DEVELOPMENT AND APPLICATION OF A MOVING - BASE VISUAL FLIGHT SIMULATOR INCLUDING THE DESIGN OF HYDRAULIC ACTUATORS WITH HYDROSTATIC BEARINGS

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Abstract

A brief historical synopsis of flight simulator developments at Delft University of Technology and a survey of current manned flight simulation research at the Department of Aerospace Engineering is presented.

Special emphasis will be placed on the assessment of the dynamic characteristics of flight simulator motion systems, resulting in a uniform measurement and analysis method as suggested by AGARD.

The design of hydraulic actuators for flight simulator motion systems at the Department of Mechanical Engineering will be discussed in the second part of the paper. A performance diagram presents the connection between specifications, e.g. max. excursion, max. velocity, max. acceleration, bandwidth, and the design parameters. For a smooth motion of an asymmetric actuator an asymmetric servovalve will be required. To eliminate Coulomb-friction, resulting in the well known "reversal bump", hydrostatic bearings providing for a permanent oilfilm between the sliding surfaces inside the actuator are applied. A summary of the design method yielding the dimensions of conical hydrostatic bearings will be presented.

I. Introduction

The importance of the flight simulator as a tool in the research of aircraft handling qualities can - to a certain extent - be compared to that of a windtunnel in aerodynamic research. Just as the quality of the windtunnel is of utmost importance to the results obtained, so is the quality of a piloted flight simulator of paramount importance to the fulfilment of its purpose. This is equally true if the simulator is not used for research but for development or training.

Since 1966 the Departments of Aerospace and Mechanical Engineering of Delft University of Technology are working together on the development of a moving-base visual flight simulator to be used for research purposes.

The first part of this paper presents a survey of manned flight simulation work carried out at the Department of Aerospace Engineering and one of the research applications, e.g. the measurements of the dynamic characteristics of flight simulator motion systems, is described.

In the second part of the present paper, the

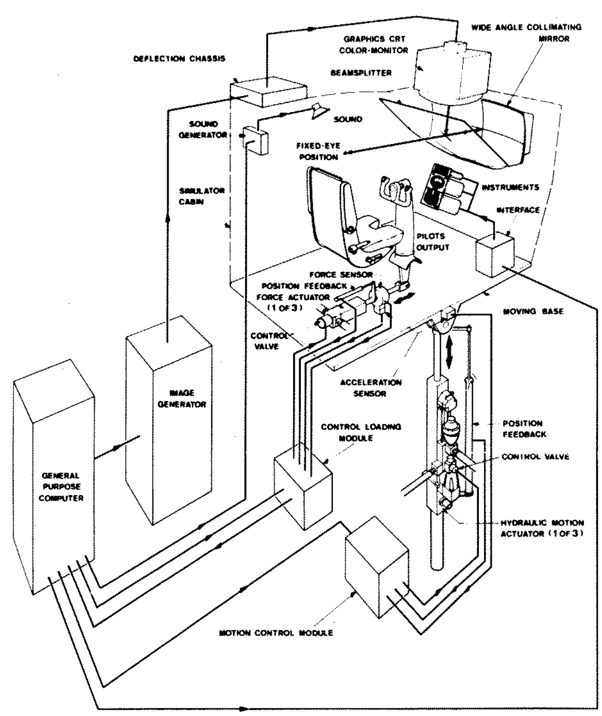

Fig. 1 Simplified lay-out of flight simulation components.

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design of hydraulic actuators with hydrostatic bearings for flight simulator motion systems will be discussed. The design example concerns one of the actuators of a large six-degrees-of-freedom synergistic motion system under current development at the Department of Mechanical Engineering.

II. Flight simulation research at the Department of Aerospace Engineering

II. 1 Brief history

At the Department of Aerospace Engineering flight simulation activities started in 1955 with the arrival of a university built analog computer. In 1967 the fixed-base flight simulator, developed since 1962, was provided with a so called contact analog visual display of a runway outline, providing the possibility for studies on approach and landing simulation(1). This visual display, using a high-intensity projection oscilloscope, was replaced in 1980 by a CGI-night visual system, using a beam-penetration color CRT and collimating optics, see Figure 1.

Research on flight simulator motion started in 1969 with the introduction of a three-degrees- of-freedom motion system (roll, pitch, heave) using hydraulic actuators with hydrostatic bearings(2). It was the first motion system using this now well-known and accepted technique to improve the dynamic characteristics of flight simulator motion generation systems. Responsability for the design of this motion system, see Figure 2, was with the Department of Mechanical Engineering, where the first author of the present Paper started development work on hydrostatic bearings in 1966. Results of these efforts are reported in Section IV.

II. 2 Nature of the work

Research within the Department of Aerospace Engineering is carried out by the staff with a very essential participation of our students, doing thesis-work on different subjects, not only in the field of flight simulation but in other areas as well. This modus operandi generally applies also to the Department of Mechanical Engineering.

The flight simulator is used in various research programs of the Department, covering the following topics :

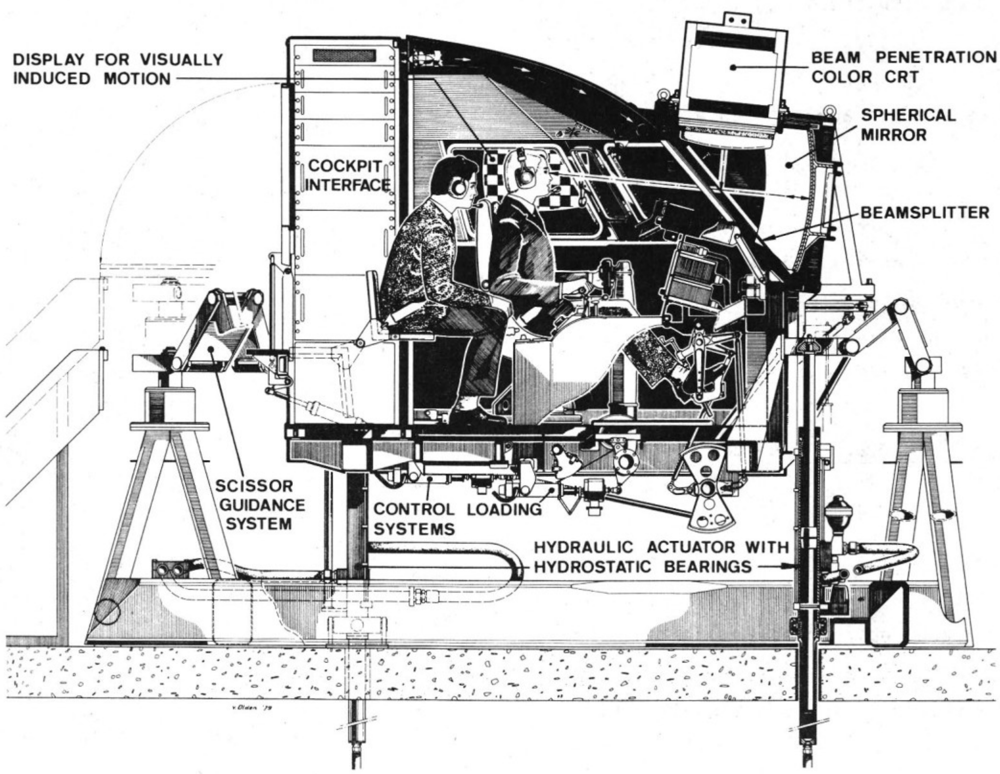


Fig. 2 Delft University moving-base visual flight simulator.

