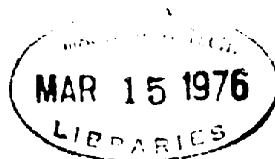HYDRAULIC DAMPER DESIGN
FOR A KNEE PROSTHESIS

by

Eric Hershler Bott

Submitted in Partial Fulfillment
of the Requirements for the
Degree of Bachelor of Science
at the
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
February, 1975

Signature of Author..
        Departments of Mechanical Engineering and
                        Physics, February 23, 1975

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Accepted by...
        Chairman, Departmental Committee on Theses

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ABSTRACT

Damping characteristics for the optimum performance of
various tasks by the users of knee-leg prostheses are found
to be incompatible, which indicates an ultimate need for a
damping mechanism which is controlled by the amputee rather
than by a state function of a specific task (e.g. cyclical
muscle-firing in the stump, or weight bearing of the prosthe-
sis, during the walking cycle.)

A damping mechanism which will be utilized in a proto-
type knee joint to experimentally determine optimum damping
characteristics for all phases of the walking cycle and other
functions (stairway and ramp negotiation, controlled sitting
and kneeling, etc.) has been designed. Overriding design
considerations have been found to be: Damping range from full
joint lock to full joint freedom, dynamic response over the
damping range to 30 Hz, low power consumption by the actua-
tion mechanism, and adaptability to a light weight, low bulk
production design. Hydraulic damping, and historical design
of prosthetic knees is discussed. The force-balanced poppet

valve is introduced, and the prototype valve and actuator design is presented.

The prototype double poppet valve design is found to exhibit excellent properties for use in above knee (A/K) prosthetics, but the actuator design does not meet the response criterion. An improved design, founded on the knowledge gained from experimentation with the prototype, and which it is believed will meet the original design criteria, is offered.

ACKNOWLEDGEMENTS:

In every human endeavor, results are forged with the help
of human interaction- the pool of human experience, thought,
labor, encouragement, criticism, love, hate, etc. is stirred
once again, and answers rise from the murky depths. My
parents showed me the pool and how to use it. They gave
me the chance to learn to use it, and have contributed so
much to it-- Thank you, George and Elizabeth.

Professor Woodie Flowers, whose approach to problems in
life and design I admire so much, has helped me to under-
stand this one, and has given me guidance, assistance and
encouragement in my attempt at its solution.

I offer my thanks to Ralph Whittimore and Max Donath for
their help most generous in procuring materials and bending
them to my will, and to J. "Tiny" Caloggero and Sammy
Marcolongo for their aid.

TABLE OF CONTENTS:

INTRODUCTION: "We have the technology..."

The human locomotion system is extremely complicated, and is one in which each part interacts with the others. The interactions are caused by conditions of geometric compatibility imposed by the rigid bones and the joints with specific degrees of freedom, and by the momentum transfers necessitated by the human form of locomotion. The human muscular system has developed in concert with these restrictions and interactions. The two joint muscles, spanning the hip-knee and ankle-knee joint systems, give excellent efficiency in motion, and coordinate the motions of the various locomotor parts, while the single joint muscles and ligaments lend stability to the system. Obviously, when a mechanical device is used to replace some part of the locomotor system, its function must duplicate that of the replaced part as nearly as possible. Materials research has enabled the replacement of rigid members in the system. However, when a part that plays an active role in the system needs replacement, problems arise.

In particular, prostheses designed to replace the lower leg and knee joint cannot as yet duplicate the interactions with the rest of the locomotor system of the original parts. The normal interactions are of three types: 1. Geometric

constraint and interaction through the hip joint, 2. Momentum transfer from the lower leg to the body through the pelvic girdle, and 3. Energy input (flexion or extension) at the knee or ankle.

If the designer wishes to limit the scope of the problem to one of designing a prosthesis for the function of walking only, some simplification of the interaction assumptions can be made. It has been determined that the net energy input to the knee and ankle joints during the walking cycle in the natural leg are zero. This means that the designer need not incorporate an energy source into his device for the purpose of exciting these joints- he need only incorporate potential energy storage devices, properly designed with respect to impedance matching, to replicate the walking energy interactions of the natural leg. Momentum interaction is similarly easy to design for, since the individual centroids of masses of the parts of the leg remain approximately constant within the parts for all normal muscular activity.

Geometric compatibility is the form of interaction which is tough to design for properly. This is a consequence of the extreme complication of knee and ankle flexures and extensions required by the human form of bipedal locomotion for proper efficiency. During the walking cycle, the knee must first be flexed, when the leg is in the "swing" phase,

to clear the ground. It must then extend almost to lock just prior to contact with the ground. The knee must flex slightly as the weight is born on the leg, to prevent excessive hip rise and shock, and must finally re-extend to maintain hip height as the hip proceeds away from the foot-ground interaction point.

Normal leg motion through one walking cycle is shown in Figure 1, while flexures of the hip, knee and ankle joints are shown in Figure 2. Each leg undergoes two distinct phases per cycle: 1. The "swing" phase, comprising about 35% of the cycle time, when the foot is not in contact with the ground, and 2. The weight-bearing or "stance" phase, comprising the remaining 65% of the cycle. One can see that the lower leg undergoes fairly severe angular acceleration and deceleration about the knee joint, and that the knee joint must support full body weight in both a fully extended and a partially flexed state in order to maintain the natural hip motion. It is here that traditional knee-leg prostheses generally fail to perform naturally. In most such devices, the knee locks when it bears weight or reaches full extension after the swing phase, and does not flex during the hip walk-over (stance) phase. Hip height is kept to reasonable limits by mechanisms at the ankle and the sole of the prosthetic foot. This causes an ungainly walking motion, requiring more energy than does natural gait, and gives the user and his associates continual and embarrasing reminder
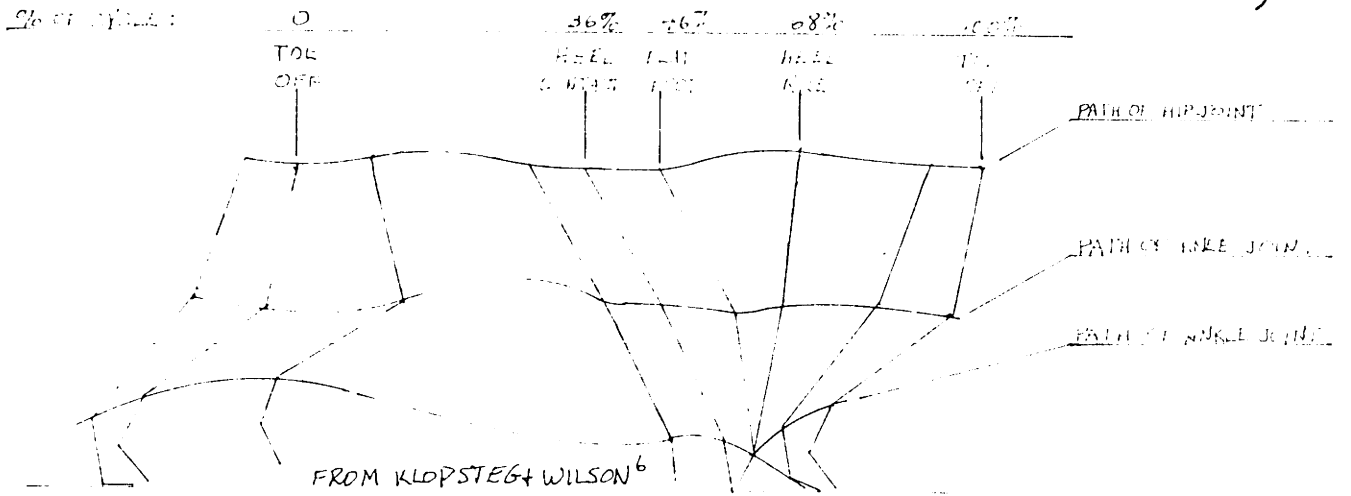
FROM KLOPSTEG+WILSON[6]

