

Modeling of an Excavator System - Semi Empirical Hydraulic Pump Model

2011-01-2278

Published
09/13/2011Paolo Casoli and Alvin Anthony
Universita di Parma

Manuel Rigosi

Copyright © 2011 SAE International

doi:[10.4271/2011-01-2278](https://doi.org/10.4271/2011-01-2278)

ABSTRACT

This paper describes the preliminary results of a study focused on the semi empirical modeling of an excavator's hydraulic pump. From the viewpoint of designing and tuning an efficient control system, the excavator is a very complex nonlinear plant. To design and tune such a complex control system an extremely good nonlinear model of the plant is necessary. The problem of modeling an excavator is considered in this paper; a nonlinear mathematical model of an excavator has been developed using the bond graph methodology realized in the AMESim® simulation software to replicate actual operating conditions. The excavator model is described by two models: a hydraulic model and a kinematic model. At this stage of research the hydraulic model deals solely with the model of the main hydraulic pump, which has been conceived as a semi empirical model. The pump model has been conceived as a grey box model; where the flow compensator and pressure compensator have been modeled as white box models while the actual flow characteristics of the pump as a black box model. The other excavator hydraulic system components comprising of valve block and actuators have been created using generic mathematical models. This approach has been followed to enable the study of the pump's dynamic behavior and interaction with a completely developed kinematic model of the excavator. Dynamic loading of the system has been realized through a 2D kinematic model of the excavator's body elements. The kinematic model comprises of the boom, arm and bucket whilst their respective motions have been defined for a cycle of operation. The dynamic parameters of each element are continuously calculated during an excavation cycle, thereby providing a platform to study the pump's behavior. At this phase of model development the

semi empirical model of the variable displacement pump has been validated on the basis of a set of experimental data collected on a test rig at particular experimental operating conditions. The final objective of this study would be to develop a complete mathematical model of the Excavator Hydraulic Control System and in turn facilitate the study of alternate control strategies towards energy efficient systems. This paper presents the results of this study.

KEY WORDS

Hydraulic Excavator, Semi Empirical Modeling, Variable Displacement Pump

INTRODUCTION

The development of control systems for complex mechanical systems such as large manipulators, mobile cranes and excavators requires the use of powerful software development tools. This is particularly true, where a short commissioning period is desired. Here, the complete control-system must be developed and tested by simulation methods. This requires a comprehensive software package for mechatronic systems, consisting of a modeling and simulation part and a control development part. The simulation of the kinematics and dynamics of multi-body systems is a topic of increasing importance in many industrial branches, due to its potential for reducing costs and that it provides insight into inherent effects governing the systems behavior. At a more detailed level, simulation offers enhanced product assessment, the potential of early stage conceptual testing, and a virtually unlimited spectrum of "what if" analysis [7]. Towards realizing this objective a model comprising of a hydraulic and kinematic systems have been developed. The need to model these systems is

attributed to the inherently nonlinear hydraulic drive, used to achieve precise motion and power control. This choice of drive is imputed to the superior power density of hydraulic systems in comparison to electrical or mechanical drives; regrettably the indigent energy efficiency of these systems is a major drawback. The need for energy efficient systems demands that this potent drive be self adjusting to meet actual load requirements [12]. A method of adjusting these systems to meet load requirements is by controlling the flow of a pump. In this context, variable- displacement axial piston pumps are often used, whereby the displacement of the pump can be varied by tilting a swash plate. This can be achieved fast enough to meet the dynamic demands affected by multiple loads. In this research, a nonlinear pump model has been developed, which takes into account the essential nonlinearities of the system and can be easily adjusted to pumps of different displacement sizes in the same model range. A practical pump model must enable one to examine the dominant characteristics influencing the behavior of the pump. In a load sensing pump this would include the response of the pressure and flow compensators in addition to the sensitivity of swash plate motion which defines the pump's displacement. This model can be conceived in a number of ways either as purely mathematical, empirical or a mix of the two as a semi empirical model. The approach of semi empirical modeling of the pump has been adopted in this paper, thereby providing a basis to rapidly examine different pump sizes to vary the flow gain of a complete excavator with the objective of achieving desired design characteristics. The pump displacement is controlled by the highest pressure feedback generated from the excavator's actuators, in an operating cycle. To subject the pump to these varying forces a detailed model of the kinematics has been realized. Due to the complexity of the whole system, dynamic forces are required for computing the load on the system which is achieved by the kinematic model of the excavator. Previously published research comparisons describe the differences between these static and dynamic forces [7]. The nonlinear effects occurring during the excavation cycle such as the bucket-soil interaction and the nonlinearities of the hydraulic system complicate the control of the pump. These factors have to be taken into account in hydraulic modeling and control. The above details provide a glimpse into the complexity of variable forces that a system would experience during an operating cycle for which the variable pump compensates its displacement. Thus far, having described the objective of this research, the following sections will detail the modeling of the pump, the excavator kinematics and validated results of the pump model.