- aircraft stability and control research
- research on flight simulation techniques.

In aircraft stability and control research, the following projects can be distinguished :

- aircraft handling quality investigations, including the response to atmospheric turbulence and windshear[3], and the acceptability of new approach and departure techniques[4],

- studies of human pilot behaviour in flight, including physiological measurements on vestibular thresholds of perception[5], and the interaction of visual and vestibular perception of aircraft motions[6],

- research on digital control techniques, where the flight simulator is used as a test bed for a digitally controlled flight director/auto pilot, prior to the installation of the equipment in the De Havilland DHC-2 "Beaver" laboratory aircraft.

To solve the problem of how good does the simulator need to be and what kind of visual and motion systems are necessary for each of the various missions to which the simulator may be applied, research on flight simulation techniques is carried out. Under this heading the following topics can be distinguished :

- the formulation of mathematical models for flight simulation, programmed in modular form, see Figure 3,
- visual simulation of the outside world, using CGI-techniques, see Figure 4,
- research on the assessment of the dynamic characteristics of flight simulator motion systems, see Section III,
- the generation of motion cues on aircraft motion simulators[7],
- description and simulation of windshear and non-Gaussian (i.e."patchy") atmospheric turbulence[8],
- control force ("feel") identification and simulation using hydraulic actuators with hydostatic bearings in the flight simulator control loading systems[9].

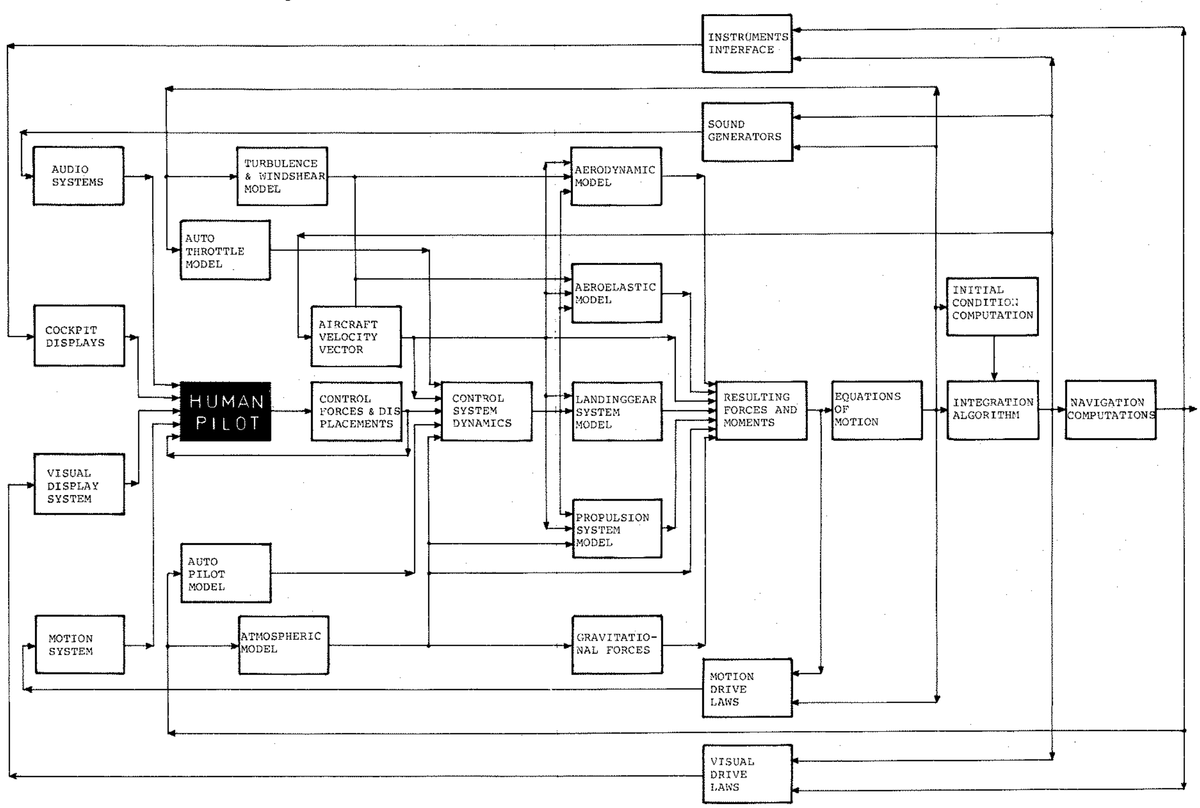


Fig. 3 Block diagram of hardware and software modules for piloted flight simulation.

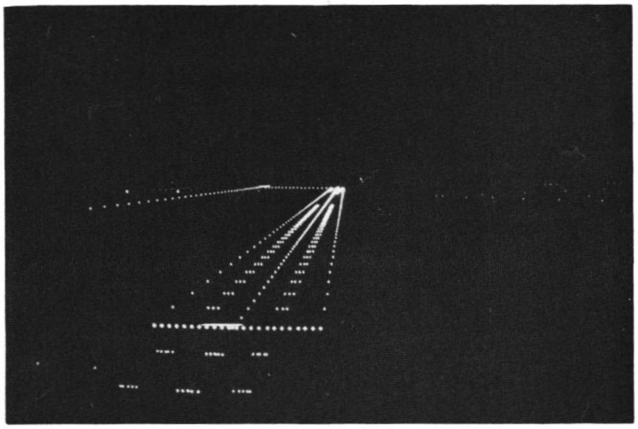


Fig. 4 Visual simulation of the outside world, e.g. the airport.

