VALVES AND ACTUATORS FOR HYDRAULICALLY POWERED PROSTHESES

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Summary

There are several potential advantages in using hydraulically powered arms in comparison with existing pneumatic systems. These include improved system response and controllability, stability and stiffness together with increased safety. Recent successful development at University College London of a prototype miniature hydraulic power unit indicates that portable hydraulic systems are now a viable proposition. One of the remaining major difficulties associated with the implementation of completely hydraulically powered prostheses is the development of a suitable control valve.

Current work at University College London is concentrated on developing micro-miniature hydraulic servo valves capable of controlling small fluid flows whilst operating at the low pressures found in prosthetic systems. To conserve energy such valves must have negligible leakage at null and no external leakage. Operating forces also need to be very small.

The paper describes the development of a valve satisfying these requirements, and experimental results of the laboratory tests of a valve actuator system will be given.

A simple nomogram has been developed to enable specifications of actuator stroke and area, and linkage ratio from the requirements of range of movement, load capacity and maximum velocity. The nomogram also enables variations in system pressure and maximum flow through the valve to be assessed.

Complete design of the valve actuator system must also include assessment of the effects of flexibly coupled loads, and large inertial forces both of which will be present in prosthetic systems; and must consider the clinical implications. These aspects of the application of servo hydraulics to prostheses are discussed.

Introduction

The multi-functional externally powered arm prostheses currently available in the U.K., suitable for fitting to children suffering from bilateral dysmelia, are all powered by compressed carbon dioxide gas /1/. Arm articulation is accomplished by using three-way control valves in association with differential area piston actuators, or two three-way valves mounted back to back with equal area double acting actuators; in closed loop position controlled servomechanisms.

Several advantages are associated with the use of hydraulic fluid as the powering media for such devices /2,3/. These mainly accrue from the relative compressibility of the two media. The virtually incompressible nature of hydraulic fluid results in safe operation of systems at higher working pressures and consequently results in better power to weight ratios. Thus hydraulically powered position control systems, designed to conform to the prosthetic requirements of safety, compactness and lightness, are able to meet load, response and stability specifications which exceed the capabilities of their pneumatic counterparts. The output stiffness of such hydraulic servos is also greatly increased.

The construction of a hydraulically powered arm prosthesis has not been possible to date due to the lack of a suitable power source. However, recent successful development of a prototype miniature hydraulic power unit indicates that portable hydraulically powered systems are viable /4,5/. Energy storage for this hydraulic supply is accomplished by electric batteries which offers great advantages in ease of recharging or replacement compared with compressed carbon dioxide storage cylinders.

A remaining area of difficulty in implementing a hydraulic arm prosthesis was the development of a suitable control valve. The paper describes the requirements of such a valve and the development and testing of a successful prototype.

Control Valve Development

Specification

The requirements of a control valve for use in a hydraulic servomechanism for prostheses can be enumerated.

1. The valve must be a four way device as it is desirable to use equal area actuators to obtain the same load capacity, response, stability and saturation velocities in both directions of motion.
2. It is to operate at working pressure of approximately 30 bar which are compatible with the best efficiency of energy conversion obtainable from the portable power supply to be used.
3. The maximum flow through the valve, which will dictate the saturation velocity in the actuator and hence the maximum angular velocity of the particular arm function, is to be of the order of 0.6 litre/min. This corresponds to an elbow flexion/extension velocity of 5 rad/s with a

load torque capability of about 6 Nm, or 3 rad/s with 10 Nm available; depending on the specific actuator and linkage arrangement used. Thus the valve is required to provide proportional control of flows of hydraulic fluid over the range 0 - 0.6 litre/min.

4. The valve must be as small and as light as possible to enable use in a prosthetic system where size and weight are of primary importance.
5. The operating forces required to control the valve position and hence to operate a particular arm function must be as low as possible so that a suitable interface with the patient may be effected.
6. Finally the valve must have no null leakage of hydraulic fluid and a minimum internal leakage between supply and drain, as these leakages constitute energy losses from the system. Also the valve must be totally sealed from its environment to eliminate hydraulic fluid loss.

