ON OPTIMIZING RESPONSE AND EFFICIENCY IN HYDRAULIC ACTUATION OF POWERED PROSTHESES

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Summary

The conflicting requirements of design for good response or for high efficiency in engineering systems is discussed with reference to powered prosthetic control systems. It is seen that of the three practical actuating media, pneumatic, electric and hydraulic, the latter lends itself most readily to a design in which response and efficiency are optimised. The approach reported to effect this optimisation involves a unique design of hydraulic actuator with several effective working areas available. The device is compact and differs only in length from a standard double acting actuator capable of exerting the same maximum force and having the same stroke. The results of an analogue computer simulation incorporating two working areas are presented in which response, efficiency, and stability are investigated in the context of prosthetic applications with variable loading. The potential of using this actuating strategy in prostheses is evaluated.

Introduction

Several independent movements are needed in most powered artificial limbs, and most workers accept that closed loop position control provides the best control strategy, except in the case of prehension. In order that the operator be unaffected by delays, it is desirable that the response time be fast. In a conventional design, this inevitably means low efficiency in most operations. Davies and Lambart [1] show comparisons of hydraulic, pneumatic, and electric actuators in which the former gives the highest efficiency and the best response.

For these reasons our work has been concentrated on hydraulic control systems, and we set out to avoid the compromise between response and efficiency and to optimize both in a novel actuator design. This involved consideration of constant supply pressure, variable output force devices such that demands involving high velocity or heavy loading were met at a certain cost (volume of fluid used); but that less stringent demands were met to the same response standard but at a lower cost by producing a smaller available force using less fluid. It was recognized that these demands change dynamically during even simple single movements. Design studies led to the consideration and rejection of variable ratio gear and lever systems with a conventional actuator. Their main deficiencies are weight, deterioration of output stiffness, and the absence of a datum piston position corresponding to a datum limb position. Attention was then turned to a multi-area piston cylinder arrangement which is described in detail in the next section.

This design also goes some way towards optimizing stability and in fact the appearance of a single small overshoot in the results shown for heavy loads is intentional. Dr Rudd's studies on response parameters in hydraulically operated man-machine control systems [2, 3] show that a transient response with a single small overshoot is acceptable. Heavier damping is not necessary from the point of view of man-machine operation, and its introduction would cause excessive energy dissipation .

Actuator Design

The actuator can be considered basically as being a standard double acting piston-cylinder arrangement having a thick ram (R), (Fig. 1). In each side of this ram are two cylindrical bores (B)

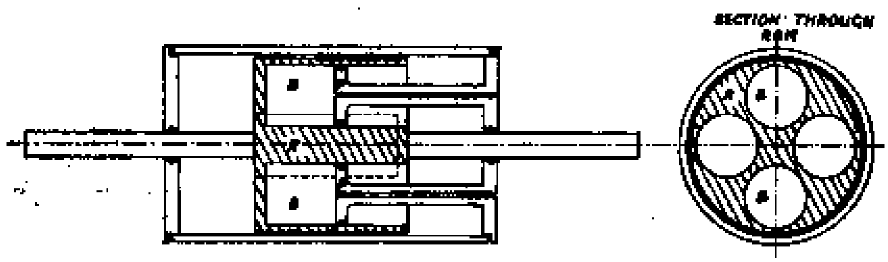

Fig. 1. The actuator

which are swept by small pistons (P) mounted diametrically opposite each other on the actuator end plates. The four cylindrical bores, two let into each face of the ram, are arranged to be positioned symmetrically about the ram rod when the actuator is viewed from the end. Three inlets for hydraulic fluid are let into each end plate, one to supply the volume surrounding the small pistons, and one led up the centre of each of the small piston rods to supply the cylindrical bores.

The two small bores (B) in each side of the ram constitute the small area of the actuator, and these with the addition of the area surrounding them, (i.e. the total ram area) constitutes the large area. For the small area to operate, pressurised fluid is supplied to the two small bores on one side of the cylinder, whilst the area surrounding them, and the whole of the area on the opposite side of the ram are connected to the fluid drain. This avoids the occurence of hydraulic lock. Two symmetrically positioned cylinders, rather than a single eccentric one, are used for the small area to avoid the production of a couple on the ram rod and the associated wear and leakage at the sliding seals.