II. 3 Co-operative programs

Experience obtained in operating the moving-base visual flight simulator resulted in the subsequent co-operations of Delft University with partners outside the University :

- Assistance in the development of a four-degrees-of-freedom moving-base visual flight simulator for the National Aerospace Laboratory (NLR) in Amsterdam, operational since 1973[10].
- Advice on the selection of CGI-visual displays by KLM, Royal Dutch Airlines, for their flight training simulators in 1974.
- Improvement of both hardware and software of KLM's B-747 flight simulator in 1975[7], leading to a significant reduction of flight training in the real aircraft.
- Specification of the motion system hardware and software, and of the atmospheric turbulence model of two wide-body transport flight simulators presently on order by KLM.
- Participation in or support of several Working Groups of the Advisory Group for Aerospace Research and Development (AGARD) on different aspects of fidelity of flight simulation, including motion and vision.[1][11][12]

It would be outside the scope of the present paper to go into the details of all the subjects mentioned above. In the next section special emphasis will be placed only on the assessment of the dynamic characteristics of flight simulator motion systems.

III. Assessment of the dynamic characteristics of flight simulator motion systems

III. 1 Introduction

Since many years a lot of emotions are generated within the flight simulation community on the issue of the usefulness of flight simulator motion. Many evaluations of the addition of motion on different types of flight simulators have been made. In several of these the conclusion was reached, that the addition of motion to the simmulation exercise had no influence on pilot performance or even worse, e.g. "motion off is better than motion on".[13][14][15]

In this connection a relevant question seems to be : What were the dynamic characteristics of the motion systems used in these evaluations and how were the motion cues generated ?
Until recently very little information has been published on the actual dynamic characteristics of the motion system hardware as it is applied to existing flight simulators in military and civil aviation. Therefore the above conclusions appaer to be based on flight simulation experiments, where the dynamic characteristics of the motion systems used were not, or perhaps insufficiently established.

The subjective impression, that the dynamic characteristics of the Delft University motion system were different in some respects if compared to those of several other motion systems, lead to the development of a measurement method capable of determining the relevant dynamic characteristics of flight simulator motion systems. Such a method was initially developed by den Hollander[16]. Subsequent work by a Working Group established by the Flight Mechanics Panel (FMP) of AGARD resulted in a much improved and extented method[11]. The following deals with these efforts.

III. 2 Definitions

The dynamic characteristics of flight simulator motion systems can roughly be divided into two main parts :

- the "classical" characteristics, such as maximum travel and bandwidth of operation in each degree of freedom

- the "smoothness" of motion and the levels of interaction between the various degrees of freedom, giving an impression of the dynamic characteristics of the motions produced by the system.

Therefore the important dynamic motion system hardware characteristics, which determine the rather illusive notion of motion cue fidelity, were identified as :

- the maximum excursion, rate of displacement and acceleration in each degree of freedom,

- the motion system describing function for each degree of freedom, see Figure 5,

- the spectral power distribution and the RMS and peak-values of the acceleration noise and parasitic accelerations of the motion system,

where acceleration noise is defined as the perturbations from the nominal specific force and angular acceleration outputs of the motion system. Acceleration noise in the non stimulated degrees of freedom is called parasitic accelerations, e.g. pitch and/or roll accelerations due to heave motion, expressing the levels of interaction between the various degrees of freedom.

III. 3 Measurements

Two types of forcing functions were applied in the measurement procedures resulting in different levels of acceleration noise[16]:

1. The constant speed test signal, resulting in constant nominal specific force outputs (-1g), or zero nominal angular accelerations of the motion system, giving the possibility of a "quick look" at its acceleration noise.

2. Sinusoidal test signals, approximating real input signals to the motion system during flight simulation, where apart from the acceleration noise and parasitic accelerations, motion system control errors are measured and its describing functions and performance are determined.

Digital registration and analysis techniques are applied to meet the accuracy requirements necessary to quantitatively determine the above defined dynamic characteristics of the motion system.

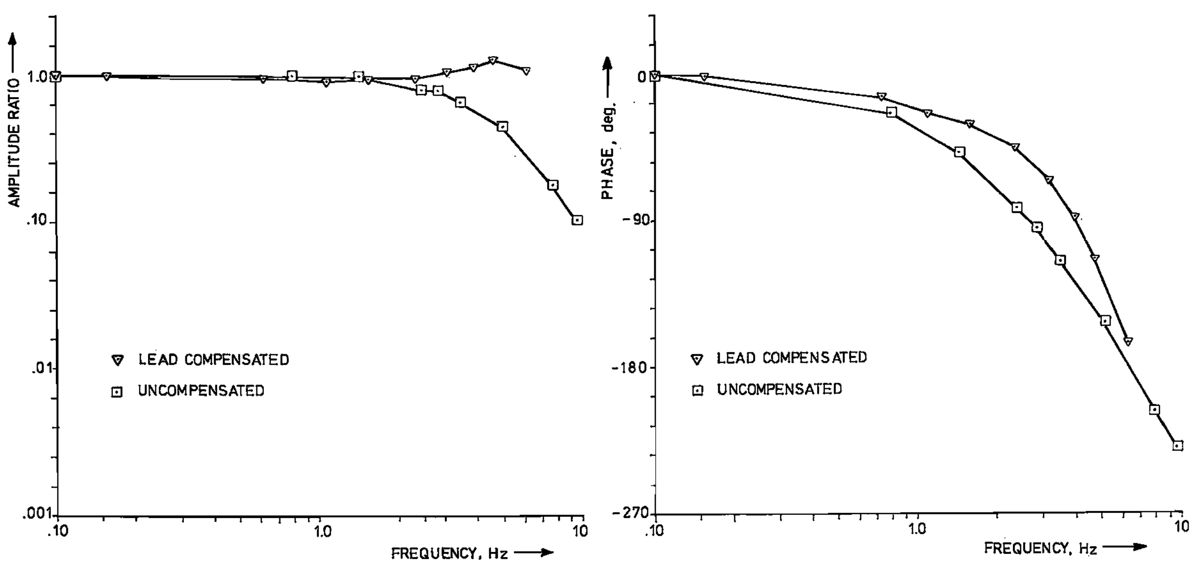


Fig. 5 Describing function for heave operation of DUT flight simulator (from[16]).

III. 4 Uniform measurement and analysis method

This work was continued in a contribution to the activities of a FMP-Working Group of AGARD in 1977 [11]. The aim of this Working Group was to establish a systematic and uniform method of measuring the dynamic characteristics of flight simulator motion systems. The expected benefits of such a measurement method are threefold :

1. Direct comparison of the dynamic characteristics of different kinds of motion systems, even in terms of smoothness, becomes possible, see Figure 6, where the performance diagrams for heave motion of two different motion systems can be directly compared.

