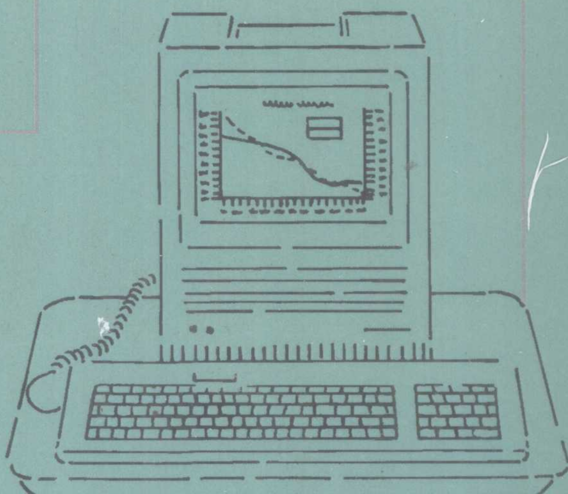
HYDRAULIC AND ELECTRO-HYDRAULIC CONTROL SYSTEMS

SECOND ENLARGED EDITION

R.B. WALTERS



KLUWER ACADEMIC PUBLISHERS

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Second Enlarged Edition

by

R.B. WALTERS
Engineering Consultant, Wembley, U.K.

KLUWER ACADEMIC PUBLISHERS
DORDRECHT / BOSTON / LONDON

Library of Congress Cataloging-in-Publication Data

Walters, R. B. (Ronald B.)
Hydraulic and electric-hydraulic control systems / by R.B. Walters.-- 2nd enl. ed.
p. cm.
Rev. ed. of: Hydraulic and electro-hydraulic control systems. c1991.
ISBN 0-7923-6537-2 (hc. : alk. paper)
1. Hydraulic servomechanisms. I. Walters, R. B. (Ronald B.) Hydraulic and electro-hydraulic control systems. II. Title.

TJ857 .W3 2000
629.8'323--dc21

00-060528

ISBN 0-7923-6537-2

Published by Kluwer Academic Publishers,
P.O. Box 17, 3300 AA Dordrecht, The Netherlands.

Sold and distributed in North, Central and South America
by Kluwer Academic Publishers,
101 Philip Drive, Norwell, MA 02061, U.S.A.

In all other countries, sold and distributed
by Kluwer Academic Publishers,
P.O. Box 322, 3300 AH Dordrecht, The Netherlands.

Printed on acid-free paper

The first part of this book has been reprinted from the first edition.

Printed in the Netherlands.

Preface to the First Edition

Force and motion control systems of varying degrees of sophistication have shaped the lives of all individuals living in industrialized countries all over the world, and together with communication technology are largely responsible for the high standard of living prevalent in many communities. The brains of the vast majority of current control systems are electronic, in the shape of computers, microprocessors or programmable logic controllers (PLC), the nerves are provided by sensors, mainly electromechanical transducers, and the muscle comprises the drive system, in most cases either electric, pneumatic or hydraulic.

The factors governing the choice of the most suitable drive are the nature of the application, the performance specification, size, weight, environmental and safety constraints, with higher power levels favouring hydraulic drives. Past experience, especially in the machine tool sector, has clearly shown that, in the face of competition from electric drives, it is difficult to make a convincing case for hydraulic drives at the bottom end of the power range, specifically at fractional horsepower level. A further, and frequently overriding factor in the choice of drive is the familiarity of the system designer with a particular discipline, which can inhibit the selection of the optimum and most cost-effective solution for a given application.

One of the objectives of this book is to help the electrical engineer overcome his natural reluctance to apply any other than electric drives. Another difficulty often encountered among all types of engineers is the unwillingness or inability to tackle the dynamics of hydraulic control systems in view of their relative complexity as compared with electric drives. Owing to the compressibility of the working fluid and the non-linear characteristics of hydraulic control devices, dynamic system modelling involves the manipulation of non-linear, high order differential equations. This fact can have a daunting effect on all but the more analytically inclined engineers, and has contributed to the wide gap that exists between the control engineer and the average hydraulic application engineer. It has often led to the oversimplification of hydraulic system identification, which has frequently necessitated costly re-design and even resulted in litigation.

