DESIGN AND CONSTRUCTION
OF A CABLE-CONTROLLED,
PARALLEL LINK MANIPULATOR

by
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DESIGN AND CONSTRUCTION
OF A
CABLE CONTROLLED PARALLEL LINK
MANIPULATOR

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DESIGN AND CONSTRUCTION
OF A
CABLE-CONTROLLED, PARALLEL LINK
MANIPULATOR

by

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ABSTRACT

A manipulator of novel configuration for undersea use has been de-
signed, modeled, simulated, built and tested. The control linkages are
cables, spooled by hydraulic motors, which operate in a parallel manner
to determine the position and attitude of the end effector. These work
in concert with a single telescoping spine in passive compression. At
the time of writing, a simplified, three-degree-of-freedom mechanism is
complete, operating under joystick control.

Time response and loading tests have been conducted both of indivi-
dual actuators and the manipulator as a whole, to evaluate its perfor-
mance in following input commands. Still in its infancy, the arm has
nonetheless proven the novel actuating principle on which the system is
based, demonstrating the power, speed and smooth action which can be re-
alized by the floating linkage in which no rigid member exists between
base and end effector. The final load capacity will be in excess of
1000 lb, slew speed over 200 in/sec, and incremental step size less than
.0005 in.

Thesis Supervisor: Dr. Thomas B. Sheridan
Professor of Mechanical Engineering and
Applied Psychology
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Chapter 1
INTRODUCTION

1.1 PURPOSE

The original task was to develop a manipulator to be fitted to the remotely-operated submersible SEAGRANT I donated by Perry Oceanographics Co. The purpose of the manipulator is to perform work such as cleaning, inspection, simple repair and assembly, and undersea welding. Through the use of different control regimes, namely direct manual operation, pre-programmed task execution and a hybrid of these two, supervisory control, the arm is to demonstrate in useful operation various control strategies developed in previous work by members of the Man Machine Systems Laboratory. A newly developed touch sensor will be used to supplement, and in murky water replace, camera vision for environmental feedback. The goal is to effectively guide the arm through tasks in worksite environments about which the human operator/programmer may have very imperfect descriptive information. A good example is the cleaning and inspection or welding of a seam in turbid waters, in which an accurate trajectory must first be established and then followed in the absence of visual feedback to the supervisor.

1.2 CONTENT AND ORGANIZATION

This thesis describes the analysis, synthesis and testing of a manipulator which has been developed to perform general undersea tasks. Due to its unconventional linkage geometry and, to our knowledge, entirely novel actuating mechanism, the arm is of interest in its own right, from the point of view of kinematic analysis, mechanical control
and dynamic performance. It is a fast, accurate and powerful positioner suitable for high-speed assembly and other dynamically demanding tasks.

The organization of this thesis and an outline of its contents are as follows:

(i) A set of target design specifications to assure satisfactory performance is derived. These include strength, reach, accuracy and speed criteria for the manipulator.

(ii) Two paths by which to meet these specifications are examined. A comparison is made between the "serial" and "parallel" linkage geometries, and motivation to explore the parallel-link configuration is given.

(iii) Three distinct physical implementations of the concept are presented and evaluated to arrive at the final design path.

(iv) An outline of the forward and inverse kinematics for the six-degree-of-freedom parallel linkage is presented.

(v) Discussion of the actual design process then begins with a description of a physical model which was constructed to demonstrate the actuating principle behind the chosen design solution.

(vi) Static loading analysis of the manipulator is presented, which later is used to size the base and actuating components to meet performance specifications.

(vii) Dynamic simulation is then described, wherein the interactive forces between the seven active links are simulated to estimate whether such coupling might cause instability.

(viii) Discussion of the hardware design, including the choice of hydraulic actuators, construction materials, system components, sealing
and protection methods, are presented in informative but not exhaustive detail.

(ix) Control of the manipulator is discussed in both wide and fine focus. The overall flow chart of command input, input processing and manipulator response is presented, along with a discussion of various command strategies. Then the dynamics of the individual servomotors are investigated to arrive at an estimate for the system time response and behavior under load.

(x) Looking beyond the realm of undersea tasks to the domain of high-speed precision assembly some simple strategies to improve system performance are discussed. The aim is to increase the manipulator speed and accuracy, while adding flexibility in the form of a variable load capacity.

(xi) The performance of the manipulator is evaluated according to the initial design criteria. The dynamic response and accuracy of the arm operating both freely and under load are investigated in detail.

(xii) Applications of the manipulator are discussed.
Chapter 2
DESIGN SPECIFICATIONS

2.1 SPECIFICATION TOLERANCES

Given the wide band of indeterminacy in both worksite conditions and tasks to be performed under the water, a good deal of initiative was required in deriving a set of specifications to assure overall satisfactory performance of the manipulator. Environmental vicissitudes such as violent turbulence or even seaweed impose incidental loading upon the manipulator which is difficult to estimate, while neither the precise nature of the tasks nor the tools to be used in their execution has been specified. The design specifications derived below are based on rough calculations and personal experience as a mechanic in working with tools and obstinate machinery under adverse conditions. The criteria for strength, reach, accuracy, speed, weight and depth serve as targets with reasonable tolerance margins rather than rigid constraints on the mechanical design.