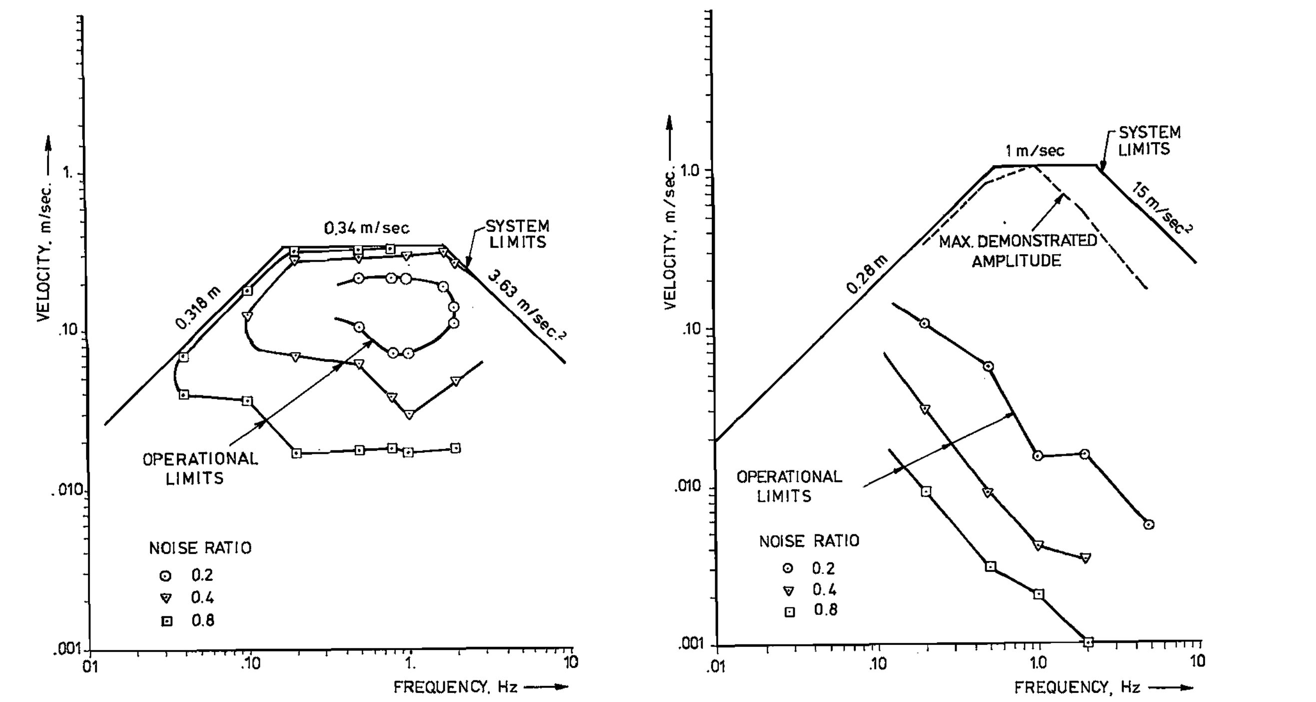
2. Diagnostic techniques become available to the engineer, designing and tuning motion systems or with some minor modifications, also control loading systems and T.V.-camera transport systems used in C.C.T.V.-model board visual displays.

3. A careful specification of motion system hardware, in particular the required dynamic characteristics, becomes possible.

The recommanded motion system characteristics to be measured according to the final report[11] of the FMP-Working Group are summarized below :

a. excursion limits for single degree of freedom operation, where the excursion limits are subdivided into system limits and operational limits, see Figure 6,
b. describing function for each degree of freedom,
c. linearity and acceleration noise,
d. hysteresis,
e. dynamic threshold.

The characteristics a, b, c, and d are measured by applying to the motion system a digitally generated sinusiodal input signal, standardized in frequency and amplitude.[11]
The measured output acceleration signals are, after Discrete Fourier Transform (DFT), partitioned into



SYSTEM AND OPERATIONAL LIMITS FOR HEAVE OF A POSITION CONTROLLED THREE-DEGREES-OF FREEDOM HYDRAULIC MOTION SYSTEM

SYSTEM AND OPERATIONAL LIMITS FOR HEAVE OF A POSITION CONTROLLED THREE-DEGREES-OF-FREEDOM HYDRAULIC MOTION SYSTEM WITH HYDROSTATIC BEARINGS

Fig. 6 Direct comparison of the performance of two different hydraulic motion systems (from[11]).

the components shown in Figure 7. In this manner the describing function (A), the low frequency (B) and high frequency nonlinearity (C), the acceleration noise (B, C, D) and the roughness (C,D) are identified using a limited set of measurement runs.

Measuring the dynamic threshold (e) results in an indication of the time delay introduced by the motion system hardware, see Figure 8. For hydraulic motion systems with hydrostatic bearings, correctly compensated for dynamic lag, this time delay is reduced to below 50 msec for acceleration step inputs larger than .01g.[12]

It was outside the Terms of Reference of the Working Group to make a judgment on the acceptability of the measured characteristics.

A logical next step in assessing the required dynamic characteristics of flight simulator motion systems, considering the dynamics of the airplane simulated and the task at hand, might be to set standards of acceptability for some or all of the characteristics defined.

In the next section the design of hydraulic motion system actuators will be discussed in detail. As a design example the actuator of a large synergistic six-degrees-of-freedom motion system, under current development at the Department of Mechanical Engineering, is dealt with.

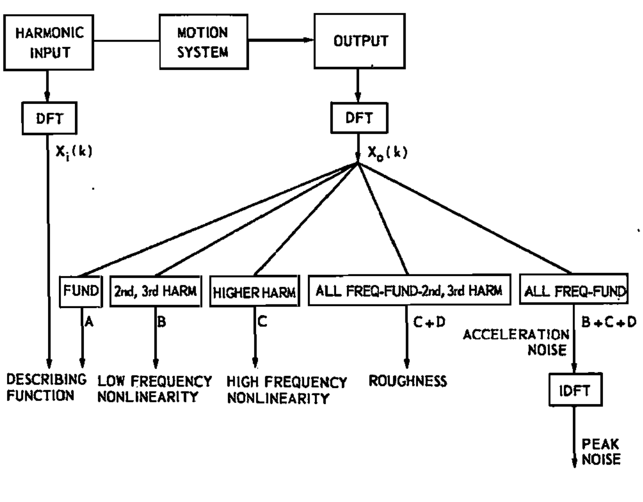


Fig. 7 Harmonic analysis methodology for flight simulator motion systems (from [11]).