It is hoped that this book will help in bridging the gap between the academic and the application engineer and thereby make some contribution towards the wider application of hydraulic control systems.

Partly due to its relative complexity, hydraulic control system analysis is an ideal hunting ground for the mathematically biased engineer. Several analytical methods have been developed over the years and every specialist in this field has his own preference. The conventional methods can be briefly summarized as:

(1) Non-linear analysis in the time domain.
(2) Linearized small perturbation analysis using the root-locus (pole-zero) approach.
(3) Linearized small perturbation analysis using the frequency response approach.

The approach adopted in this book is based on method (3), with an extension into the time domain, facilitating the modelling of system transient response to any given duty cycle. This concept, which permits system optimization in the frequency domain, has been developed and successfully applied over more than 15 years, with close correlation between predicted and actual performance over a wide range of applications, which is, in the final analysis, the ultimate criterion of credibility.

This book is based on an earlier version, *Hydraulic and Electro-hydraulic Servo Systems*, published in 1967. The original publication was aimed at an engineer using a slide rule, graph paper and other manual aids, whereas the present edition is focused on an engineer with a personal computer at his disposal. This underlines the considerable advance in communication technology that has taken place over the past twenty years. To bring the contents into line with this monumental change in analytical capability, now readily available to every engineer, the bulk of the text had to be re-written. It will become apparent to the reader that a meaningful dynamic analysis of a complex electro-hydraulic control system is not feasible without the aid of a computer.

In that context it is of interest to note that whereas the manuscript for the original edition was laboriously handwritten and subsequently typed and all graphs were manually drawn, the text for the latest edition was typed directly into a word processor and all graphs were generated and plotted by means of a specially adapted computer simulation program.

Thus time marches on!

Preface to the Second Enlarged Edition

A 'hands-on' exercise to familiarise the reader with the programs contained in the enclosed disk is a logical sequence to Part 1, which is an in depth study of component design and system analysis.
It will have become apparent to a reader of Part 1 that a meaningful dynamic analysis of a complex electro-hydraulic control system is not feasible without the aid of a computer. This leaves us with two alternatives, i.e. to write our own set of programs tailored to our requirements, or to use a suitable proprietary software package, preferably PC compatible, dedicated to the analysis and performance prediction of hydraulic and electro-hydraulic control systems.

It is of interest to list the required expertise to undertake these tasks. Considering first the user of a proprietary software package, minimum requirements can be summarised as:

1. Basic knowledge of hydraulic control concepts.
2. Ability to formulate a Performance Specification.
3. Access to a PC.

The Engineer choosing to write his own software programs would have to satisfy the following additional requirements:

4. Detailed knowledge of hydraulic control concepts.
5. Working knowledge of control theory.
6. Ability and time to write and debug software.

The majority of Application Engineers working for hydraulic equipment manufacturers, distributors or end users would fall into the first category, whereas some System Designers would have the capability to write their own programs., although frequently, in view of the considerable amount of effort required to write and debug programs,pressure of work would preclude this.

The objective of presenting part 2 in the form of an Operating Manual is two-fold, on the one hand to provide some guidelines to those Engineers prepared to write their own programs, and on the other hand to highlight the features and versatility of a proprietary dedicated knowledge-based performance prediction software package with special emphasis on the presentation of performance characteristics in the graphical form of single and multiple plots.

In contrast to Part 1 of this textbook, Part 2 does not contain any equations or algorithms, underlining the earlier stated requirement for a user of a suitable software package, which does not include the need for any analytical or programming skills.

Acknowledgment

The author wishes to express his thanks to Vickers Systems Ltd, a Trinova Company, and Flotron Ltd for permission to use some of the material and illustrations.