PHYSICAL MODELING

PUMP MODEL

The pump model described in this paper is that of a load sensing variable displacement axial piston pump. This is a standard line production pump developed by Casappa SpA and belongs to the MVP series. The study of optimal displacement control of variable displacement pumps has been a topic of interest for fluid power researchers' world over and continues to provide scope for development with electro hydraulics and recent advances in robust control strategies. To set the pace of study, this paper describes the current stage of model development and will detail the actual pump being produced, i.e. with classical hydraulic feedback. As depicted in Fig.1, the pump model comprises of three sub-models a flow compensator, pressure compensator and the flow characteristics. The pump model has been conceived as a grey box model, i.e. a model based on both insight into the system and experimental data as can be seen in Fig.2. The flow and pressure compensators have been modeled as white box models and that of the flow characteristics as a black box model. The grey box model of the pump correlates the control piston pressure and the system pressure to provide the equilibrium of forces that defines the swash plate angle. This model approach has been adopted to provide the manufacturer with the flexibility of selecting pumps with discrete maximum displacements to vary the complete system's gain.

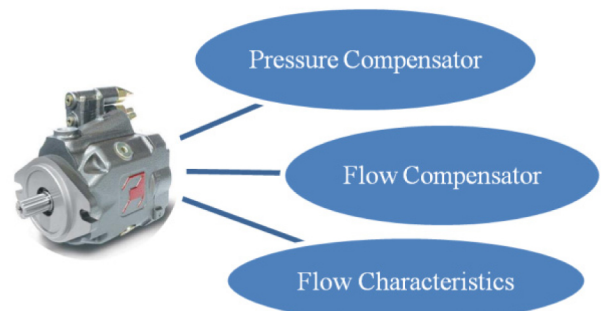


Fig. 1. Model breakup of the MVP series, Load Sensing Variable Displacement Pump

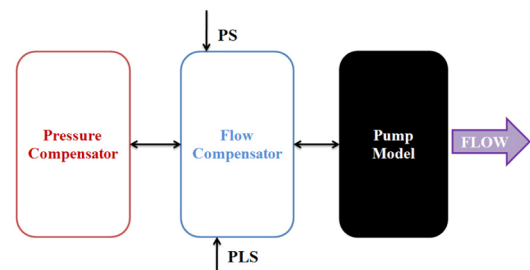


Fig. 2. Grey Box modeling methodology

The mathematical models of the flow compensator, pressure compensator and flow characteristics have been developed using the bond graph methodology realized through the AMESim® simulation software. This methodology uses the transfer of power between elements to describe the dynamics of the system. The basic idea being - the direction of power flow at any moment in a system is invariant. Power may be expressed as a multiplication of two factors - generalized effort and generalized flow. Bond graphs are far more powerful in modeling complex systems which involve the interaction of several energy domains [2].

Flow Compensator Model

The flow compensator (FC) has the most important function of offsetting the pump displacement for a set preload by regulating the swash plate angle. This component has been modeled and verified in great detail [10].

The logic integrated in the FC is to compare the dominant load pressure (PLS) with the pump's output pressure (PS) to modulate the flow through the FC, hence regulating the pump's displacement. The objective of the FC is to maintain a fixed differential pressure across the control orifice accomplished by modulating the pump's flow. The FC's spring comprises of two springs one being a snubber spring. This arrangement has been adopted to provide a snubbing function - thereby resiliently biasing the spool from the first operating position to the second operating position, when the fluid pressure in the LS chamber is increased from a first pressure level to a second pressure level. This feature has an attribute of controlling pressure spikes more carefully due to the precise metering characteristics of the valve, thus reducing the oscillations on the swash plate.

