertia, receives also the contribution of the spring and of the control actuator. The variation of the swashplate position is instantaneously conveyed to the piston elements and all the cinematic variables are updated as a consequence.

The main geometrical parameters describing the valve plate, the cylinder block, the pistons are defined through a pre-processor module.

Equations (8) shows the displacement s_p, speed and acceleration of the piston: they are function of the shaft angular position ϑ_i and of the swashplate angular position β, calculated for a certain constant angular speed ω.

$$\begin{aligned}
s_{p,i} &= R_{cp} \cdot (1 - \cos\theta_i) \cdot \tan\beta \\
\dot{s}_{p,i} &= R_{cp} \cdot (\sin\theta_i) \cdot \tan\beta \cdot \omega + \frac{R_{cp} \cdot (1 - \cos\theta_i) \cdot \dot{\beta}}{\cos^2(\beta)} \\
\ddot{s}_{p,i} &= R_{cp} \cdot \omega^2 \cdot \cos\theta_i \cdot \tan\beta + \\
&+ \frac{2R_{cp} \cdot \sin\theta_i \cdot \dot{\beta} \cdot \omega + R_{cp} \cdot (1 - \cos\theta_i) \cdot \ddot{\beta} + 2R_{cp} \cdot (1 - \cos\theta_i) \cdot \dot{\beta}^2 \cdot \tan\beta}{\cos^2(\beta)}
\end{aligned} \tag{8}$$

Piston-slipper assembly undergoes the action exerted from the pressure within the chamber, $F_{p,i}$ from the inertia contribution $F_{m,i}$, from the viscous friction that arises in the piston-cylinder block gap $F_{v,I}$, from the swashplate reaction $R_{z,i}$.

$$\begin{aligned}
F_{p,i} &= p \cdot \frac{\pi}{4} D_p^{\ 2} \\
F_{m,i} &= -m \cdot \ddot{s}_{p,i}, \\
F_{v,i} &= -\frac{\pi \mu D_p le}{h_{pi}} \dot{s}_{p,i}
\end{aligned}$$

$$\Rightarrow R_{z,i} + F_{p,i} + F_{m,i} + F_{v,i} = 0 \tag{9}$$

Equation (10) describes the swashplate dynamic motion: I the moment of inertia, T_c is the torque due to the actuator action and spring, T_{sw} is the torque of the piston slipper assemblies on the swashplate, as shown in equation (11), T_v the viscous friction torque, as expressed in equation (12) where c is the viscous friction coefficient.

$$I \cdot \ddot{\beta} = T_c + T_{sw} + T_v \tag{10}$$

$$\begin{aligned}
T_{sw} &= \sum_{i=1}^{N} T_{sw,i} = \sum_{i=1}^{N} F_{sl-sw,i} \cdot l_{swi} \\
l_{sw,i} &= \frac{R_{cp} \cdot \cos\theta_i}{\cos\beta} \\
F_{sl-sw,i} &= -R_{z,i}/\cos\beta
\end{aligned} \tag{11}$$

$$T_v = -c \cdot \dot{\beta} \tag{12}$$

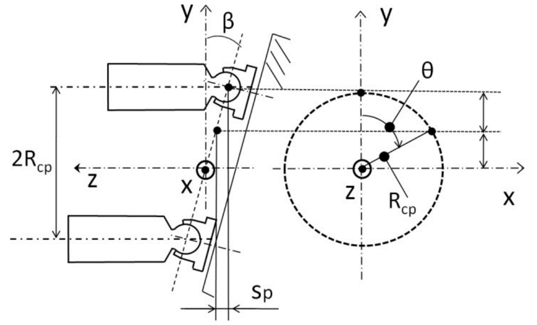

Figure 2 *Frame of reference and schematic view of an axial piston swashplate pump.*

Finally, detailed modelling of the flow and pressure compensators and of the control and bias actuators are added to the model of pump in order to be able to describe the dynamic variation of the machine displacement during pump operation.

3 *Alternative Architectures*

Variable Pump Margin (VPM)

The load-sensing variable displacement pump is a very known and studied component, but in recent years it is undergoing substantial changes for the introduction of advanced electronic for the control of the hydraulic power generation.

The electronic control allows an infinite range of adjustments and controls, but a good integration with fluid power components has to be studied and developed in order to guarantee the performances and robustness of the system.

In a typical LS power supply unit, the pump operates at a fixed pressure level higher than the pressure actually required by the user.

