DYNAMICS OF VARIABLE DISPLACEMENT PUMP CONTROLS

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"What you don't know would make a great book"
Rev. Sydney Smith

ABSTRACT

For a medium pressure axial piston unit installed in a test circuit where two variable restrictors excite the system transients, a dynamic model explores the performances of individual controls. For two case-studies ("pressure compensator" and "flow compensator") a reference configuration is treated in some detail and checked for sensitivity. Multiple controls are then introduced by giving two examples of "free" structures, identifying the general class of "modular" structures and outlining few interaction problems.

KEYWORDS: Axial Piston Pump; Pump Displacement Control; Dynamic Modelling; Simulation

INTRODUCTION

What For

Variable displacement pumps gain a growing recognition in the development of energy effective and more productive systems, particularly in mobile hydraulics. The specific interest in displacement controls arises from their being key points in design of applications and selection of hardware (the advent of electronics even makes the statement stronger).

The scenery offered by the current relevant literature (really not great in number) suggests the image of several small streams of water still waiting to merge into a main river-bed. A unified cultural background is still to come. Then, the project described here in its early stages starts one more stream, aiming to develop first hand know-how for education and design purposes.

What About

The analysis is restricted to automatic controls suitable for open circuit axial piston pumps. According to the broad classification given in Fig.1, individual hydraulic controls are mainly addressed (pressure and flow compensators), complemented by some hints on combined controls.

The functional structure of any automatic control features: (i) a pilot stage, that accepts information signals coming from the outside world and outputs one or more control signals; (ii) a power stage, that changes the swashplate attitude according to the control signal/s. Four feasible power stage schemes are sketched.
Fig. 1 - Classification of automatic pump displacement controls

in Fig. 2. Scheme 1 is the most frequent in commercial products, schemes 2 and 3 coming next; scheme 4 is rare. The present paper deals with scheme 2 (inclusive of scheme 1 as a limit).

(Absolute) Pressure Limiter

The absolute pressure limiter, commonly named pressure compensator (PC), is the "king" of pump automatic controls, as the relief valve is the "queen" of control components.

Inside Model

The conceptual cutaway view of a pressure compensated pump is found in Fig. 3. The relevant dynamic model describes three physical conditions, briefly outlined hereafter (without analytical details):

1) Swashplate Equilibrium - Basically the approach is the same as in (1). A number of active torques are well defined, though not necessarily precise. They are due to: swashplate inertia, average pumping pistons inertial terms, swashplate weight, bias piston and spring forces, control piston force. Two more terms are less defined and require a decision: (a) all frictional effects are lumped into a viscous term proportional to the swashplate angular speed; (b) torque resulting from pressure transients within the pumping chambers comes from ideal (matched) transitions driven by damping grooves on the timing plate. Sooner or later assumption (b), excluding the effect of pump speed, is likely to be superseded by an "angle oriented" approach as for instance in (2);

2) Control Chamber Continuity - The compressibility flow is balanced by three terms: (i) in- and out-flow through a circular hole metered by the control spool edges (Fig. 3); (ii) flow induced by the

Fig. 3 - Pressure compensated pump schematic (1 = swashplate; 2 = bias piston; 3 = bias spring; 4 = timing plate; 5 = control spring; 6 = control spool; 7 = control piston)
control piston motion; (iii) external leakages through piston clearances. Though leakages are sometime referred to be dependent on the square of pressure difference, e.g. (2) and (3), the conventional linear dependence is retained;
3) control spool equilibrium - The spool & spring inertia is balanced by the delivery pressure action, the spring reaction and a friction resistance proportional to spool speed. Flow forces are absent, because of marginal modelling problems.

Outside Model

According to ref.(4) the pressure compensated pump to be tested is installed in a simple resistive circuit (Fig.4). The dynamic model implies two continuity conditions (bulk modulus being always held constant): (a) the front volume is a node point for pump flow modelled as in (2), flow to control piston, flow through the front restrictor and possibly flow through a relief valve set according to (4); (b) the rear volume is simply a node point for flows through the two restrictors.