The functioning of the FC as depicted in Fig.3 and Fig.4 is such that the pump pressure enters the FC valve through the chamber A. The pressure entering this chamber acts on the area of spool 1 to create a force. This is realized in the model by means of a transformer element, in the form of spool 1, where the modulus of the transformer is the ratio between the spool - piston and rod area. The force created by the pump pressure on spool 1 is countered by the force created by the springs 5 and the LS pressure which is sensed in chamber D. The sum of these two forces i.e. spring 5 and the LS pressure acting on the area of piston 4 is the force value used to maintain the force balance of the spool. When the pump pressure is greater than the force on piston 4, the spool is displaced to the left creating a flow path between chamber A and chamber B, referred to as the intermediate chamber. The flow through this chamber has the function of varying the pump's swash plate angle, and in this way the pump displacement is regulated. Chamber C is connected to tank and its function is to drain the oil from the intermediate chamber. Spool 2, has been modeled as a transformer element with the modulus being the area ratio between the spool -

piston and rod. Leakage models have been included as depicted in 3 (Fig.4) to describe internal leakage between chambers and to increase the system damping.

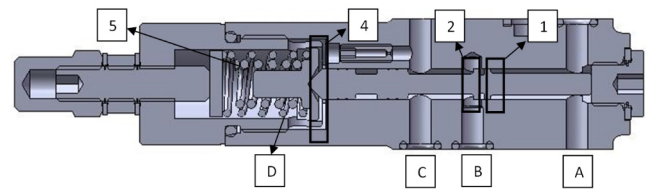


Fig. 3. CAD model of the Flow Compensator

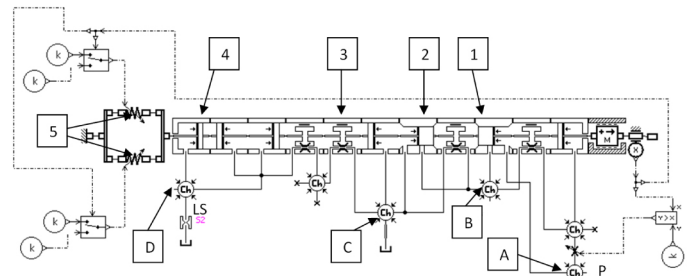


Fig. 4. AMESim® model of the Flow Compensator

The governing equations are described by the interaction between a fluid-dynamic model (FDM) and a mechanical-geometrical model (MGM). The FDM calculates the pressures inside the chambers and the flow rate between adjacent chambers, while the MGM calculates the forces acting on the spool and determines its dynamics and the flow areas.

The FDM is based on a lumped parameter framework. The pressure inside each control volume is assumed uniform and time dependent, and is determined by the pressure-rise rate equation:

$$\frac{dp_i}{dt} = \frac{\beta}{\rho_i} \frac{1}{V_i(x)} \left(\sum \dot{m} - \rho_i \frac{dV_i(x)}{dt} \right) \quad (1)$$

The model assumes a constant value of fluid temperature. The fluid density is evaluated as a function of pressure as described in [8]. The summation term represents the net mass flow rate entering or leaving the volume. This is obtained by considering the contribution of all orifices connected with the considered volume. The mass exchange occurring through the orifices is calculated using the generalized Bernoulli's equation under quasi-steady conditions, Eq. (2):

$$\dot{m} = C_d A(x) \sqrt{\frac{2 |\Delta p|}{\rho}} \quad (2)$$

The user sets an appropriate saturated value for the coefficient of discharge of each connection, on the basis of experimental data or using values reported in literature, such as [8, 9]; thereafter the instantaneous coefficient of discharge value is evaluated as a function of Reynolds number, to account for partially developed or fully turbulent conditions. Annular leakages past spool bodies have been evaluated using Eq. (3) as reported in [8, 9]:

$$\dot{m} = \rho \frac{\pi \cdot R \cdot h^3}{6 \cdot \mu} \cdot \frac{|\Delta p|}{L} \quad (3)$$