2.2 STRENGTH

The first step was to estimate the strength which the manipulator must have to do its work. This refers both to the force and moment capacity of the arm. The dead weight of the tools to be used by the arm can be compensated by the addition of syntactic foam floatation, which has a bouyancy of 40 lb/cu.ft. Hence it is the operating loads involved in the use of the tools, such as torque and thrust from a drill, reaction to a high-velocity water jet (used for cleaning welds), moment from a wrench, or the recoil force of the stud welder, which determine the
strength specification for the arm. To such loads one must add a safety factor to account for drag due to water turbulence, which could be high in the vicinity of an offshore drilling rig, a jammed drill bit, or a rock which has fallen in the way. The dry weight of the stud welder, probably the largest tool, is seven pounds, while its recoil load will not exceed two pounds. Nominally, a snug-tight bolt requires ten ft-lb of torque, and the reaction torque to a 1/2" drill or cleaning brush is well below that figure. Twenty pounds force, twenty ft-lb torque was set as the strength requirement for the manipulator. These specifications allow a factor of safety of at least 100% for unforseen loads and a hostile environment. Note that load flexibility in the form of both a higher maximal capacity (opening a rusted valve or holding the submarine in a fixed position), and lower maximal capacity (work with delicate instruments), would be to great advantage. The arm which has been developed incorporates this feature of "programmable strength" in a natural way.

2.3 REACH

Next to be determined was the reach, or working envelope of the manipulator. First note that six degrees of freedom is the minimum number required to arbitrarily position an object in space: three degrees to locate a given point (e.g. \((X_0, Y_0, Z_0)\) coordinates of the center of mass), and three more to determine the object's orientation within the reference frame (e.g. \((\alpha, \theta, \psi)\) Euler angles describing rotation). In the interest of economy, "position" will be used to denote both position and orientation.

In one sense, the entire submarine can be viewed as a manipulator -
in itself capable of five degrees of freedom. From this viewpoint the arm need be nothing but an appendage either rigidly attached, or with one degree of freedom which simply grasps a tool while the sub manoeuvres to achieve the desired position. For certain tasks, such as cleaning or gathering samples, this regime is adequate. Practically, there are three problems with this simple scheme in carrying out the more complex tasks outlined in the introduction:

(i) Control over the submarine is not advanced to a sufficient degree as to achieve and maintain within useful tolerance (±0.5") the desired position of the end effector.

(ii) Even if such control be available the arrangement is clumsy: the sub pivots about its own center of mass, not about the manipulator wrist. Imagine painting with wrist, arm and shoulder in a cast.

(iii) Use of the sub's thrusters to effect minute changes in end effector position is highly inefficient.

Hence, for the majority of tasks to be undertaken, the strategy will be to first moor the sub to a reference position using anchor cables or a grabber arm, then utilize a dexterous manipulator to complete the work at hand. A natural scale of manipulator reach to sub length is 1:2, the human proportion. Why not make use of several thousand years of natural selection to fix the balance point between reach and strength of our general-purpose arm? Given the submersible length of six feet, a workspace characteristic dimension of three feet seemed suitable.

2.4 ACCURACY

The design specification for accuracy refers to two distinct char-
acteristics: absolute accuracy, and repeatability. Absolute accuracy is the dimensional tolerance which must be allowed the manipulator in responding to a command given in reference frame coordinates, i.e. the maximum discrepancy between actual position and command position. Repeatability is the maximum discrepancy between responses of the manipulator to identical position instructions. It measures how faithfully the arm will track a pre-taught trajectory. Note that the absolute accuracy serves as a lower bound for the repeatability specification.

Due to the imperfect knowledge one normally has of worksite configuration, and the positional indeterminacy of the submarine, where the tolerance is on the order of feet, the absolute accuracy of the manipulator is not paramount. One necessarily relies on visual or tactile feedback to position the manipulator. Nonetheless, one can conceive of touching three known points of a fixed undersea structure and thereby establishing a reference coordinate frame.

With respect to these coordinates absolute accuracy has meaning: one could drill holes, weld studs or follow seams at predetermined locations on the structure in the absence of vision. Of the tasks for which the arm is intended, that requiring greatest locational accuracy is the stud-welding operation: tolerance is on the order of 1/8". Regarding orientation, angular alignment of a drill or welding electrode is most critical: typically in the range of 5° solid arc. Hence for absolute accuracy, the target specifications were set at: positioning within 1/8", orienting within 5° solid arc. The repeatability was set at: 0.1" linear, 2° angular tolerance. This was dictated by the process of mating the arm with interchangeable tools to be kept in a tool rack.
within the arm's reach. The arm will serve as a dexterous socket with hydraulic/electronic interface to power a drill, brush cleaner, cutting tool or gripper.

2.5 SPEED

The final characteristic to be specified was speed. There being no high-speed undersea assembly tasks projected for the manipulator, and given the generally slow and deliberate pace of underwater activity, a high-speed lower bound of 80 in/sec was set. This figure is based on the submersible's top speed of 40 in/sec. The 2:1 ratio allows the manipulator to perform certain functions such as cleaning or object retrieval on the fly. Top speed, however, is of secondary importance compared with the manipulator's slow-speed characteristics.

