HYDRAULIC CONTROL SYSTEMS DESIGN

Fengyu Yang

LANZHOU UNIVERSITY PRESS

PREFACE

HYDRAULIC CONTROL SYSTEM DESIGN fills the education gap between practical texts that emphasize hardware and applications at the expense of theory, and theoretical fluid dynamics texts that have little relationship to the real world of functioning systems.

Blackburn, Lee and Shearer's magnificent book *Fluid Power Control* gives an almost inexhaustible source of fundamental information. The best book is that of Merritt, *Hydraulic Control Systems*, Wiley, New York, 1967. This book of 350 pages is directed more specifically to hydraulic control problems. Except for the last chapters, it is excellent. However, it does not cover design problems.

This textbook's main source of contents comes from:

- *Analysis, Synthesis and Design of Hydraulic Servosystems and Pipelines*, by Taco J. Viersma,1980.
- *Control System Principles and Design, by* Ernest O. Doebelin,1985.
- *Modern Control Systems*, by Richard C. Dorf and Robert H. Bishop,2002.

Analysis, Synthesis and Design of Hydraulic Servosystems and Pipelines discuses many design topics and experimental results. There are various case studies and special topics in *Control System Principles and Design* which make the study become enjoyable. *Modern Control Systems* supplies update information, for example topics in robust control systems and in digital control systems.

Combining these splendid contents, with an emphasis on the analytical technique and the designing methods, is the main aim of *HYDRAULIC CONTROL SYSTEMS DESIGN*.

Fengyu Yang

Lanzhou University of Technology

March, 2004

CONTENTS

1. TOOLS FOR THE ANALYSIS OF HYDRAULIC SERVO'S

2. ANALYSIS OF HYDRAULIC SERVOSYSTEMS

3. DYNAMIC ANALYSIS AND SYNTHESIS

4. PERFORMANCE DIAGRAM AND DESIGN OF SERVOMTOR

5. ROBUST CONTROL SYSTEMS

6. DIGITAL CONTROL SYSTEMS

CHAPTER 1 TOOLS FOR THE ANALYSIS OF HYDRAULIC SERVO'S

1.1 CONTROL SYSTEM DESIGN PROCEDURES

Just as for any other engineered product or service, the overall design of feedback control systems proceeds in systematic fashion through a sequence of steps as in Fig.1.1. Rather than discussing this general process we will consider those aspects peculiar to feedback controls, which apply to the portions of the overall design process so designated on Fig.1.1. Due to the closed-loop nature of the system configuration, the input of each component is affected by all the other components, requiring a design/analysis approach that considers the entire system simultaneously. Fundamentally, the system is described by differential equations, the most general and highly developed theory being based on ordinary linear equations with constant coefficients, in which time is the independent variable.

Before one can write and solve the system differential equations to obtain system performance data, one must first conceive a system configuration and operating principle. New systems generally utilize combinations and/or variations of well-known principles, thus a designer must be familiar with this background. In addition to familiarity with basic principles, practical designers must also maintain up-to-date knowledge of available hardware in their field of application. Trade journals, where manufacturers present advertisements and technical papers explaining the latest developments, are a major source of such information. Although new hardware is continually being developed, there are many classes of equipment that have been in wide use for decades and that show no signs of disappearing. Since the philosophy of this text attaches great importance to the benefits obtained from a concurrent treatment of theory and practice, hardware of such enduring nature will also be discussed at some length.

Once a tentative system concept has been developed through combination of basic principles and hardware familiarity, components must be "sized" to meet the needs of the specific application. For example, in the tracer lathe, use of a servovalve and hydraulic actuator represents the designer's choice (from several available alternatives) of a particular form of drive system; however, further decisions must be made as to the size of valve and actuator. These are determined from consideration of specifications such as the desired maxima of actuator force and speed required in the machining operation. Tentative choices of the form and "size" of all system components can be made in this way, and once this phase is complete, the form of the components' differential equations (and the numerical values of coefficients in these equations) become available, the simultaneous set of all these equations becoming the overall system description. Actually, at this stage, one numerical value, the loop gain, is left open to choice. Usually, loop gain is composed of several "bits and piece" of gain, one for each component, so its adjustment to some desired value presents no practical difficulty. The "proper" value for loop gain is the one that meets the system accuracy specifications while maintaining adequate stability margins. A "brute force" approach to finding this value would be to repeatedly solve the system differential equation for the controlled variable (with the desired command/disturbance