Several valve systems were investigated to satisfy the requirements. None were commercially available; in fact the smallest commercially available electrohydraulic valve, the Series 30 Moog, has a quiescent power drain which exceeds the continuous rates supply from the portable pump.

Description

The prototype valve which has been developed is a miniaturised version of the conventional four way hydraulic valve. It is shown schematically in Figure 1, and a photograph of the valve, dismantled to show its various components, is shown in Figure 2. The valve is approximately 22 mm long by 12 mm diameter; and the spool is approximately 3.2 mm diameter. The spool was made from a hardened steel needle roller, and the required profile ground to achieve an overlap of about 12 μ to eliminate leakage in the null position. The drain lands were made over twice the width of the drain port to reduce leakage and its associated energy loss. The spool was made so as not to obscure the load ports at full opening in either direction. The valve body was manufactured from Aluminium Bronze which provides a low friction sliding contact with the spool and also overcomes problems of differential thermal expansion during valve operation. The body was bored and lapped to give a 5 μ diametral clearance for the spool, thus minimising leakage across the spool lands and achieving low operating forces. The valve ports were drilled and reamed approximately 0.8 mm diameter and positioned in the body so that symmetrical lap to supply and exhaust was obtained. The ports were also positioned symmetrically around the circumference of the valve to reduce radial dynamic pressure unbalance forces on the spool, which would increase the operating force requirement. Leak paths back to drain were provided in the body to prevent hydraulic locks forming at the spool ends. The end caps were manufactured of aluminium alloy and designed to incorporate physical stops for the spool at full opening in either direction. They were machined to take standard O-ring seals to prevent leakage of hydraulic fluid to the environment. The spool shaft is itself sealed to prevent external leakage. Special screw in connectors were used to ensure reliable operation at the working pressures.

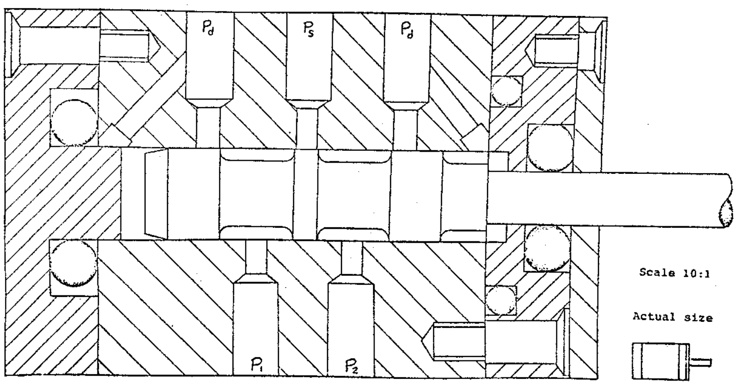


Fig. 1. Schematic diagram of miniature hydraulic spool valve

The main problems associated with making the valve are in accurately positioning the ports in the body to achieve the required lap conditions. This is fairly critical as too small a lap will result in leaks and too large a lap introduces a dead zone and hence non-correlation between input and output of the position servo; the lap must also be symmetrical to avoid hydraulic locks forming in the system. Work is in progress on a modified version of the valve which uses a laminar construction principle and this should obviate the port location difficulties.

