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Development and application of co-simulation and "control-oriented" modeling in the improvement of performance and energy saving of mobile machinery

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Abstract

Due to rising energy costs and tighter emissions restrictions from law regulations, mobile machinery and off-road vehicles manufacturers are forced to develop and exploit new techniques for the reduction of fuel consumption and pollutant emission. The main focus in this direction is the optimization of the matching between the fluid power circuit and the thermal engine to improve the efficiency of the hydraulic system and reducing the fuel consumption. A specific research activity has been started in this field by the authors to define methods and techniques for the mathematical simulation of off-road vehicles, where usually hydraulic systems are powered by internal combustion engines. The models proposed in the paper and the related results clearly show how these simulation tools can be used to improve the energy efficiency of the overall system, leading to an interesting reduction in fuel consumption by merely changing the engine rotational speed instead of adopting a constant-speed strategy.

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Keywords: Hydraulic system; Diesel engine; co-simulation; control-oriented.

1. Introduction

In the field of mobile machinery new techniques to reduce fuel consumption and pollutant emission are increasingly required, and many research centers are focusing their activities on this topic [1-4]. One of the main

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target in this direction is the optimization of the matching between the fluid power circuit and the thermal engine in order to improve system overall efficiency and to reduce fuel consumption. Up to now, few proposals reported in the scientific literature [5,6] are based on a proper integration of the behavior of the hydraulic system and of the internal combustion engine in order to make the best use of their operating characteristics. The research activity presented in this paper is focused to define methods and techniques for the mathematical simulation of mobile machinery where hydraulic systems are powered by internal combustion engines (ICE). The know-how gained by the authors in the last years in the fields of mathematical modeling of both fluid power systems (with AMESim®, e.g., [7,8,9]) and ICE (with Simulink®, e.g., [10,11,12]) allowed to set up a mathematical tool for the simulation of the behavior of complex systems (also in transient conditions) with short calculation time. The final aim of the work is the development of "control oriented" models able to run real time simulations for designing and testing of control and diagnostic strategies (even through Model-, Software- and Hardware-in-the-Loop systems (MiL/SiL/HiL)). This approach will lead on one hand to significant savings in time and costs in the development and testing of the control system allowing to start trial run and debugging phases of the control strategies and ECU prototype simultaneously with the design and development of the controlled system. On the other hand, a reliable simulation tool of the whole system will represent an efficient way to define its optimized architecture given the ability to simulate the interaction of various components (hydraulics, engine, kinematics) in real operation conditions. The ultimate target of this research is to propose customized control strategies able to improve overall performance of the system (e.g., energy efficiency, fuel consumption, etc.) for specific applications.

The paper presents a first application based on the matching of a naturally aspirated Diesel engine with an axial piston pumps. By carefully handling causality, I/O parameters and co-simulation issues, a mathematical model of the system has been set up allowing the simulation of steady and transient behavior of the considered sub-system (i.e., engine and pump) faster than real time. To this extent, different integration time steps have been defined for the two sub-models taking account of their differing numerical "stiffness". In the paper results given by the simulation model of the engine-pump system in typical operating steady-state conditions are reported. In this first stage of the analysis, only steady state operating conditions are accounted for since a virtual driver, needed to follow a transient duty cycle, has not been yet developed up to now. Nevertheless, the results obtained by a steady state analysis could be employed as first benchmarks and optimal target for a further dynamic analysis. The firsts results reported in the paper show clearly to what extent the proposed approach could be beneficial, leading to the concurrent design of integrated control strategies compared to an independent optimization of engine and fluid power system strategies.

