Proceedings of the Third International Fluid Power Symposium

TURIN 1973

Proceedings of the 3rd International

FLUID POWER SYMPOSIUM

May, 1973

Turin, Italy

organized by

BHRA Fluid Engineering

in conjunction with

ITM, Politecnico di Torino

Edited by: H.S. Stephens
Christine A. Richardson.

When citing papers from these proceedings, the following reference should be used :-
Title, Author, Paper No., Pages, Proc. 3rd International Fluid Power Symposium, Turin, Italy. BHRA Fluid Engineering, Cranfield, (May, 1973).

Printed and Published by

THE BRITISH HYDROMECHANICS RESEARCH ASSOCIATION
CRANFIELD, BEDFORD, ENGLAND.

ISBN 0 900983 30 2

OPENING ADDRESS

Many problems have been posed in the last few years concerning engineering and technology and the development of applied science. Man seems to be astonished and, to a certain extent afraid, of the rate of technological progress.

Astonished because this development is more and more rapid and the time for man to grow accustomed to a certain new situation is very short; new developments immediately take place.

Afraid because it is sometimes very difficult to control developments and their effects.

Different techniques have been studied in the past: each of them brought important achievements. The exchange of results gave new possibilities of progress.

Now the modern approach is "system engineering", due to the application of mathematical models and computers to the study of engineering or industrial problems as a whole.

I said "modern approach" but we should remember that the concept of system engineering was applied by the Ancient Egyptians and Babylonians with their exceptional systems of irrigation and canalisation; the same was applied by the Romans with their net of roads and water pipes.

Later on the examples of system engineering cover many fields; electric energy, gas and oil distribution etc.

Today system engineering development seems to be due to electronics and electronics plays a big part in this development because this means computers and their applications to make the calculations, when introducing figures in the mathematical models. What is the position today of all the techniques dealing with fluids?

I mean especially water, oil and air.

In many fields these hydraulic techniques cannot be displaced by others and in many applications oil-hydraulic techniques give better results, which are more economical.

In order to discuss these problems, we have organised this Conference together with BHRA in Torino; this is the second time we have jointly organised a Conference.

Two sessions of the symposium deal with systems, just to underline this important and new aspect of technology; the problems concerned with control and stability of oil hydraulic systems will be discussed, including transient phenomena; the oil-hydraulic components and equipment will be considered including chatter and noise.

The applications must always be considered with special attention; it may be useful to remember the advantages of oil hydraulic applications in comparison with some solutions to big engines, machine tools, machines for earth moving jobs and so on; for their good reliability, life and resistance to heavy working conditions.

Even the economic side must be considered, both from the production point of view and from its use.

The manufacture of oil-hydraulic components with new materials (e.g. plastic) and mass production methods gives good results even from the production cost point of view.

New components have been recently produced as micropumps, new pneumatic actuators, electro-sensitive fluids and fire proofs fluids.

During the conference all these technical aspects and trends are to be considered.

I now take this opportunity of welcoming to my home city all the delegates, who have come from 19 countries: Austria, Belgium, Canada, Cuba, Denmark, France, German Federal Republic, Greece, Hungary, Ireland, Italy, Netherlands, Norway, Spain, Switzerland, Sweden, U.K., U.S.A., Yugoslavia.

In opening the technical sessions of this Symposium on behalf of BHRA Fluid Engineering, whose director, Mr. Adler, is here with us today, and the Politecnico di Torino, I hope that each of you will find something of interest and value in the technical programme.

G.F.Micheletti,
Politecnico di Torino, Italy.

Proceedings of the 3rd International FLUID POWER SYMPOSIUM

CONTENTS

Discussion

3rd INTERNATIONAL FLUID POWER SYMPOSIUM
9th–11th May, 1973

REVIEW PAPER,

A REVIEW OF HYDRAULIC DRIVES AND SERVODRIVES SOME ACTUAL PROBLEMS

Prof. Ing. R. Chiappulini
CEMU, Italy.

Held at the Unione Industriale, Turin, Italy.
Sponsored and organised by BHRA Fluid Engineering, Cranfield, Bedford, England, in association with the Politecnico di Torino.