This additional pressure, also called Pump Margin, is always fixed by the spring preload of the flow compensator, giving rise to energy losses, depending on the duty cycle or the operation carried out.

Here, the nominal value of 2.5 MPa assigned to Pump Margin is the minimum possible value of system pressure in the standard configuration in order to overcome the losses caused by valves and the actuators connected to it when the flow rate is at its maximum value.

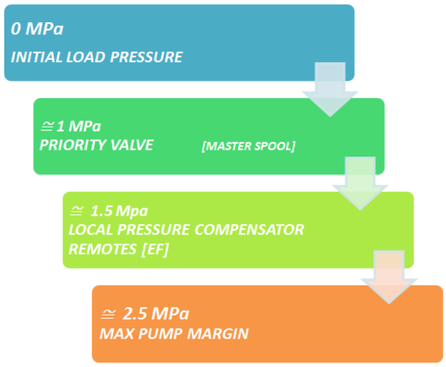

Figure 3: *Contribution to build up the pump margin*

The estimation of the Pump Margin, as shown in the Figure 3, is due to two fundamental components of the hydraulic circuit: the priority valve and the local pressure compensator of the auxiliary distributors.

The resulting energy losses are due to an high pressure drop across the two valves priority & auxiliary distributors.

The VPM strategy is based on the reduction of the pressure difference between the delivery pressure of the pump and the actual pressure of the generic actuator, by adjusting the spring preload of the flow compensator that equips the axial piston pump, using a solenoid valve and an opportune strategy on the ECU (Electronic Control Unit) to control it.

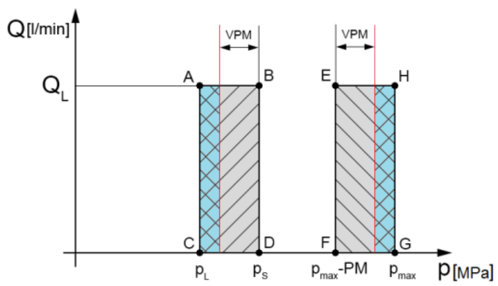

Figure 4 *Variable Pump Margin Concept*

Let's consider a request of flow rate Q_L corresponding to a pressure p_L, if the regulation of the spring preload of the pump flow compensator is equal to p_M pressure, the main pump is working at pressure p_S given by the sum of p_L and p_M.

The power dissipated in standard working conditions, is represented by the area ABCD between the levels of pressure p_S and p_L. Under pressure saturation condition and considering the maximum pressure p_{max} allowed in the system, the power loss is represented by the area EFGH, between pressure levels p_{max} and p_{max}-p_M.

For the implementation of the VPM control strategy, it was decided to use a commercial valve, called EFx (EFx is a commercial valve that replace the flow compensator for the axial piston pump. For related and detailed note refer to http://www.boschrexroth.com), which replaces the hydraulic compensator with an electro-hydraulically flow compensator controlled by a specific control unit.

The pressure margin can vary between 2.5 MPa to 1.5 MPa due to the action of this valve.

The VPM control strategy provides two types of control that can be implemented on the ECU: fixed point strategy and dynamic variation strategy.

The fixed point strategy is based on the setting of a single value of reduced pump margin for the pump to be regulated when possible and without any possibility of dynamic adjustment.

This strategy is possible only in the case of standby condition of the machine: in fact, in a generic operating condition the lowering of the pressure imposed by the VPM fixed point strategy to the minimum level, will not be able to meet the flow requests coming from the auxiliary and other users connected.

A second control strategy for the VPM is introduced: through the use of pressure sensors (dislocated on the discharge line of the pump and on the main working lines)and reading via CAN net the control signals at the auxiliary directional valves , it is possible to create an operating map for the control unit (ECU).

For a first analysis, a linear dependence has been used between the total required capacity (easily obtainable through the Electro Hydraulic signals of the auxiliary distributors available via CAN network) and the current that has to be provided to the solenoid for the dynamic variation of the Pump Margin.

The VPM dynamic strategy, allows to obtain a continuous variation of the Pump Margin as a function of the required flow and pressure from by the users.

The dynamic variation of the pressure, allows to increase the energy saving of the system in all the duty cycle stages, immediately adapting the Pump Margin to the new conditions of load.

Despite this solution seems quite promising, the minimum level of Pump Margin assignable in dynamic conditions, is limited by the presence of the priority valve. Hence a combined optimization process will have to be developed to obtain the maximum energy saving contribution from this solution.