FIGURE 1

NATURAL LEG ATTITUDE DURING WALKING CYCLE

HIP ROTATION

KNEE ROTATION

ANKLE ROTATION

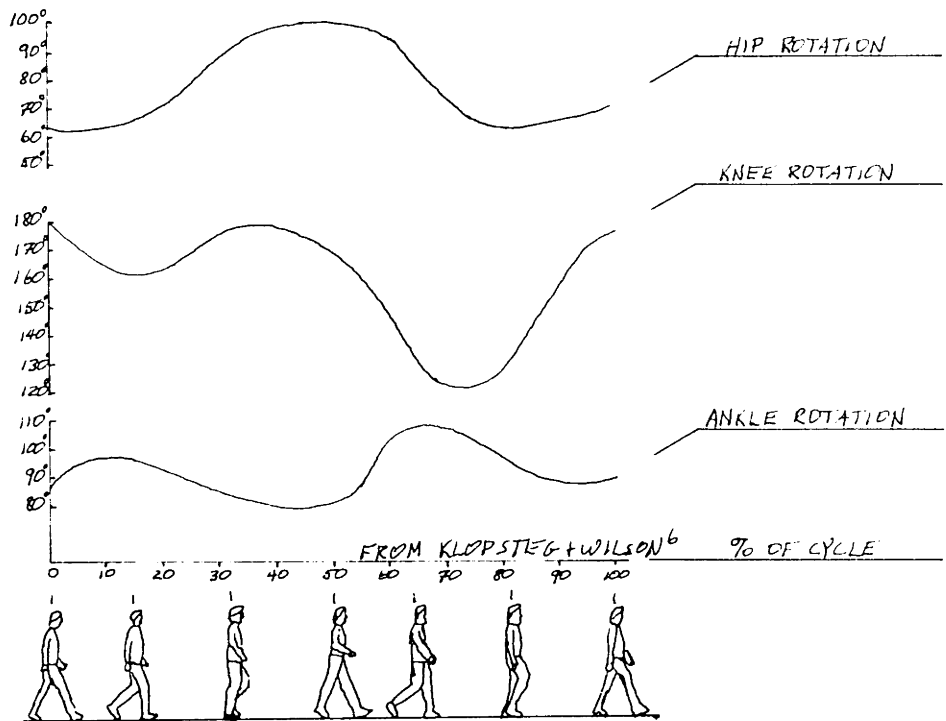FROM KLOPSTEG+WILSON[6]   % OF CYCLE

FIGURE 2

ROTATION OF LEG JOINTS IN FORWARD-VERTICAL PLANE

that the device is present. In extreme cases of poor design, compensatory hip and/or trunk motion may be required to maintain balance, which might ultimately lead to further medical problems. Further, the lower leg ends the swing phase very abruptly at its travel limit, introducing the possibility of shock transmission to the stump, while giving a very artificial look to the gait.

Recent design work has been done on the problem of damping the lower leg at the end of the swing phase, so that the prosthesis undergoes angular deceleration about the knee more naturally, coming to a smooth, shock-free halt at swing phase end. This has great cosmetic and comfort appeal to amputees who are being fitted to an artificial above knee (A/K) prosthesis. The damping is traditionally made a function of position of the lower leg in the swing phase: i.e. damping becomes a state function of the walking cycle. It has been found that subtle changes in this function can yield enormous changes in the fatigue experienced by the user of the device, and vast amounts of time and money have been expended in the developements of mechanisms which are intended to exhibit less energy consuming damping patterns.

Unfortunately, these devices, even when fitted with damping mechanisms, are still controlled by events in the walking cycle: their characteristics are functions of this cycle, and are in general not suited to the performance of other tasks. For example, when properly "tuned" for walking,

such devices become very difficult to use on stairs or ramps,
where the user must vault over a locked knee to progress.
This is at best ungainly; it can be disasterous when descend-
ing stairs. Further, the user cannot use the prosthesis to
assist the natural leg in gravity aided tasks, such as sit-
ting, if the knee is locked during weight bearing.

Flowers[5] has suggested that voluntary control of knee-
flexure damping would broaden the spectrum of uses of the
A/K prosthesis and simultaneously permit more natural gait.
The user would be able to perform gravity aided tasks very
much more naturally by allowing the artificial knee to flex
in a controlled fashion. Gait would become more natural
because the knee could be flexed slightly during the hip
walkover part of the stance phase, allowing more natural
hip-pelvis movement. Complexity could be reduced from that
of current devices, which must sense and process various
signals of the walking cycle (e.g. swing, contact and weight-
bearing, heel-off and toe-off) to produce their response.
Finally, the user could educate himself to be able to use the
prosthesis over a much greater range of gait-speeds than
current prostheses are capable of.

Thus it is hoped that by ultimately providing the am-
putee with a knee in which he can continuously vary the damp-
ing at will from full lock to full freedom, rather than with
a knee in which the damping is controlled by a state function

of the walking cycle, the walking motion can be made more natural, and the scope of the prosthesis can be extended readily by the user, without "retuning", to tasks other than walking.

The objective of my thesis is to design a prototype hydraulic valve-piston assembly (see Figure 3) that can provide the necessary continuous damping characteristics, can be activated with minimum energy expenditure, is reliable and cheap to manufacture, and can be made light and non-bulky. The prototype will be used with a computer link-up to study the optimisation of knee damping characteristics during the walking cycle and during the performance of various other tasks.
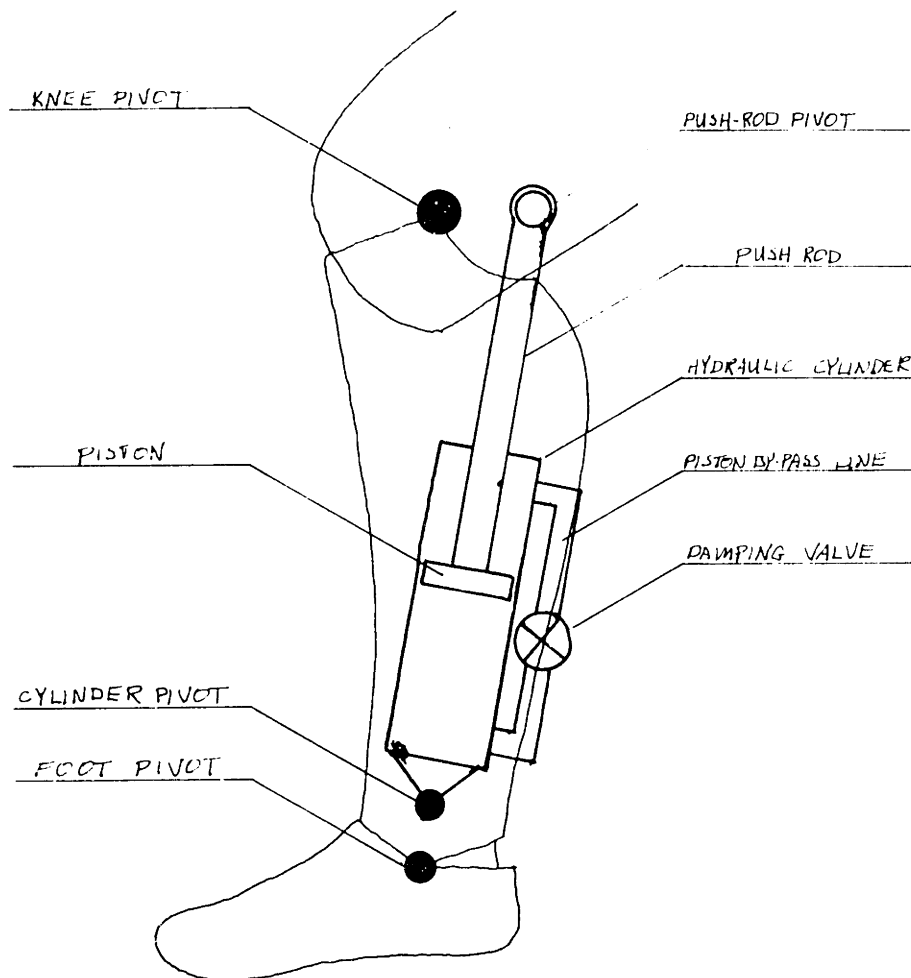
KNEE PIVOT