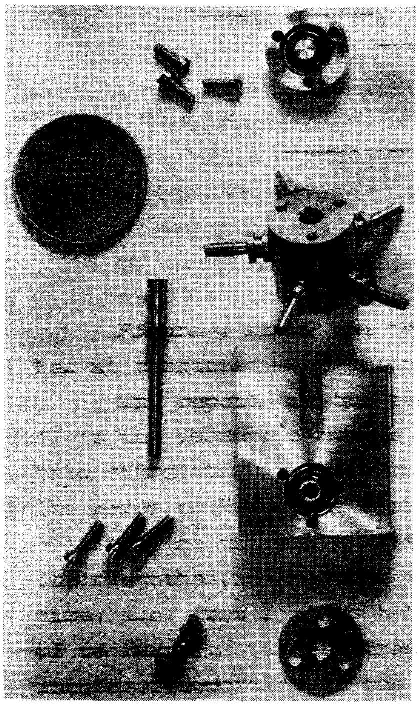


Fig. 2. Photograph of valve showing components

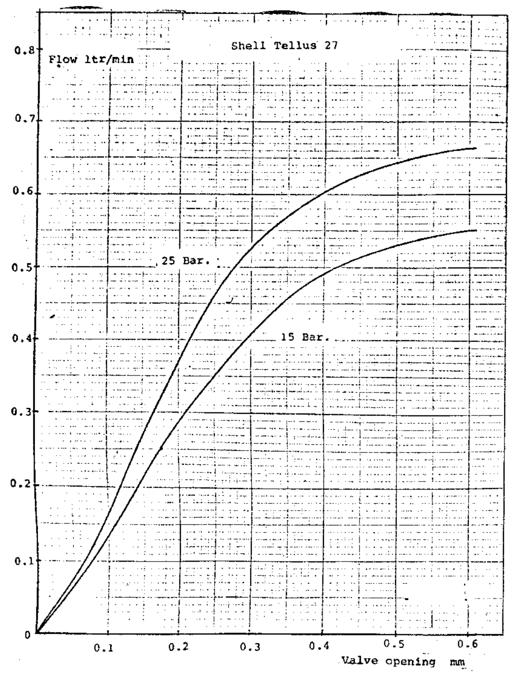


Fig. 3. Flow/displacement valve characteristic

Experimental Evaluation

Flow Tests

Flow tests were conducted on the valve at supply pressures of 15 bar and 25 bar. The valve spool displacement was measured by a micrometer, and the flow of hydraulic fluid from one of the load ports was measured using a rotameter float type flow meter. A graph of valve opening in mm against fluid flow rate in litre/min is shown as Figure 3. It shows the typical choked flow curve characteristic of spool valves having a considerable linear range of almost 50% of the operating movement at the higher pressure, together with a low valve flow gain; both of which are desirable for good controllability of the servo. The design flow requirement of

0.6 litre/min is obtained, and the valve flow gain, normalised for pressure variations, of 0.4 $\frac{\text{litre/min}}{\text{mm}\sqrt{\text{KN/m}^2}}$ is in agreement with the theoretical design prediction.

General valve characteristics which were observed during these and subsequent tests were that the leakage to drain was less than 1 cc/min, that is less than 0.2 per cent of the full rates flow. So with five valves in a multi-functional arm, supplied by a pump continuously rated at 0.25 litre/min, there would be an energy loss of the order of 2 per cent. There was no external leakage observed throughout testing, either through the end cap seals, spool shaft seal or from the connectors. The valve was easy to move from one position to another showing very low stiction effects. The operating forces were less than 1N, but these are mainly due to friction in the shaft seal and this has not yet been optimised. Reducing the seal squeeze will reduce operating friction forces whilst maintaining satisfactory sealing of the valve. Also because of the sensitivity of the valve, the valve input movements will probably have to be geared up to the patient, which will result in greater forces being available to operate the valve.

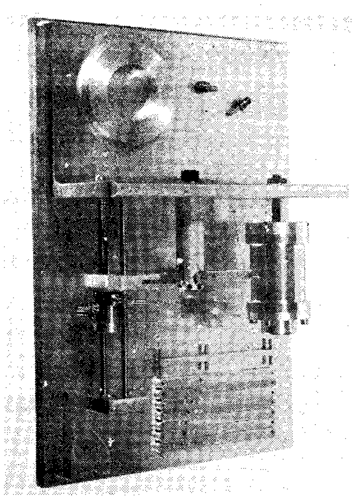


Fig. 4.

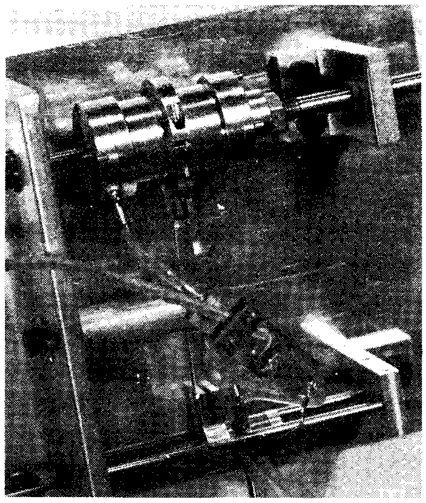


Fig. 5.