In seam welding or assembly operations, smooth, jerk-free motion is paramount. This is particularly important in the manual control mode, wherein a human operator must develop a sense of feel for his remote appendage. This symbiotic relationship will be inhibited by stick-slip behavior and backlash in the manipulator. The term 'smoothness' really translates into two specifications:

(i) minimum steady speed

(ii) smallest step increment from rest.

On the basis of simple experiments with pencil and paper in tracing over a complex curve (connect the dots), a low-speed threshold of 0.1 in/sec and incremental step size of 0.1 inch seemed adequate.

2.6 DEPTH AND SIZE

The remaining design criteria are the depth rating, rest size and
weight of the manipulator. 500 ft was the nominal depth specification: the sub's 1000 ft tether comfortably allowed this much. Syntactic foam adds bulk even if it subtracts weight, so 100 lb wet weight seemed a rea-
sonable bound for the arm. No firm limit was placed on the rest size of the manipulator, but allowing it to protrude farther than a foot or two from the sub's steel frame when inactive, without a protective cage, would risk damage to the hardware—particularly while the sub is being loaded and unloaded from the mother ship. A retractible cover for the arm seems a good idea.
2.7 SUMMARY

To summarize the design target:

STRENGTH - 20 lb force, 20 ft-lb torque

REACH - Three to four feet.

ACCURACY - ABSOLUTE: linear= 1/8", angular= 5° (solid angle)

REPEATABILITY: linear= 0.1", angular= 2°

SPEED - MAX: 80 in/sec

THRESHOLD: 0.1 in/sec

STEP INCREMENT: 0.1 in

DEPTH RATING - 500 ft

WEIGHT (immersed) - 100 lb
Chapter 3

CONCEPTUAL DESIGN: PARALLEL vs SERIAL LINKAGE

3.1 DECISION TO EXPLORE

A set of functional specifications having been determined, the task was then to generate various feasible design solutions. These solutions could fall into one of two categories:

(i) A redesign of existing hardware, following established lines but incorporating useful modifications.

(ii) A completely new design approach, well off the beaten path, which might expose, along with new problems, advantages in strength, dexterity and simplicity.

My impression was that the anthropomorphic, serial-link type of manipulator has, quite exclusively, been developed to a high level of sophistication. Indeed, a brief survey of the literature revealed that many companies, including G.E., Kraft and Deep Ocean Technology offer manipulators for undersea use which would satisfy most of our performance criteria. On the curve of design improvements vs labor invested, however, the state-of-the-art seems well into the asymptotic region of diminishing returns: by and large, the arms have poor weight-to-strength ratios and suffer from joint stiction.

The decision was to explore a new path: jump to a new design curve right at the origin, where returns are steepest. There is a saying, "in the mind of the expert there are few possibilities, in the mind of the beginner anything is possible."1

1 Suzuki, Zen Mind, Beginner's Mind
My mind was very much the beginner's: eschewing the literature on the subject, I began investigating other means of positioning an object in space with the requisite force, range, speed and accuracy.

3.2 PARALLEL vs SERIAL

There is a different type of linkage capable of this manipulative function which fascinated my colleagues and me. If the human arm be referred to as serial, then the analogous term for this configuration would be parallel.²

To clarify: serial refers to a multi-stage mechanism built of successive links, usually hinged at rotary joints, which are actuated in a coordinated fashion to position the end effector (the final stage), Figure 1. Grossly, this is the configuration of the human arm, wherein the shoulder, elbow, wrist and knuckles are the joints and upper arm, forearm, palm and fingers are the links.³

Most manipulators have been fashioned after this pattern, with variations as to link length, degrees of freedom per joint, and actuating scheme (rotary vs linear) to control the joint angles. In contrast to this is the parallel configuration in which the actuator/link assemblies are not staged one atop the other, but function in tandem to determine the position of the end effector. Each link serves a role equal to its neighbors, as opposed to the staged, hierarchical structure of the seri-

² Marvin Minsky, Manipulator Design Vignettes
³ This is a simplification: some of the links, such as palm and forearm, behave as compound joints.
FIGURE 1 & 2
MANIPULATOR GEOMETRIES

(1) SERIAL

(2) PARALLEL
al configuration. It is a socialist arm! See Figure 2. There is nothing new about this configuration: a camera tripod and guy-wired radio antenna are commonplace examples. More commonplace, but less obvious, this is really the actuating principle behind our human muscles and joints: tendons and muscles acting in opposed pairs to twist or rotate a bone. The twist of the forearm-wrist linkage in turning a doorknob is a good example: there is no isolated pivot point, but rather a continuous twist due to the simultaneous contraction of many muscles along the arm. The human neck serves as a 5 degree-of-freedom (D.O.F.) manipulator, with head as end effector. Figure 3 shows the parallel configuration of the five muscles which determine its position and orientation.

In the manipulator which has been developed, a telescopic hydraulic cylinder replaces the rigid spine, thus adding a sixth D.O.F. There are advantages in strength and suppleness to this action over a link-joint-link assembly. Cockpit positioners for flight simulators have used this principle in the form of six telescopic links in parallel, and a 1968 M.I.T. B.S. thesis describes the design of a mechanical wrist, in essence a miniaturization of the flight simulator; see Figure 4.4

3.3 ADVANTAGES AND DRAWBACKS OF PARALLEL LINKS

Some advantages and disadvantages of the parallel configuration are as follows.