Contents

PART 2. System Simulation Package HYDRO ANALYST

1

Introduction

All control systems can be reduced to a few basic groups of elements, the elements of each group performing a specific function in the system. The division into groups of elements can be carried out in a number of different ways, but selecting the following four groups forms a convenient structure for the definition of hydraulic and electro-hydraulic control systems.

(1) The power source.
(2) The control elements.
(3) The actuators.
(4) The data transmission elements.

The power source consists invariably of a pump or combination of pumps and ancillary equipment, e.g. accumulators, relief valves, producing hydraulic energy which is processed by the control elements to achieve the required operation of the actuator.

The control elements can be valves of one type or another, variable displacement pumps or variable displacement motors. Some control systems contain a combination of all or some of these control elements.

The actuator converts the hydraulic energy generated by the power source and processed by the control elements into useful mechanical work. An actuator producing linear output is referred to as a cylinder, jack or ram, whilst an actuator giving continuous rotation is a hydraulic motor and an actuator giving non-continuous rotation is usually called a rotary actuator.

The control elements act on information received from the data transmission elements; in a 'simple' hydraulic control system the data transmission elements are mechanical linkages or gears, but in 'complex' systems data transmission can take many forms, i.e. electrical, electronic, pneumatic and optical, or combinations of these types of data transmission. Although 'simple' or mechanical–hydraulic control systems are still in use, they are being progressively replaced by the more versatile and flexible electro-hydraulic control system, using electronic data transmission.

Control systems can be subdivided into two basic types: on–off, or 'bang-bang', and proportional. A typical example of the former is an electro-hydraulic system controlled by solenoid-operated directional valves, where actuator velocity is pre-set but not controlled, whereas an example of the latter would be a velocity control system controlled by a solenoid-operated proportional valve controlling the flow to the actuator and hence its velocity. This book will confine itself to proportional systems, i.e. to control systems where a functional relationship exists between the controlled output quantity and the demand signal.

2

Hydraulic Power Source

In hydraulic control systems the pump supplies fluid either at substantially constant pressure which is independent of the external load acting on the actuator or, alternatively, at a supply pressure which is a function of the external loading.

In systems which have the supply pressure maintained at a constant level, the hydraulic power source can be either a fixed or variable displacement pump. In either case, as pressures normally encountered in power hydraulic systems are relatively high, the pump would be one of the three positive displacement types, that is gear, vane or piston. Piston pumps can have either axially or radially mounted pistons. The choice of pump depends mainly on the maximum pressure and rate of flow required for the operation of the system.

Three typical arrangements of a constant pressure supply are shown in Figs 2.1–2.3. In its simplest form, shown in Fig. 2.1, the power unit consists of a fixed displacement pump, a pressure relief valve and a reservoir. As the

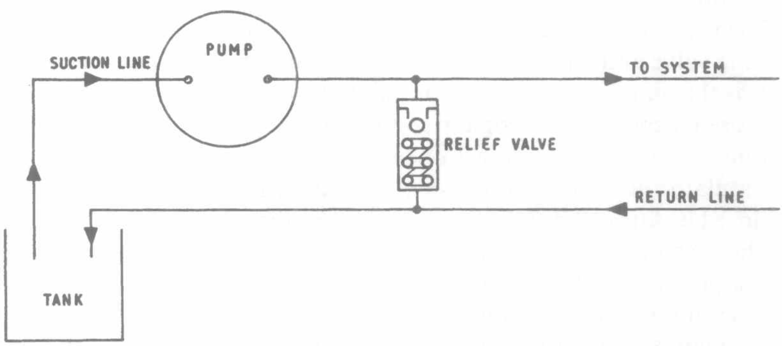

Fig. 2.1 Fixed displacement pump with relief valve.