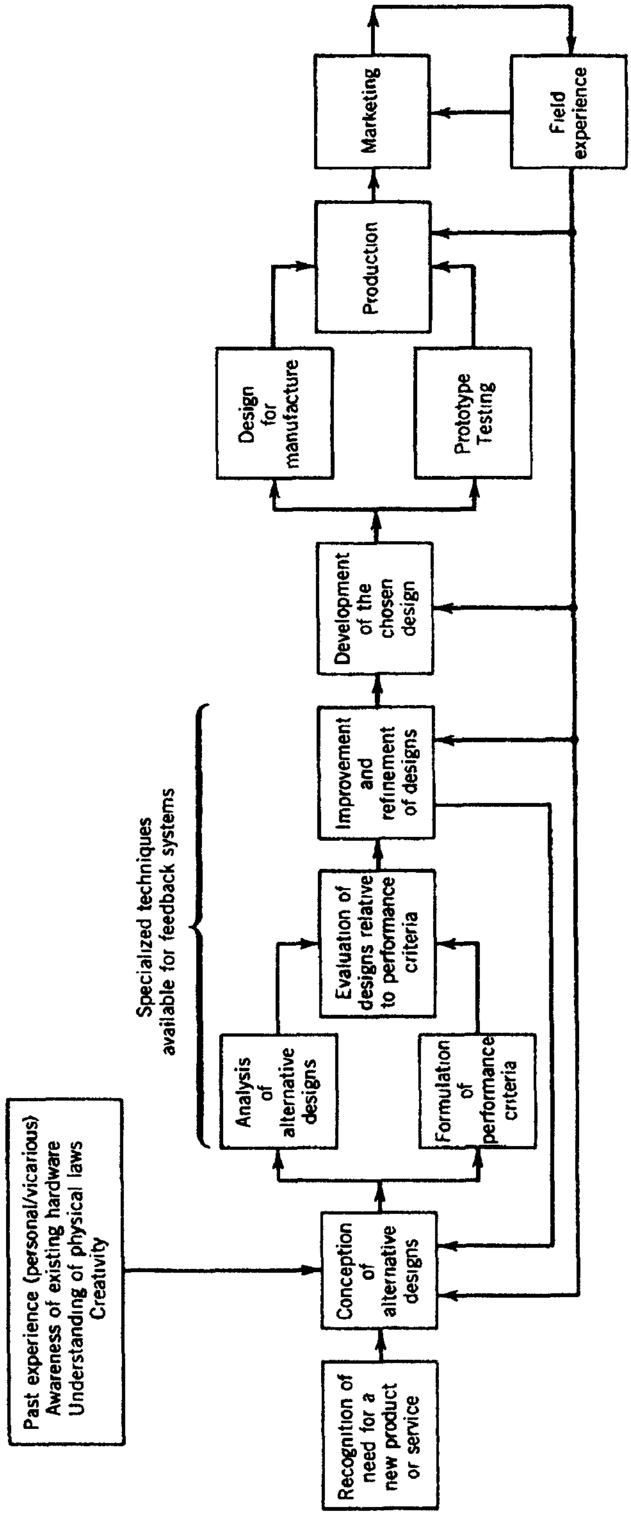

Fig.1.1 Product design process outline inputs applied) for a range of gain values and select the best. Computer-aided-design (CAD) techniques can make such a trial-and-error technique quite cost effective. However, classical control theory provides more direct (and usually one-step) methods for solving this most basic system design problem of gain setting. The two most successful practical methods, frequency response and root locus, will be explained later in this text.

Although every feedback system must have its gain set, there is no guarantee that the optimum gain for the initial choice of components will meet all system specifications. When this failure occurs, changes other than loop gain must be tried. These include:

a. Improved component dynamics ($\tau' s, \omega_n' s$, etc.)
b. Addition of dynamic compensation elements.
c. Change of controller type (on-off, proportional, integral, etc.)
d. Totally new system concept (change from electric drive to hydraulic drive, for example)

Fortunately, the frequency-response and root locus design methods include techniques that provide guidance on the type and amount of change needed to help satisfy a particular unmet specification. The modified system must then again have its gain set, and it is hoped that sufficient iterations of this modify set gain/evaluate performance procedure lead to an acceptable design. Both the frequency-response and root locus approaches were originally implemented as "manual" graphical procedures, but they are now available as software packages for rapid computer-aided design on many computer systems. In fact, the control systems field was the leader in developing CAD procedures, which are now spreading rapidly to other fields.

1.2 INTEGRATOR WITH PROPORTIONAL FEEDBACK

Complex electro-hydraulic servomechanisms involve so many facets, elements and components and so many disciplines that a systematic approach from the beginning may take many specialized chapters before some understanding of the essential characteristics of a hydraulic servo is reached.

In an attempt to discover directly some important control features, we shall derive in this introduction a very simple model of a hydraulic copying system, showing that a hydraulic servo system behaves like an integrator with proportional position feedback. To show the relationship between mechanical design and control behavior the influence of motor and valve configurations on the loop gain is discussed from the beginning.

In Fig. 1.2 a) very simple asymmetric "motor" with a so-called three-way valve is shown. Fig 1.2 a) can be simplified to give b), while c) gives the normalized presentation in which the mechanical valve configuration is not included.

Assuming an area ratio of 2:1, at one side of the piston the constant supply pressure P_S prevails, while at the other side of the piston the pressure is $P_1 = 0.5P_s$ when there is no external or frictional load. Thanks to Newton's law, $P_1 = 0.5P_s$ not only at a piston speed $\dot{y} = 0$ but also at $\dot{y}$ =constant. Evidently, the magnitude of speed $\dot{y}$ of the piston (without lode) is determined by the inlet opening a_1 and the outlet opening a_2, which in turn are controlled by the displacement x of the spool valve. Assuming an inlet flow Q_1 and an outlet Q_2, we find

$$\dot{y}A = Q_1 - Q_2 = a_1 v_1 - a_2 v_2$$

where v_1 and v_2 are the speed of the oil through a_1 and a_2, respectively. The flows Q_1 and Q_2 depend on the magnitude of the corresponding openings as well as on the pressure drop across these openings.