PUSH-ROD PIVOT

PUSH ROD

HYDRAULIC CYLINDER

PISTON

PISTON BY-PASS LINE

DAMPING VALVE

CYLINDER PIVOT

FOOT PIVOT

FIGURE 3

SCHEMATIC OF VALVE AND CYLINDER PLACEMENT

IN PROSTHESIS

DESIGN CONSIDERATIONS:

The application of a damping unit on an A/K prosthesis im-
poses severe design limitations on the mechanism. The most
physically evident of these are size and weight limitations.
Braune and Fisher[6] have reported that the combined weight
of the skank and foot comprises 8%, and the thigh 12% of the
total body weight. This means that the total supportive
structure and damping mechanism, with the required foot
mechanism must weigh on the order of 8-10 pounds if the
prosthesis is to be universally applicable. While no defin-
itive work has been done at this time on the optimum moments
of A/K prosthetic components, making it impossible to quote
hard figures, a further restriction on such components is
that their moments about the knee be somewhat less than that
of the natural shank and foot, because there exists no poss-
ibility for power input to the knee for the purposes of
flexure and extension. For cosmetic reasons, the A/K pros-
thesis should be as compact spatially as the limb it replaces.
This means that both the structural members and the damping
device must be designed within the dimensions of a normal leg.

The above considerations impose power consumption restric-
tions on the damping actuator design, since power storage
devices involve considerable weight and storage space commit-

ments. This implies that proper design will entail an actuator which requires no power to maintain a steady-state damping level, and a configuration which does not tend to lock up under load.
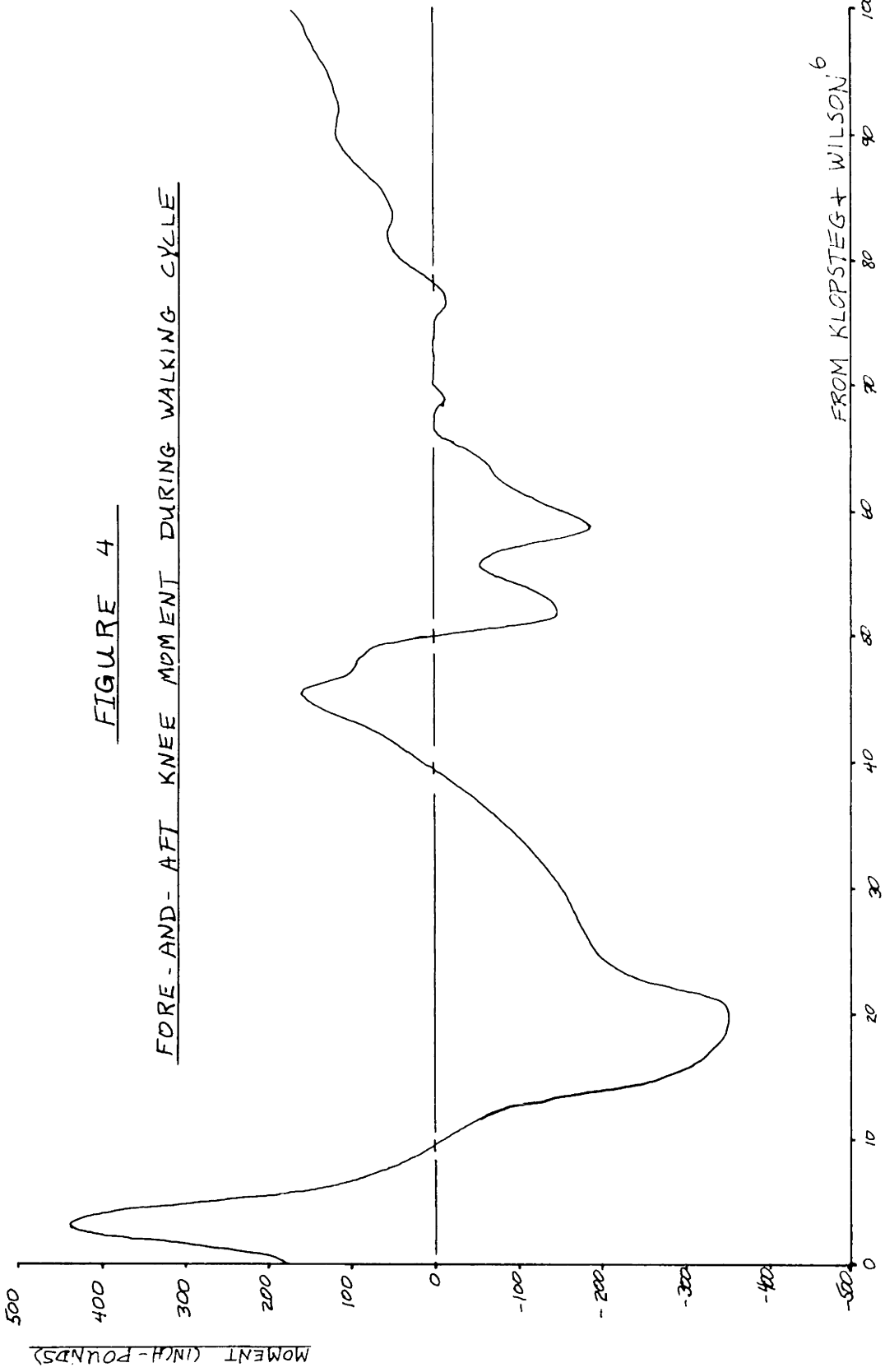
Figure 4 shows the moment which the normal knee must supply in order to reproduce a typical walking gait. Graphical differentiation indicates that to reproduce this curve accurately, assuming a .6- .7 second cycle time, a frequency response of at least 25-28 Hz. must be attainable by the damper actuator, and that the device provide a dynamic range of zero to 450 inch pounds moment about the knee. Other activities, especially extranormal ones, such as stumble recovery, demand that the device be able to supply (and sustain) much higher moments. Finally, the requirements of standing necessitate that the knee be able to "lock" fully.

It is desired to use a prototype damping device to study the optimisation of damping moments for various tasks, and to determine if conscious control of the prosthesis by the amputee is feasible in terms of training time and utility. Thus, it is required that the actuating system be compatible with some form of computer interface, so that a computer may be used to interpret command signals from the wearer, and actuate the device.

The final overall design criterion is that the device have the potential for developement into a production model.

FIGURE 4

FORE-AND-AFT KNEE MOMENT DURING WALKING CYCLE

MOMENT (INCH-POUNDS)

500
400
300
200
100
0
-100
-200
-300
-400
-500

0   10   20   30   40   50   60   70   80   90   100

% OF CYCLE, STARTING FROM HEEL CONTACT, AS IN FIGURE 2.

FROM KLOPSTEG + WILSON.[6]

16

This entails low cost of manufacturing, and minimum complexity. Due to the scarcity of repair stations for prosthetic devices, and due to the fact that failure of the prosthesis would generally be socially and physically "catastrophic", reliability of design is of paramount importance.