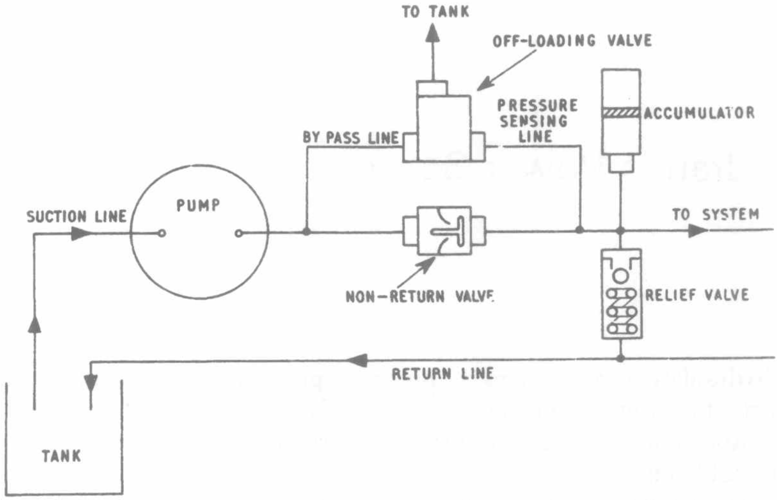

Fig. 2.2 Fixed displacement pump with accumulator and off-loading valve.

pump delivers fluid at a constant rate, the supply pressure is determined by the setting of the pressure relief valve which diverts pump flow in excess of system demand back to tank. The energy dissipated through the relief valve is not recoverable and is converted into heat energy causing a rise of temperature of the fluid in the system, which can be counteracted to some extent by increasing the amount of fluid circulating in the system, although normally, other than for very low power levels, some form of cooling would have to be provided.

A more efficient supply system, employing some additional components, is shown in Fig. 2.2. The additional components are: an accumulator, an off-loading valve and a non-return valve. On start-up the bypass line is closed and, as the pump charges the accumulator, system pressure rises; when the pressure reaches a value corresponding to the high level setting of the off-loading valve the bypass line is opened, thereby off-loading the pump. The accumulator now maintains system pressure with the non-return valve in its closed position. Since the flow required by the system has to be provided by the accumulator, the supply pressure will drop as the accumulator discharges. When the pressure reaches a value corresponding to the low level setting of the off-loading valve, the bypass line is blocked and the pump again supplies the system; the accumulator is re-charged and supply pressure rises. The sequence of operations is then repeated, the relief valve

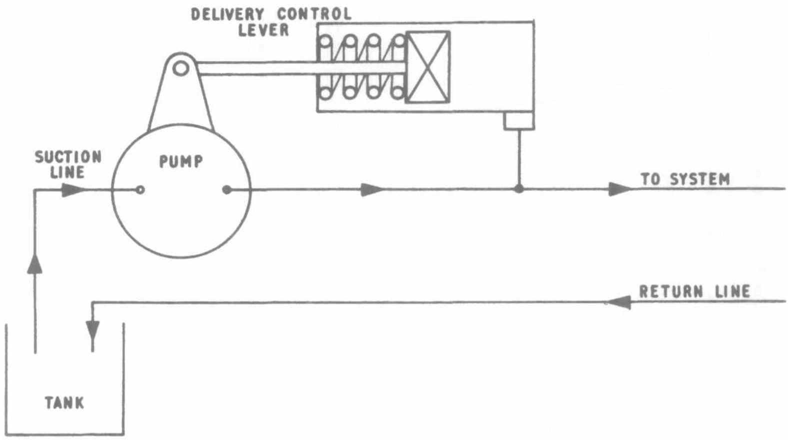

Fig. 2.3 Pressure-compensated variable displacement pump.

acting solely as a safety device and remaining seated under normal operating conditions.

This arrangement is particularly suitable for systems operating for extended periods under idling or low flow conditions. The rating of the pump and size of the accumulator depend on the duty cycle; therefore, if in addition to long idling periods the system is required to operate for extended periods at peak or near peak demand, the pump should be rated at the maximum flow, enabling a small accumulator to be used to maintain supply under idling conditions. Energy dissipation and the consequent cooling problem is considerably reduced by using a power unit of this type rather than the simpler arrangement previously described.