NOMENCLATURE

a/b	(...)	=	reduction ratio of the transmission lever between stylus and valve spool
b	(m)	=	peripheral width of valve distributing ports
$d = b/\pi$	(m)	=	effective valve diameter
d_t	(m)	=	inner diameter of laminar throttling pipe
e	($kg^{-1}.m^2$)	=	oil compressibility = 1/bulk modulus
f	($kg.m^{-1}$)	=	stiffness of the elastic floating damping piston
g	(m)	=	(double) valve underlap
h	(m^2)	=	section (surface) of the floating elastic damping piston
k	($m.amp.^{-1}$)	=	servovalve spool electric constant
i	(amp.)	=	excitating (input) current to the servovalve.
$j = \sqrt{-1}$	(...)	=	imaginary unity
l	(m)	=	length of the laminar throttling pipe
m	($kg.m^{-1}.s^2$)	=	mass of the moving slide
p	($kg.m^{-2}$)	=	useful (load) pressure
p_o	($kg.m^{-2}$)	=	pump pressure
$p_L = F/A$	($kg.m^{-2}$)	=	differential (load) pressure on the actuating piston
q	($m^3.s^{-1}$)	=	useful (absorbed) load oil flow
r	(m)	=	radial clearance of the valve
$s = j.\omega$	(...)	=	complex frequency
v	($m.s^{-1}$)	=	speed (feed rate)
w	($kg.m^{-1}.s$)	=	friction coefficient of guide ways (load)
y	(.m.)	=	servovalve spool displacement (deflection)
z	(...)	=	damping factor of the servosystem
A	(m^2)	=	actuating piston section (surface)
A_{eff}	(m^2)	=	surface (section) on the valve spool for the pressure feed-back
Amp.	(m)	=	oscillation amplitude
C	(m)	=	effective piston stroke
D	(m^3)	=	oil motor displacement
E	(m)	=	valve opening = error = signal

F	(kg)	= force (resistance) exercised by the actuating piston
I	(m)	= input amplitude
J	($kg.m.s^2$)	= moment of inertia of oil motor
K	($amp.m^{-1}$)	= amplifier and inductosyn constant
K_m	($kg.m^{-1}$)	= stiffness centering spring of valve spool
L	($kg^{-1}.m^5.s^{-1}$)	= leakage coefficient between cylinder chambers
$M = O/I$	(...)	= constant amplification circles
N	($kg.m.s^{-1}$)	= hydraulic power
O	(m)	= output amplitude
P	($kg.m^{-2}$)	= pump pressure
P	(m)	= slide travel per revolution of oil motor
Q	($m^3.s^{-1}$)	= oil flow
R	(kg)	= friction or resisting force
R_o	(kg)	= Coulomb friction force
R_1	(kg)	= continuous friction force
V	($m.s^{-1}$)	= speed, feed rate
Ω	(m^3)	= effective (elastic) oil motor oil volume
α	(...)	= stability coefficient (Nyquist diagram)
α_o	(s^{-1})	= speed (kinetic) gain
α_2	(s)	= (damping) coefficient of the 2nd degree term of the differential equation (transfer function)
α_3	(s^2)	= coefficient of the 3rd degree term of the differential equation (transfer function)
β	($kg.m^{-1}.s$)	= damping coefficient (slope) of the friction - speed characteristic
γ	(...)	= slope of the Nyquist curve at the intersection point with the negative real axis
δ	(m)	= limit cycling amplitude at rest
μ	($kg.m^{-2}.s$)	= kinetic oil viscosity
ω	(s^{-1})	= exciting frequency (rad/sec)
ω_n	(s^{-1})	= natural frequency (rad/sec)

INTRODUCTION

The exhaustive treatment of hydraulic drives and servodrives would require more than a single paper; so that we shall limit ourselves here to hydraulics as applied to machine tools and to deal only with particularly important and actual problems not yet fully resolved and to the resolution of which we offer here some original and new contributions.

We have divided the argument into three chapters: A) some significant problems referring to hydraulic drives and controls (open loop systems); B) some special problems associated with linear positional hydraulic servodrives (closed loop systems); and, finally, C) non-linear positional hydraulic servodrives not yet completely understood.

A) HYDRAULIC DRIVES AND CONTROLS

We shall deal here with only one type of hydraulic drive (structurally analogous to the more common type of hydraulic servodrive): a frictionless drive with a symmetric cylinder (Fig. A1), fed by a constant-delivery pump (with relief valve) and controlled for speed and power by a nonlinear (zero lap) proportional spool valve with four pilot ports and in theory without radial clearance (the rolling-type slide guides being frictionless and the cylinder practically frictionless when impressing motion to the piston).