DETAIL DESIGN:

Hydraulic damping was chosen over mechanical damping for several reasons. Among the most important were:
1. Hydraulic damping need not incorporate wear surfaces which require periodic adjustment. 2. Large dynamic damping ranges may be designed for, up to and including full lock. 3. Small actuating forces are required by properly designed valves due to large gains possible with fluid devices.

Some success has been had with a piston- two-port cylinder of 7/8" diameter acting on a two inch moment arm about a simply-pivoted knee joint (roughly depicted in Figure 3.) The data from Figure 4 indicates that with this geometry one might expect a working pressure differential across the piston of up to 375 psi in walking. However, any system designed around these parameters must have a large safety factor for handling non-normal loads, as found in stumble recovery. 1000 psi was chosen as a reasonable peak load, as any greater moment applied to the stump (1000 psi⇔1200 inch pounds) might result in pain or damage to the stump. If pressure failure occurs in the damper due to pressures higher than 1000psi, the problem may best be elim-

inated by placing an appropriate check valve around the piston.

Referring to the piston- moment arm geometry, and to Figure 2, which indicates that peak-to-peak knee deflection is nearly $55°$ during the walking cycle, one sees that the damping valve and associated plumbing must pass 1.15 cubic inches of hydraulic fluid in approximately 30% of the walking cycle. At an average gait (.7 seconds per cycle) this implies 5.5 cubic inches per second maximum flow rate. Noting from Figure 4 that the maximum flow rate occurs at a part of the cycle in which relatively little moment (30 inch pounds average) is applied at the knee, one can calculate the required freedom of the system. A moment of 30 inch pounds implies 25.2 psi across the piston. Experimental evidence from Breen[2] indicates that plumbing with 3/8" line would yield the required freeness up to 24" maximum plumbing length. A criterion of the valve design would then be that it contribute less damping in its free state than a suitable length of 3/8" plumbing.

A review of commercially available valves indicates that the particular criteria of freeness, dynamic response, and actuation power consumption demanded by prosthetic application are not offered in a valve in off-the-shelf form. Servovalves exist which might be modified to suit the task, but would become prohibitively expensive. Thus, there is a very real

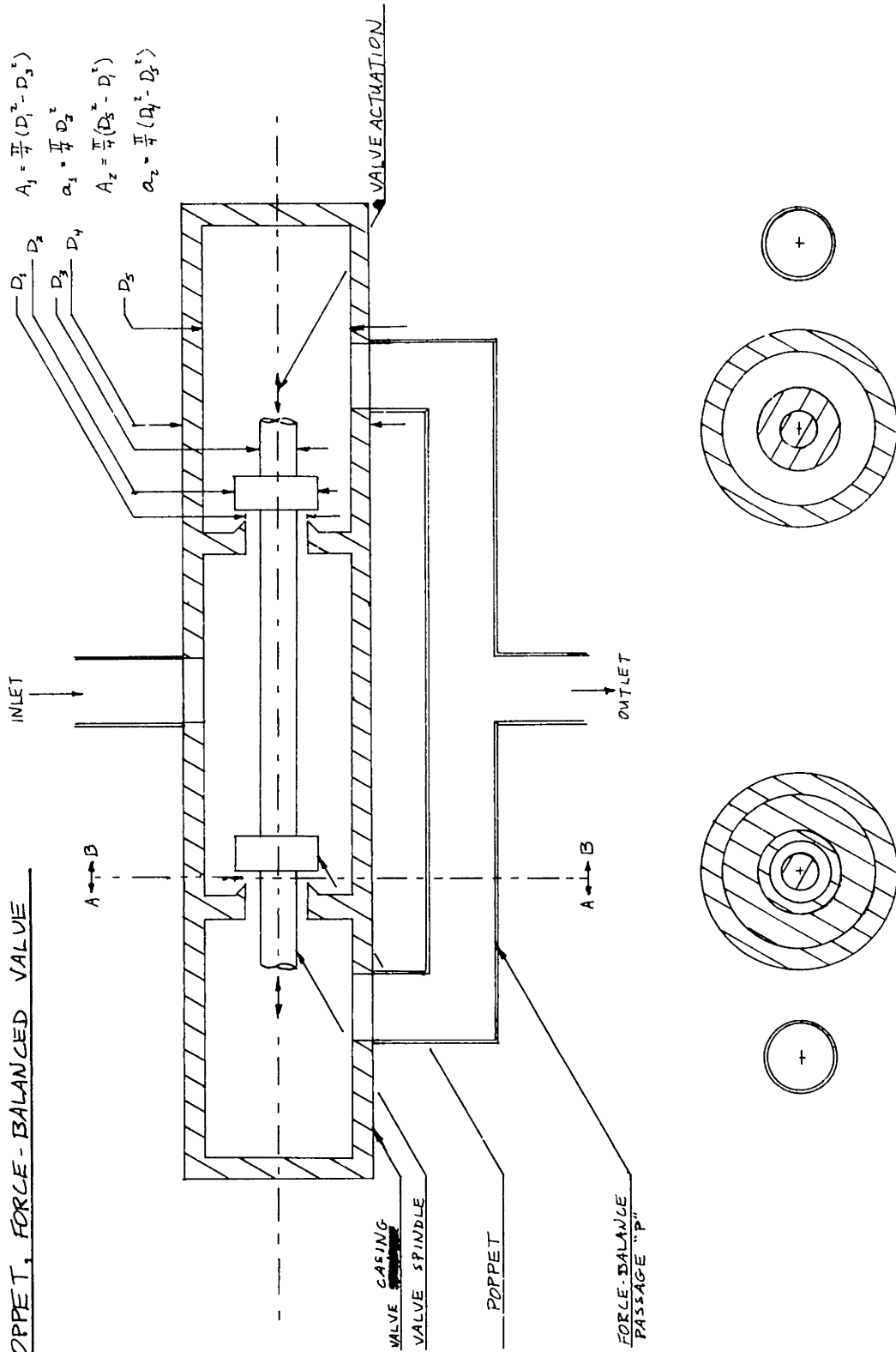need for a suitable valve design for this application.

Attempts have been made to design such a suitable valve.
The attempts seem to have been subject to two modes of fail-
ure. When a pressure sealing surface is used as a bearing
surface, a trade-off exists between large clearance for low
bearing friction and small clearance for low seal leakage.
This trade-off in an A/K prosthesis is certainly critical;
leakage and friction must both be severely restricted.
When a valve is designed with an objective of light weight
and low bulk, particular attention must be paid to compon-
ent warpage due to pressure. Seisure of adjacent parts may
occur, even when proper bearing clearances are specified
for the unloaded case.

The double-poppet valve allows elegant solution of
these design problems (see Figure 5) while offering several
distinct advantages. It will be seen that this type of
valve has no inherent frictional characteristics. With the
valve seat design as in Figure 5, such a valve can be made
insensitive to pressure warpage by making the ratios A_1/E_1a_1
and A_2/E_2a_2 equal (where E_1 and E_2 are the Young's Moduli of
the valve spindle and casing material respectively), thereby
insuring that at closure, the pressure will elastically de-
form the spindle and casing equally.