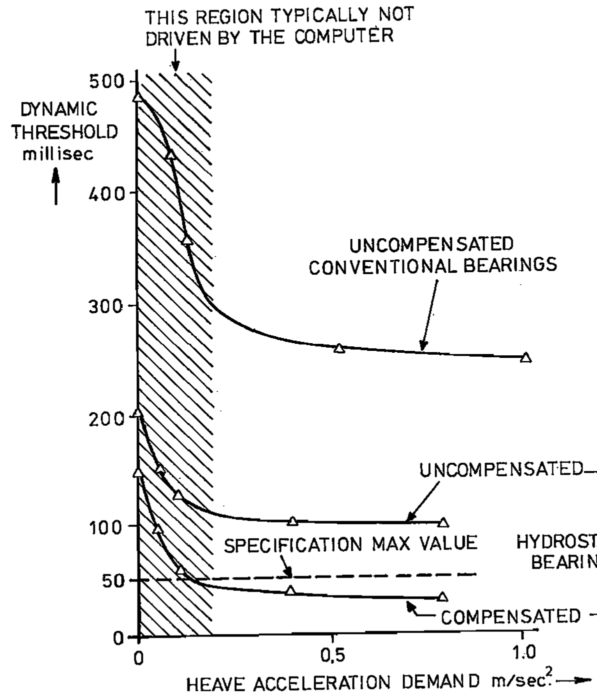


Fig. 8 Dynamic thresholds of flight simulator motion systems (from [12]).

IV. Developments and new designs in hydraulic actuators

IV. 1 Performance diagram and design of actuators

There is a direct relation between the specification of flight simulator motions and the hydraulic actuators with corresponding servovalves and oil supply pressure. This will be illustrated by the performance diagram of one single symmetric servosystem actuated sinusoidally with an amplitude $\hat{Y}$.

Specification of the actuated mass M, the stationary load F_{st} (for example due to gravity), the maximum stroke $S = 2\ Y_{max}$, velocity $\dot{Y}_{max}$, acceleration $\ddot{Y}_{max}$ and the bandwidth ω_b must and will determine the following design parameters :

- . piston area A (in m^2),
- . servovalve capacity Q_{max} (in m^3/s),
- . oil supply pressure P_s (in N/m^2).

In a well designed and controlled symmetric servo the bandwidth ω_b approaches very well the natural frequency ω_0 of the hydraulic system, determined by the actuated mass M and the stiffnes $C_0 = M\,\omega_0^2$ of the oil colums on both sides of the piston[17]. Therefore :

$$\omega_b \approx \omega_o = \sqrt{\frac{C_o}{M}}, \quad C_{o\,min} = \frac{4\,AE}{S} \qquad (1)$$

Here S is the maximum stroke, A the piston area and E the bulk modulus of the oil. Consequently :

$$\hat{Y} = \frac{S}{2} = \frac{2\,AE}{C_{o\,min}} = \frac{2AE}{M\,\omega_o^2} \approx \frac{2\,AE}{M\,\omega_b^2} \qquad (2a)$$

or:

$$A = \frac{M\,\omega_b^2\,\hat{Y}}{2\,E} = \frac{M\,\omega_b^2\,S}{4\,E} \qquad (2b)$$

So specifying S, M, E and ω_b the piston area A of a symmetric actuator is determined. Logically, a compromise between the specifications and the design parameter A must be made.

At very low frequencies the amplitude $\hat{Y}$ according to eqn. (2a) could be effected, but at higher frequencies the servovalve capacity Q_{max} will limit the amplitude. With $Q_{max} = A\,\hat{\dot{Y}}_{max} = A\,\omega\,\hat{Y}_{max}$ we find :

$$\hat{Y} \leq \frac{\hat{\dot{Y}}_{max}}{\omega} = \frac{Q_{max}}{\omega\,A} \qquad (3a)$$

Combining eqns. (2b) and (3a) gives :

$$Q_{max} = \frac{M\,\omega_b^2\,S\,\hat{\dot{Y}}_{max}}{4\,E} \qquad (3b)$$

Again the specifications lead to a vital design parameter to be selected : the valve capacity Q_{max}.

The frequency range in which eqn. (3a) is valid is limited to the frequency where the hydraulic system becomes overloaded. Dominant components of the load are the inertia load $M\,\ddot{Y}$ and the stationary load F_{st}.

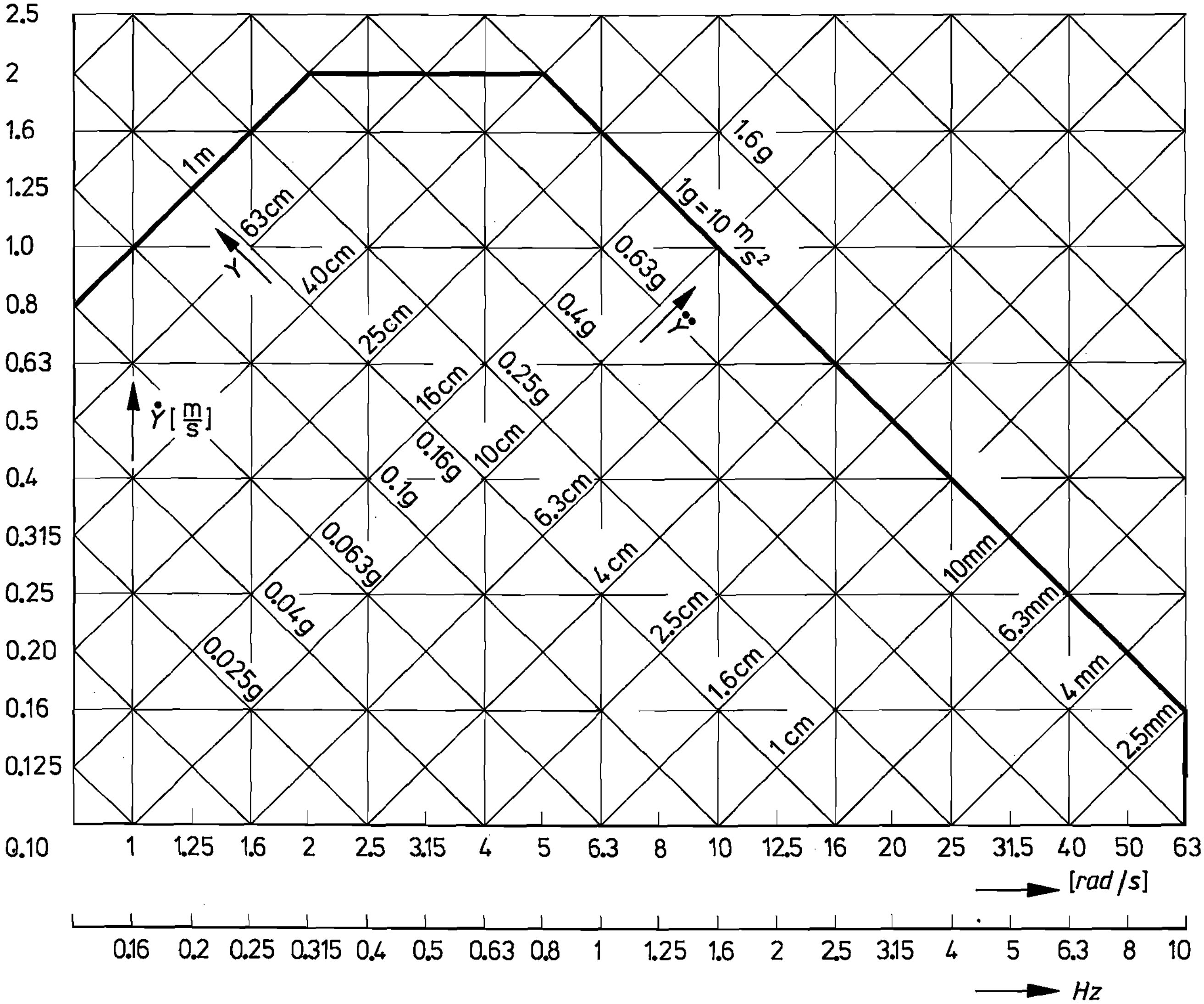


Fig. 9 Performance diagram.