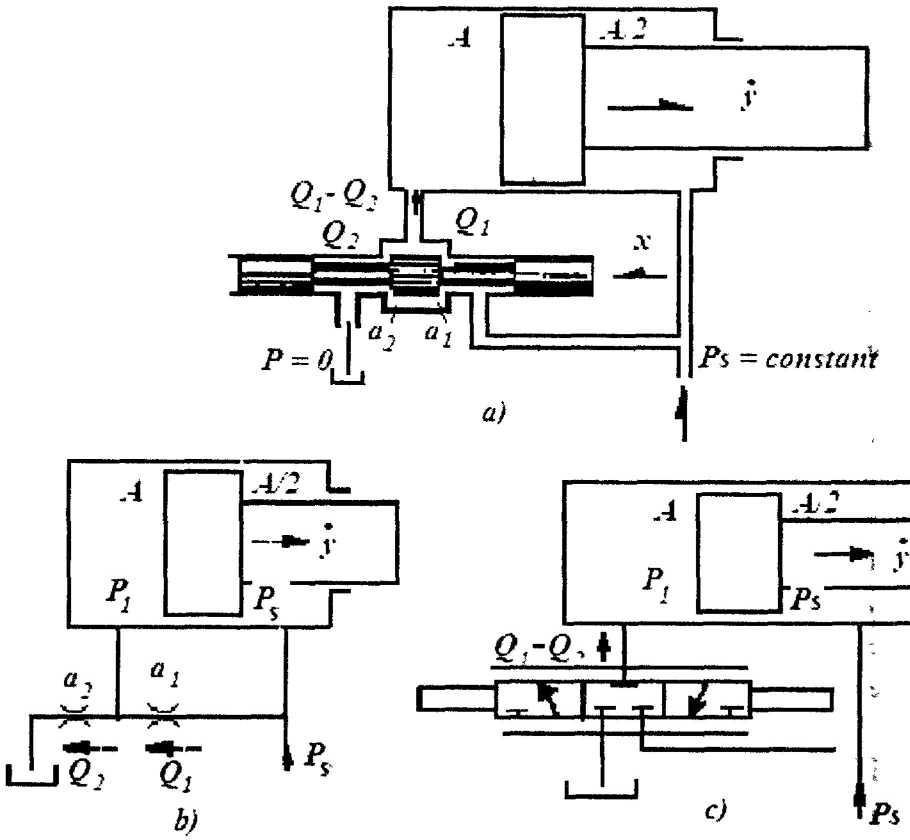

Fig.1.2 Three-way-valve-controlled asymmetric motor

Assuming turbulent flow, the (average) speed v of the oil through an opening is determined exclusively by the pressure drop across that opening. Since the pressure drops across inlet and outlet openings are equal ($P_s - P_1 = 0.5P_s$ and $P_1 = 0.5P_s$ respectively), the speeds in the no-lode situation are also equal: $v_1 = v_2 = v_0$. Thus $\dot{y}A = v_0(a_1 - a_2)$, and for the valve configuration of Fig.1.2

$$\frac{\partial \dot{y}}{\partial x} = \frac{v_0}{A} \cdot \frac{\partial (a_1 - a_2)}{\partial x} = K_m = \text{constant.}$$

Within a limited valve travel the velocity gain K_m of our motor really is a constant, so $K_m = \partial \dot{y}/\partial x = \dot{y}/x$, and the input to output transfer function of the open loop system becomes

$$\frac{y}{x} = \frac{K_m}{s}$$

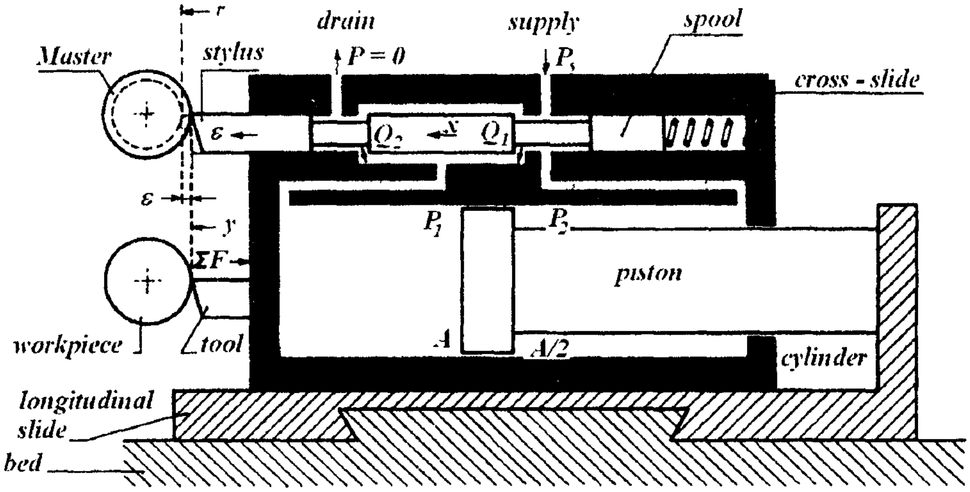

Fig.1.3 Copying lathe

where s is the differential or Laplace operator.