2. Mathematical models

2.1. Diesel Engine

The simulation library built up by the authors in the last decade has been conceived to create "real-time" models of automotive engines relying on Mean Value or Crank Angle approaches [13,14,15,16]. Based on Filling and Emptying (F&E) and quasi steady flow (QSF) methods, and developed within MATLAB®/Simulink® environment to improve portability and flexibility, the library has been organised in a hierarchical structure so that sub-model blocks can be found, picked up and assembled following the desired system layout. Dedicated procedures have been defined for the identification of each block [12]. Intake/exhaust components and cylinders are modelled either as volume components (i.e., capacitances) through an F&E approach based on mass and energy conservation differential equations (e.g., manifolds), or as non-volume components (i.e., resistances) with a QSF methodology (e.g., valves, compressors, turbines, etc.) employing algebraic equations. Processes which take place in the cylinder have been modelled through a Crank Angle method, following a single-zone approach applied to an open thermodynamic system in the well-known form of mass and energy conservation equations (F&E), where the working fluid is assumed to be a mixture of ideal gases. Temperature and mass within the cylinder can be evaluated as functions of time (or crank angle) by integration of the previously mentioned equations. Heat flux rate through walls is evaluated through the classical heat exchange equation, evaluating the heat exchange coefficient from Woschni's equation [17], while the heat released during the combustion is evaluated through Watson's formulation of the fuel burned fraction (*FB)* [18], with the ignition delay evaluated through Hardenberg and Hase correlation [17]. Changes in gas composition are evaluated assuming a perfect oxidation to CO_2 and H_2O. An original algorithm has been developed for the integration of mass and energy conservation equations in the cylinder with a suitable time step, while keeping a larger overall time step. A variable integration time step has been used for cycle simulation in order to keep an angular step of approx. 1deg CA, taking account of engine speed (assumed constant during each overall time step). The algorithm has been implemented in the in-cylinder block model to catch fast dynamics of related processes without compromising real-time capabilities of the comprehensive engine model. Valves on the other hand are considered as components in which no accumulation of energy or mass is possible [10,19] and are therefore modelled through QSF techniques.

2.1.1. Engine model calibration

In the present work, a four cylinder direct injection, naturally aspirated, midsize Diesel engine has been considered, whose major characteristic parameters are reported in Table 1. The engine is equipped with an electric governor which detects the speed of the rotating shaft and compares it with the desired reference value. The quantity of the injected fuel is thereby regulated by the controller in order to reduce the measured error, guaranteeing that, with a constant speed target, engine torque is equal to the requested load. In Figures 1, 2 engine characteristic in terms of output power, brake torque, and BSFC are shown.

Table 1. Engine characteristics parameters (Yanmar®)

Type	four-strokes, inline, water cooled, Diesel
No. of cylinders – Bore x Stroke [mm]	4 –98 x 110
Combustion system	direct injection
Compression ratio	18.5
Displacement [cm^3]	3319
Rated output [kW] @ 2200 r/min	46.3
Specific fuel consumption [g/kWh] @ 2200 r/min	240
Max Torque [Nm] @ 1400 r/min	228
Fuel injection timing [deg] @ 2200 r/min	13

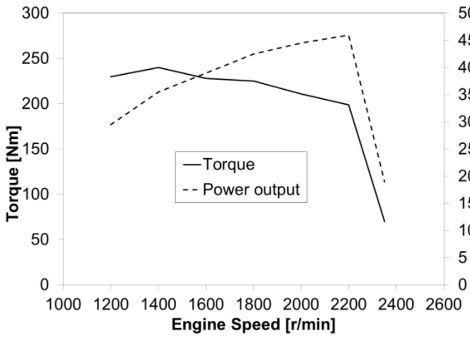

Fig. 1. Engine power and torque vs. engine speed.

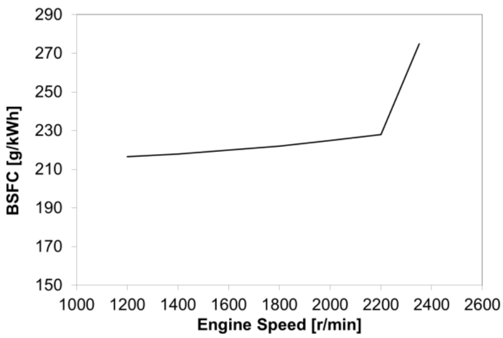

Fig. 2. Brake specific fuel consumption vs. engine speed.