This type of valve, with proper seat geometry, can be
made to be force-balanced by means of a passage connecting

FIGURE 5

DOUBLE - POPPET, FORCE - BALANCED VALVE

$$A_1 = \frac{\pi}{4}(D_1^2 - D_3^2)$$
$$a_1 = \frac{\pi}{4} D_2^2$$
$$A_2 = \frac{\pi}{4}(D_3^2 - D_1^2)$$
$$a_2 = \frac{\pi}{4}(D_4^2 - D_5^2)$$

VALVE ACTUATION

D_1 D_2
D_3 D_4
D_5

INLET

OUTLET

A B
A B

VALVE CASING
VALVE SPINDLE
POPPET
FORCE-BALANCE PASSAGE "P"

SECTION "B-B"

SECTION "A-A"

the two outer valve chambers (Passage "P" in Figure 5).
Blackburn, Reethof and Shearer[1] indicate a rather complicated
and critical seat geometry to yield force balancing. (See
Figure 6A) This geometry involves extreme nonlinearity of
flow vs. aperture. I felt that a seat design less critically
dependent on exact machining tolerances, and with better
small aperture linearity might be designed. The semi-torroid-
al/seat shape shown in Figure 6B, is very durable and excellent
in terms of leakage, although nonlinearity persists.

The advantage of simplicity is evident in the principle
of the double-poppet valve; in manufacture only one set of
dimensions is critical: The distance between poppet faces
must be extremely close to the distance between seat faces.
Experimentation has shown that leakage becomes acceptable
only when the error between these two dimensions is reduced
to .0005" or less.

The gain characteristics of the poppet valve, especially
with faces of the type shown in Figure 6B are quite attract-
ive. Valve gain will be increased as the square of the seat
diameter, by increasing the ratio of aperture area/actuation
distance similarly. Valve sensitivity may be decreased by
reducing the seat diameter.

The design of the prototype valve was intended to be
quite heavy, to preclude any misalignment which might occur
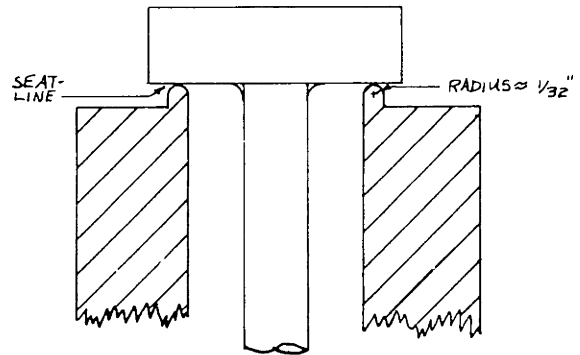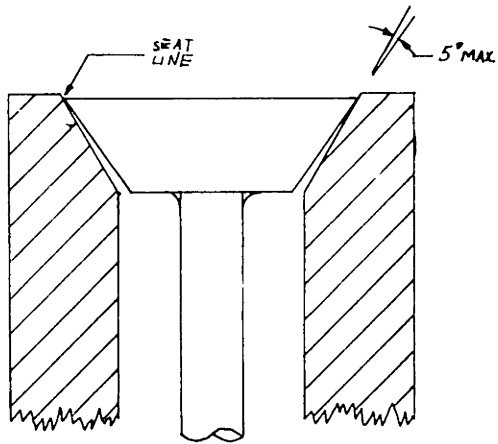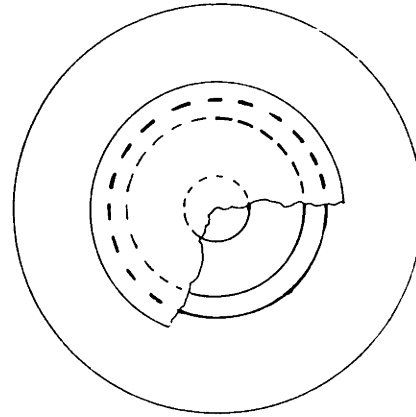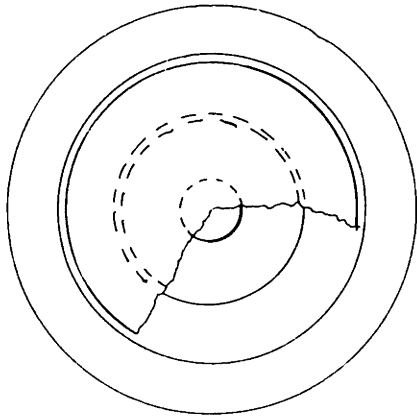upon misuse or repeated assembly and disassembly. It was

SEAT
LINE

5° MAX.

FIGURE 6-A

FORCE-BALANCE VALVE-SEAT DESIGN,
BLACKBURN, REETHOF + SHEARER

HIGHLY NON-LINEAR FLOW VS. OPENING

SEAT-
LINE

RADIUS ≈ 1/32"

FIGURE 6B

PROPOSED FORCE-BALANCE
VALVE-SEAT DESIGN

decided to maintain overall simplicity by utilizing a thread-
ed shaft to drive the valve spindle, and by utilizing avail-
able fittings and actuator parts where feasible. This led
to the unfortunate necessity of including a rotary O-ring
seal around the actuator screw, which introduced severe
friction problems. A Portescap Micromotor #28PL21-413 (See
Appendix for specifications) was used to power the drive
screw. The final valve configuration is shown in Figure 7.
External plumbing was fitted to achieve force balancing and
to couple the valve to a 7/8" diameter cylinder with 4" of
piston travel. The distance between the poppet faces was
fixed; final adjustment on the seat seperation dimension
was done by inserting various thicknesses of aluminum shim
material between the mating faces of Valve Seat 'A' and the
middle valve casing until acceptable internal leakage was
obtained. Valve gain (seat diameter) was chosen as small as
possible, commensurate with the necessary hydraulic freedom
in the valve-open state.

FIGURE I

DRAWING OF PROTOTYPE VALVE
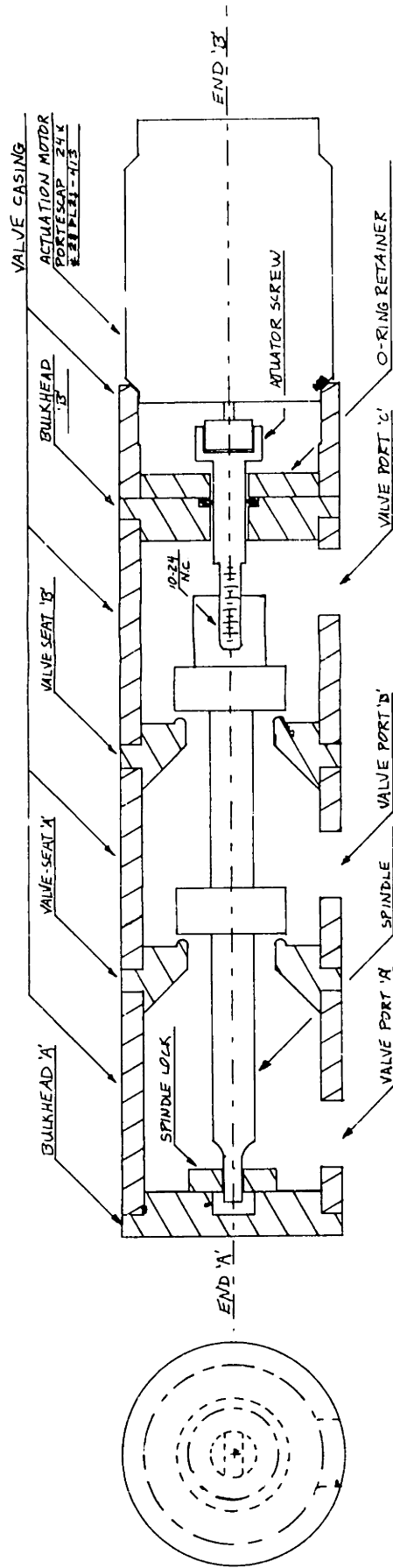
DRAWING IN SCALE (1:1)

ALL PARTS: STAINLESS STEEL, EXCEPT SPINDLE LOCK + ACTUATOR SCREW: BRASS
ALL PARTS EXHIBIT CYLINDRICAL SYMMETRY EXCEPT SPINDLE + SPINDLE LOCK

TESTING AND EVALUATION:

The testing of the valve was intended to determine the
suitability of the double-poppet valve (static testing) and
of the actuator mechanism (dynamic testing). Briefly, the
valve design was a success, proving itself excellently suit-
ed to the task, while the actuator design was a failure due
to seal friction.