No-Load Tests

A rig was constructed to enable testing of the valve as a component of a mechanically signalled hydraulic position servo and photographs of this apparatus are shown as Figure 4 and 5.

The valve was mounted in linear bearings to enable it to move horizontally, and the actuator was mounted such that it would execute parallel motions, the piston of the actuator being fixed by its rod at one end and the actuator body being free to move through a fluoroscint bearing. For ease of construction the piston is not in fact of equal area. There is a 5 per cent difference in the area of the two sides due to the area of the piston rod. The motion of the actuator is fed back to the valve body via a mechanical lever arrangement providing closed loop control, the feedback lever ratio can be varied from 1 to 5 up to 5 to 1 simply by using the other levers shown in the photograph. This enables the gain of the servo to be altered and the effects of these gain changes can be studied.

A schematic diagram of the system is shown in Figure 6 and the operation of the servo can be described by reference to this. If the valve spool is moved in the x positive direction this will open the actuator chamber A to supply and allow chamber B to exhaust. The pressure differential will cause the actuator body to move in the y + ve direction and as it does so this motion is transmitted back to the valve to shut off. Thus a new stable position is reached with the output position an exact correlation with that demanded at input. Similarly, spool displacements in the y negative direction will be reproduced at the output.

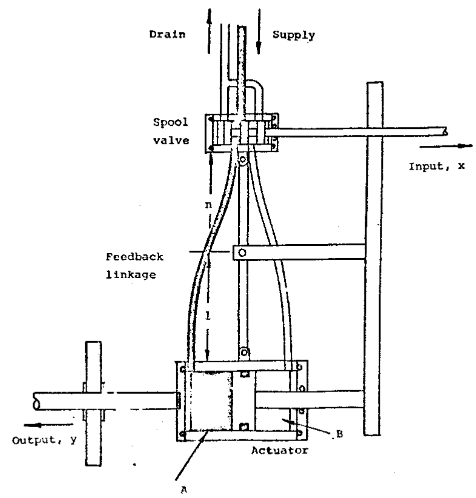
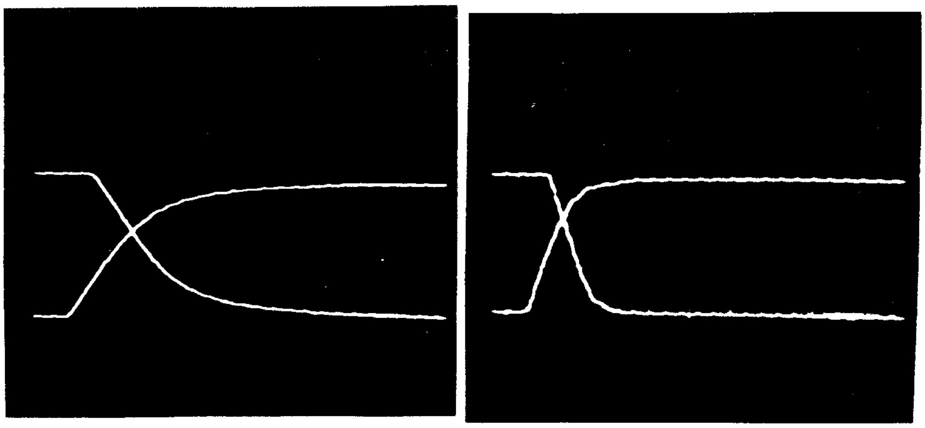


Fig. 6. Schematic diagram of mechanically signalled hydraulic position servomechanism