From the control point of view the system is an integrator, meaning that the output signal y can be derived by integrating the input signal x.

The physical implication of this is that the motor is pushed by flow and not by force. This is not a contradiction of Newton's law, since external, frictional and inertial forces should be seen as disturbance signals influencing the input of the integrator. This will be clarified later.

Giving the integrator a proportional position feedback, the system described above becomes a real servosystem able to control displacements. By way of illustration, Fig.1.3 shows very schematically the hydraulic control system of a copying lathe.

Here the piston is stationary, the cylinder is movable, while the spool valve can move relative to the cylinder. The input signal r of the control system is given by the master, the (rotary) workpiece to be copied. In perfect copying the tool displacement y (shaping the workpiece being manufactured) equals the master dimension r. Any different $\varepsilon = r - y$ will result in a corresponding spool valve displacement $x = \varepsilon$ causing cylinder movements which try to make $x = \varepsilon = r - y = 0$ or $y = r$. Here the feedback has been obtained in a purely mechanical way.

The complete cross-slide together with the hydraulic servosystem, is traveling with a constant speed along the bed of the lathe (perpendicular to the plane of drawing). The stylus will follow the variable contours of the master and controlled by the valve-the cylinder will follow according to

$$\dot{y} = K_m x,$$

where

$$x = \varepsilon = r - y.$$

Of course it is not necessary that $x = \varepsilon$, as there could be a lever between stylus and spool or even an electro-hydraulic servovalve, making $x = K_s \varepsilon$ and

$$\frac{y}{\varepsilon} = K_s \frac{y}{x} = \frac{K_s K_m}{s} = \frac{K_v}{s}.$$

But this does not change the essential character of our servo, which is that of an integrator with proportional position feedback.

1.3 DOMINANT PROPERTIES OF A SIMPLE SERVOSYSTEM

Disregarding complications concerning stability and load, one could characterize any servosystem as an integrator with feedback.. Its most important properties have been summarized in Fig.1.4. Figure 1.4 a) shows the open loop transfer function of the servo

$$\left.\begin{array}{c} \dot{y} = K_v \varepsilon \\ \text{or} \\ y = K_V \int \varepsilon dt \end{array}\right\} \overset{0}{H}(s) = \frac{y}{\varepsilon} = \frac{K_V}{\varepsilon}$$

The closed loop transfer function is that of a first order system

$$\frac{y}{r} = \frac{\overset{0}{H}}{1+\overset{0}{H}} = \frac{\frac{K_v}{s}}{\frac{K_v}{s}+1} = \frac{1}{\frac{s}{K_v}+1}.$$

K_v is called the velocity gain of the integrator. An important implication is the so-called velocity error: in order to maintain a constant speed $\dot{y}$ of the servo, fundamentally a control error ε has to be present. The control error $\varepsilon = \dot{y}/K_v$ is called the velocity error.

The well-known closed loop step response and frequency response are show in Figs.1.4 c) and d). Figs 1.4 b), c) and d) make clear that a large velocity gain K_v is desirable to obtain a

- small velocity error
- fast step response
- large bandwidth.

In practice, a desirable increase in K_v is limited by considerations regarding stability and damping.

The effect of a given load has been neglected so far. In the simplest case one may state that a load F (opposing the motion y) gives the open loop servo a proportional offset in the speed $\dot{y}$. In the motor of Fig.1.2 a) this can be explained easily. A leftward force F will result in an increased pressure P_1, causing the pressure drop across the inlet opening to decrease and that across the outlet opening to increase. Q_1 and Q_2 will change accordingly and a speed (in the direction of the load being applied) will result. This has been summarized in Fig.1.5 a), showing that an extra error signal is needed in the closed loop to compensate the load. Its consequences are shown in b) and d). From a) we derive