The model of the engine has been developed assembling the components of the original library set up by the authors, following the causality scheme reported in Figure 3. In this first step of the analysis exhaust gas recirculation has been neglected in order to cope with a Single Input Single Output (SISO) model (Figure 4). Input parameters of the engine model are ambient temperature and pressure (assumed as constants), engine rotational speed and injected fuel mass. Even if the model allows the evaluation of every state parameter accounted (output power, exhaust temperature and composition, etc.), only shaft torque and rotational speed are requested to match the engine with the pump in the considered layout. As shown in Figure 4, the engine controller define the mass of injected fuel taking account of the torque requested by the load. As mentioned before, to reduce the complexity of the model in this first step of the study, the EGR system was not considered (i.e., EGR valve was closed).

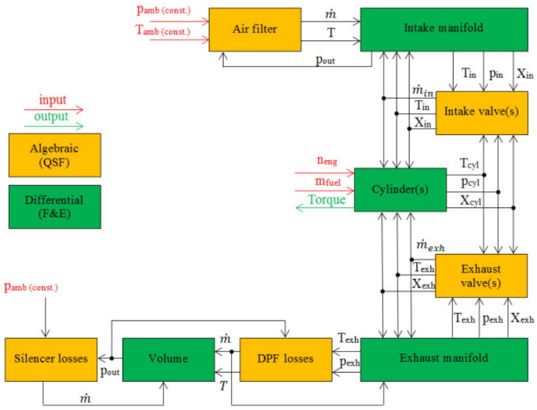

Fig. 3. Causality scheme of the four-cylinder Diesel engine.

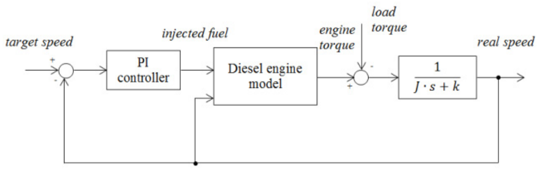

Fig. 4. Engine control schematic.

The model has been identified on the basis of steady-state experimental data from the OEM (Figures 1 and 2), at fixed values of rotational speed and fuel flow (extrapolated from reported BSFC and outlet power). Parameters from Watson formulation of Fuel Burn Rate have been tuned in order to obtain the peak of engine torque at 2200 r/min corresponding to a start of injection (SOI) of 13 deg before top dead centre (TDC), as reported on the engine data sheet. In other operating conditions, still keeping the obtained values of Watson's formula parameters, SOI has been adjusted to the minimum value corresponding to the maximum output torque, as shown in Figures 5, 6, according to standard procedures in engine tuning in order to obtain maximum efficiency [20]. A comparison between the results obtained from the engine model and the data given by the OEM and reported on the engine technical sheet is illustrated in Figure 7, showing in general a satisfactory agreement, with errors below 5% and trends adequately reproduced.

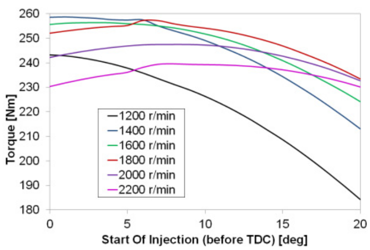

Fig. 5. Output torque versus SOI at different shaft speeds.

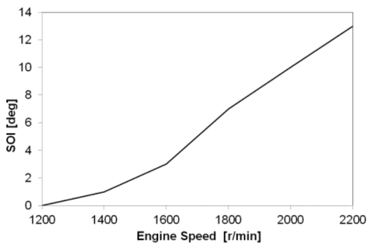

Fig. 6. Defined values of SOI vs. engine speed

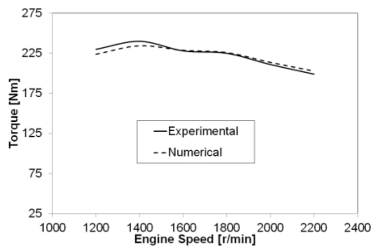

Fig. 7. Comparison of engine torque characteristic obtained from the model and from the technical data from the OEM.