A series of small signal step tests were conducted to observe the transient response of the system. The valve spool was moved using a solenoid actuation system through its full operating range of 0.75 mm; the solenoid being switched by an oscillator controlled relay so that continuous testing at various frequencies was possible. Figure 7 shows the response, i.e. actuator displacement against time, of the servo at various feedback link ratios. It can be seen that the servo is extremely stable, showing no tendency to oscillate and responds in an exponential manner, the rise time of the exponential curve can be taken as a measure of its performance. Figure 8 shows the variation of 60% rise time against feedback linkage ratio n, where n is as specified in the schematic diagram (Fig. 6). It can be seen that the time constant reduces from 80 ms at an n of 0.33 to 20 ms at an n of 3.0. However there will naturally be a loss in system stability corresponding to the performance improvements although this cannot be seen in the case of the unloaded system tested. It appears that a feedback ratio of unity may well give the best compromise between response and stability and here the rise time was 35 ms.



Feedback link N = 0.33	Feedback link N = 1.0
Time base = 50 ms/Div	Time base = 50 ms/Div
Displacement = 0.72 min/Div	Displacement = 0.24 min/Div
60% rise time = 80 ms	60% rise time = 35 ms

Fig. 7. Response to 0.75 mm step inputs

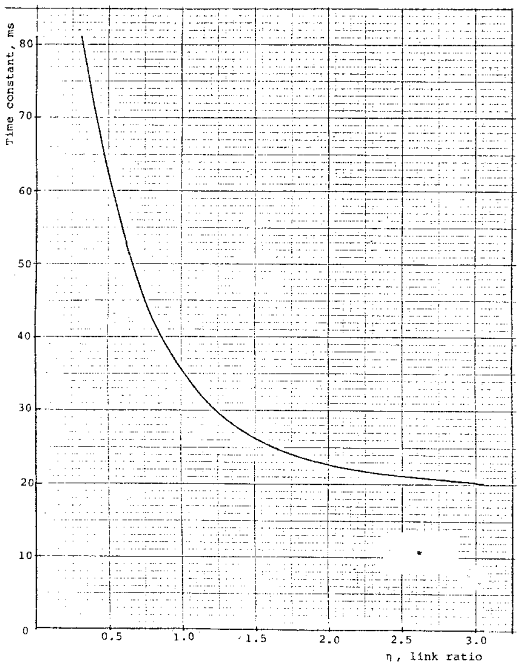


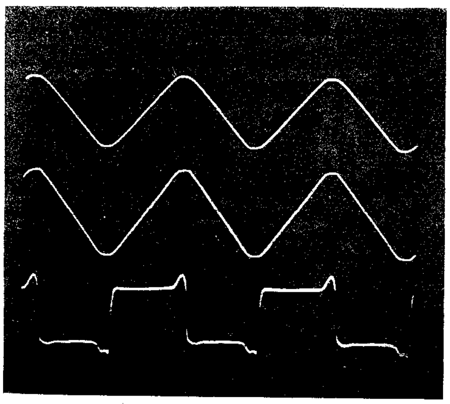
Fig. 8. Variation of 60 per cent rise time with feedback linkage ratio

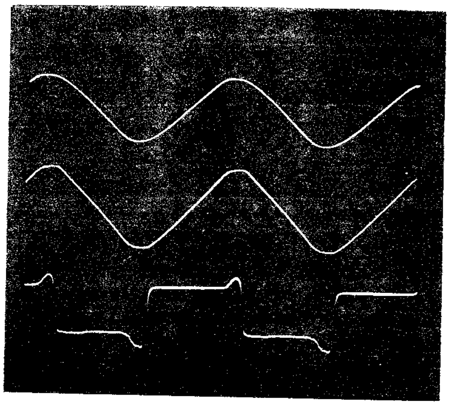
Pushing the valve fully open and allowing the actuator to cover full stroke enabled the saturation velocity of the system to be measured as 20.5 mm/s in the y positive direction and 19.5 mm/s in the y negative direction; the 5 per cent difference being entirely due to the area differential.

The output stiffness of the system was very high as is expected with hydraulic servos. With a working pressure of 30 bar the actuator developed a stall force of about 1 kN.