Static Testing:

Static testing was performed to evaluate four criteria:
1. Valve leakage in the closed state. 2. Valve freedom in
the open state. 3. Valve linearity of flow vs. aperture.
4. Valve force balancing in the closed state and in conditions
of steady flow.

The test results for criteria 1-3 are shown in Figure 8,
which shows flow vs. time in various states of valve freedom.
The data show that, with proper tuning, the valve design
exhibits excellent closed-state leakage characteristics.
Testing was done with 24.9 psi across the valve. Data was
taken using a constant voltage applied through a linearly
actuated potentiometer, driving a Hewlett-Packard strip
chart recorder. Figure 9 shows the experimental set-up.
Recorder deflection was proportional to total flow through
the valve, and the recordings were graphically differentiated
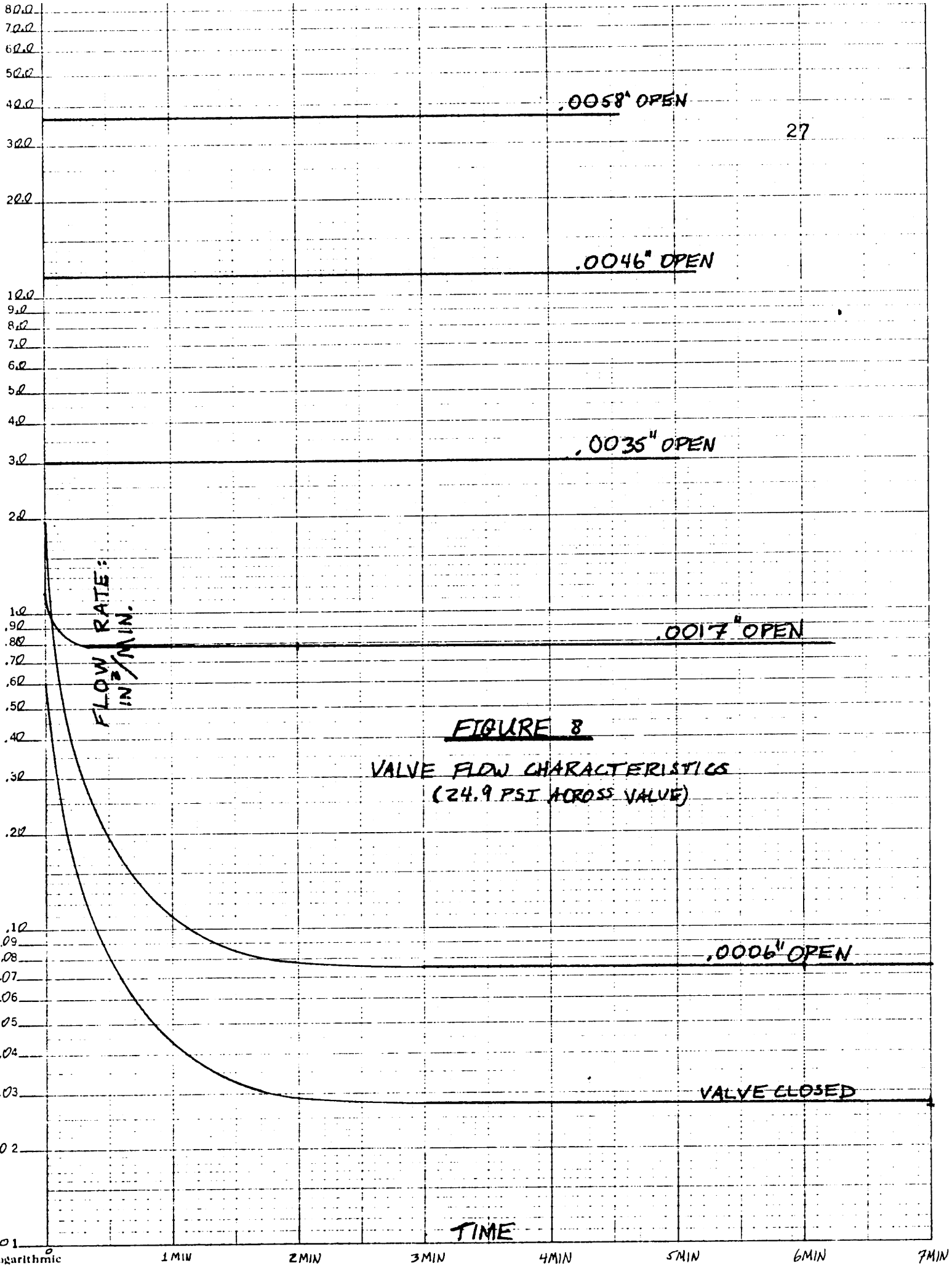
FIGURE 8

VALVE FLOW CHARACTERISTICS
(24.9 PSI ACROSS VALVE)

Flow rate curves labeled: .0058" OPEN, .0046" OPEN, .0035" OPEN, .0017" OPEN, .0006" OPEN, VALVE CLOSED

Y-axis: FLOW RATE: IN³/MIN.
X-axis: TIME — 1MIN, 2MIN, 3MIN, 4MIN, 5MIN, 6MIN, 7MIN

27

Semi-Logarithmic
Cycles x 10 to the inch

15 lbs force.