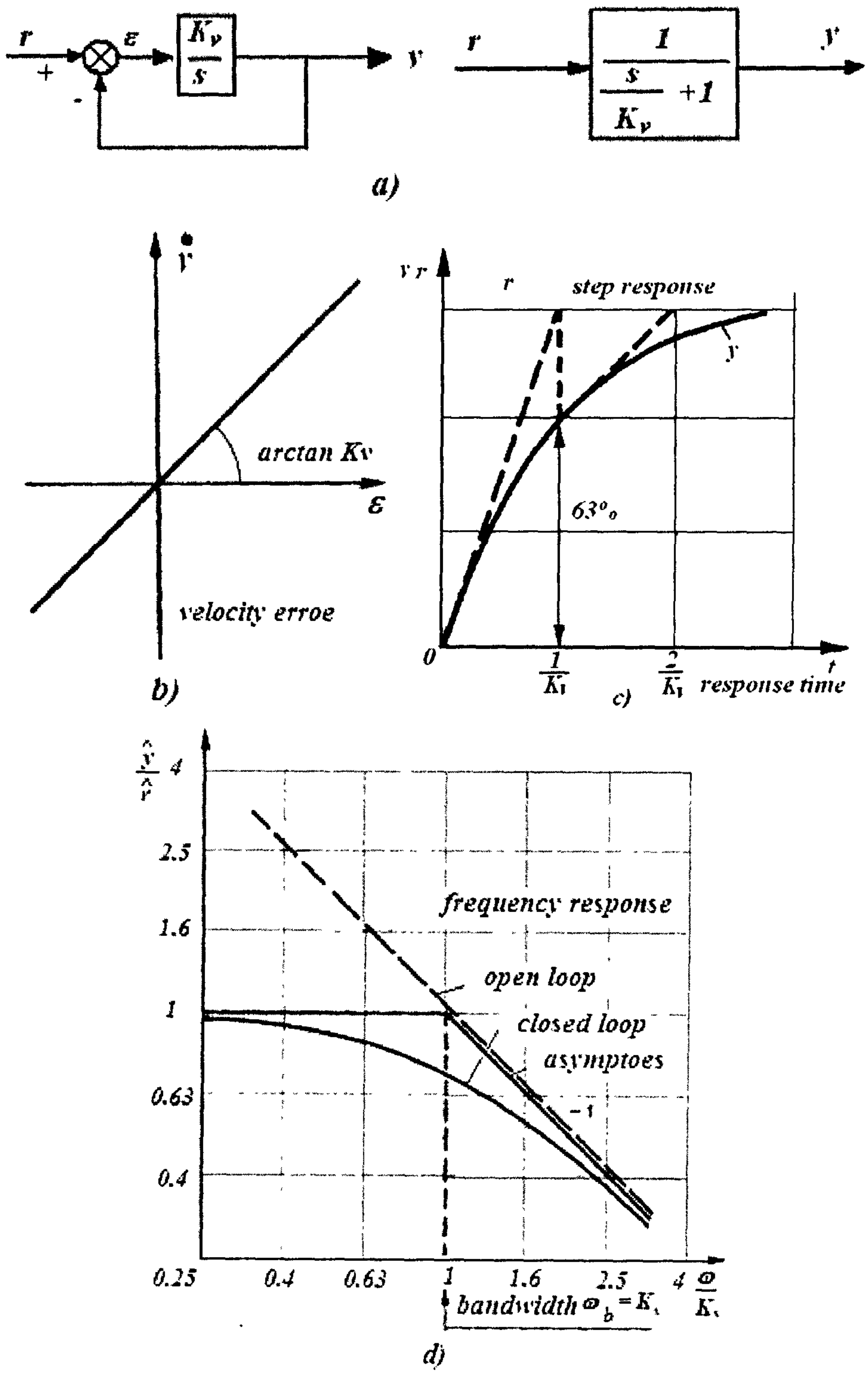

Fig.1.4 Integrator with proportional feedback

$$\varepsilon = \frac{\dot{y}}{K_v} + \frac{\sum F}{c} = \frac{\dot{y}}{K_v} + \frac{F_e}{c} + \frac{F_c}{c}.$$

Here c is the load stiffness and $\sum F$ the "total" load including external forces F_e (pay-load) and the (always present) Coulomb friction, see Fig.1.5 c). $\sum F/c$ is called the load-error, which must be added to the velocity error $\dot{y}/K_v$ to obtain the complete control error ε.

The effect of the Coulomb friction on a servosystem with load stiffness c is shown in Fig.1.5 d). Coulomb friction, reversing its sign with the speed $\dot{y}$, introduces a dead zone, which poses a serious threat to the accuracy of a servosystem.

Unpredictable variations in Coulomb friction will cause corresponding variations in the control error and could have fatal consequences for the "smoothness" of operation, particularly with regard to the speed $\dot{y}$ and the acceleration $\ddot{y}$ of the servosystem.