The so-called load pressure

$$P_L = \frac{M\ddot{Y} + F_{st}}{A} = \frac{M\omega^2 \hat{Y} + F_{st}}{A}$$

should stay well below the hydraulic supply pressure P_S in order to avoid cavitation etc., so we choose $P_L \leq \frac{2}{3} P_S$.
Thus we find :

$$\hat{Y} \leq \frac{\hat{\ddot{Y}}_{max}}{\omega^2} = \frac{\frac{2}{3} P_S A - F_{st}}{M\omega^2} \qquad (4a)$$

Combination with eqn. (2b) finally yields

$$P_s = 6\,E\,\frac{M\,\hat{\ddot{Y}}_{max} + F_{st}}{M\,\omega_b^2\,S} \qquad (4b)$$

Equations (2a), (3a) and (4a), stemming directly from the specifications, together constitute the complete performance diagram and eqns. (2b), (3b) and (4b) directly show the "consequences" A, P_s and Q_{max} of the specifications.

In Figure 9 an example is worked out. At M = 1500 kg , F_{st} = Mg = 15000 N, E = 10^9 N/m^2, ω_b = 10 Hz = 63 rad/s, S = 2 m, $\hat{\dot{Y}}_{max}$ = 2 m/s and $\hat{\ddot{Y}}_{max}$ = 10 m/s^2 = 1 g we find :
A = 3.10^{-3}m^2 = 30 cm^2, Q_{max} = 6.10^{-3}m^3/s = 6 dm^3/s and P_s = 15.10^6 N/m^2 = 150 bar. The "peak" supply power is $N_s = P_s \cdot Q_{max}$ = 90 kW.

This (extreme) example should make very clear that there should be a well-coordinated give and take between specifications and design parameters to be accepted.
Equation (2b), for example, shows that large excursions (S) of the actuator and a large bandwidth (ω_b) are more or less controversial. An "optimal" choice should be based on fundamental knowledge of man-machine systems. However, relevant knowledge in this field is scarcely present.

To apply the above results to flight simulators, first of all the asymmetry of the hydraulic actuators should be taken into account. Straightforward derivations for asymmetric actuators have been publisched by Viersma(17), resulting in formulas, essentially having the same nature as eqns. (1) - (4).

A multi-axes motion system essentially is a highly nonlinear multi-variable control system

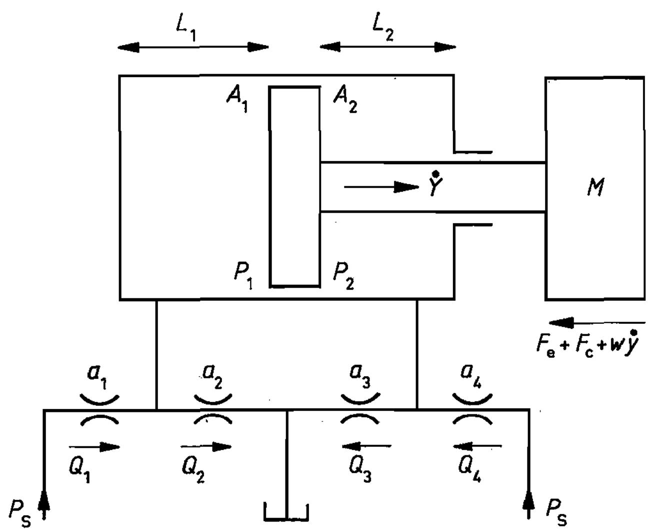


Fig. 10 Asymmetric hydraulic actuator with a critical center valve (eiter $a_1 = a_2 = 0$ or $a_2 = a_4 = 0$).

which could be extremely complicated by interaction, compliance etc. In many cases interaction could be reduced effectively by a proper design, enabling a rough single-axis-of-freedom analysis. Essential is the mass M to be taken into account. In simulators with six degrees of freedom (and six actuators) some of the actuators could eventually face a "virtual" mass equal to one third (!) of the simulator mass, enabling the analysis, synthesis and design procedure of this section.

IV. 2 Servovalve asymmetry

A moving-base flight simulator is actuated mostly by hydraulic jacks with one piston rod only, meaning the actuator is asymmetric : $A_1 \neq A_2$, see Figure 10. For a smooth control it is a must to adapt the asymmetry of the servovalve to the asymmetry of the actuator. To prove this we assume a *critical-center* valve, having either (at a positive velocity $\dot{Y}$) $a_2 = a_4 = 0$ or (at $\dot{Y} < 0$) $a_1 = a_3 = 0$ (see Figure 10). Assuming perfect turbulent port flow and neglecting for a moment the compressibility of the oil (i.e. E = ∞), we get

$$\text{either} \quad \dot{Y} = \frac{a_1}{A_1} C_d \sqrt{2\,\frac{P_s - P_1}{\rho}} = \frac{a_3}{A_2} \cdot C_d \sqrt{2\,\frac{P_2}{\rho}} > 0 \qquad (5a)$$

$$\text{or} \quad \dot{Y} = -\frac{a_2}{A_1} C_d \sqrt{2\,\frac{P_1}{\rho}} = -\frac{a_4}{A_2} C_d \sqrt{2\,\frac{P_s - P_2}{\rho}} < 0 \qquad (5b)$$

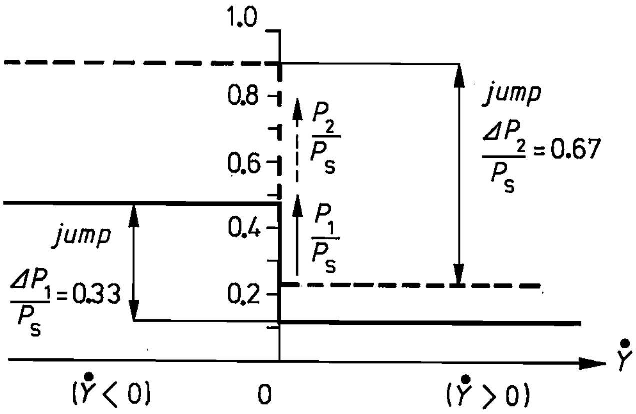


Fig. 11 Pressure jumps in asymmetric actuators ($A_1/A_2 = 2$) controlled by symmetric servovalves ($a_1 = a_3$ and $a_2 = a_4$).