An alternative solution, that overcomes the inefficiency of the first arrangement, is to use a double pump unloader system incorporating a sequence valve to unload the larger pump.

The most efficient hydraulic power source is a variable displacement pump. Figure 2.3 shows a diagrammatic arrangement of a pressure compensated variable displacement pump which supplies fluid to the system at constant pressure and varying flow. In a variable displacement pump the rate of flow is controlled by operating the delivery control lever which in turn varies the displacement of the pump from nominal zero to its maximum rating. Different methods are used to obtain variable displacement, but most designs use either the principle of variable swash-plate angle or provide means for varying the angular position of the cylinder block assembly in relation to the drive-shaft axis.

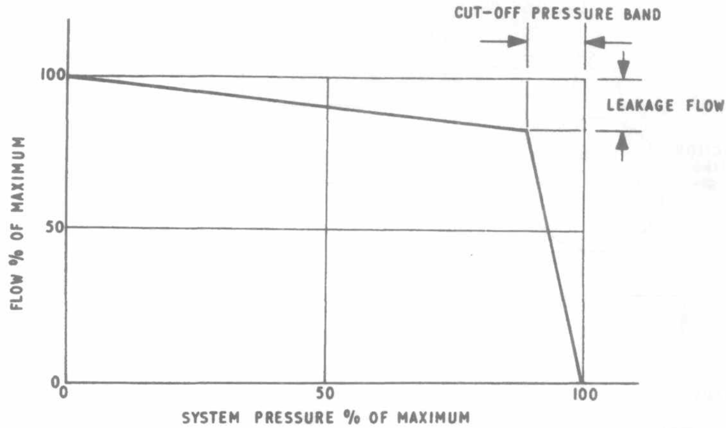

Fig. 2.4 Pressure-flow characteristics of a pressure-compensated variable displacement pump.

The pre-load of the bias spring in the pressure control unit determines the supply pressure. If the pump delivers fluid in excess of that required by the system, supply pressure will momentarily rise, unbalancing the load acting on the piston of the delivery control actuator which will then reduce the displacement of the pump until flow equilibrium is restored. If the delivery control lever overshoots the equilibrium position, supply pressure will drop below the nominal setting, unbalancing the control piston and thereby increasing pump flow until equilibrium is regained. Typical pressure flow

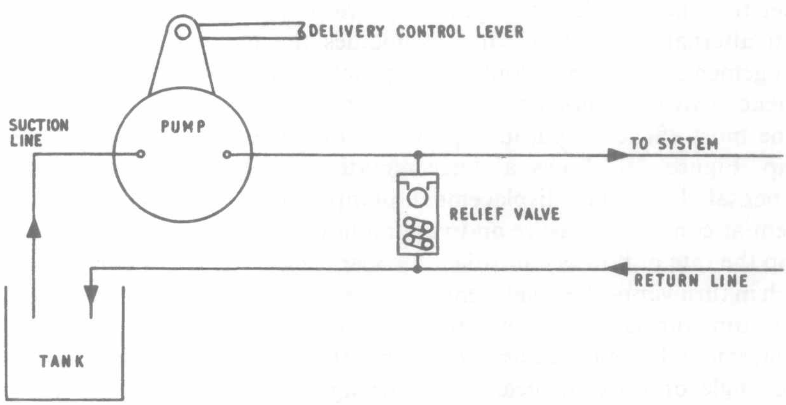

Fig. 2.5 Variable pressure supply system.

characteristics of a pressure-compensated variable delivery pump are shown in Fig. 2.4. Unless the pump is allowed to saturate, working pressure is always within the cut-off pressure band, the design of the pressure control unit determining pressure flow characteristics. Normally supply pressure at maximum flow is 90% or more of the pressure at zero flow.

In hydrostatic transmissions the velocity or position of the actuator is directly controlled by varying the flow of a variable displacement pump. In such a system the supply pressure does not remain constant but varies as a function of the external loading. A schematic arrangement is shown in Fig. 2.5. Under normal operating conditions the relief valve is seated and acts as a pressure limiting safety valve.