Simulation

The complete set of model equations is integrated by means of an implicit numerical scheme derived from (6). Implicit methods are frequently applied to hydraulic systems because of their inherent "stiffness". Ad hoc programming features handle two special limit conditions: (1) both swashplate and control spool are to be maintained within specified end stops; (2) flow equations through orifices and restrictors are better shifted from the standard quadratic to a linear form when the pressure derivative tends to infinity.

Though pressure compensators can work in two modes (relieving and regulating) the former only is considered, mainly because the boundary conditions are better defined. The typical simulation experiment has the following sequence:
- the front restrictor is moved rapidly from a wide open position to a totally closed position in 30 ms (off-stroke transient);
- after the deadhead steady state is reached, the front restrictor is moved in the opposite direction to get the starting position again (on-stroke transient).

The concise parameters chosen to qualify the system behaviour are defined in Fig.5 along a pressure record. Similar descriptions are stored for other state variables (e.g. swashplate angle and spool position).

Reference Configuration

To run a simulation experiment thirty-four model parameters are to be defined. Most of them are dimensionless, since in the opinion of authors the benefits of this approach generously recompose for the preparation work. The chosen group of "reference" (not optimal!) parameters mostly come from two sources: (1) a 40 ccu/rev commercial unit for geometrical and mechanical measurements (a selection in dimensional form is listed in Tab.1); (2) literature reports (7,8) for "invisible" factors (e.g. friction coefficients).

The control performances can be evaluated from several points of view:
- a) looking at the overall system behaviour, Fig.6 maps both response and recovery time over a two-dimensional grid of pump speed and PC setting values. As a complement, Tab.2 collects the extremes of positive and negative pressure gradients;
- b) looking at the detailed pump behaviour, Fig.7 illustrates swashplate angle, spool position and delivery pressure during the
Table 1 - Selection of reference data
(pressure compensated pump)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>A(1)</td>
<td>2.27E-4</td>
</tr>
<tr>
<td>A(2)</td>
<td>1.13E-4</td>
</tr>
<tr>
<td>A(3)</td>
<td>4.01E-4</td>
</tr>
<tr>
<td>A(4)</td>
<td>3.85E-5</td>
</tr>
<tr>
<td>C(1)</td>
<td>7.00E-1</td>
</tr>
<tr>
<td>C(2)</td>
<td>1.40E-1</td>
</tr>
<tr>
<td>J(0)</td>
<td>4.50E-3</td>
</tr>
<tr>
<td>K(2)</td>
<td>1.18E+4</td>
</tr>
<tr>
<td>K(3)</td>
<td>7.71E+4</td>
</tr>
<tr>
<td>M(0)</td>
<td>2.45E00</td>
</tr>
<tr>
<td>M(1)</td>
<td>6.30E-2</td>
</tr>
<tr>
<td>M(4)</td>
<td>3.10E-2</td>
</tr>
<tr>
<td>R(1)</td>
<td>3.35E-2</td>
</tr>
<tr>
<td>R(3)</td>
<td>6.50E-2</td>
</tr>
</tbody>
</table>

It is difficult to make a comparative evaluation of point a) results, because informations in literature and data sheets refer sometimes to pressure transients and at other times to swashplate transients, rarely specifying the relevant test conditions (e.g. capacitance). In any case point a) and c) results could pretend to be of some help towards a unified representation of performances. Point b) results (or similar) allow a better insight into the internal pump processes: in particular, they advise the review of the "averaging" model assumptions (one pump revolution takes 42 ms at 150 rad/s and 25 ms at 250 rad/s).

Sensitivity

Following the usual common people acceptation, sensitivity is here the analysis of the performance changes occurring when system parameters are shifted from their reference value one at a time (a formal approach is found in (8)). Four problem areas are worth introducing:

1) in (8) and elsewhere it is concluded that some parameters are little influential on the compensator dynamics. Accordingly, the swashplate inertia and the control chamber capacitance have been halved and doubled, whereas the swashplate friction coefficient has been divided and multiplied by five. When checked in the corner points A, B, C, D of Fig.6, the conclusion is essentially
confirmed: the maximum deviations found from reference are 5 or 6 ms. They occur in four experiments (out of 24), always at low setting levels;
2) no discussion exists about the impact (on PC performances) of the testing environment: capacitance of volumes, restrictor closing & opening time, starting pressure. Fig.8 shows how the four corner points of Fig.6 relocate when the rear capacitance only or both capacitances are doubled. The neglected influences can be appreciated to a certain degree by qualitative reasoning;
3) coming to the internal structure of the model, two parameters are considered among those not readily available from geometric measurements (invisible factors): (i) by nullifying or reversing the torque on swashplate due to pumping chambers pressure transition, a moderate increase in response time is experienced (up to 20 ms) with minor average benefits on recovery time; (ii) by halving the spool friction coefficient, the effect is almost negligible, whereas the opposite (doubling) produces in point B oscillations at least up to 40 s;
4) to show the impact of non-linearities on system behaviour, Fig.9 displays what happens (with the same inputs of Fig.7) when an end stop bounds the movement of the control spool to 50% of the companion hole diameter (all previous runs had such bound set at 90%).

DIFFERENTIAL PRESSURE LIMITER

As an additional example of individual control the differential pressure limiter, commonly named load sensing or flow compensator (QC), is almost a must. Firstly because of its increasing popularity, secondly because of its formal congruency with the preceding PC.

Model And Simulation

On a qualitative basis, the only thing to be changed is the active force on the control spool, coming now from the difference between the (apparently) direct acting pump delivery pressure and the rear volume pressure (external pilot) explicitly transmitted through a damping orifice. An additional equation is then requested to model the continuity condition in the pilot chamber.

As to simulation, the testing sequence has many degrees of freedom because the control dynamics is affected by both restrictors in Fig.4: the front restrictor to modify flow setting, the rear to modify the opposing load. Only two experiments are simulated, leading to a differential pressure history equivalent to that of Fig.5:
experiment #1 - actions are parallel to the PC case-study (front restrictor completely closed), the only difference being that the initial pressure drop across the front restrictor is set at 75% of the QC setting;
experiment #2 - front and rear restrictor are adjusted to have at start an intermediate swashplate angle (8-9 deg); then, the rear restrictor moves up to a high opening in 30 ms (off-load transient) and comes back (on-load transient).

Reference Configuration

Thirty-nine dimensionless parameters are to be defined to run a simulation. Most are not changed; in comparison with Tab.1, M(2) is slightly lower, C(2) equals C(1), whereas K(3) is 20% lower. The reference differential setting is chosen at 1.5 MPa; the damping orifice diameter is

![Fig.8 - Effect of test circuit capacitances on response and recovery times (corner points only)](image1)

![Fig.9 - Effect of an end stop active on control spool (to be compared with Fig.7)](image2)
high enough that its effect is almost negligible.

Few resulting QC performances are the following:

a) response and recovery time from experiment #1, plotted in Fig.10 for three values of pump speed (150 - 200 - 250 rad/s) and absolute starting pressure (5 - 15 - 25 MPa), do not identify a grid but a nearly straight line. The relevant differential pressure gradients are relatively low: the maximum positive is 171 MPa/s and the maximum negative is -376 MPa/s;
b) response and recovery time from experiment #2 are reported in Tab.4 for three values of pump speed and two values of starting pressure. When plotted, results for the same pressure are almost aligned. The differential pressure gradients change considerably: the maximum positive turns to be 333 MPa/s, the maximum negative -34 MPa/s.

Sensitivity
Among the innumerable possibilities, two are considered, with reference to experiment #1:
1) in the same Fig.10 four points are plotted, pertaining to a lower QC setting (1.0 MPa). They are still aligned and exhibit higher response times. Generally, the same trend applies to all other characteristics, excepting one condition (150 rad/s and 25 MPa) that deserves additional evaluations;
2) in (9) it is concluded that a small difference in control spool active areas contributes a relevant damping effect to load-sensing controllers. By applying a 2% area reduction on the pilot pressure side and simulating four extremal conditions (corresponding to A, B, C, D in Fig.10): (i) the maximum gain in response time, settling time and recovery time is 17 ms; (ii) undershoots are almost unchanged; (iii) the maximum overshoot reduction is from 880% to 811%. Really, available data are far from supporting a definite opinion; for instance, few scattered runs of experiment #2 exhibit significant improvements, mainly during the on-load transient.