Electronic Displacement Control - EDC

A further alternative solution related to fluid power supply group, is based on the use of bigger pumps coupled with the internal combustion engine, in order to being able to deliver the same flow at lower speed, thus optimizing the pump-engine coupling and having the chance of a more flexible management of the power and torque characteristics of the engine.

To do this, an electro-hydraulic valve is introduced, able to set a limit of the maximum swash plate angle of the axial piston pump. The valve is screwed directly into the seat of the actuator control of the swashplate and through the regulation current allows to vary the maximum value of the stroke of the swash plate, thereby adjusting the maximum flow supplied by the main pump.

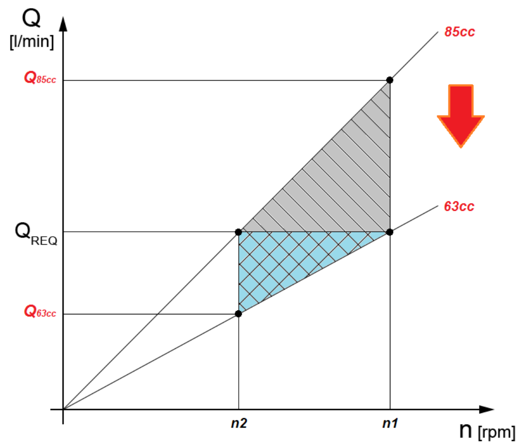

Figure 5 *Electronic Displacement Control Concept*

Using this concept, the EDC solution allows to obtain the same flow, delivered by the power supply unit, at different speeds of rotation of the combustion engine connected to the power supply.

Setting the level of flow in the standard conditions, a rotational speed n_1 of the pump is obtained. Following the linear characteristics of the machines, the required flow Q_{REQ} can be achieved at a lower engine speed, equal to n_2, using the machine with higher displacement and the EDC valve.

This may for example allow to switch from the maximum power to the maximum torque operating condition.

4 *Duty*

Thanks to the collaboration between the University of Modena and Reggio Emilia and the manufacturer CNH Industrial S.p.a, collection of data from the field has led to the definition of a duty cycle that describes the usage of the tractor during its average life, identifying four main phases of work: transport, standby, work in low speed & work at high speed. With reference to the tractor life, the duty cycle defines four main phases of work that occupy the whole lifetime of the tractor as described in the following: 25% transport, 20% standby, 25% work with low engine speed and 30% work with high engine speed. The following subsections will describe the details of how each work phase has been defined and then simulated with the model.

Transport

In this phase the portions of the hydraulic circuit that are working, beside the fluid power group, are the steering system and eventually the trailer system; the value of pressure and flow rate taken as reference in each working phase are non-dimensioned with respect to the maximum pressure and flow of the hydraulic system:

- steering phase: the driver acts continuously on the steering to make corrections of the vehicle trajectory while driving. For this reason, the steering is quite stressed and it is considered that its usage covers the whole transport phase (100%). The operating conditions in terms of flow rate and pressure level at the steering are: flow rate 14 $\%Q_{max}$; pressure 24 $\%p_{max}$.

- trailer system: during transport, a trailer may be attached to the agricultural tractor. From experimental data however the usage of the trailer is quite limited and covers only 1.25% of the transport phase under operating conditions equal to: flow rate 21 $\%Q_{max}$; pressure level 87 % p_{max}.

The tractor speed during transport is considered varying between 12 and 40 km/h.

Standby

In this work phase actually the system is not working with the hydraulic actuators, the axial piston pump is at its minimum displacement, delivering only the flow needed to compensate leakages and to allow regulation of the flow and pressure compensators. The operating conditions in terms of flow rate and pressure level are: flow rate $4\%Q_{max}$, Pressure 12% p_{max}. The engine speed is at its minimum value.

Work at high engine speed

The work phase at high engine speed involves the simultaneous use of the hydraulic actuators controlled by the auxiliary distributors and the steering system:

-steering: in this case the tractor is working on field, hence the contact between the tires and the ground is significantly different from the transport phase. As a consequence, the duty cycle involves higher pressure level and flow rate as operating conditions: flow rate $14\%Q_{max}$; pressure 38% p_{max}. The usage of the steering system occupies the 50% of the phase of work at high engine speed.

- auxiliary users: the auxiliary distributors are ones of the most dissipative components of the system and are used for approximately the 7.5% of the phase of work at high engine speed, also considering the simultaneous working of more than one actuator. The operating conditions are: flow rate $100\%Q_{max}$; pressure 48% p_{max}.