4 Bennett, A Mechanical Wrist For A Robot Arm
FIGURE 3
HUMAN NECK
FIGURE 4: BENNETT WRIST

KAZEROONI

WRIST

HYDRAULIC CYLINDERS
ADVANTAGES

(i) Very high strength/stiffness-to-weight ratios can be achieved because actuating links bear no moment loads, but are simple tension-compression members.

(ii) Because the actuators act in parallel, rather than series, to position the end effector, the force and moment capacity of the manipulator is much higher than that of the individual servomotors. A serial link arm is no stronger than its weakest link or joint actuator.

(iii) Manipulator inertia is minimal. Bulky links and massive motors are not being waved about in space—only the end effector. This results in economy of power and superior dynamic performance.

(iv) High accuracy results from the geometry: actuator errors are not multiplied by long linkage arms to determine endpoint tolerances: rather, the end effector offset is normally of the order of actuator offset, due to their direct connection.5

(v) Simplicity results from the direct connection between the actuators and effector. No complex drivetrains are necessary. This pays dividends in cost, reliability, and performance—friction and backlash are minimized.

(vi) The inverse kinematics are easy, allowing control of the manipulator with a small microprocessor.

DISADVANTAGES

5 Generally true, but not without exceptions. Near singular points (vanishing of the inverse kinematics’ Jacobian determinant) small perturbations in cable length may give rise to relatively large offsets of the end effector.
(i) The principal disadvantage of the parallel arrangement is the difficulty in reaching around a corner. This type of work, however, is not common, and the deficiency can be partially compensated by a "snout" extension attached to the end effector. Common examples of this are a socket wrench, bottle cleaner, carburetor wrench.

(ii) At extended reach, near the aforementioned singular points, loading of the manipulator can give rise to high tensile and compressive loads in the actuators. A loading analysis of the parallel manipulator is given in Chapter 6.

The advantages of the parallel geometry, together with the desire to explore new territory, persuaded us to base the manipulator design on that concept, with a novel modification as explained below.
4.1 FEASIBLE ALTERNATIVES

Examining the flight simulator and wrist model, Figure 4, one readily perceives the limitation in workspace imposed by the use of single-stage hydraulic cylinders as links. The stroke:length ratio of less than 1/2 severely restricts the working locus of the positioner: indeed, the mechanism is more of an orientor than manipulator. Three design alternatives to this actuation scheme were considered. They alternately replace the hydraulic cylinders with:

(i) Telescopic hydraulic cylinders

(ii) Telescopic rotary ball screws

(iii) Flexible cables used in conjunction with a central compression member.

Telescopic cylinders, Figure 5, seemed a natural choice, preserving simplicity while allowing stroke:length ratios of up to 3:1. However, they are available from stock in dump-truck size only, and were priced @$8,000 per unit on a special order basis.

Telescopic rotary screws, Figure 6, seemed a good idea except for the bulk of the motor required at the base or tip. This is a problem because each of the six screws must pivot freely through a solid angle of about 120°.

The cable linkage, Figure 7, required an additional member to provide preload, and introduces appreciable compliance into the system. Nonetheless, the drawbacks inherent in the other two schemes, along with
FIGURE 5
TELESCOPIC HYDRAULIC CYLINDER
FIGURE 6
TELESCOPIC ROTARY BALL SCREW
FIGURE 7
PARALLEL CABLE LINKAGE
the following advantages of the cable drive, made this the design choice.

4.2 ADVANTAGES OF A CABLE LINKAGE

(i) Simplicity. No cumbersome gimbals are necessary on the upper plate, to which an end effector will mount, save one at the center for the compression member.

(ii) There is little interference between the six cables as the manipulator moves about. Both other schemes pose greater clearance problems between adjacent links.

(iii) Though some price is paid in terms of stiffness, the low inertia of the cables as actuating links presents a great advantage not only over the two competing designs, but also with respect to standard serial-link arms, which normally include actuators at each joint.

(iv) This arrangement permits mounting of the main actuators beneath the manipulator base. This not only results in a simpler, more lightweight structure, but also a better protected manipulator. In the event of a collision or other fatal mishap, our actuators are not out on a limb. Cable and even "spine" replacement would be a fast and inexpensive repair. (The spine might consist of a series of single-stage hydraulic cylinders.)

(v) There is less friction inherent in the cable drive than either the linear hydraulic or screw systems, both in terms of the actual drive mechanism (seal stiction in hydraulic cylinders, coulomb friction in the screws) and the joints at either end of the actuators. This will translate into a smoother mechanism with a lower speed threshold and smaller incremental step size (no coulomb "lurch").
(vi) The cable links, stored on spools, collapse to a very small rest size while permitting great extension of the active manipulator. At rest, the spine can simply fold down onto the base.