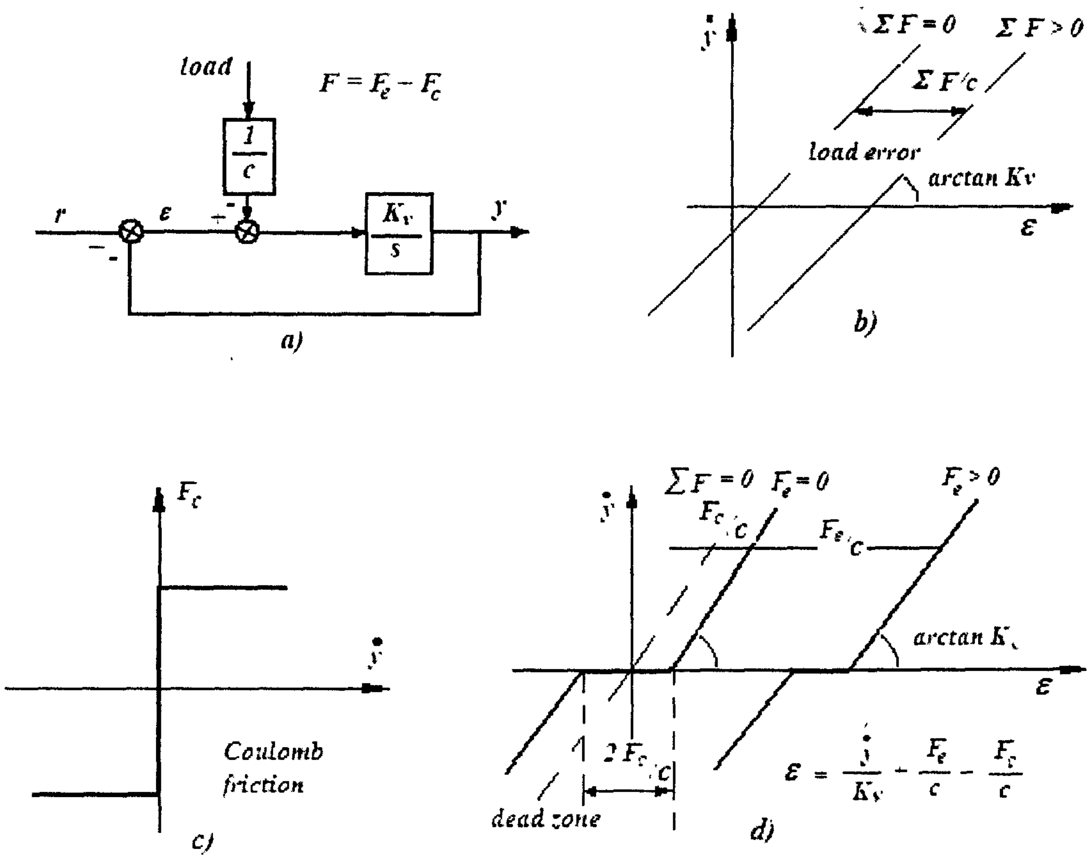

Fig.1.5 Influence of external and frictional load

To measure the performance of the servosystem one is inclined in many cases to measure the result, the output signal y. But, assuming the input signal r to be perfect, it sometimes make more sense to measure the error signal ε during operation or testing. Thus machine tool copying systems

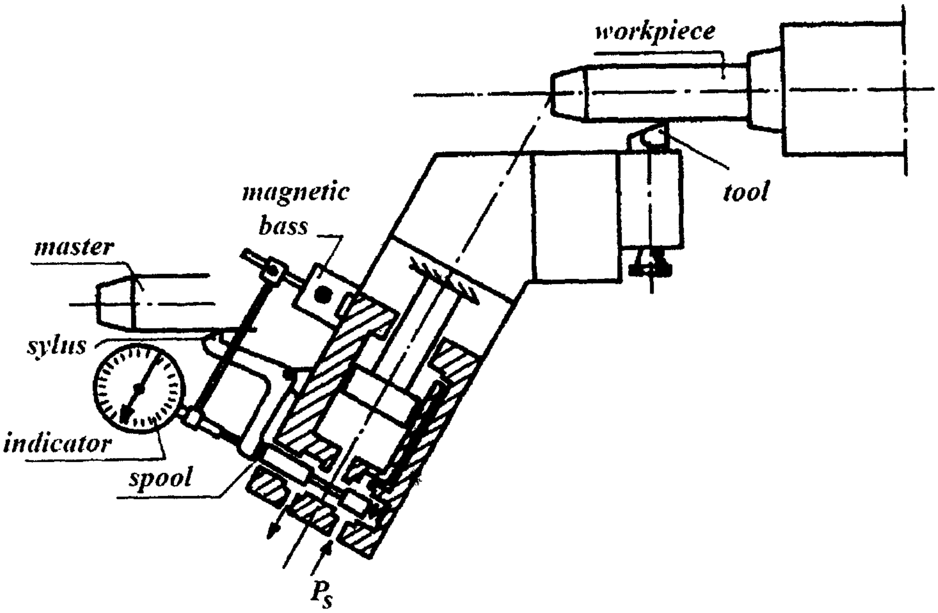

Fig.1.6 Measuring set-up

should not be tested primarily by measuring the workpieces that have been manufactured, but by measuring the error signal ε directly, as is shown in Fig.1.6, where a micrometer (attach by means of a magnetic clamp) indicates the spool displacement with respect to its housing. In the closed loop situation several points of the characteristic of Fig.1.5 d) could be measured easily, determining the velocity gain K_v and load stiffness c, as well as the dead zone. The value of such simple tests, in which relevant system parameters are measured (which is the aim.), can be illustrated by the fact that some firms are using mainly the measurement of the dead zone as a criterion in selecting copying machines to be installed in there workshops. Without any preparation, the measurement of the dead zone of a copying system could – according to Fig.1.6 – be carried out in a few minutes. It is therefore surprising to observe at machine tool exhibitions that neither sales managers nor machine tool operators understand the essentials of the experiment.

1.4 VELOCITY GAIN K_v AND VALVE CONFIGURATION

In the previous sections we have found that

$$K_v = \frac{\partial \dot{y}}{\partial \varepsilon} = \frac{v_0}{A} \cdot \frac{\partial (a_1 - a_2)}{\partial \varepsilon} = K_s \frac{v_0}{A} \cdot \frac{\partial (a_1 - a_2)}{\partial x}.$$

Thus the velocity gain K_v is proportional to the slope of the characteristic, in which the difference between inlet and outlet opening ($a_1 - a_2$) is plotted as a function of the valve displacement x. That characteristic depends completely on the valve configuration and its dimensions. In Fig.1.7 some examples are given. Fig1.7 a) shows an open-center 3-way valve and

b) a critical-center valve, both with cylindrical port openings. The corresponding characteristics are given also. It appears that the open-center valve has within the underlap-region a gain twice as outside, while the critical-center valve has a constant gain.