The tractor speed during this phase is varying between 5 and 14 km/h.

Work at low engine speed

Also in this phase, a simultaneous usage of auxiliary distributors and steering is considered but the operating conditions are changed.

-steering: the use of the steering system covers the 20% of the phase under the following operating conditions: flow rate $14\%Q_{max}$; pressure 48% p_{max}.

-auxiliary users: the auxiliary distributors work occupies the 7.5% of this phase, the operating conditions are equal to the previous ones: flow rate $14\%Q_{max}$; pressure 48% p_{max}.

The tractor speed during this phase is varying between 0.05 and 5 km/h.

The Figure 6 shows the duty cycle subdivision just described.

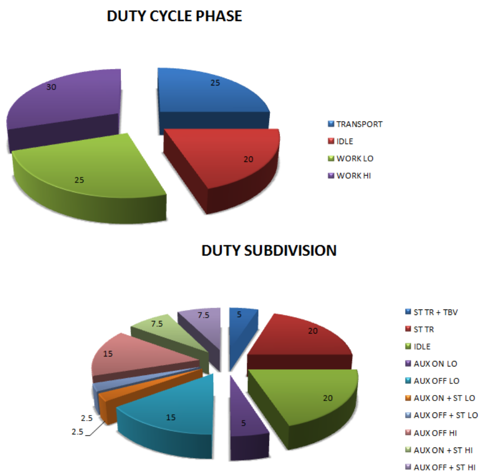

Figure 6 *Duty Cycle specification*

5 *Results*

The alternative configurations introduced in the previous sections have been modelled in LMS Imagine.Lab AMESim environment and compared with the standard configuration. The comparison has been done using the duty cycle previously described, chosen accurately and representative of the average life cycle of an agricultural tractor.

Results are presented making reference to the power consumption at the fluid power generator group shaft in percentage, viewed as the power consumption of the alternative architecture with respect to the power consumption of the standard architecture (100%); the power saving is the difference between the percentage power consumption of the standard version and the percentage power consumption of the alternative architectures.

Variable Pump Margin - VPM

The power saving achieved by the VPM fixed point strategy in the standby phase is very low and does not justify the installation on the vehicle.

As can be seen from the Figure 7 the power saving achievable by this configuration is about 0.13%, which is irrelevant for the chosen duty cycle.

This is due to the fact that in the standby phase the flow discharged by the pump is low and the incidence of lowering the pump margin is very little.

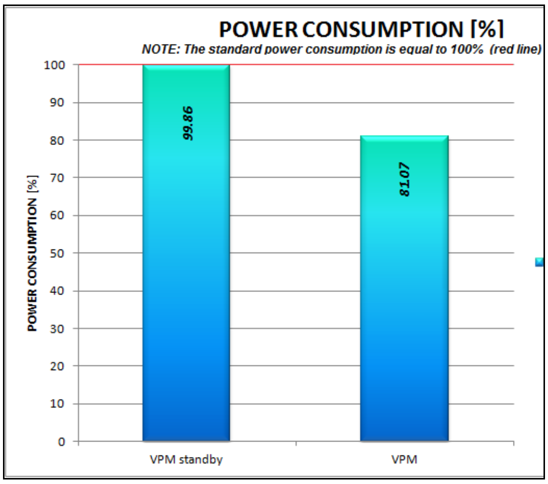

Figure 7 *Power consumption: VPM fixed point strategy*

In order to increase the amount of power saved, the best solution relies on using the control strategy with dynamic variable pump margin. To show this, a duty cycle where only the auxiliary distributors are working has been supposed. The auxiliary distributors have been proportionally opened in order to gradually rise the flow request till the maximum value. the Figure 8 shows the correspondent percentage of power saving as function of the flow rate.

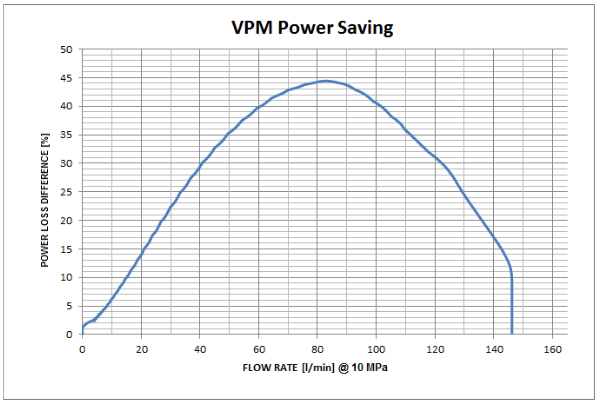

Figure 8 *VPM Dynamic Solution*

Using the linear mapping performed in the control unit (ECU), the maximum available power saving is achieved at intermediate flow rate request.