In both cases ($\dot{Y} > 0$ and $\dot{Y} < 0$) the pressures P_1 and P_2 on both sides of the piston can be solved directly by using

$$P_1 A_1 - P_2 A_2 = \Sigma F \qquad (6)$$

Here Σ F is the "total" load, including inertial, frictional and external load. Solving P_i and P_2 from the eqns. (5) and (6) yields the very surprising result that P_1 (as well as P_2) jumps when the velocity reverses ($\dot{Y} > 0 \Rightarrow \dot{Y} < 0$) even if the total load Σ F is constant ! Combination of eqns. (5) and (6) with ΣF = constant and assuming :

a_1/a_3 (at $\dot{Y} > 0$) = a_2/a_4 (at $\dot{Y} < 0$) yields :

$$\frac{\Delta P_1}{P_s} = \frac{P_1\ (\dot{Y} > 0) - P_1\ (\dot{Y} < 0)}{P_s} = \frac{1 - (\frac{A_1}{A_2})^2 (\frac{a_3}{a_1})^2}{1 + (\frac{A_1}{A_2})^3 (\frac{a_3}{a_1})^2} \qquad (7a)$$

$$\frac{\Delta P_2}{P_s} = \frac{P_2\ (\dot{Y} > 0) - P_2\ (\dot{Y} < 0)}{P_s} = \frac{A_1}{A_2} \cdot \frac{\Delta P_1}{P_s} \qquad (7b)$$

In Figure 11 an example for a symmetric valve (i.e. $a_1 = a_3$ and $a_2 = a_4$) in combination with an asymmetric actuator ($A_1 = 2\ A_2$) is presented, showing pressure jumps up to Δ P = 0.67 P_s!!

In practice real pressure jumps will not occur thanks to the compressibility of the oil (a pressure jump in a given volume with a compressible medium is implying an infinitely large oil flow which, of course is impossible). So the phenomenon of pressure jumps will be smoothed out somewhat. Nevertheless, sudden changes in flows and pressures will occur while reversing the velocity, implying considerable disturbances in the acceleration.

To avoid this phenomenon the servovalve should be "tuned" to the actuator. Equation (7) shows that pressure jumps can be avoided only by the following condition :

$$\frac{a_1}{a_3} = \frac{A_1}{A_2} = \frac{a_2}{a_4} \qquad (8)$$

So the inlet opening a_1 of one cylinder compartment should be in area ratio proportional (A_1/A_2) to the outlet opening a_3 of the other compartment. The same holds for a_2 and a_4. In other words : the servovalve should be asymmetric, tuned to the asymmetry of the piston. Those valves nowadays become available (on special request), being somewhat more expensive of course. But these extra costs imply a far better performance!

IV. 3 Disturbances due to Coulomb friction.

Possibly the most important factor disturbing the smooth operation of flight simulators is Coulomb friction being present inside the actuators. Coulomb friction in actuators sometimes far exceeds 10% of the maximum load capacity. However, reliable data and evidence is lacking completely.

Yet, already in the 1950's there were publications of Sweeney[18] and others concerning "hydraulic lock", indicating how serious Coulomb friction can spoil the system performance, introducing dead zones[17], slip-stick-motions or even complete halting of motions. Its message is containing the extremely serious warning that classical bearings and sealings at pistons and piston rods provide Coulomb friction which is not negligible at all.

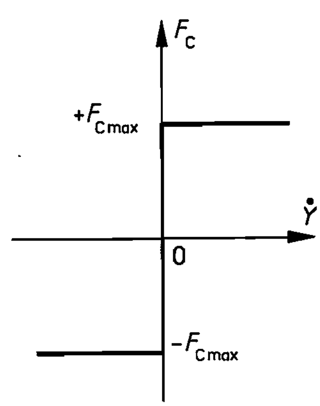


Fig. 12 Simplified model of Coulomb friction.

How Coulomb friction affects motion quality, especially when the piston is reversing its velocity, becomes clear from Figure 12 where the most "primitive" model of Coulomb friction is presented. A smooth reversing of the velocity $\dot{Y}$ implies a sudden change in the frictional force :

$$\Delta F_c = 2 F_{c\,max}.$$

Regardless of the control system and the presence of any feedback this frictional jump ΔF_c will cause a sudden change in the acceleration $\ddot{Y}$ of the actuated mass M :

$$M \Delta \ddot{Y} = \Delta F_c \qquad (9)$$

So :
$$|\Delta \ddot{Y}| = \frac{|\Delta F_c|}{M} = \frac{2 F_{c\,max}}{M} \qquad (10)$$

If for example, $F_{c\,max} \approx 0.1$ Mg (which is by no means an exceptional assumption) then :

$$\Delta \ddot{Y} = 0.2 \text{ g}$$

At a human percenption level of about 0.02 g this is enormous. Consequently the so-called reversal bump could be noticed in any flight simulator, disturbing realistic simulations considerably.

But also at very low and pseudo-constant velocities sudden changes in Coulomb friction (due to minimal changes in cylinder cross-section from place to place) might cause disturbances

$$\Delta \ddot{Y} < \frac{F_{c\,max}}{M} \qquad (11)$$

A radical remedy for these disturbances is provided by the introduction of hydrostatic bearings for pistons and piston rods, eliminating the Coulomb friction almost completely.

IV. 4 Hydrostatic bearings in hydraulic actuators.

To avoid metal-to-metal contact and Coulomb friction between surfaces moving relative to each other (like a piston in a cylinder or a piston rod in its bearing) hydrostatic bearings should guarantee a permanent oil film between the two sliding surfaces. The bearing should introduce a *centering force* more or less proportional to the eccentricity (of the piston or the piston rod) to compensate for eventual side loads. As a source of hydraulic energy or pressure the hydraulic supply pressure and/or the cylinder pressures can be exploited, avoiding any extra energy supply device at all.