Several other test results were obtained for the system at unity feedback ratio when excited with ramp and harmonic inputs. For these tests an electro-mechanical closed loop position servo was used to drive the valve spool with whatever waveform was required. Typical results of the ramp and harmonic tests are shown in Figure 9, in each case the top trace shows the input waveform, the centre trace shows the output response and the bottom trace shows the pressure fluctuations in the small area side of the actuator. Excellent correlation between input and output is obtained over the normal range of operating frequencies.

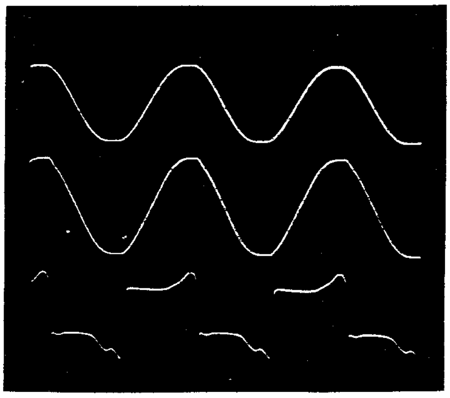
Unity feedback link ratio
Input 2.4 mm/DIV
Output 2.1 mm/DIV
Pressure 3.5 mm/DIV

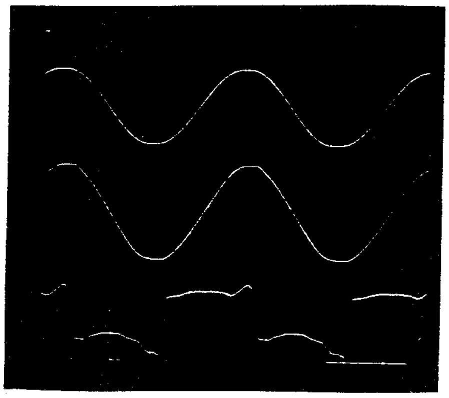




Ramp inputs
0.5 Hz
0.5 s/DIV

1.0 Hz
0.2 s/DIV





Harmonic inputs
0.5 Hz
0.5 s/DIV

1.0 Hz
0.2 s/DIV

Fig. 9. Response to ramp and harmonic inputs

Load Tests

The destabilising effect of loads on servo systems is well known, and this may be expected to be pronounced in prosthetic systems, where due to power limitations the ratio of inertial and steady loads to the stall capability of the actuator will be large compared with normal hydraulic servomechanism practice. A test facility to enable the effects of realistic prosthetic loads on the response of the hydraulic servo described above is just completed and a photograph of the system is shown as Figure 10. Here the valve is directly mounted on the actuator giving a unity feedback system, the actuator drives a tubular arm which can be loaded to simulate any desired prosthetic situation. The arm can be driven either at $\pm$ 60° about the horizontal or $\pm$ 60° about the vertically down position thus enabling most prosthetic shoulder or elbow orientations to be simulated.

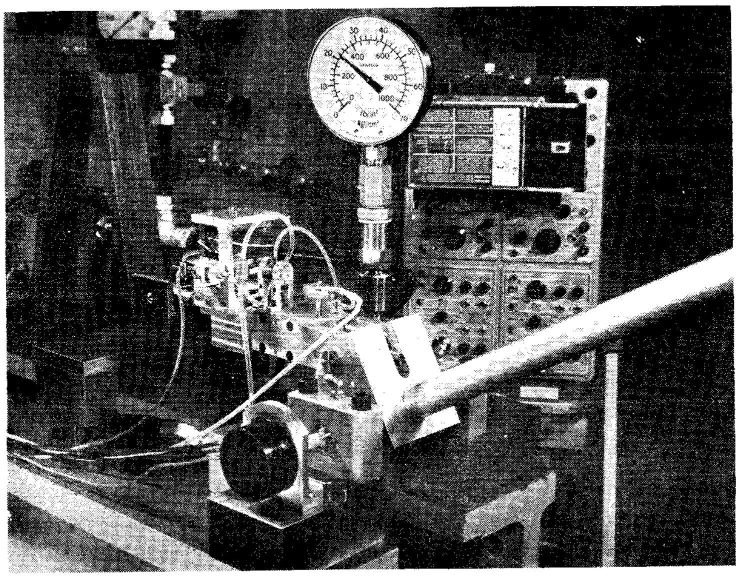


Fig. 10. Photograph of load arm rig

Currently,a series of tests are being performed on this rig and the results of these investigations will be reported.