The steady-state functioning of this drive is given by the fundamental equation linking in a clearly nonlinear manner the three fundamental function variables (speed V, force F exercised by the piston and valve opening E):

$$V=(E.\,024d\sqrt{p_o} - F/A)\;/A \qquad (1)$$

where:
p_o = (relief) pump pressure
A = cylinder section
d = b/π = effective valve diameter
b = peripheral width of valve ports
p_L = F/A = differential pressure on piston.

Figs. A2 and A3 illustrate Formula (1). To approach still more the structure of a linear servodrive, we have (Fig. A4) given the valve an underlap g/2 on all four distributing ports; also we have bypassed the two cylinder chambers with a laminar duct and fitted the slide with a brake of linear friction coefficient w. In the range of the valve openings $E \leq \pm\, g/2$ due to underlap g/2, this system presents a stationary state map practically linear for the three fundamental variables (Fig. A5) and its included formulae (2), (3), (4). Fig. A6 tabulates and describes the constant coefficients a_o and a_2 of the terms in V, E, F, (1); the above table also contemplates the hydroscrew drive. As evidence, the explicit expression of the inverse force gain for cylinder drives is

$$\frac{1}{\text{force gain}} = \frac{g}{4p_o A} + \frac{L}{0.47dA\sqrt{p_o}} + \frac{weC}{1.88md\sqrt{p_o}} \qquad (5)$$

which clearly shows that g/2, L and w reduce the force gain (in servodrives they increase damping, that is the dynamic stability).

In a cylinder drive, the explicitated formula (2) becomes:

$$V = E.\,\frac{0.47d\sqrt{p_o}}{A} - \frac{F}{A^2}\cdot\frac{(0.12gd}{\sqrt{p_o}} + L + \frac{weCA)}{4m} \qquad (2')$$

clearly evidencing the speed reduction caused by F (combining with g/2, L and w). However, the underlap is necessary for linearizing the drive over a large functioning range.

Fig. A7 repeats the formulae (5) and (2') and illustrates the explicit expressions (3') and (4') of the formulae (3) and (4), again for the cylinder drive.

To conclude this first part dealing with hydraulic drives:- In the drives (formula (2) to the cause, or signal, corresponds the proportional but not homogeneous effect, or output, "speed V" (modified by the proportional, secondary effect of the resisting force F). This non-homogeneity between effect V and cause E, and the imperfect (F) numerical proportionality accompanying it, are the secondary characteristics (defects) of the drives forming open-loop drive and regulating systems; a principal characteristic is the fact that the effect (output) has no influence (feedback) on the cause (input) or signal (opening of the valve).

A very important, but still uncertain problem of linear hydraulic drives is that of stick-slip when the guides (or cylinder or hydromotor) are not of the rolling or hydrostatic types but of the simple sliding type with the little understood, capriciously variable friction-speed characteristic. This is a real nonlinearity possessing, in general, a first speed-range with a negative slope (main cause of stick-slip at low speeds).

The given observations are only qualitatively useful; we shall however here illustrate particular concepts regarding damping, in general not yet sufficiently explained by the technical, scientific and experimental literature.

The above-said (a_2) permits at once the conclusion that the linear valve ($g < 0$) exerts a damping effect on the system equivalent to that of a linear brake on the slide with a linear friction coefficient w equal to:

$$w_V = (0.47\, mgd^V) / (AeC \sqrt{p_o} \tag{6}$$

Likewise, it is easy to verify that the actuator by-pass leakage L is equivalent to a linear damping coefficient w_L on the slide of the value:

$$w_L = L.4m/AeC \tag{7}$$

It is wonderful to see (not intuitively) that the damping coefficients w_V and w_L are proportional to conductances and not to resistances, contrary to direct w on the guides (in fact, gd and L are equivalent to conductances).

It is also observed that w, w_v, w_L are constants, independent of speed V and amplitude Amp of possible oscillations, which appear during the stick-slip phenomenon examined here.

The damping effect of the sliding guides is, on the contrary, a function of the speed V and amplitude Amp of the oscillations, according to the following mechanism.