VALVE

PLUMBING

CYLINDER

PLUMBING

PISTON, 7/8" DIA

FIGURE 9
TEST APPARATUS SCHEMATIC

205 V. D.C.
SUPPLY

STRIP CHART
RECORDER

LINEARLY
ACTUATED
POTENTIOMETER
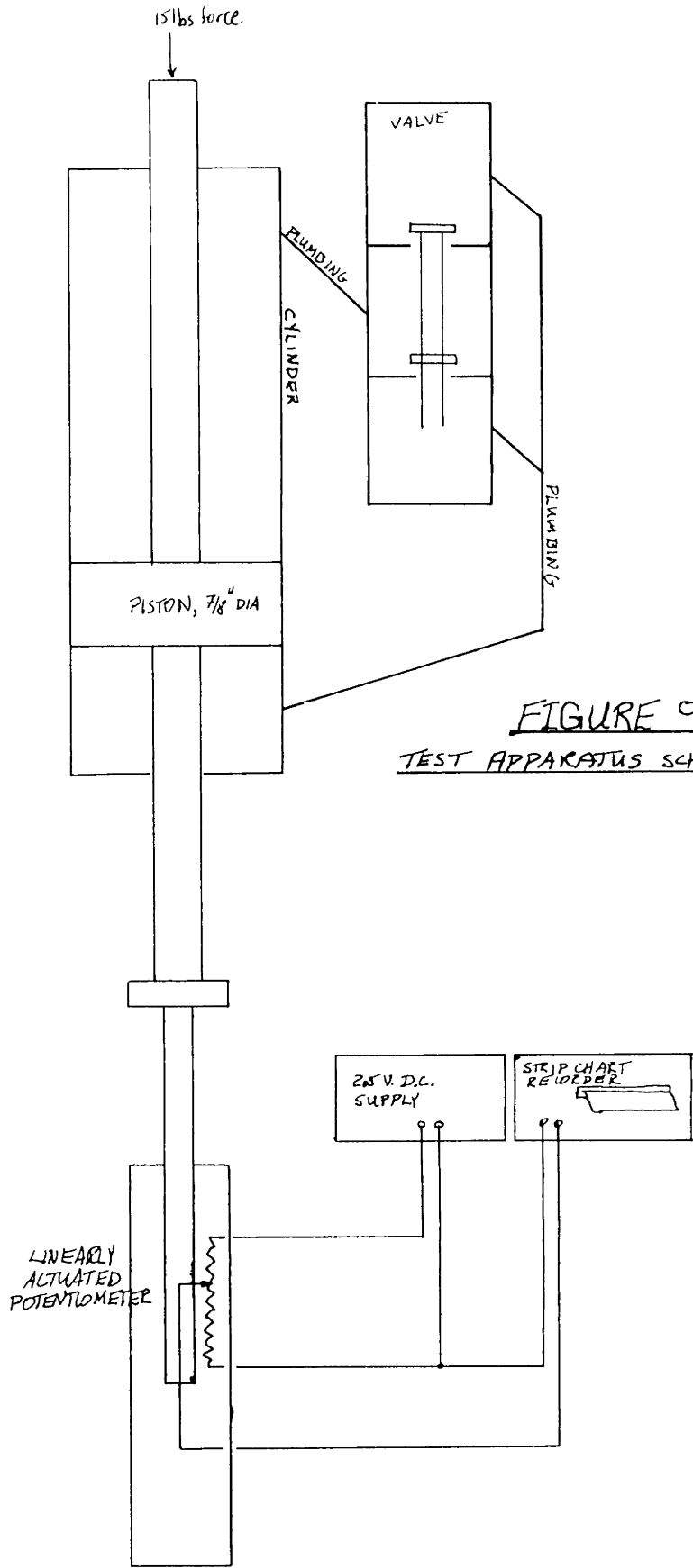
to yield flow rates. A relatively viscous oil (diffusion pump oil) was used in this set of tests. However, subsequent tests with lighter oil (Mobil hydraulic fluid) showed that leakage rates were also entirely satisfactory for less viscous fluids. There was some initial time nonlinearity of the flow rate in this test due to air trapped in the system. However, the data levelled off rapidly, and the steady-flow data are representative of the fluid freedom of the valve.

Due to the viscous nature of the oil, maximum flow rate was less than anticipated. Tests with the spindle removed from the valve casing confirmed that, in the open state, the flow limiting factor of the system was in the plumbing and/ or fittings: virtual valve freedom was obtained at .011" between the poppet faces and the valve seat faces. The valve is very nonlinear in regions of small aperture. Although a knife edge seat as shown in Figure 5 would tend to linearize the flow in the small aperture regime, it is felt that the increased fragility would be prohibitive.

Blackburn, Reethof and Shearer[1] give a treatment on axial forces exerted on spool-valve spools due to steady-state flow, and indicate that the analysis may be extended to other geometries. For a spool-valve, with two orifaces in series, the axial force on the spool is:

$$F_A = 2Q\sqrt{\rho\, P_V}\, \cos\theta, \text{ where Q is the flow rate,}$$

ρ the fluid density, P_v the pressure drop across the valve, and θ is defined as in Figure 9. With the valve seat geometry as designed, it is evident that in the double-poppet valve, θ will always be greater than in the spool valve, and will also be more constant with valve opening. Using the viscous oil, with the maximum flow rate calculated in the DESIGN CONSIDERATIONS section, one finds that the axial force on a spool-valve spool would be about .2 pounds for a reasonable value of x/C_r. Thus, one would expect the axial force due to steady-state flow on the double-poppet valve spindle to be less than .2 pounds. It was found in testing that the O-ring seal friction completely dominated all other forces acting on the spindle; the axial flow force was unmeasurable. The axial O-ring seal forces measured .518 pounds (stiction).

Dynamic Testing:

Analysis showed that, with the required actuation distance (.02" maximum), component weight, and available motor torque, but with frictionless bearings, the valve would have a dynamic response up to 19 Hz., which would have been adequate for a slow walk at best. As indicated, however, the O-ring seal was not frictionless, and no dynamic response for the valve was obtained. Experimentation showed that the running torque exerted by the O-ring was 1.02 inch ounces, while the static torque was on the order of 2.9 inch ounces.
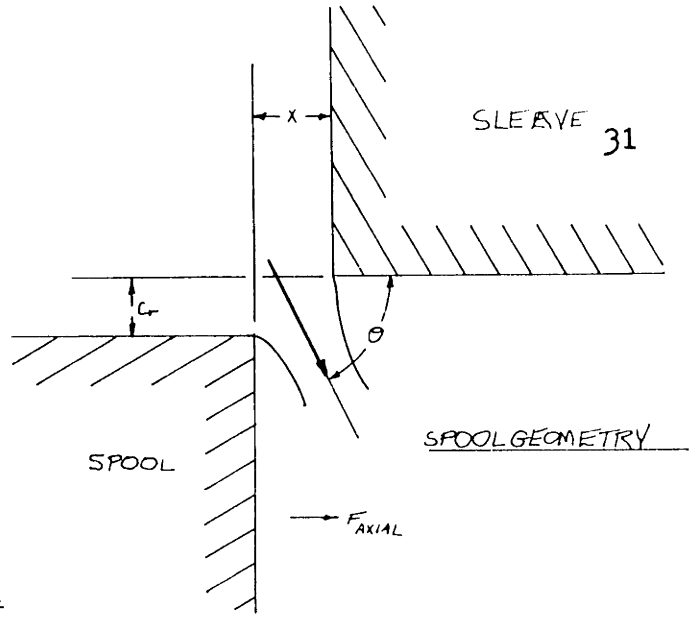
SLEAVE 31

X

Cr

θ

SPOOL

F_{AXIAL}

SPOOL GEOMETRY

cos θ vs. X/Cr
(EXPERIMENTAL)

FROM BLACKBURN, REETHOF + SHEARER

cos θ

1.0
.8
.6
.4
.2
0

0   10   20   30   40   50
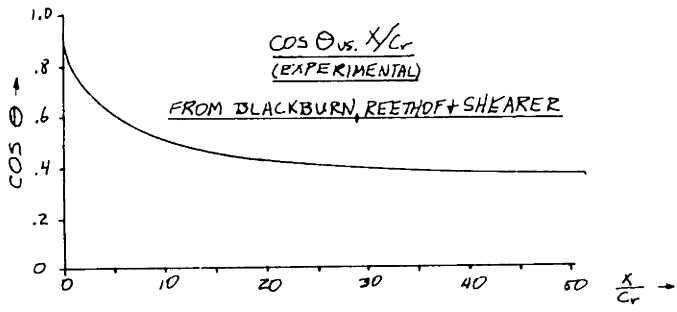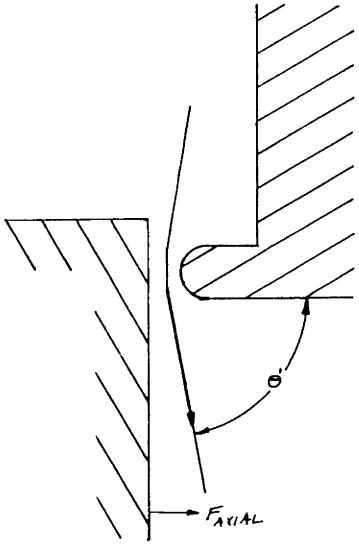
$\frac{X}{Cr}$ →

FIGURE 10

AXIAL FORCES ON SPINDLE
DUE TO STEADY STATE FLOW

θ

F_{AXIAL}

PROTOTYPE SEAT GEOMETRY

While the motor would open and shut the valve on a steady
state command, it would not perform dynamically at even 1 Hz.
Two additional sources of problem have been identified:

1. As the actuator screw turns within the O-ring, the friction
skews the O-ring rotationally. Inordinately high torque
must be applied to the actuator screw to break the resultant
O-ring 'set', unless the O-ring set has had time to relax.
This relaxation apparently takes on the order of seconds to
occur.[7] 2. The actuator screw threads tend to bind under
high load application (valve seating), causing start-ups to
require high torque.