The electric governor of the engine has been assumed as a simple PI controller which has been calibrated following Ziegler and Nichols method [21] and integrated with a tracking anti windup strategy [22] in order to properly handle saturation condition. The control is also bound to respect some stated specifics of the electric governor equipped, which in this case are: a maximum overshoot of 12% in engine speed over the rated value of 2200 r/min, and a steady state error lower than 6.8%, with a load variation from 100% to 0%; a maximum recovery time of 6 seconds is permitted (Yanmar®).

2.2. Hydraulic Pump

The pump model has been set up with reference to a variable displacement axial piston pump manufactured by Casappa Spa. An ISO schematic of the pump is reported in Figure 8.

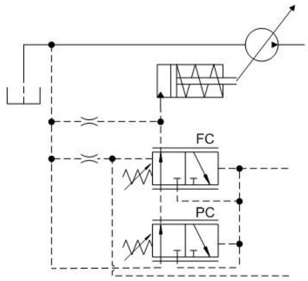

Fig. 8. ISO schematic of the hydraulic pump.

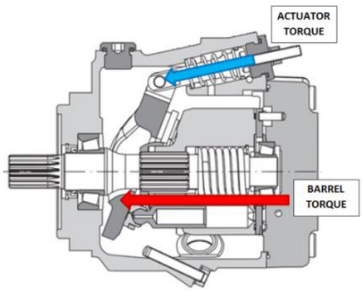

Fig. 9 Swash plate free body diagram.

The pump is controlled by two compensator: the pressure compensator (PC) which acts as a relief valve reducing the pump displacement and thus limiting the maximum pressure of the system, and the flow compensator (FC) which modulates the outlet flow in order to obtain a specific pressure difference between pump outlet and the pressure in the load sensing line. The functioning of these compensators are described in detail in [8].

The pump model presented in this paper is an improvement of a previously developed model [8, 10]. The grey box model includes a white box model of the flow and the pressure compensators and a black box model of the piston torque acting on the swash plate of the pump. The mathematical model of the pump has been developed using AMESim® simulation software. At this stage of development the mathematical model has been carefully validated at steady state conditions, while few dynamic tests have been carried out for a preliminary validation in transient condition.

2.2.2 Flow characteristics model

The basic idea of this model is to consider the pump as a simple flow generator which output is a function of both the given rotational speed (usually a constant value set in the range of 1200 ÷ 2300 r/min) and the instantaneous displacement, geometrically related to the swash plate angular position. Thus the volume flow rate is defined as:

$$\dot{V}_{th} = V_d(\alpha)\, n \tag{1}$$

where $\dot{V}_{th}$ is the theoretical instantaneous volume flow rate, α the swash plate angular position, V_d the instantaneous displacement, and n is the shaft rotational speed. The flow characteristic of the pump is thereby closely correlated to the equilibrium of the swash plate, which, referring to Figure 9, can be calculated as follows:

$$J_{EQ}\ddot{\alpha} + c\dot{\alpha} = \sum_{n=1}^{N} T_n + T_{ACT} \tag{2}$$

where J_{EQ} is the term that summarize the swash plate inertia, $\sum_{n=1}^{N} T_n$ is the barrel torque, resulting from the contribution of each n-th single piston force acting on the swash plate, T_{act} is the torque from the pump control actuator and c is a term summarizing the viscous friction contribution. In order to estimate the parameter c dynamic experimental activities have been performed. The swash plate position dynamic response, toward a step variation of the flow control orifice area on the pump outlet, allowed to evaluate the viscous friction parameter *c*. Further dynamic analysis will be developed in the next future. While the evaluation of the control actuator torque has been successfully achieved through the development of a white box model of the compensators, the determination of the barrel torque has proven to be quite complex, requiring the identification of a large number of parameters (e.g., pressure of pumping piston, damping effects, pistons masses, lengths and diameters, valve plate geometry, etc.) [23]. Therefore, a black box model has been preferred in the determination of the barrel torque, in order to provide a fast but still reliable model of the pump [8]. This has been achieved by measuring the pressure in the pump actuator chamber (i.e., the control actuator torque) at given values of pump outlet pressure, swash plate angular position and shaft rotational speed. At steady-state conditions since $\dot{\alpha} \cong 0$, Eq. (2) guarantees the equivalence $\sum_{n=1}^{N} T_n = T_{act}$. Thus, given the area of control actuator piston and the arm of the actuating force, a linear correlation subsists between pump actuator pressure and barrel torque under steady-state conditions (given by Eq. 3), where A_{ACT} is the pump actuator piston area and d_{ACT} is pump actuator arm:

$$\sum_{n=1}^{N} T_n = p_{ACT}(p_D, \alpha, n) \cdot A_{ACT} \cdot d_{ACT} \tag{3}$$

Therefore, a correlation has been defined through a surface fitting tool between the pressure in the pump actuator chamber and both swash plate angular position and pump outlet pressure for a variation of the pump shaft rotational speed between *850 r/min–2300 r/min* (Figure 10). Taking account of the influence of the shaft rotational speed is the main improvement with reference to the model presented in [13].

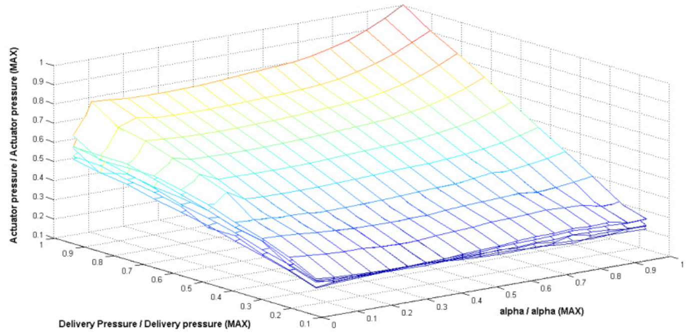

Fig. 10. Correlation giving the actuator pressure as $p_{ACT} = f(p_D, \alpha, n)$.

Experimental determination of hydraulic-mechanical and volumetric efficiencies has been also carried out according to the ISO4409-1986 standard: Figures 11, 12 represent the pump overall efficiency for different displacement.

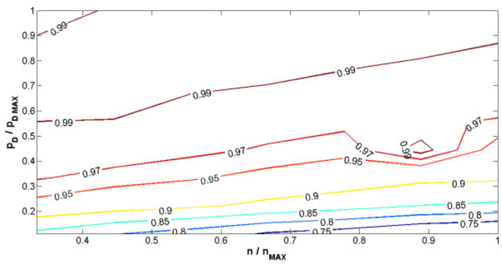

Fig. 11. Pump normalized overall efficiency η_g/η_{gMAX} at $\alpha/\alpha_{MAX} = 0.5$

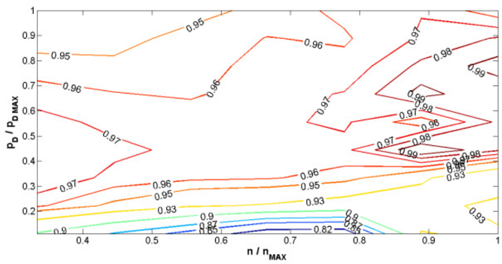

Fig. 12. Pump normalized overall efficiency η_g/η_{gMAX} at $\alpha/\alpha_{MAX} = 1$.