The stationary (R;V) characteristic is in general a decreasing-increasing one (Fig. A8). We neglect here the secondary effects of time on the unstationary real phenomenon (i. e. the effect of the duration of the stops etc. or "hereditary mechanics") because they are inessential and severely complicate the treatment. This characteristic can be divided into two components: the Coulomb one R_o(Fig. A9) and the continuous one R_1 (Fig. A10). Here we declare at once that the Coulomb friction component R_o is "always" damping or indifferent (always stabilising or indifferent in servodrives), while the continuous component R_1 is, in general, exciting (or unstabilizing), having usually a negative slope at low speeds.

When we consider the energy per oscillatory cycle destroyed by the Coulomb component R_o, we find it to be equivalent to a laminar friction coefficient w_{Ro} on the guides equal to:

$$w_{Ro} = \frac{4R_o}{\pi\omega_n Amp} \cdot \sqrt{1 - (V/\omega_n \cdot Amp)^2} \tag{8}$$

as shown in Fig. A11. The natural frequency of the system ω_n is, in the case of a cylinder-type drive:

$$\omega_n = \sqrt{(\text{stiffness/mass})} = \sqrt{4A/eCm}$$

The slopes of the continuous component R_1 are the (essentially negative) damping coefficients $\beta = dR_1/dV$, which are generally variable with the speed V and the cycling amplitude Amp. From the "generical path of R_1 of Fig. A10, the "generical" family of curves dR_1/dV (V;Amp) (Fig. A12) can be derived. The analytical treatment is impossible because of the non-analytic and non-repeatable nature of R_1 and dR_1/dV so we have chosen a qualitative graphical analysis by superposing the familiity of curves $-\beta$ of Fig. A12 (dashed) over the curve family $(w_t + w_{Ro})$ of Fig. A11 (full lines). The points of intersection of curve pairs of equal amplitude belonging to the two families give the stick-slip oscillation amplitude Amp as a function of the speed V as shown in Fig. A13. This checks at least quantitatively the law Amp $\simeq V/\omega_n$, due to and essentially governed by the Coulomb friction and already otherwise determined by the Author [1], [3], [4]; a law which, at least at very low speeds (where we have β = constant $= \beta_o$ as the value assumed by β for $V = 0$ and Amp = 0) can also be analytically proven from the evident equation:

$$w_{Ro} = \beta_o - (w + w_v + w_L) = \beta_o - w_t \tag{10}$$

We shall not repeat here the analytical determination, based on the possibility of developing as a binomial the square root

$$\sqrt{1 - 0.25\ (\pi V/R_o)^2 \cdot (\beta_o - w_t)^2}$$

for values of the second term very much smaller than unity.

The critical or threshold speed V_{cr} at which stick-slip (for decreasing speeds) is beginning (also shown by Fig. A11) is substantially near the speed V_o (being essentially conditioned by V_o), where $(dR_1/dV)_{Amp=0} = 0$, and decreases with the appearance and growth of the remaining dampings w_t. The complicated qualitative research so far illustrated proves the usefulness of elevated ω_n by the law Amp $\simeq V/\omega_n$, giving in Fig. A11 values of Amp that are only slightly decreasing with increasing essential R_o. It also indicates the weak point (more than the neglect of the hereditary facts) of having taken the stick-slip oscillations to be harmonic and of frequency ω_n also at minimum speeds. It has, nevertheless, a certain value of physical intuitivity.

Fig. A14 gives the principal formulae of this analytic-graphical treatment.

Another important problem concerning hydraulic systems in general is that of noise and low efficiency (elevated heating) attributed with ever growing vehemence to such systems and forming a heavy handicap when compared to the fast-moving modern progress of electric systems.

Excluding the efficiency of pump and actuator (given and rather difficult to be radically improved) we shall consider the ratio between the hydraulic power absorbed by the actuator p x q and that supplied by the pump P x Q, a ratio which essentially depends on the type of circuit (and the units composing it) connecting pump and actuator, this ratio can be improved (toward unity) by improving the type and structure of the circuit.

We shall review ever more rational circuits with regard to installed power, efficiency (heating) and noise (proportional to the installed power).