A Simplified Approach to the Design of Hydraulic Systems for Prostheses

The problem faced by an engineer designing an actuator for a prosthetic system may be stated as follows. For a given hydraulic system specified by a supply pressure P and a maximum valve flow q it is necessary to specify values of actuator area a, stroke L and linkage radius arm r, which enable the requirements of an arm function to be met. Where any arm function may be defined by load torque capability T, angular velocity ω and range of movement θ.

The simplified relationships between the parameters of this system may be stated as:

$$L = r.\theta \qquad (1)$$

That is, the actuator stroke will be defined by the range of movement required and the linkage radius arm

$$T = r.a.P \qquad (2)$$

The stall torque of the arm function will be the product of the actuator stall force, a.P. and the radius at which it acts, r. Obviously this is the ideal torque capability of the arm but during dynamic operation of the prosthesis it will be reduced by pressure differentials required to maintain fluid flow through the valve, accelerate the system inertial loads and overcome coulomb friction in the actuator seals.

$$\omega = \frac{q}{r.a} \qquad (3)$$

This final relationship simply states that the linear velocity of the actuator is equal to the fluid flow velocity in the actuator q/a. Here this simplification ignores the secondary effects of compressibility of the fluid, leakage of fluid past actuator seals and dilation of the pipes and actuator, all of which constitute flow losses and will reduce ω slightly.

It is obvious therefore, by using the three relationships, that for any given constant P and q a solution can be found for the three unknowns r, a and L which satisfies certain specifications of any arm function defined by T, ω and θ.

A nomogram has been developed for the case of a simple linkage such as a chain and sprocket, cable and pulley or geared drive system and is shown as Figure 11. This enables actuators to be specified in accordance with defined prosthetic requirements. To construct the nomogram the pressure scale should be detatched and the windows and slots cut in the main sheet as indicated, the pressure scale can then be moved under the main sheet enabling any pressure between 15 and 40 bar to be set in the upper window. An example will illustrate its use.

1. Suppose the system pressure has been set at 30 bar as shown in Figure 12.
2. Select valve flow q and observe possible combinations of ω and T which are available, note any desired ω will infer a given T value. Say for $q = 0.6$ litre/min; if ω is to be 3 rad/s, T is fixed at 10 Mn. This is denoted by line I on the nomogram.
3. Select range of movement θ; this will define possible combinations of radius arm r and stroke of actuator L for a simple linkage. Say $\theta = 2.5$ rad. is required; this can be met by an r of 20 mm and an L of 50 mm as shown by line II.
4. Finally for the T resulting from 2 and the r from 3 a certain actuator area is defined, i.e. joining 10 Nm on the T-scale and 20 mm on the r-scale as line III will intersect the a-scale in 167 mm^2.

Another example is shown in Figure 13 where a system working at 20 bar and 0.4 litre/min is to be designed for

$$T \geqslant 5 \text{ Nm}$$
$$w \geqslant 2 \text{ rad/s}$$
$$\theta = 3 \text{ rad}$$

A solution could be for $\omega = 2.5$ rad/s and $T = 5.33$ Nm: $L = 40$ mm; $r = 13.3$ mm; $a = 200$ mm^2.

This process can be repeated for any other system pressure and valve flow and it is thus possible to synthesize a system very quickly to meet a considerable range of performance specifications.