The overall efficiency η_g is given by the volumetric efficiency η_v times the hydraulic-mechanical efficiency η_{hm} (both not shown here for brevity) thus its trend is affected by both of them. Obtained results, after a wide experimental investigation, are typical for this type of pump [23]. A correlation between η_{hm}, η_v and rotational speed, delivery pressure and swash plate displacement has been implemented in the pump model. The real flow outlet is thereby calculated as:

$$\dot{V} = \eta_v(p_D, n, \alpha) \cdot \dot{V}_{th} \tag{4}$$

while the torque required at the shaft is:

$$T = \frac{\Delta p \cdot V_d(\alpha)}{2\pi} \cdot \frac{1}{\eta_{hm}(p_D, n, \alpha)} \tag{5}$$

where Δp is the difference between pump outlet pressure and pump inlet pressure.

3. System Model Co-Simulation Setup

The pump and engine models are connected through a shaft accounting for the overall inertia; a schematic of the resulting system, resembling the classical Load Sensing layout is reported in Figure 13, with a valve operating at a constant pressure difference (i.e. the difference between load and pump delivery pressure). The flow rate supplied is therefore univocally determined by the position of the valve spool, due to the regulation on pump delivery pressure actuated by pump flow compensator. This simple layout provides the basis of many different types of mobile machinery, and provides therefore a basis for preliminary efficiency improvements considerations. As the two models were originally developed in different simulation environments, the model of the hydraulic system has been reduced to a “S-Function” and imported into Simulink® environment. The equations describing the system are solved using Euler fixed step method with the adoption of two different sample time: the one competing to the engine model has been set equal to 2 *ms* (roughly the same sample time of an electronic control unit), while the one competing to the hydraulic circuit model has been reduced to a smaller size (about 80 *ns*) in order to avoid numerical instability. In fact, due to the high bulk modulus of the used fluid, stiff differential equations are associated with the hydraulic circuit simulation. In order to build a model suitable to run in real time and to MiL/SiL/HiL applications, the implementation of specific ODE solution methods for stiff equations is not possible, as they are based on a variable step approach, incompatible with the straightforward causality of a real time model. Lower time steps for the resolution of hydraulic circuit model equations are therefore needed. With these assumptions, the overall model runs on a 2.2 GHz Quad Core Pentium PC with 8 GB RAM with a satisfying ratio of computation time over physical time always lower than 0.9. The complete model has been at this point used to obtain an extended map of the overall fuel consumption of the system in different operating conditions. In this first stage, only steady state operating conditions are accounted as no virtual driver, needed to follow a transient duty cycle, has been yet implemented in the model.

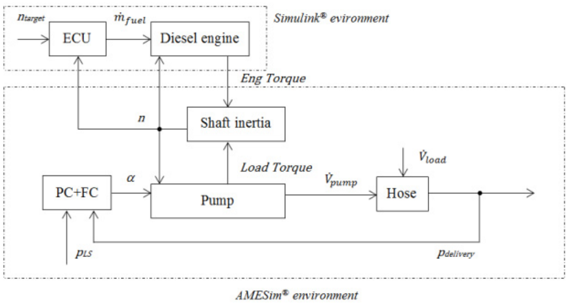

Fig. 13. Schematic of the complete system model.

Nevertheless, the results obtained by this steady state analysis will be employed as a benchmark for a further dynamic analysis. As shown in Figure 14, the overall system model has been run for different load pressures (p_{LS})

and outlet flow rates: load sensing pressure varies from a minimum of 5 *bar* to a maximum of 250 *bar*, while outlet flow rate varies from 15 *l/min* to 180 *l/min*, thus covering the whole operating range of the pump; the margin pressure has been set equal to 15 bar. The absence of working points at the top right hand side of the figure 14 is due to a limit on the maximum power, in order to meet the engine characteristics. Even if usually in these applications a very simple constant-speed strategy is used, every steady state point condition from Figure 14 has been simulated with different engine target speeds, in order to understand the impact of target speed on the overall fuel consumption. For this reason, for each steady state point condition target speed varies from a minimum to a maximum velocity: the maximum velocity has been fixed at *2200 r/min*, while minimum velocity, usually set at *1200 r/min*, is eventually increased in order to guarantee the required flow rate at the pump outlet (as, from Eq. 1, maximum flow at maximum displacement is proportional to pump speed) or to guarantee the required power output from the engine (Figure 1).