(vii) The cable system with opposing telescopic spine appears to be an entirely new actuation scheme. Unlike tendons pulling in pairs against a bone, there are no rigid members connecting the end effector to the base. It is a floating configuration with cables pulling against each other and the force of the central spine. The spine compression can be preset at a given value, or varied on-line, to provide a preload to the manipulator. This preload, independent of supply pressure to the cable actuators or their controller gains, serves in some ways as a stiffness parameter, greatly influencing the loading characteristics and dynamic response of the manipulator. Apart from its function as an undersea manipulator, the control and dynamic behavior of this arm stand on their own as interesting subjects for investigation. The decision having been made to explore this new path, the manipulator project became equal parts pragmatic design and investigative experimentation.
Chapter 5

KINEMATIC ANALYSIS

5.1 FORWARD AND INVERSE KINEMATICS

There are two mappings which characterize the geometry of a manipulator. In order to translate reference frame commands into the language of the manipulator, and to investigate the dynamic behavior of the system, the two mappings must either be found in closed form or synthesized by computer search. These mappings, known as the forward and inverse kinematics, relate the natural coordinates of the manipulator to reference, or absolute coordinates. The forward kinematics transform actuator coordinates into the reference coordinates of the end effector; this map tells you where you are if the attitude of each motor is known. The inverse mapping says how far to turn each motor if you want to reach a given position. Interestingly, the nature of the kinematic analysis of the parallel linkage is in many respects the dual of that of the serial configuration.

For a serial arm with N links, the natural coordinates are the set of joint angles:

\[ \theta = \{ \theta_1, \theta_2, \ldots, \theta_n \} \]

which are directly determined by actuator position. Hence the forward kinematics is the mapping:

\[ F: \theta \rightarrow X \]

where \( X \) is the six-tuple representing the position of the end effector. This representation depends on the choice of reference coordinates.
In Cartesian form, using Euler angles to specify orientation,

\[ X = \{x, y, z, \phi, \theta, \psi\} \]

The inverse mapping is:

\[ G: X \rightarrow \theta \]

The forward map \( F \) is a straightforward series of linear operations involving matrix transformation to account for link rotation, and vector addition to specify the position of consecutive joints. \( N \) iterations of this process yields the position of the end effector at the end of the chain. The inverse map \( G \), determining joint angles from position, is less straightforward. For a general serial linkage the map \( G \) cannot be specified in closed form, but only approximated by a search algorithm implemented on a computer. The map is plagued by singular points which may have multiple solutions or none at all. An exception to this occurs when all rotational axes are either parallel or intersect: indeed, many manipulators are purposely constructed under such restrictions to render \( G \) more tractable!

The natural coordinates for the parallel linkage are the link lengths, Figure 7.

\[ L = \{L_1, L_2, \ldots, L_6\} \]

will be used to denote the six cable lengths. Note that the extension of the central compression member, hereafter known as the spine, is not needed in order to specify the state of our system: it simply serves to maintain tension in the cables. The kinematics of this configuration are the dual of the serial in that here it is the forward map which is not closed, while the inverse is straightforward arithmetic. We outline both mappings, beginning with the idealized inverse kinematics.\(^6\)

\(^6\) For simplicity, the discussion concerns an idealized model where the six links terminate in pairs at three points on the base and upper plate. This is a reasonable first approximation, as the actual tie point separation is 1", compared to the 16" equilateral base triangle in which the three tie point groups are arranged. Accounting for the actual separation merely involves a little additional arithmetic.
First, some notation must be introduced. Let \( X \) be given in spherical–Eulerian coordinates

\[
X = \{ r, \alpha, \beta, \phi, \theta, \psi \}
\]

This representation lends itself naturally to the working envelope of the manipulator, which is a conical section between two concentric spherical shells, Figure 8. Coordinates \( r, \alpha, \beta \) specify the position of the midpoint \( Q \) of the upper plate, while \( \phi, \theta, \psi \) are the Euler angles describing the orientation of a coordinate frame \( Qxyz \) rigidly attached to this plate. Let \( OXYZ \) be the global, inertial reference frame, with origin \( O \) at the base midpoint, Figure 9.

The links in our idealized model originate at three points \( I, II, III \) on the base and connect to the upper plate at \( A, B, C \). All points of connection behave as spherical joints, allowing the links to pivot and twist freely. In the physical system these links are taut cables, and the spine pivots freely on gimbals at \( O \) and \( Q \).

5.2 PARALLEL INVERSE KINEMATICS

\[
G: \quad X \rightarrow L
\]

\[
G = \{ G_1, G_2, \ldots, G_n \} \quad \text{where} \quad G_i(X) = L_i
\]
FIGURE 8
CONICAL WORKSPACE OF MANIPULATOR
FIGURE 9
REFERENCE FRAMES
The following is an outline of the inverse kinematics for the parallel linkage, whereby the cable lengths required to achieve a given position are calculated. The simplicity and speed of this algorithm are among the principal advantages to the parallel geometry.

(i) Given a command position \( \mathbf{X} \), form the Euler rotational transformation \( T_{\phi\theta\psi} \) which describes the orientation of the plate-fixed reference frame \( \mathbf{Qxyz} \).

(ii) Compute the plate vertex vectors \( \mathbf{A} \), \( \mathbf{B} \), \( \mathbf{C} \):

\[
\begin{align*}
\mathbf{A} &= T_{\phi\theta\psi} \mathbf{Q} + \mathbf{Q} \\
\mathbf{B} &= T_{\phi\theta\psi} \mathbf{Q} + \mathbf{Q} \quad \text{eq(i)} \\
\mathbf{C} &= T_{\phi\theta\psi} \mathbf{Q} + \mathbf{Q}
\end{align*}
\]

where, e.g., \( \mathbf{QA} \) is the constant vector, expressed in coordinates \( \mathbf{Qxyz} \), connecting the plate midpoint \( \mathbf{Q} \) to vertex \( \mathbf{A} \).