For high flow rates all the available pump margin pressure is needed and therefore there is no possibility of obtaining a benefit with respect to the standard configuration.

With reference to the duty cycle defined previously, Figure 9 shows there is no appreciable difference between the control strategy with VPM fixed point (VPM standby) and the standard version, while obtaining a power saving of 19% in the case of using a dynamic VPM adjustment.

During the transport phase, the pressure and flow rate requests are intermediate, so it is possible to achieve the maximum power saving, as shown in Figure 9.

In all other work phases, pressure and flow rate levels are such as to minimize the power saved adopting of the VPM strategy.

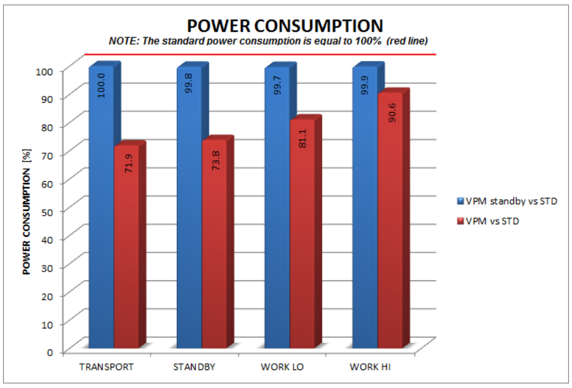

Figure 9 *Power consumption during tractor lifetime: VPM Solution*

The EDC strategy allows to introduce in the system an additional degree of freedom in order to provide the efficiency of the internal combustion engine-pump coupling. The user may decide to limit the displacement of the main pump in order to get the same flow rate of working fluid at different engine speeds. Generally, the introduction of a larger pump which rotates at a lower engine speed, to keep fixed the flow, involves a reduction in the efficiency of the same pump. The lowering of efficiency of the main pump can be considered relatively low compared to the possible benefits coming from an optimized management of the engine speed. However, with this particular duty cycle the power saving was about 1-2%.

6 *Conclusions And Future Work*

The motivation of this work derives from the need to reduce energy consumption in hydraulic circuit of agricultural tractors. To do this an intense research work to evaluate the actual situation has been carried out and from the results obtained suggestions for the improvement of the hydraulic energy production and usage in the vehicle are expected. In the target vehicle analyzed the hydraulic circuit has a typical hydraulic load sensing multi-users architecture where hydraulic power is provided by a group of fixed and variable displacement pumps and in this paper the attention has been focused on the variable displacement pump. A detailed lumped parameter model has been developed for the whole hydraulic circuit of the tractor, focusing in this paper in particular on the analysis of the pumps together with the flow compensator. The study of the standard hydraulic layout highlighted that many critical issues are represented by the excessive pressure drop across the valves and an improper management of the pressure levels between the main pump and the principal loads. Some alternative solutions are hence suggested. The standard architecture of the fluid power supply group in the tractor has been compared in terms of power consumption with some simple alternatives which involve the implementation of an electronic control of the variable displacement pump. In relation to the fluid power supply unit, the alternative solution proposed is based on the analysis of the pump margin pressure of the axial piston pump. The "intelligent management" of this additional pressure between pump and load, leads to an energetic improvement of the entire system: adopting a dynamic variation of the pump margin as function of the requested flow rate by means of an electronic flow compensator, it is possible to achieve the maximum energy saving.

With the aim of defining a term of comparison of the standard and alternative architectures, an actual duty cycle has been defined considering the entire life of the vehicle.

From the comparison, depending on the type of the flow rate request and following a linear characteristic within the ECU, it is possible to reach the maximum energy saving for the intermediate pressure and flow rate levels by using the Dynamic VPM strategy (19%).

In case of use of the Dynamic VPM strategy a critical aspect is the difficulty in some operating conditions to ensure the correct flow rate levels: to solve this problem a new architecture, involving a flexible priority valve, able to maintain the correct pressure levels and flow rate levels request by the users, is under investigation.

These alternative solution of the fluid power supply unit will be experimentally tested on a prototype vehicle thanks to the manufacturer CNH Industrial SpA.

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