Normal servovalves can be used to operate the actuator. If two valves are used, one supplying the small area and one the area surrounding it, then the small and large areas can be operated by different criteria. For example, the small area could be error operated whilst the large area is utilised according to load criteria. If an error criterion is to be used for both areas then a single servovalve, with an extra pair of ports, is sufficient.

Analogue Simulation

The analogue computer flow diagram (Fig. 2) represents the equations given in the Appendix, but does not include any area changing or input circuits. The three relays which effect the area change are fired by the same comparator, and for simplicity are shown separately on the flow diagram. Because of the size of the 24-amplifier computer certain simplifications to the equations were necessary. The system's coulomb friction was combined with the leakage term in the equation to form an "equivalent leakage" term in the manner developed, and justified experimentally by Lambert and Davies [4]. The load on the system was assumed to be purely inertial and of constant magnitude for the duration of any response. The inertial load represented could be varied between responses by altering certain potentiometer settings.

The actuator was chosen to operate as an elbow flexor and the arm geometry and mass used for this simulation were the same as those used by Lambert and Davies [5], that is, a forearm of length 15 in with a distributed mass of 1 lb connected to the actuator by a 1 in. lever arm, and a hand or hook of 1 lb holding a mass of up to $2^1/_2$ lb. With the 1 in. lever arm a 1 in/sec ram velocity produces a 1 rad/sec arm velocity. As discussed by Lambert and Davies the actual load on the actuator will be a combination of both steady and inertial loads, and its value will vary with the arm position and the mass carried in the hand. The accelerations expected from the actuator for different hand loads and areas were calculated using this actual loading, and always with the arm in the position corresponding to maximum loading on the actuator. The purely inertial load value used for the simulation was then calculated by consideration of these arm accelerations and of the force available from the actuator. This ensured that actuator loading was always overestimated rather than underestimated.

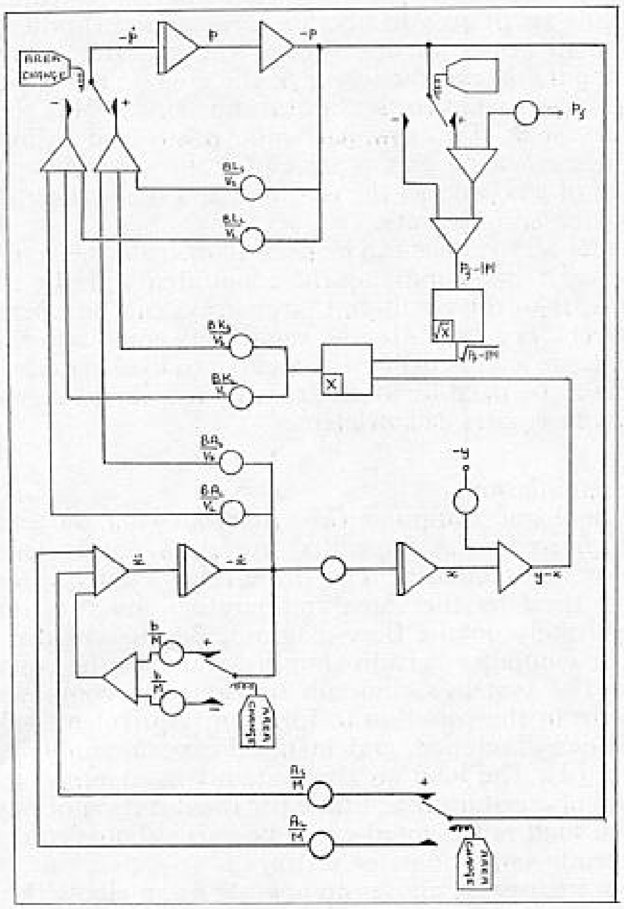

Fig. 2. Analogue Computer flow diagram.

Realistic values were assigned to certain parameters in the equations, again with reference to experimental values obtained by Lambert and Davies [4]. A supply pressure of 500 p. s. i. was chosen for the simulation. If realistic portable power units are found to impose a lower supply pressure the actuator's operation would be unaffected, but its areas would have to be increased for the same force outputs. The large area of the actuator was calculated so as to be capable of producing an acceleration of 200 rad/sec^2 with the mass of an unloaded arm, an acceleration of the same order as the maximum available from a human arm. The geometry of the actuator limits the maximum size of the small area, and in this case a small area of $^1/_4$ of that of the large area was chosen. The areas chosen, along with other parameter values, are given in the Appendix.