Furthermore, it should be noted that the open–center valve implies an oil supply (from the pump) that is particular independent of the spool position in the underlap-region. With a critical-center valve there is no oil consumption at $x=0$. This absence of "leakage flow" (at least theoretically) could be an advantage, especially in systems (such as aircraft) with a large number of motors connect to one pump set.

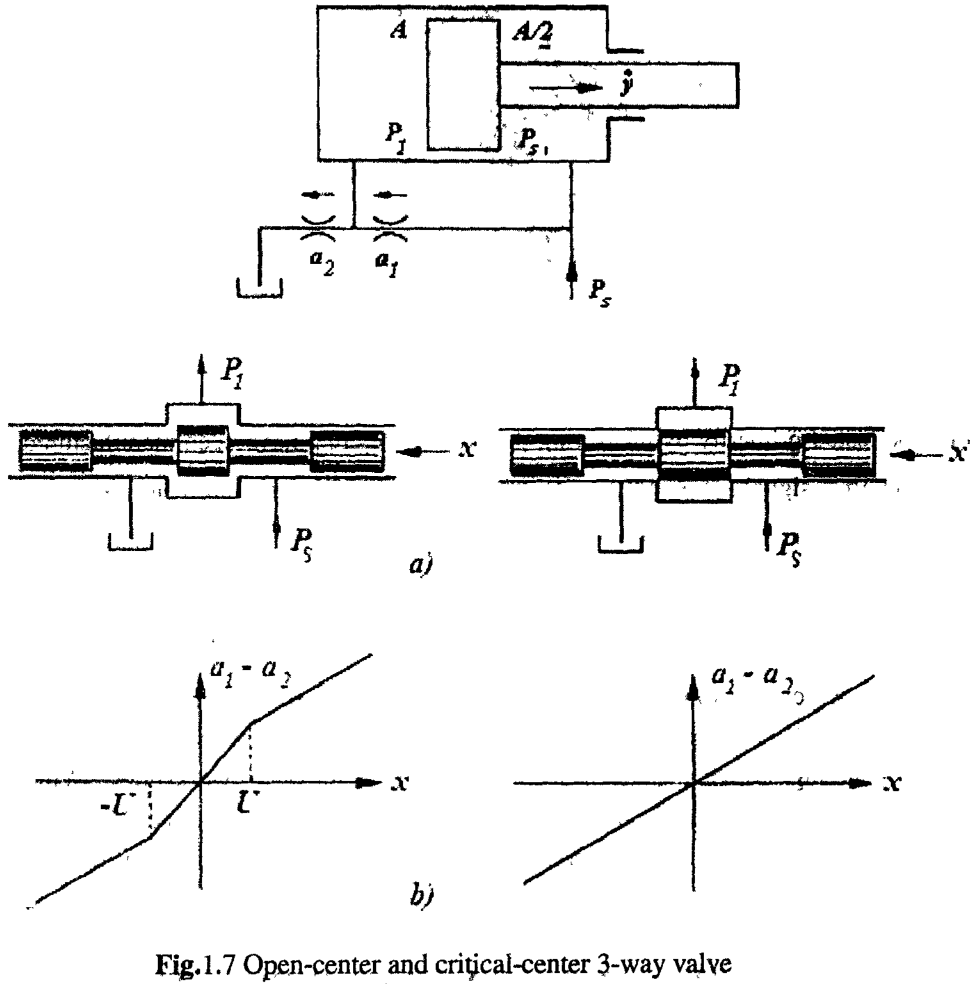

Fig.1.7 Open-center and critical-center 3-way valve

1.5 MULTIPLE-HOLE VALVE

In Fig.1.8 a very special 3-way valve is shown, a "multiple-hole" valve, where the inlet a_1 and the outlet a_2 consist of a large number of circular holes arranged in one, two or more rows. Displacing the spool, either the inlet holes (a_1) or the outlet holes (a_2) will be opened. When the "underlap" U is properly chosen, see Fig.1.8 b), the characteristic of $a_1 - a_2$ versus x (determining K_v.) is rather straight, see c). Disclosing one single hole with diameter d over a distance h , one finds an opening