(iii) Calculate the six norms

\[
\begin{align*}
L_1 &= |\mathbf{A} - \mathbf{I}| \\
L_2 &= |\mathbf{A} - \mathbf{II}| \\
L_3 &= |\mathbf{B} - \mathbf{II}| \\
L_4 &= |\mathbf{B} - \mathbf{III}| \quad \text{eq(ii)} \\
L_5 &= |\mathbf{C} - \mathbf{I}| \\
L_6 &= |\mathbf{C} - \mathbf{III}|
\end{align*}
\]

The inverse map \( G \) is thus a continuously differentiable mapping, defined for all \( \mathbf{X} \).\(^7\)

\(^7\) A mapping \( F: \mathbb{R}^n \rightarrow \mathbb{R}^n \) is said to be continuously differentiable on a set \( U \) in case each of the partial derivatives \( \frac{\partial F_i}{\partial x_j} \) is continuous in \( U \) for \( i = 1, 2, \ldots, n \), \( j = 1, 2, \ldots, n \).
5.3 PARALLEL FORWARD KINEMATICS

\[ F: \mathbf{L} \rightarrow \mathbf{X} \]

The forward mapping from link lengths to position are neither globally well-behaved nor defined. Indeed, it is not obvious that the six link lengths \( \mathbf{L} \) uniquely determine the position of the plate. Grubler's criterion suggests that, at least locally, the linkage is rigid:

\[
\text{Number of D.O.F} = \text{six}^8
\]

\[
= 6\times(n - 1) - 3\times P - 6; \quad "P" \text{ for pinned (ball) joints}
\]

\[
= 6\times(8 - 1) - 3\times 12 - 6
\]

\[
= 0
\]

Local rigidity, however, does not imply global uniqueness; multiple solutions \( \mathbf{X} \) corresponding to a given \( \mathbf{L} \) are possible. Two further mathematical constraints are necessary to ensure uniqueness:

(i) The \( z \)-coordinate of the end effector is positive (upper plate must be above the base).

(ii) The scalar product between the radius vector \( \overline{OQ} \) and upper plate normal \( \hat{\mathbf{n}} \) must be positive (upper plate cannot pivot so far as to

---

\(^8\) Subtracting the degrees of freedom corresponding to twist in each link about its centerline, which is of no consequence to plate position.
hit the compression member: its normal can be inclined at most 90° to
the direction of the spine).

Note that both of these criteria are also physical constraints on the
hardware, not just mathematical artifacts.

Before proceeding further with a practical description of the forward
kinematics, a purely mathematical observation concerning this map-
ning as the inverse of map \( G \) provides some useful insight. Consider a
state \( x_0 \) of our system where the Jacobian determinant is
non-vanishing:

\[
\left| J \right|_{x_0} = \begin{vmatrix}
\frac{\partial g_1}{\partial r} & \frac{\partial g_1}{\partial \alpha} & \ldots & \frac{\partial g_1}{\partial \phi} \\
\frac{\partial g_2}{\partial r} & \frac{\partial g_2}{\partial \alpha} & \ldots & \frac{\partial g_2}{\partial \phi} \\
\vdots & \vdots & \ddots & \vdots \\
\frac{\partial g_6}{\partial r} & \frac{\partial g_6}{\partial \alpha} & \ldots & \frac{\partial g_6}{\partial \phi}
\end{vmatrix} \neq 0
\]

Then the fundamental inverse function theorem guarantees the existence
of a continuously differentiable inverse \( F = G^{-1} \) in some neighborhood
\( W \) about \( L_0 = G(x_0) \).\(^9\)

This guarantees the local existence and differentiability of \( F \), the for-
ward kinematics, in some neighborhood of a non-singular state. This is
a more rigorous demonstration of local rigidity than Grubler’s criteri-
on. It also demonstrates that locally, the six links \( L_1 \) may be inde-
dependently specified: they don’t fight one another. The identifica-

\(^9\) See, e.g., Taylor, Mann: \textit{Advanced Calculus}, Chapter 12.
of all singular points is a topic for further work: however, an example of such degenerate positions, where a change in position may be effected with negligible change in cable length (the manipulator loses physical rigidity) is shown in Figure 10.

5.4 PRACTICAL IMPLEMENTATION OF FORWARD KINEMATICS

This analysis guarantees the existence of F locally, but provides no insight into how it may be determined generally. Practically, the solution of the forward kinematics involves solving the six simultaneous quadratic equations (ii) together with three constraint equations. These specify that the upper tie points A, B, C are vertices of an equilateral triangle of side p:

$$|A - B| = |A - C| = |B - C| = p \quad \text{eq(iii)}$$

Two techniques were considered for determining the position $X$ of the manipulator end effector from the link lengths $L$. The first is a direct integration:

Given $dL = J^*dX$

$$dX = J^{-1}*dL$$

Hence $X = \int_{L_0}^{L} J^{-1}*dL + X_0$

The position may thereby be estimated to high accuracy, albeit at the expense of successively calculating $J^{-1}$.