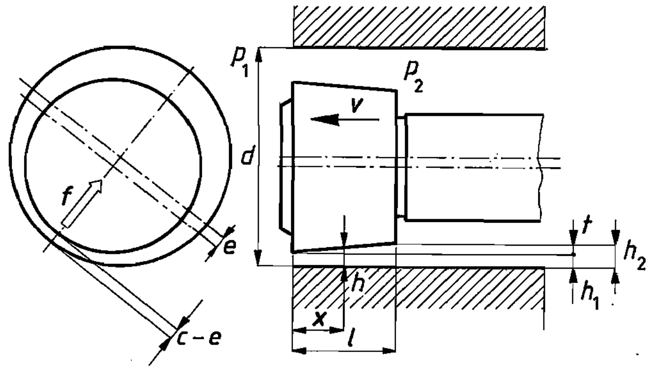


Fig. 13 Dimensions of a tapered piston. In the case $P_2 > P_1$ the conical gap is acting as a hydrostatic bearing providing a centering force f increasing with the eccentricity e.

Hydrostatic bearings can be made, in principle, by giving the piston itself (or the piston rod bearing) a small taper in such a way that at the wide end of the gap the hydraulic pressure is larger than at the narrow end, see Figure 13. Based on the assumptions

- pure laminar flow,
- no circumferential flow (i.e. axial flow only),
- no relative velocity between piston and cylinder,

Blackburn(19) gave already in 1960 :

$$\frac{f}{d\,l\,\Delta P} = \frac{\pi}{4} \cdot \frac{t}{e}\left[\frac{t + 2C}{\sqrt{(t + 2C)^2 - 4e^2}} - 1\right] \qquad (12)$$

Here f is the centering force, d and l the diameter and length of the bearing respectively, $\Delta P = P_2 - P_1$ the pressure drop across the bearing, e the eccentricity and t the taper of the piston, and C the clearance at the narrow end of the gap at e = 0, see Figure 13.

Obviously there is an optimal taper ratio t/C. Assuming a small eccentricity ratio e/C, the optimum becomes t = 2C. Thus we find

$$\lim_{\substack{e \Rightarrow 0 \\ t \Rightarrow 2C}} \frac{f}{d\,l\,\Delta P} = \frac{\pi}{2} \cdot \frac{t\,e}{(t + 2C)^2} = \frac{\pi}{16} \cdot \frac{e}{C} \qquad (13)$$

Equations (12) and (13) at least give a too optimistic idea about the side load stiffness and capacity of a hydrostatic bearing.

Elaborating the basic principle, however, reliable data could be obtained only by taking the circumferential flow as well as the relative velocity into account. Observing a maximum eccentricity e = 0.75 C, Blok [20] came to the following combination of maximum side load and velocity :

$$\frac{F}{\bar{F}} + \frac{V}{\bar{V}} = 1 \qquad (14)$$

where :

$$F = \frac{4\,f}{\pi\, d^2\, \Delta P}\ , \quad V = \frac{\mu\, d\, v}{c^2\, \Delta P}$$

Values of $\bar{F}$ and $\bar{V}$ ($\approx$ 0.15 T/L) are presented in Table 1 as a function of the design parameters L = l/d and T = t/C at E = e/C = 0.75.

L	T	$\bar{F}$	$\bar{V}$
0.5	1	0.090	0.352
	2	0.095	0.622
	3	0.089	0.900
1.0	1	0.129	0.182
	2	0.149	0.317
	3	0.146	0.454
1.5	1	0.133	0.120
	2	0.165	0.210
	3	0.170	0.300

Table 1. Data for the design of single hydrostatic bearings, using eqn. (14).

The axial leakage flow in an apropriate designed hydrostatic bearing normally can be neglected completely with respect to the maximum servovalve flow.

The irony of history can be illustrated by the fact that originally eqn. (12) was derived to explain the phenomenon of "hydraulic lock" based on a de-centering force, occurring at $P_1 > P_2$ in Figure 13 and causing e = C. Fundamental understanding of the nature of the phenomenon ultimately lead to the succesfull development and applications of hydrostatic bearings (by making $P_2 > P_1$!) at Delft University of Technology in the late sixties, spreading around the world in the late seventies.

Introducing hydrostatic bearings in normal hydraulic double acting actuators, there are three basic problems to be solved :

a - the pressure drop across the tapered pistons and bearings must be always "positive", introducing serious problems with double acting rams,

b - the velocity (in one direction) limits the load bearing capacity drastically,

c - the design and manufacturing.

In Figure 14a and 14b the solution to the problems (a) and (b) are shown : introducing the so-called double-bearings both the problem of the pressure drop across the piston as well as the problem of the velocity will be solved. In principle, the two bearing halves together compensate for the influence of the velocity completely. Detailed derivations and instructions for design and manufacturing are given by Viersma [17] and many successful applications in various fields have been published by Blok [20].

V. Concluding remarks

Experience in the development and application of the Delft University flight simulator has shown that, for effective, high fidelity flight simulation, all the cues presented to the pilot must be of a consistently high quality and, particularly, must be free from roughness and unnatural lags. Especially the availability of the present motion system with its outstanding dynamic characteristics, made research in the area of motion simulation possible. The motion system could be obtained

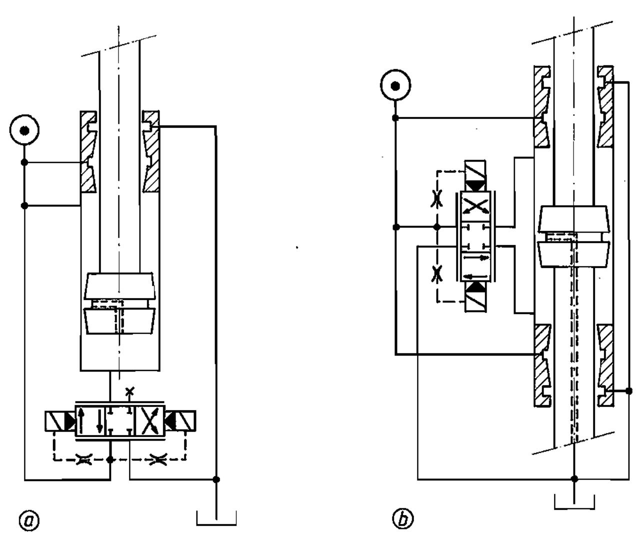


Fig. 14 Hydraulic actuators controlled by three-way (a) and four-way (b) servovalves, respectively. Pistons and piston rods both have double-bearings.

as a result of the close co-operation between two departments of the university, each with its own contribution, interest and background knowledge, including a very essential participation of their students. The research is supported by the University Board providing both adequate funding and manpower. This resulted in continuity over many years.

After the introduction of flight simulators with hydraulic motion and control loading systems provided with hydrostatic bearings at Delft University and at the National Aerospace Laboratory (NLR) in Amsterdam, the interest in such hydraulic systems gradually became international.
Simulator operators such as KLM, Fokker, Lufthansa and Boeing stimulated flight simulator manufacturers in Great Britain, Canada and France to meet higher requirements regarding flight simulator motion and control loading performance based on hydrostatic bearings. The result has been an innovative breakthrough in this very specialized field of application.

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