Fig. A15a shows the essential, functional part of a drive for low power systems (feeds), often used because of its simplicity and low cost despite its extreme energy dissipation. In fact, the generic Fig. A15b shows in the dashed zones the elevated fractions of installed power P x Q that are destroyed into heat by both the relief valve $\{P\ (Q - q)\}$ and the speed regulating throttle valve $\{q(P - p)\}$, leaving as useful residue the power at the user p x q. A more detailed analysis is required of the mechanism controlling these losses with the change of the operating conditions (useful pressure p = R/A and speed V = q/A) and for various types of speed regulating valve. Fig. A15c, with a rigid type throttling valve (adjustments corresponding to nominal flows q_{nom}), shows the very high power lost as heat and the low useful powers resulting from variation of the resisting load p = R/A; the straight dashed lines ideally corresponding to various effective flows q_{eff} are also given.

Fig. A15d indicates the improvements that can be obtained when using pressure-compensated throttling valves (especially at high loads p) with, as before, varying load p = R/A.

The effects of the variation of the speed V = q/A are given in Figs. A15e and A15f and are equal for both the fixed throttle valve and the pressure-compensated one (at equal load p, obviously).

A first decisive improvement is obtained by using the self-regulating, practically constant-pressure pump (eliminating the relief valve) of Fig. A16a); the generic energy characteristics become that of Fig. A16b (the hatched area, corresponding to heat losses, being limited to that due to the throttling valve).

Repeating the four cases as above of the variations of the load p (with two types of throttling valves) and flow (speed) q (again with two types of valves), we obtain Figs. A16c, A16d, A16e, A16f, resulting in great, general advantages relative to the fixed pump and relief valve, especially in the effects of minor power losses, but also in the useful power. In particular, we find again the better efficiency of the pressure-compensated valve relative to the fixed one, and especially relative to useful power at high loads p.

The final step of totally eliminating the power destroyed in heat (dashed areas), that is, the constant equality of the furnished power N and the useful power N_u, can be obtained by abolishing also the throttle valve and using a pump of the self-regulating flow (displacement) and pressure type, thus taking over the functions of the two omitted valves. We shall now describe some type (or rather sub-types) of such self-regulating pumps. The first one with flow regulation (up to zero) by the user's pressure (principal variable), largely used in forming machine tools (Fig. A17a). Here, the law of dependence of flow q from useful pressure p (Fig. A17b) is unique, as is also the path of the powers with the variation of useful pressure p (Fig. A17c). The zero value of the powers, even at the maximum of the load p, are also important.

Very similar is also the "about constant" power pump (Fig. A18a) with unique bilinear (or trilinear) law governing the dependence of the flow from the principal variable, the pressure p (Fig. A18b), obtained by suitable spring trains acting against the pressure. This approximates the rectangular hyperbola giving the real power constancy, the path of power vs. load p (Fig. A18c) being in this case flatter than the preceding one and approaching the desired constant value.

The two last quoted limitations, that is, the uniqueness of the power curve with varying load and its zeroing also at the extreme maximum P of the resistant pressure p, can be overcome by using a pump that is adjustable for displacement q independently of the load p (Figs. A19a, A19b) by independent means. These may be manual (direct or hydraulically assisted) or direct electromechanical or electro-hydraulic, etc., in the case of open-loop controlled systems. In servo systems, the regulating means can use a positional mechanical error signal or a tachymetric one (power-amplified, for example, by a hydraulic follower); or a positional or tachymetric electrical signal error (power-amplified, for example, by an electrohydraulic servovalve) and in this case the pump may be called a "servopump"; it is obvious, that the servopump (in the case of positional servodrives) must have (between the first and second stage) a feedback, so that to each input signal intensity corresponds a proportional value of the eccentricity (flow) not of its rate. That is, the servo-pump acts as pump and also as power servovalve without throttling. In addition, the servopump acts as pump and also as power servovalve without throttling. In addition, the pump acts as the original relief valve and directional valve (until now not considered in the schemes, because unessential for energy reasoning).

With these last two types of adjustable pump, the flow can be chosen (or fixed) as desired, independent of the load and not sensibly modified by a casual variation of the load (on the contrary, not modified at all when acting as servopump in a servosystem). The obtainable powers are therefore of a double infinity (Figs. A19c, A19d) and may reach very high values, proportional both to the flow q (speed) and load p.

To conclude, the servopump constitutes the oilhydraulics' vanguard in its competition with electric and electronic systems. However, in the fields of numerical control and copying, the best servopumps available today are still characterized by too high response times; the realization of servopumps having response times at least 5 times lower than the best today available will be, therefore, very important for further research and development.