The second method is a search based on a geometrical interpretation of (ii) and (iii) as follows:

Equations (ii) describe six spheres of radius $L_i$, centered two each at the fixed base vertices I, II, III. Intersecting these spheres yields three circles whose centers are on lines containing the base triangle I,
FIGURE 10
"SNAPPING" INSTABILITY
II, III and which lie in planes perpendicular to these lines. The upper plate must be positioned with each vertex lying on the corresponding circle, Figure 11. This position will be $X$, the kinematic inversion of $L$. One strategy to find this position is to form a sphere of radius $P$ with center $Q$ on one of the circles, and observe the distance between intersection points of this sphere with the other two circles as $Q$ moves along the first. There may be two, one or no point of intersection with each. A candidate solution is found when the intersection points are a distance $p$ apart, for then the center $Q$ will form an equilateral triangle with each of the two intersection points. Note that the candidate solution, through satisfying the constraint equations, is not necessarily physically possible, e.g. to each $X$ there corresponds a mirror solution with the plate underneath the base. In addition to multiple roots is the possibility of no solution at all, e.g. for

$$L_1 > L_2 + s$$

where $s$ is the separation of base points I, II. Homayoon Kazerooni implemented this search algorithm on the computer. Generally, if the search "seed", or initial guess, is in the neighborhood of a physical solution (e.g. the last position of the manipulator) then the algorithm quickly converges to a unique, realizable solution. The forward mapping $F$ is locally well behaved. Globally, however, this nice behavior does not hold. Unrealizable solutions appear regularly, so that "truth tests" for candidate position solutions are necessary.

10 Homayoon Kazerooni is a member of our laboratory who developed the software for the computer simulation of the manipulator.
FIGURE 11
FORWARD KINEMATICS
"SEARCH"
Of the two mappings, forward and inverse, it is the inverse kinematics which must be computed on-line for real time trajectory control of the manipulator. Hence the tremendous computational advantage of the parallel configuration over the serial geometry: a microprocessor will be adequate for on-line trajectory control of this manipulator. For the dynamic analysis of the manipulator, however, both forward and inverse maps are necessary. The final section of the next chapter discusses a dynamic simulation of this system.
Chapter 6
DESIGN MODELING, ANALYSIS AND SIMULATION

The first three steps towards the realization of a functional manipulator were construction of a physical model, a static loading analysis, and a simple dynamic simulation of the system. These were logical steps taken to anticipate and design around problems which would inevitably arise in the hardware implementation of the parallel configuration; in particular those associated with the novel actuation scheme of a "floating" end effector.

6.1 PHYSICAL MODEL

A simple, working model was built to demonstrate the principle behind a parallel linkage, and the feasibility of its physical realization in cable form, Figure 12. Frankly, most colleagues were doubtful of the whole approach. The model consisted of a plywood base and top plate separated by a rigid spine pivoting on universal joints at either end. Bicycle control cables were kept in tension by constant-force springs (S), and locked in position by a quick-release braking device (B). The cables were guided from the spring to nylon base blocks (N) via bicycle brake cable housing. The model was very instructive in the following respects:

(i) Demonstrated the high load capacity, both force and moment-wise, of the configuration.

(ii) Impressive stiffness was achieved in spite of the light cable links.

(iii) Though of limited mobility due to the rigid spine, the model
nonetheless illustrated the large workspace and dexterous movement admitted by the configuration.

(iv) Helpful in giving me something to look at while writing equations. Through hands-on experience with the model, I gained an intuitive feel for the mechanism which was helpful when designing the hardware for the actual manipulator.

Two negative aspects of the model were in some ways the most instructive lessons. When the model was first constructed, the spine pivot was above the plane containing the points on the base from which the cables emerged. As illustrated in Figure 13 this caused a "snapping" instability at certain limit orientations which caused considerable alarm until the problem was diagnosed: singular points in the kinematics' Jacobian J (discussed in Section 5.3) had inadvertently been introduced, admitting multiple solutions to the forward mapping. The top plate was snapping precipitously from one root to another! Fortunately, the solution to this problem was simple: the singularities disappeared when nylon base blocks were added to raise the plane of the cable tie points above the spine pivot. It is fortunate that these singularities were discovered at an early stage!

The second instructive annoyance was the high friction in the cable guidance system: the non-linear disturbance would spell disaster for any control regime aimed at smooth position command of the manipulator. Second only to simplicity, low friction was of the highest priority in design of the hardware.

6.2 STATIC LOADING ANALYSIS

Having demonstrated the physical feasibility of a cable-controlled
FIGURE 13
MULTIPLE ROOTS
parallel manipulator, the next step was to construct a mathematical model to investigate the static loading characteristics of the system. A moment's reflection, or a few minutes play with the physical model, reveal that a given load induces stresses in the cables and structure which depend greatly upon the position of the arm. Given the design specifications of working reach and load capacity, what cable, spine preload, and resultant structural stresses should one expect?

The mapping:

\[ S: \{X, D, P\} \rightarrow \{T\} \]

yielding cable tensions for a given loading and position of the manipulator and fixed preload in the spine had to be described in order to size the hardware.

\[ X \text{ denotes the manipulator position,} \]
\[ D = (F_x, F_y, F_z, M_x, M_y, M_z) \]
denotes the resultant of all external forces and torques at the upper plate midpoint. \( P \) is the (controlled) preload of the spine and

\[ T = \{T_1, T_2, \ldots, T_6\} \]
denotes the six cable tensions. Note that, at this early stage in the design process, two more "dimensions" could be profitably included in the domain of \( S \):

(i) Absolute dimensions \((R,r)\) of the base and upper plate, Figure 14.