The mechanical model calculates the instantaneous position and velocity of the spool using Newton's second law:

$$\sum_i F_i = ma \quad (4)$$

The forces acting on the spool, Fig.4, are: hydrostatic forces; spring force; friction forces; hydrodynamic forces. Static and dynamic friction forces are evaluated by use of the Karnopp friction model and considering the Stribeck effect; static and dynamic friction coefficients are assumed constant; the hydrodynamic forces are proportional to the orifice flow sectional area and pressure drop across the orifice, the model implements the equation:

$$F = 2 \cdot C_d \cdot A \cdot \Delta p \cdot \cos \theta \quad (5)$$

where the jet or flow angle θ is affected by chamber geometry, orifice clearance and sharpness; for spool valves with a sharp edged orifice and no clearance between spool and sleeve, θ can be assumed equal to 69° [3]. The other assumptions are that fluid inertia is neglected; springs are assumed linear.

Pressure Compensator Model

The function of the pressure compensator (PC) is that of a relief valve and its function is to limit the maximum pressure of the system, Fig.5. The relief valve provides an alternate flow path to tank while keeping the system pressure at the relief valve setting. The relief valve realized in the PC is that of the direct operated type, it operates with a spring to preload the valve spool. Since a small flow rate is passed through the spool, the set pressure can be maintained with nearly no effect on the pressure flow characteristics of the

valve. The functioning of the PC is such that the system pressure enters the PC through chamber E (Fig.6) and the pressure acts on the area of spool 6 to create a force, which resists the force created by the spring 8 and spool 9, creating a relief setting pressure. When the system pressure is greater than the relief setting pressure, the spool 6 is displaced and creates a flow path to the swash plate actuating piston and to tank through an orifice S1 (Fig.6), which is housed in the PC valve. The governing equations describing the physical behavior of the PC are the same as those described in the section on the modeling of the flow compensator.

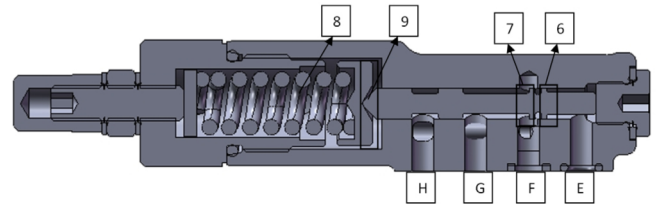


Fig. 5. CAD model of the Pressure Compensator

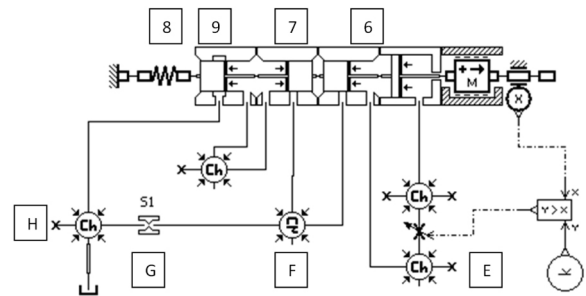


Fig. 6. AMESim® model of the Pressure Compensator

Flow Characteristics Model

A model of the pump's flow characteristics must permit the examination of dominant characteristics influencing the pump's behavior. In a load sensing pump this would include the response of the pressure and flow compensators in addition to the sensitivity of swash plate motion which defines the pump's displacement. Due to intricacies encountered in the control, design and implementation of this type of pump, it is advantageous to have a comprehensive model of the pump. Such a model would include determining the motion of the pump's swash plate based on the instantaneous operating conditions. To achieve this, the model must include the effects of friction acting on internal components, accurate determination of pressure in the pumping pistons and the effects of the swash plate motion on the control actuator. The model must reflect both the supply flow characteristics of the pump as well as the dynamic behavior associated with the internal components in the pump itself.

In practice the accurate prediction of swash plate motion is the exclusive parameter required to represent an axial piston pump, as all pump components interact with the swash plate to determine its motion. The angle of the swash plate determines the stroke of the pumping pistons, the length of which dictates the flow characteristics. The prediction of the swash plate motion is made difficult due to the exciting forces imparted on the swash plate by the pumping pistons as well as the compressibility of the fluid in the control piston. The bond graph model of the pump Fig.7, describes that the net torque acting on the swash plate comprises of the torque contributed by the swash plate inertia, pumping pistons torque, return spring torque, control piston torque and damping effects. These torque values define the swash plate angle to the transformer that modulates the pump's displacement. Regrettably the white box model approach is quite elaborate and does not afford the flexibility required to examine pumps of different sizes as all the constants (piston mass, lengths and diameters, swash plate mass, damping, valve plate geometry etc.) would have to be modified to adopt a different pump.