B) LINEAR HYDRAULIC SERVODRIVES

When transferring the distributing valve from the fixed base to the moving slide, the drive of Fig. A4 becomes a servodrive (Fig. B1); in this case, the absolute movements of the valve spool are the input, and the signal error E, or cause, is given by the difference (I-0) between input I and output, or effect, 0; i.e., the output influences (feedback) the error (E=I-0) with the tendency to cancel it; the resulting effect being that output and input are homogeneous (displacements) and also constantly and practically identical (with resultant identity of speeds). We say here "practical" identity, because the identity is ensured less the follow-up error E (required to create the speed and the accelerations currently required by the servo system as well as for overcoming the useful external resistances).

We shall now study the behaviour of the servodrive beginning with the equations already developed for the corresponding hydraulic drive.

For steady-state functioning (speed V and resistant force F constant) we can write:

$$V = (I - 0)a_o - \frac{F. a}{m^2} \qquad (1)$$

For variable functioning we shall neglect the external constant force F, because unessential relative to the dynamic stability and replace it by the inertia force $(\delta^2 0/\delta t^2)m$; the same inertia force generating not only the speed component through the permeability a_2, but also an elastic deformation of the oil in the cylinder:

$$(\delta^2 0/\delta t^2).m.(eC/4A) \qquad (2)$$

and, therefore, an elastic deformation speed:

$$(\delta^3 0/\delta t^3).m.(eC/4A) \qquad (3)$$

Putting now:

$$meC/4A = a_3 \qquad (4)$$

we obtain from (1), (2), (3), (4) the fundamental differential equation for the general non-stationary (dynamic) functioning of the servo system; which expresses a speed (flow) balance:

$$a_3(d^3 0/dt^3) + a_2(d^2 0/dt^2) + (d0/dt) = a_o E \qquad (5)$$

Cutting now the closed loop forming the servodrive (by returning the valve to the fixed base) and exciting it with a signal error of variable frequency ω equal to E. sin ω t imposed on the distributing spool, we obtain the well-known open-loop transfer function:

$$\frac{O}{E} = \frac{a_o}{j\omega(1+a_2 j\omega + a_3 j^2\omega^2)} = \frac{a_o a_2}{a_2^2\omega^2 + (1-a_3\omega^2)^2} - j\frac{a_o(1-a_3\omega^2)}{a_2^2\omega^3 + \omega(1-a_3\omega^2)^2} \qquad (6)$$

plotted in Fig. B2 according to Nyquist (assuming the signal E to be unity), giving at once the input I to the closed servosystem required to generate the same output curve (and also the same constantly unitary error E). The same Nyquist diagram gives: the coefficient $a = a_o a_3 / a_2$ = (force gain/stiffness), which must be smaller than 1 to ensure stability; the slope angle $tg\,X = \sqrt{a_3} / a_2$ = (forcegain/ a_o $\sqrt{m.stiffness}$) (as already shown by the Author in [2] ; and the constant-amplification circles M = O/I useful to determine M_{max} at the point of tangency with the curve (as well as the angular phases between I, O, E).

The transfer function (6) is more usually written in the following form:

$$\frac{O}{E} = \frac{a_o}{j\omega\left[1 + 2Z\frac{j\omega}{\omega_n} + \left(\frac{j\omega}{\omega_n}\right)^2\right]} \qquad (7)$$

where:

ω_n = natural frequency = $\sqrt{stiffness/m}$ = $1/\sqrt{a_3}$

Z = damping factor = $a_2/2\sqrt{a_3}$ = $a_2\omega_n/2$ = $1/2tgX$ = 1 / 2 Amplif.

$tg\,X$ = resonance amplification (as shown by the Author in [2]) .-

We thus obtain the three block diagrams of Figs. B3, B3a, B4, where: s = j.

The Nyquist diagram does not give "clear and immediate" instructions on how to modify the physical parameters of a servosystem to modify its performance and make it stable (a), or at least how to increase its degree of stability (M_{max}; a) to reach a desired value (giving, for example, only indirectly the damping factor Z).

Equation (6), and especially experimental results, are more easily expressed by the Bode diagram (amplitudes O/E vs. frequencies; phases vs. frequencies, less important) as shown in Fig. B5). The Bode method gives the damping factor Z (with standard template almost) and the stability criterion a right away. It does not give any immediate indication of the input I and the stability degree, that is, the maximum amplification M_{max} = (O/I). Neither does this method give a clear and immediate instruction how to manipulate the physical parameters required for varying the performances and the stability degree (a ; M_{max}) of the servosystem.