COMBINED CONTROLS
Pump controls rarely live single. In many applications both double controls (typically PC + QC or torque limiter) and triple controls (typically PC + QC + torque or speed limiter) are found. The presence of combined functions gives rise to problems about their structure and interaction.

Free Structures
Free structures arise when combined controls are designed taking advantage of principles or devices, not reducible to the same root. Two illustrative examples:
1) the load-sensing compensator can be modified to include also the pressure compensator function: as well known, this is accomplished by limiting the pilot pressure acting on the control spool by means of a small relief valve in the pilot chamber. Both control actions are shown in Fig.11 as resulting from a perturbation sequence similar to experiment #2 (the pressure compensator is set at 25 MPa);
2) the triple scheme in Fig.12, adapted from (10), implements torque limitation (valves 1a and 1b), pressure compensation (valve 2) and flow compensation (valve 3). A specific feature of the system is the "absolute" pilot pressure (modulated by the control valves) used to command the swash-plate attitude.

Modular Structures
These combined controls include a number of stages, i.e. a number of three way valves (modelled by couples of variable orifices), affecting in two possible ways the pressure in a control chamber:

<table>
<thead>
<tr>
<th>SPEED (rad/s)</th>
<th>PRESSURE (MPa)</th>
<th>TABLE 4 - Response and recovery time in ms (reference QC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>150</td>
<td>4.17/25.3</td>
<td>19.4/28.9</td>
</tr>
<tr>
<td>200</td>
<td>4.11/22.8</td>
<td>16.7/26.4</td>
</tr>
<tr>
<td>250</td>
<td>3.66/20.9</td>
<td>15.1/24.6</td>
</tr>
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<td>--------------</td>
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</tr>
</tbody>
</table>

Fig.10 - Response and recovery time performed by the reference differential limiter at two setting levels (letters label equivalent points)
pressure (ref=25MPa)

Fig. 11 - Double control transient induced by the rear restrictor when completely closed in 30 ms (X = shift point from QC to PC function)

1) priority (Fig.13 left) - All discharge orifices connecting the control chamber to tank are in series, whereas the supply orifices connecting the control chamber to high (delivery) pressure go to a number of intermediate nodes. In this case the PC function (the most critical) must be in the first position;

2) parallel (Fig.13 right) - Discharge orifices are still in series, but all supply orifices go to a common node. As a consequence all control functions share the same priority level.

The interest in modular control structures is twofold. On one side, they are representative of a number of actual implementations; on the other side, they allow a unified approach potentially leading to generally applicable results.

Interaction Problems

There are two sources of interaction problems:

a) design and tuning of combined controls do not allow a simple superimposition of individual controls. The evolution underlying Fig.11 is exemplary. Starting from the QC reference configuration, the relief add-on implies the reduction of the damping orifice (for instance down to 1 mm) in order to separate the large rear volume from the small pilot chamber. Without any additional modification the system tends to be unstable; acceptable performances are consequent upon a 50% increase of the control spring stiffness;

b) both individual and combined controls are frequently requested to cohabit with external valves, specially in load-sensing systems (10).

CONCLUSIONS

Any conclusion drawn in conventional sense would contradict the opening quotation. However, the opinion can be firmly stated that the experience gained is original. Not because "newly formed or created" (this implies tedious comparisons and distinctions), but because "able to produce new ideas" and directions of work (model improvements, optimization of the reference system design and its associated sensitivity, extension of testing, coupling with more realistic circuits, systematic approach to multiple controls, ...). Two topics are intentionally left apart:

Fig.12 - Schematic of a triple control system (T = tank; M = gauge; LS = external pilot; P = primary; R & S = setting pilots)

Fig.13 - Priority and parallel structure of combined modular controls (1, 2, 3 = stage number)
problems related to simulation techniques (because questions are more important than tools for answers) and relationship between testing on models and testing on hardware (a discussion going well beyond the coexistence of "measured" and "calculated" curves in the same diagram).

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REFERENCES