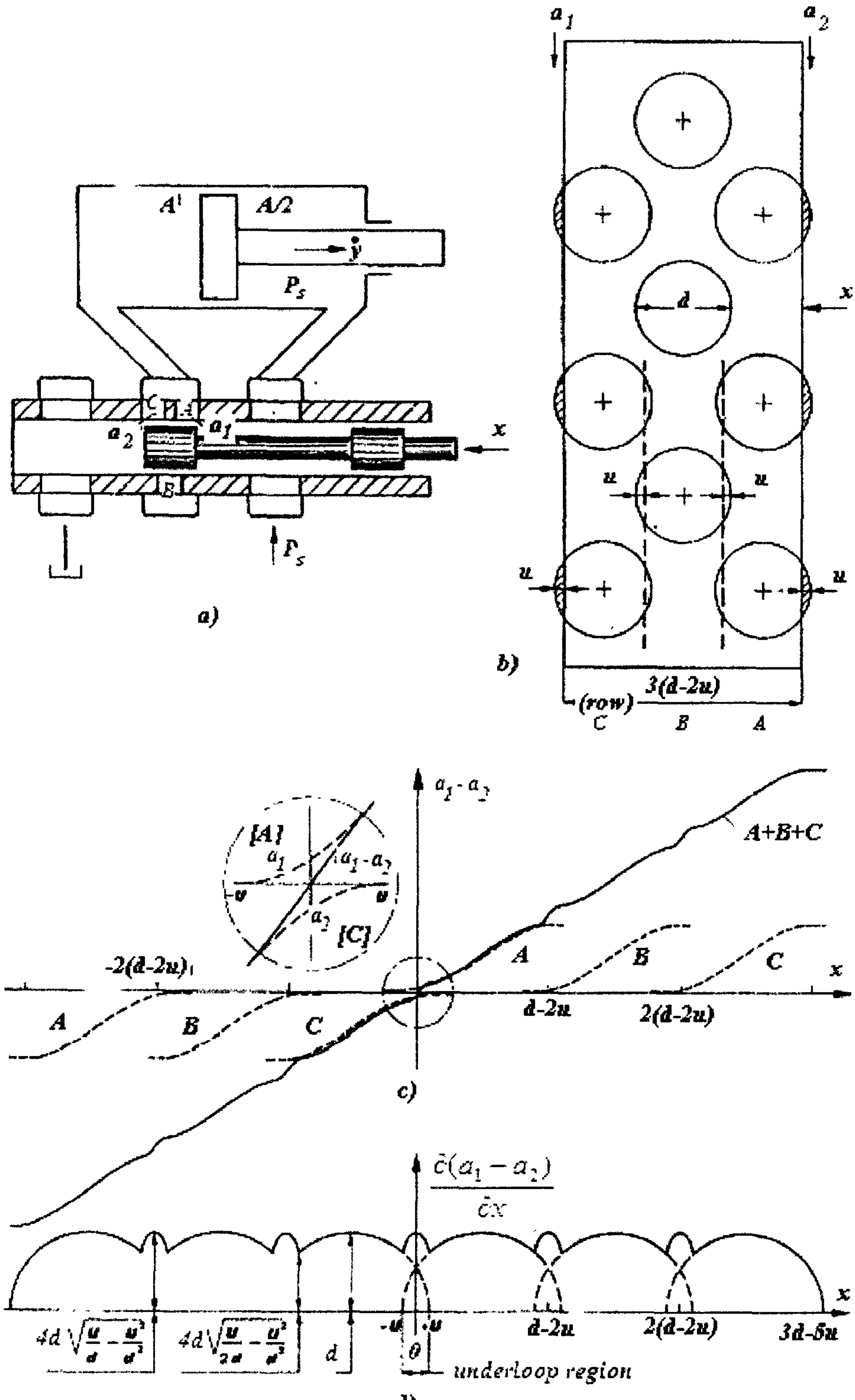

Fig.1.8 Multiple-hole valve

$$a=\frac{\pi}{4}d^2\frac{\arccos(1-\frac{2h}{d})-2(1-\frac{2h}{d})\sqrt{\frac{h}{d}-\frac{h^2}{d^2}}}{\pi}$$

and

$$\frac{\partial a}{\partial h}=2\sqrt{dh-h^2}=2d\sqrt{\frac{h}{d}-\frac{h^2}{d^2}}.$$

With an underlap U according to Fig.1.8 (b) we find inside the underlap-region at

$$|x|<U \qquad \frac{\partial(a_1-a_2)}{\partial x}=2d\left[\sqrt{\frac{U+x}{d}-\left(\frac{U+x}{d}\right)^2}+\sqrt{\frac{U-x}{d}-\left(\frac{U-x}{d}\right)^2}\right],$$

$$x=0 \qquad \frac{\partial(a_1-a_2)}{\partial x}=4d\sqrt{\frac{U}{d}-\left(\frac{U}{d}\right)^2}=\text{Maximum},$$

$$|x|=U \qquad \frac{\partial(a_1-a_2)}{\partial x}=2\sqrt{2\frac{U}{d}-4\left(\frac{U}{d}\right)^2}=\text{Minimum}.$$

Outside the underlap region we find the maximum gain at

$$x=\frac{d}{2}-U \qquad \frac{\partial(a_1-a_2)}{\partial x}=\frac{\partial a_1}{\partial x}=d.$$

The pattern of $\partial(a_1-a_2)/\partial x$ is repetitive, as is shown in Fig. 1.8 c) and d). Trying to make variations in $\partial(a_1-a_2)/\partial x$ as small as possible, we equalize the maxima inside and outside the underlap region, resulting in

$$\frac{U}{d}=\frac{1}{2}-\frac{\sqrt{3}}{4}=0.067.$$

The maximum and minimum values of $\partial(a_1-a_2)/\partial x$ then become d and 0.681 d, respectively.

Leakage flow in a multiple-hole valve depends on the number of rows with holes. In the neutral valve position (x=0), with $U/d=0.067$, the inlet opening is, in percentage of the maximum opening, 3%, $1\frac{1}{2}$% and 1% when there are, respectively, 1, 2 and 3 rows with holes, Thus the choice of number of rows depends on the amount of leakage flow leakage flow being allowed.

1.6 TURBULENT PORT FLOW

To analyze hydraulic servosystems

- Oil flow through variable port openings(valves),
- Compressibility of oil, and
- Forces and continuity

must be taken into account.