We consider the diagram (for closed loops), proposed and graphically obtained by Guillon, analytically resolved and extended by my collaborator Zuffellato and then completed and made integrally explicity (with functional and physical magnitudes) relative to coordinates and parameters by myself [2] [3] (Fig. B6) to be the most efficient instrument for the checking and dynamic design of servosystems. The explicit expressions are given in Fig. B7, which refers also to the hydraulic rotating servomotor with screw (with all the inertias taken as being concentrated as moment of inertia J on the shaft of the hydromotor and the single elasticity of the oil in the hydromotor and between hydromotor and valve). Fig. B6 reports also the % overshoots of the closed loop system after a sudden position step as input (graphically obtained by the Author from the graphic tables by Towill (8)).

The abscissae of the Guillon diagram carry the ratio α_o/ω_n between the kinetic gain and the natural frequency, the ordinates the damping factor Z, the curves parameters being the maximum amplifications $(M)_{max} = (O/I)_{max}$. The Author has added thereto the star of straight lines from the origin with the values of α as parameters. By the direct and immediate indication of α and M_{max}, the Guillon diagram gives and evidences at once both the stability criterion and the degree of stability. Due to the explicit formulation of the damping factor Z and of the abscissa by means of the functional characteristics (speed gain, force gain, natural frequency), and still more so by the elementary physical functional parameters of the system (mass; stroke; cylinder diameter and leakage; valve diameter and overlap; pressure etc.), the theoretical checking and dynamic functioning design of the servosystem becomes extremely evident and easy. Also by inspection the effect of the varying of the dynamic performance with the varying of the functional characteristics and physical and functional parameters is determined, as is shown hereinafter by a few examples.

As a premise we shall prove that contributing to the total damping factor Z are also (Fig. B7) the linear valve with the added $Z = (0.12gd/A^{3/2})\ (\sqrt{m/eCp_o})$, the by-pass leakage at the actuator with the addend $(Z_L = (L/A^{3/2})\ (\sqrt{m/eC})$) and the external brake with the addend $Z_w = (w/4)\ (\sqrt{eC/mA})$ from Fig. B6 results also, among others, the practical uselesness (or even damaging) of exceeding values of 0.7 + 1.0 of the damping factor Z.

Fig. B8 explains in detail the common generical affirmation that the increase of the mass m is contrary to stability; it is necessary to distinguish servosystems with only $Z_v + Z_L$ from those having also Z_w (the most common case in practice). An unstable servosystem with only Z_v and Z_L remains unstable and a stable system remains stable (α constant) but its stability degree decreases (M_{max} increasing). In the presence of $Z_v + Z_L + Z_w$ an unstable servosystem remains unstable, while a stable system decreases in stability (increasing M_{max}) and can even become unstable (α increasing over unity).

In Fig. B9 we analyze the effects of the increase in the effective diameter d of the valve (adjustable by means of the variation of the width of the opening $b = \pi d$), which confirm the common observation regarding the stabilizing effect of partialization or reduction of the efficient diameter d. Here too we must differ between the case with only Z_v and that having also Z_L and Z_w; repeating qualitatively the effect already observed in the preceding case, but here the effect of Z_L and Z_w on the stable servosystem is less pronounced at the lower abscissae than at the higher ones, the latter being independent of d. In Fig. B10 we analyze the stabilizing effect of an increase in the cylinder section **A**. Here the two cases are $Z_v + Z_L + Z_w$ with the same qualitatively repeated but reversed reasonings (for example, an unstable system with $Z_v + Z_L + Z_w$ can become stable and increase its stability when increasing A). The useful effects of A are however not very marked.

In Fig. B11 we analyze the unstabilizing effect of the increase of supply pressure p_o; in this case, there is no necessity to distinguish two cases according to the presence or not of one or more damping addends, because the pressure has a strong reducing effect only on Z_v.

The two last analyses show the twofold utility of keeping the product $p_oA = F_{max}$ constant by increasing A and reducing p_o in the same ratio: this is the real sense of the general observation regarding the utility of increasing A (the natural $_n$ increases, while p_o decreases).

Finally, in Fig. B12 we refer also to a variable as yet neglected. That is, in the servosystem given in schematic form in Fig. B1, a transmission lever is supposed to be interposed between the input (e.g., the stylus of a hydrocopying device) and