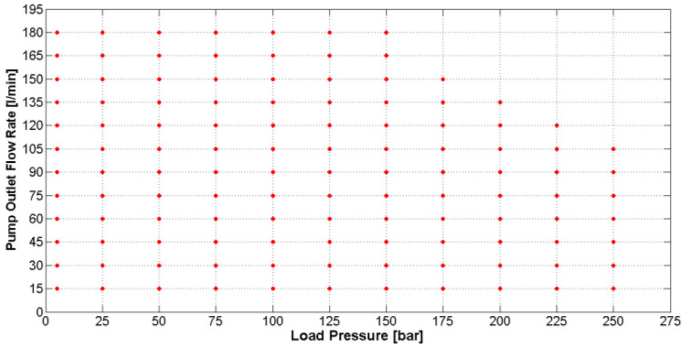

Fig. 14. Considered steady state operating conditions.

4. Results

In this section, the results obtained by the simulation of the engine-pump system operating at steady-state conditions are reported. For each working point, defined in Figure 14, the load sensing pressure and the pump outlet flow rate were fixed at constant value during the simulation. Therefore, the required power was constant too. After performing the simulation in each point with different shaft rotational speed, the minimum burned fuel mass was defined. As an example, in Figure 15 values of the burned fuel mass in a defined operating condition are reported, showing clearly a minimum.

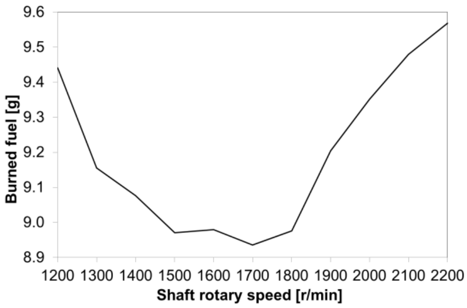

Fig. 15. Burned fuel mass for a generic working point (pump flow rate 90 l/min, load pressure 50 bar).

The simulation allowed to define the optimal shaft rotational speed minimizing the burned fuel mass for each working point. In Figure 16 the contour lines for the optimal speed in [r/min] with respect to the load pressure and the pump outlet flow rate are drawn. In the top right hand side of this plot no lines are reported because the power required by the user is higher than the maximum engine power.

Figure 17 shows the specific fuel consumption contour lines. In the left hand side of the figure the specific fuel consumption, required by the whole system, is higher than other parts of the diagram due to friction losses in the flow control valve, that are directly comparable with the hydraulic energy supplied to the fluid at the valve outlet.

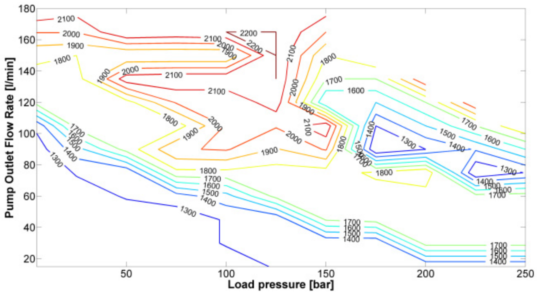

Fig. 16. Optimal engine speed contour lines [r/min].

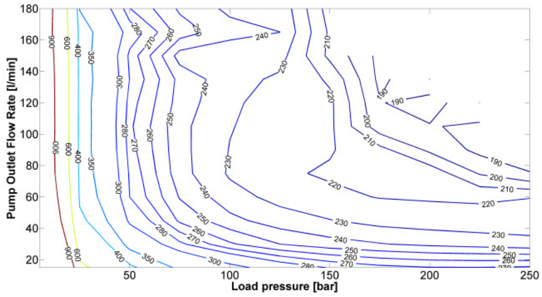

Fig. 17. Specific fuel consumption contour lines of the overall system [g/kWh].