The loop gain, structural damping, and equivalent leakage coefficients were adjusted to give the responses required for the various loading conditions, and to avoid cavitation in the higher loading cases. On the large area it was required that the fully loaded response gave only one overshoot, whilst retaining satisfactory performance with an unloaded arm. On the small area the unloaded performance was more important as it must perform most of the arms normal positioning, movements to give the greatest fluid economy. The small area's performance under heavy load or acceleration is less important since such demands would be fulfilled by the large area. In fact the small area could not support an outstreched arm with a $2^1/_2$lb mass in the hand. For this reason a load of 1 lb mass in the hand was used for the illustration of combined operation of the two areas, (see Results). The ramp input rate of 3 rad/sec was chosen for the illustration as this is the order of maximum speed attained in normal positioning movements of the human arm [6].

Results

Figure 3 shows the response of the system in three configurations to a ramp input representing 3 in/sec at the ram, saturating after a displacement of 0.9 in with an inertial load of 1 lbm at the hand. It is seen that with the small area acting alone, the required displacement is not achieved for 0.83 sec — about a half second later than the input demands: the input rate is never attained. A large overshoot occurs which would normally require compensation or additional damping which would result in a still larger response time. The large area acting alone exceeds the input rate and achieves the shortest response time. The response using combined areas shows the large area acting during the heavy accelerations at the beginning (up to 0.2 in) and end (0.9 in) of the input and the small area acting for the long middle portion of the stroke. The response is comparable with that of the more wasteful large area acting alone — the final value being achieved in 0.44 sec. Pressure and velocity traces are also shown for this configuration. There is clearly an abrupt increase in pressure difference when the large area is switched off and a fairly steady velocity throughout the action of the small area; however, there is no apparent discontinuity in displacement. A small, but tolerable, single overshoot occurs in this response as it does with the large area.

Calculation of the volumes of fluid used give a measure of the energy consumption in the three response traces illustrated. These are:

Small area (ignoring overshoot)	0.083 in^3
Large area (with overshoot)	0.346 in^3
Combined areas (with overshoot)	0.173 in^3

Thus the combined areas in comparison with the small area gives, in addition to greater stability, an improvement in response by a factor of more than three, with fuel consumption increased by only a factor of two. In comparison with the large area, it gives sensibly the same response at half the energy absorbtion.

Limiting performance under no hand load was also investigated for the two simple configurations over the same input range and it was found that the large area system could follow a ramp of 8 in/sec (8 rad/sec is the highest velocity recorded for elbow flexion in natural limbs); while the small area system was limited to

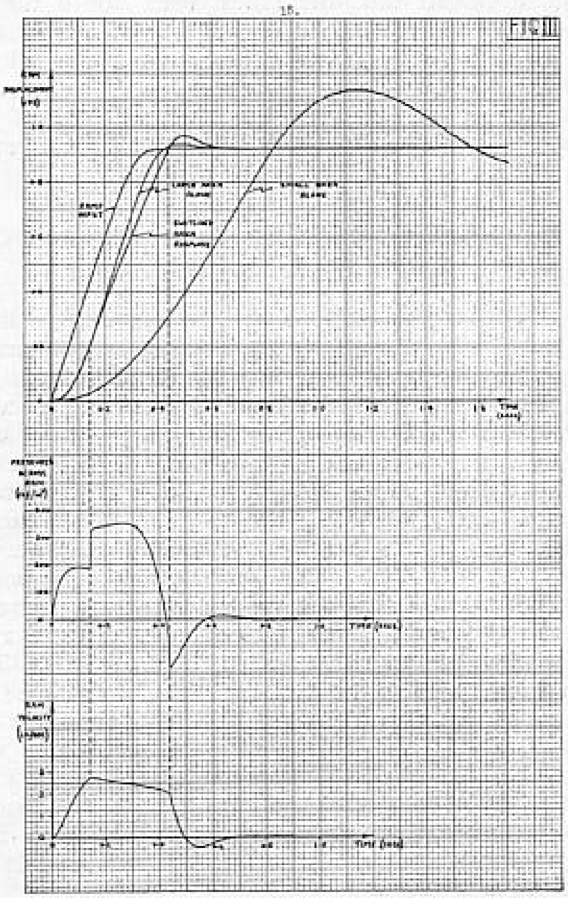

Fig. 3. Comparisons of response, and switched area case results.

2 in/sec. The latter, of course, was more stable than shown above under loaded conditions, and it is thought that many natural motions could be satisfied by this efficient operation.