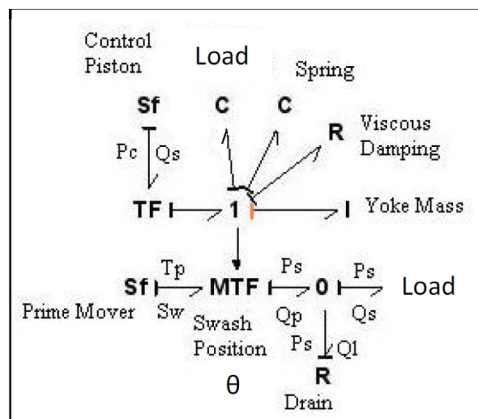


Fig. 7. Bond graph representation of the Pump

The objective of this research has been to adopt a technique for identifying the dominant characteristics of a pump and to facilitate the study of different pump sizes in a feasible manner. This flexibility has been integrated into the model to study the flow gain offered by different pumps to the system. The black box system identification was identified as the best model methodology to integrate this flexibility. This choice of modeling methodology was also attributed to the availability of an instrumented test facility and a large sample of experimental data. The black box model was realized by taking into account that the position of the swash plate could be determined by solely balancing the parameters of the control piston pressure and the system pressure. The control piston pressure is the pressure in the actuator that is used to displace the swash plate. The control piston receives its input from the flow/pressure compensator, depending on the state of the system. The system pressure is the pressure that acts on

the swash plate to determine the forces on it. The forces acting on the swash plate affected by the system pressure comprises of the forces exerted by the pumping pistons exposed to the load and the internal friction forces.

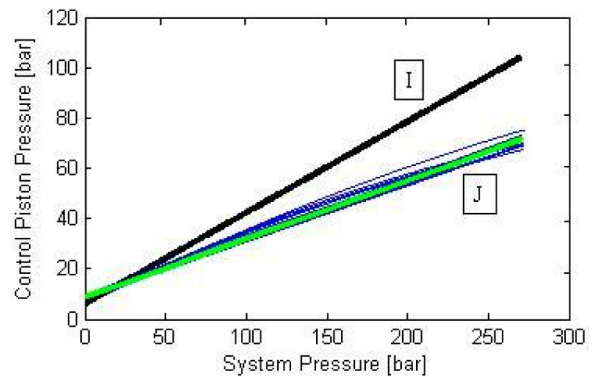


Fig. 8. Correlation between control pressure and system pressure for swash angle position at 1000 r/min

The authors have experimentally obtained a linear relation between the control piston pressure and the system pressure. It was found that the functioning of the pump in different swash angle positions can be defined by two equations. The first equation (6), as depicted in Fig. 8 curve I, describes the control piston pressure (P_C) versus the system pressure (P_S) that would be needed to initially displace the pump from its maximum swash angle position.

$$P_C = 0.362 \cdot P_S + 6.02 \quad (6)$$

The second equation (7) is the pressure balance relation for the intermediate positions. Experiments were carried out by maintaining the swash plate at different positions and varying the load. From these tests it was observed that for the various intermediate swash angle positions, the pressure relationships were found to overlap. Equation (7) was derived by interpolating the family of curves depicted in Fig.8 curve J.

$$P_C = 0.233 \cdot P_S + 8.50 \quad (7)$$

The flow characteristics model has been realized by the use of two pistons - the control piston, 10, and the barrel forces piston, 11 (Figure.10), defined by the system pressure. The relation between these two equations has been realized by a constant value in the simulation model which has been used to switch between the two equations when the swash plate is displaced from its initial position. After the swash plate angle shifts from maximum swash position, the model utilizes a correction factor of 0.644 and a different constant force to modify the pressure relationship described by equation (6) into equation (7). A transformer element in the form of a lever is used to balance the forces from the two pistons. The