Discussion

Obviously there are several constraints in the actual system parameters and they must be considered in parallel with the simple approach described. The hydraulic parameters of supply pressure and maximum valve flow will be mainly dictated by the nature of the power supplies. Higher pressures through giving better power to weight ratios nevertheless require stonger, heavier components for safe operation and a larger power pack and accumulator. The constraint on fluid flows is again related to the power supply in that higher flows enabling faster arm movements consume energy more quickly and recharge times between gross movements would cause frustration; but conversely too low a flow and hence slow arm movements may be equally undesirable. There is a narrow range of acceptable values of linkage radii as physical prosthesis dimensions will

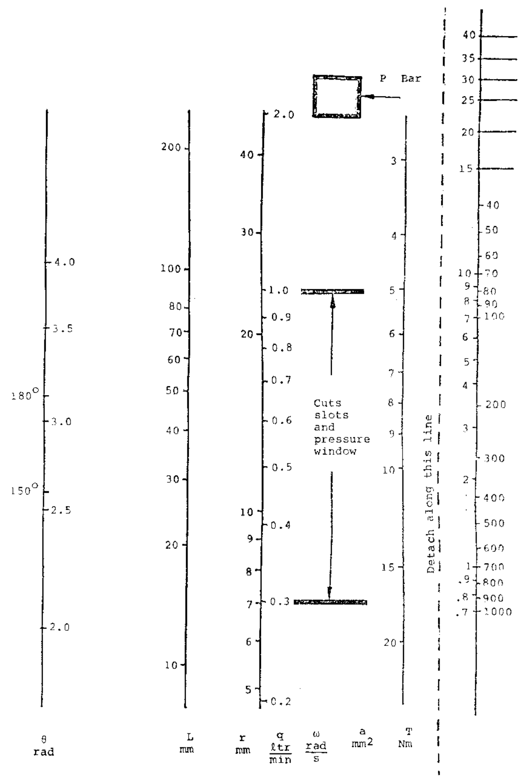


Fig. 11. Actuator nomogram

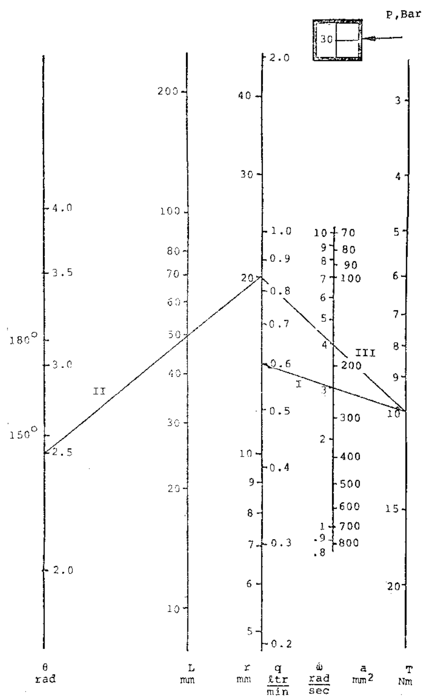


Fig. 12. Actuator nomogram

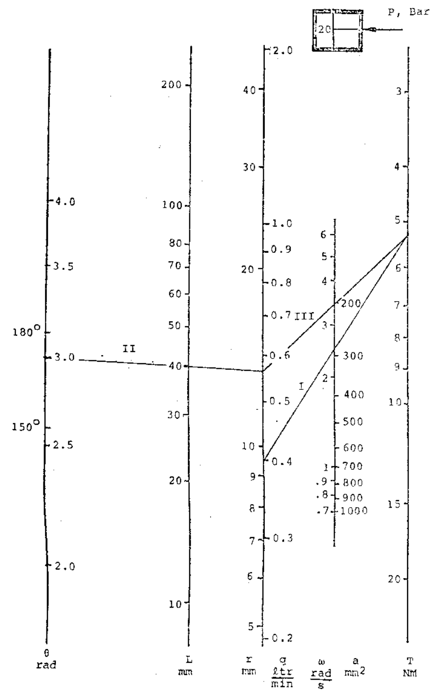


Fig. 13. Actuator nomogram

restrict the acceptable upper limit, and the smaller the radius arm the greater the effect of adverse mechanical advantage magnifying inertial and steady loads reflected at the actuator. The destabilising nature of these loading effects is to be examined on the load arm rig and this will define the lower constraint on linkage radius for acceptable operation of the servo system.

The current work on miniature hydraulic servo system components is progressing toward the production of a prototype hydraulically powered arm prosthesis.

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