Under fully loaded hand conditions (2.5 lbm) the small area was incapable of coping, as was mentioned above; but the large area system could follow a 3 in/sec input with adequate stability.

Step inputs were imposed on the system as a standard test procedure, mainly in investigating stability, but results are not thought to be of interest here since these are not realistic physiological inputs.

Discussion and Conclusions

Several criteria have been investigated for switching from one area to the other, for example error, pressure drop across the ram and ram-rod load. As yet time has not allowed the development of a switching strategy which will reproduce the performance shown in Figure 3, and in the end a combination of several criteria may be necessary.

The same type of actuator as described above can be used as a three-or four-area device by increasing the complexity of the associated valve system. For a triple area device pressurised fluid is supplied to the small area, the ram area surrounding the small area alone, or both the small area and the ram area surrounding it. For the four area device there is the additional effective area obtained by supplying pressurised fluid to the ram area surrounding the small area and also to the opposing small area. Suitable area ratios could be calculated to give the desired steps in the output force of the actuator. If the design of the actuator is modified by the addition of a second pair of small pistons acting in each direction, with the assumption that their sizes are arranged to fit the geometry, then by using different combinations of these areas both directly and in opposition 13 different effective areas are available. For artificial limb applications, where a compromise between increases in performance and added complexity and weight has to be made, it is thought that a double area actuator may be the most desirable.

On a full artificial limb there is a need for several actuators for the various joint movements. There is no reason why these dual area acuators shoul not be used in conjuction with conventional actuators on a full limb, all utilising the same pressurised fluid supply. There are also no problems which prevent this design of actuator from being used with a pneumatic supply.

A prototype actuator for use with a hydraulic fluid is to be built for experimental evaluation. If the analogue computer simulation results are shown to be accurate then large savings in efficiency can be expected along with little deterioration in response. This will lead to reductions in the size of energy stores needed for prostheses, and thus in the weight to be carried, whist at the same time making improved performance possible.

APPENDIX

The equations describing the performance of the system are of the standard form for an hydraulic servomechanism with inertial load; Lambert and Davies (4), Haines (7). The pressure representation used is that developed by Haines in his "Program V", and makes the equations non-linear. The equations are written with two letters in a square bracket associated with several of the terms. The upper letter is the constant associated with that term when the small area is operating, and the lower one when the large area is operating. A single letter in a square bracket is associated with both areas.

The equations are: —

$$\dot{p} = \begin{bmatrix} a \\ c \end{bmatrix} (y - x) \sqrt{p_f - p} - \begin{bmatrix} e \\ f \end{bmatrix} p - \begin{bmatrix} g \\ h \end{bmatrix} \dot{x}$$

$$\ddot{x} = \begin{bmatrix} i \\ j \end{bmatrix} p - \begin{bmatrix} k \end{bmatrix} \dot{y}$$

where: —

$$a = \frac{BK_s\sqrt{2}}{V_s} \qquad g = \frac{2BA_s}{V_s}$$

$$c = \frac{BK_L\sqrt{2}}{V_L} \qquad h = \frac{2BA_L}{V_L}$$

$$e = \frac{2BL_s}{V_s} \qquad i = \frac{A_s}{M}$$

$$f = \frac{2BL_L}{V_L} \qquad j = \frac{A_L}{M}$$

$$k = \frac{b}{M}$$

Subletter s refers to small area
Subletter L refers to large area

A = cross sectional area of ram
B = bulk modulus of fluid, including effect of entrapped air
b = structural damping coefficient
K = loop gain
L = leakage across ram+equivalent coulomb friction

M = mass on system + mass of ram
p = pressure difference across ram
p_f = fluid supply pressure
$2V$ = total entrapped volume of fluid
x = output displacement
y = input displacement

A dot denotes differention with respect to time.
Values used:—

$p_f = 500$ lbf/in^2
$B = 5.0 \times 10^4$ lbf/in^2
$b = 4$ lbf sec/in
$M =$ up to 1300 lbm
$K_s = K_L = 0.6$ in$^3/\sqrt{\text{lbf}}$ sec
$y = \pm 1$ in
$L_s = 3 \times 10^{-3}$ in^5/lbf sec
$L_L = 4 \times 10^{-3}$ in^5/lbf sec
$V_s = 0.120$ in^3
$V_L = 0.390$ in^3
$A_s = 0.09$ in^2
$A_L = 0.36$ in^2

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