In order to show the advantages of varying the engine shaft rotational speed instead of keeping it at a constant value in each working point defined in Figure 14, two different simulation at a constant speed of *1500r/min* and *2000r/min* have been performed. Therefore, the specific fuel consumption for this two simulation ($\dot{m}_{1500}$, $\dot{m}_{2000}$) have been compared with the specific fuel mass consumption obtained for the engine optimal speed in each working points ($\dot{m}_{nopt}$). Figures 19, 20 show the variation in specific fuel mass consumption, comparing fuel consumption at constant speeds of *1500r/min* and *2000r/min* with values reported in Figure 17.

It should be noted that in constant speed operation at the lower value (*1500r/min*), the specific fuel mass consumption variation is minimum (see Figure 18), but on the other hand the field covered by contour lines shows clearly that is not possible to guarantee the proper operation of the system in all working point, defined in Figure 14. On the contrary, with a constant speed of *2000r/min* (see Figure 19) the system can guarantee a proper operation in each point but with a higher specific fuel consumption. The final apparent results is that adopting a variable speed strategy can allow to reduce significantly the specific fuel mass consumption in all the considered working points, which are typical of real applications.

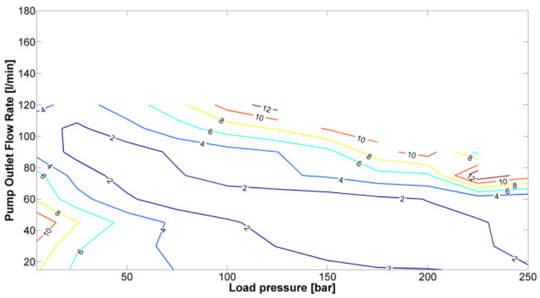

Fig. 18. Improvement in specific fuel consumption compared to constant speed (1500r/min), i.e., $\frac{\dot{m}_{1500}-\dot{m}_{n\,opt}}{\dot{m}_{1500}} \cdot 100$

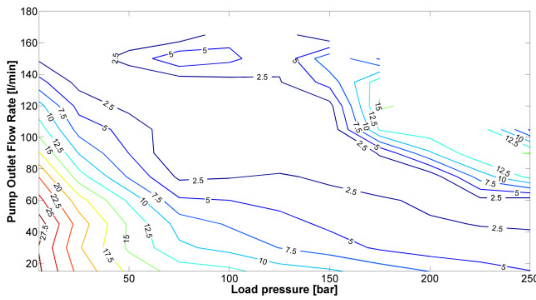

Fig. 19. Improvement in specific fuel consumption compared to constant speed (2000r/min), i.e., $\frac{\dot{m}_{2000}-\dot{m}_{n\,opt}}{\dot{m}_{2000}} \cdot 100$

5. Conclusion

The research presented in this paper is based on a proper integration of modeling techniques for the simulation of systems where hydraulic circuits and internal combustion engines are coupled in order to make the best use of their operating characteristics. The mathematical models proposed by the authors will be able to simulate in real time the behavior of complex systems also in transient conditions with short calculation time, with the aim of realizing “control oriented” models which can be useful both in the first stage of the system design (i.e., when system layout and architecture have to be defined) and in the design and testing of control systems even through the application of

MiL/SiL/HiL systems. The first application presented in the paper of the proposed methodology, with reference to the coupling between a naturally aspirated Diesel engine and an axial piston pumps, shows clearly to what extent this approach could be beneficial, highlighting how already a simple solution, based on the adoption of a variable engine speed, could be very useful to minimize the specific fuel consumption. Simulations carried out in the typical pump working range, show the possibility of a reduction of the overall system specific fuel consumption up to 10%.

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