Evaluation:

The valve designed has been shown to be very well
suited to application as an A/K prosthesis damping device.
Its flow capacity and closed state leakage are excellent
for its bulk. Force balancing was achieved, and axial
spindle forces due to steady state flow were shown to be
negligible. The valve exhibited flow vs. aperture nonlin-
earities due to viscous properties, in spite of attempts
to design for less small aperture vena-cava formation.
The valve will have to be actuated by a compensating mech-
anism, or by a nonlinear control system, if linear response
to command is desired. The valve exhibits excellent high gain
characteristics.

The valve actuator was a failure, largely due to its

frictional characteristics. O-ring seals are not suited to use in control valves for prostheses. The actuator screw design was marginal. Although it was immersed in oil, it was still a plain bearing, working against fairly high loads.

REDESIGN:

Armed with the knowledge gained from testing the original
device, a second device was designed in an attempt to over-
come the actuation problems. (See Figure 11.) In this sec-
ond design, the momentum of the moving parts has been reduced
to the minimum that will allow sufficient material for rigi-
dity. Total weight of the valve and actuator would be less
than .75 pounds. The spindle and casing have been designed
to give equal deflection at high pressure loadings. The
poppet faces have been altered for (hopefully) better small
aperture linearity and near zero axial force due to steady
state flow. The spindle is anchored by flexible bellows,
which are to be punched out of stainless steel, and must al-
low the spindle to move .025", with minimum spring constant.
The actuator is an eccentric (.025" eccentricity) coupled
directly (1:1 ratio) to a Portescap 28PL21-413 Motor, and
drives a connecting rod which drives the valve spindle. The
eccentric rides in roller bearings, and the connecting rod
rides a roller bearing on the eccentric. No reciprocating
or rotary seals infest the design at any point, and the
valve-plumbing-cylinder assembly is completely sealed. If
the eccentric drive is used within the limits prescribed in
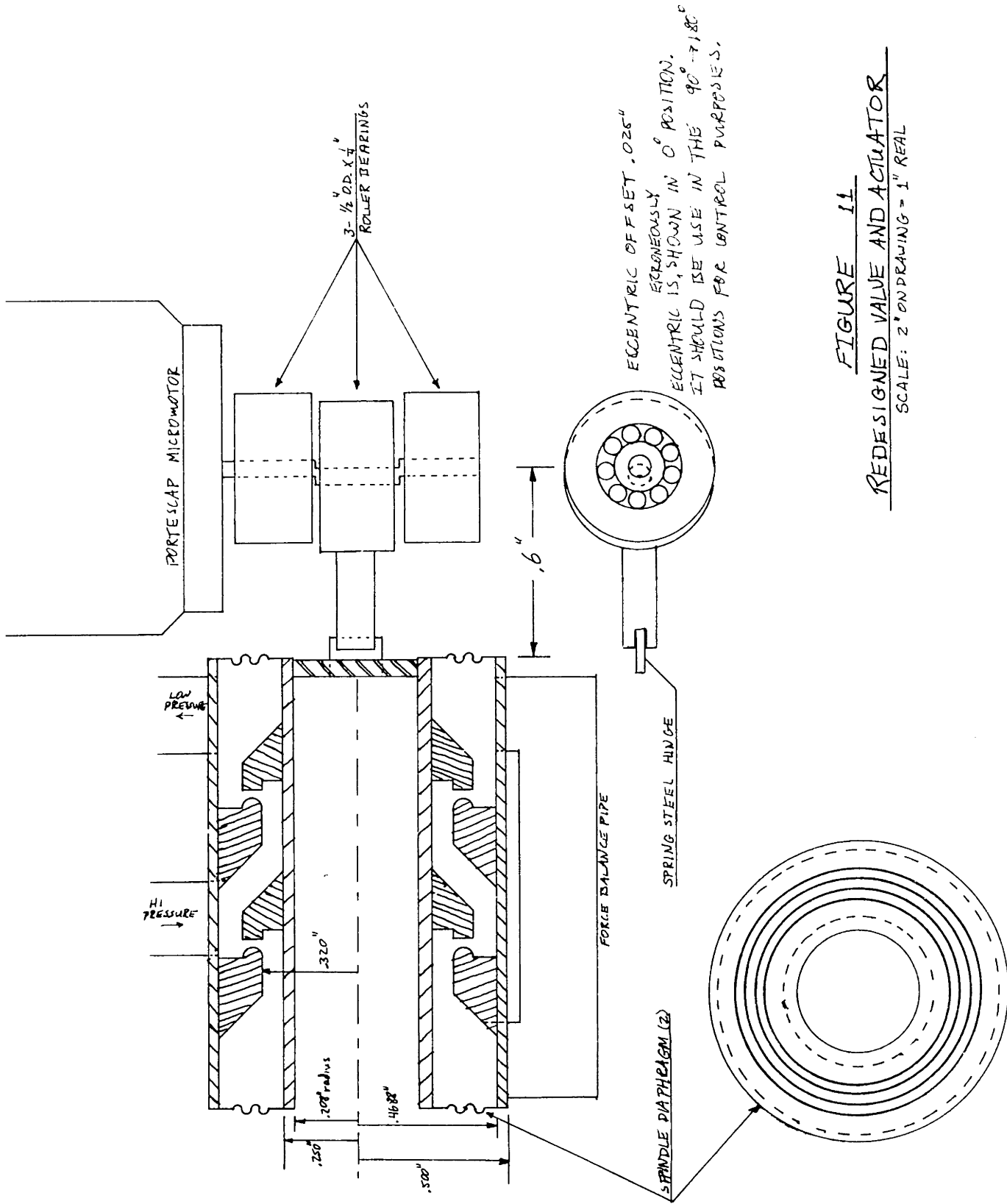Figure 11, the drive will tend to reduce the valve nonlinear-

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PORTESCAP MICROMOTOR

3 - ½" O.D. X ¼"
ROLLER BEARINGS

LOW
PRESSURE

HI
PRESSURE

.320"

.70" radius

.468"

.250

.500"

.6"

FORCE BALANCE PIPE

SPRING STEEL HINGE

SPINDLE DIAPHRAGM (2)

ECCENTRIC OFFSET .026"
EXTRANEOUSLY
ECCENTRIC IS, SHOWN IN 0° POSITION.
IT SHOULD BE USED IN THE 90°-180°
POSITIONS FOR CONTROL PURPOSES.

FIGURE 11

REDESIGNED VALVE AND ACTUATOR

SCALE: 2' ON DRAWING = 1" REAL

ities at small-apertures. With reasonable weight watching
in the fabrication of the connecting rod and bearings, the
design should give a frequency response to 35 Hz.

CONCLUSIONS:

With proper actuator design, the double-poppet valve is
eminently suited to the application of an A/K prosthetic
damping mechanism. Frictional seals of any type must be avoid-
ed in design for prosthetic control, due to energy avail-
ability limitations and response criteria. The actuator
and valve design offered in the REDESIGN section will meet
the dynamic and static criteria set forth in the INTRODUCTION,
and should be useful both in the testing of damping patterns
and voluntary amputee control of the A/K prosthetic, and
in later application to a production A/K prosthesis.

APPENDIX:

Portescap Micromotor #28PL21-413 Specifications:

Nominal voltage	24
no load speed, rpm	5500
stall torque (in. oz.)	3.96
torque constant (in-oz/amp)	5.8
mechanical time constant (ms)	30
maximum power output (watts)	3.4

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