(ii) Base configuration, Figure 15. Both the absolute and "distor-tional" geometry of these plates influence the loading characteristics. Generally, increasing base radius leads to lower cable tensions; increasing the ratio \( R:r \) increases the force:moment load capacity ratio; distorting the base (and/or its mirror image, the plate) configuration

-50-
FIGURE 14
BASE AND UPPER PLATE RADII

FIGURE 15
BASE CONFIGURATION SUITABLE FOR VERTICAL LOADING
results in asymmetrical loading capabilities. This latter would be to advantage in applications where the working load distribution has a spatial bias, e.g. predominantly vertical lifting loads, or moments due to loosening rusted bolts or values. For simplicity the analysis was restricted to a symmetrical configuration. Some trial-and-error play revealed that a base radius of about one foot and upper plate radius of 4" would maintain cable tensions below 200 lb for the given design load and working envelope while minimizing the bulk of the end effector plate.

For this fixed geometry the mapping S may be described as follows:

(i) Form the spine force vector \( \mathbf{P} = \mathbf{P} \hat{r} \), where \( \hat{r} \) is the unit vector in the direction of the spine (determined by the first three components of \( \mathbf{X} \)): \( \mathbf{r} = \mathbf{Q} - \mathbf{O} \)

(ii) Calculate \( \mathbf{F} = \mathbf{P} + \mathbf{F}_{\text{load}} \).

(iii) Form the six-dimensional load vectors

\[ \mathbf{D} = (\mathbf{F}, \mathbf{M}) \]

\[ \mathbf{P}^* = (\mathbf{P}, 0, 0, 0) \]

(iv) Form the six cable-force vectors \( \mathbf{T}_1 \) of unknown magnitude but fixed direction:

\[ \mathbf{T}_1 = \mathbf{T}_1 \times (\mathbf{I} - \mathbf{A})/|\mathbf{I} - \mathbf{A}| \]

\[ \mathbf{T}_6 = \mathbf{T}_6 \times (\mathbf{III} - \mathbf{C})/|\mathbf{III} - \mathbf{C}| \]

(v) The force balance is then:

\[ \sum \mathbf{T}_i = -(\mathbf{F} + \mathbf{P}) \]

(vi) The moment balance is:

\[ \sum p_j \times (\mathbf{T}_{2j-1} + \mathbf{T}_{2j}) = \mathbf{M} \]

where \( p_1 = \mathbf{A} - \mathbf{Q}, p_2 = \mathbf{B} - \mathbf{Q}, p_3 = \mathbf{C} - \mathbf{Q} \)
(vii) Combining the two sets of three linear equations for the six
unknown tensions $T_i$ yields the compact matrix formula:

$$S^{-1} * T = D + P^r$$

(viii) The $6 \times 6$ matrix $S^{-1}$ is then inverted to solve for the tens-
sions:

$$T = S * [D + P^r]$$

Note that the transformation matrix $S$ depends only on the position $X$
of the manipulator, $S = S(X)$. Note also the skew-linear dependence of
$T$ on the spine preload $P$.

There is an important physical constraint which must be placed on
the tension vector $T$: each component $T_i > 0$, i.e. our cables will
not support a compressive load! Hence for a given position $X$ of the
manipulator and applied load $D$, the spine preload $P$ must be increased
so that this condition is met. In fact, (viii) and (i) can be combined as

$$T = [S * D] + P[S * \hat{r}^-] ; \quad \hat{r}^- = (\hat{r}, 0, 0, 0)$$

where the quantities $[ ]$ are constants independent of spine preload.
Physically, it is clear (and can be mathematically demonstrated) that
for all points $X$ in the locus of the manipulator, the vector $S * P^r$
must have positive components: i.e., setting $D = 0$, the tension in
each cable will increase linearly with $P$.

Therefore, the minimum preload $P^*$ which is necessary to maintain
stiffness in the cables and keep our manipulator rigid is given by

$$p^* = \max \left\{ \begin{array}{c}
\text{negative component of } [S * D] \\
\text{corresponding (positive) components of } [S * r']
\end{array} \right\}$$

In case the minimum component of $S * D$ is positive, no preload is ne-
cessary in the spine.

Note further that, in general, for a fixed preload $P$ in the spine the magnitude of the applied load $D = |D|$ is limited by a maximal value $D^*$ beyond which the manipulator loses its rigidity:

$$D^* = \min \left\{ \begin{array}{c}
\text{corresponding component of } [S * P'] \\
\text{negative component of } [S * \hat{D}]
\end{array} \right\}$$

This serves as a simple loading-governor for the manipulator. It is a safety feature which will permit the use of the arm in delicate tasks wherein some maximum force $D^*$ must not be exceeded, without adjustment of controller gains and consequent degradation of the system's dynamic performance. It also adds flexibility to the system, easily enabling on-line adjustments of the manipulator's load-handling capacity by regulation of the hydraulic pressure to the spine.

Kazerouni implemented the mapping $S$, along with the inverse kinematics $G$, on the computer. For the base radius of 11" and plate radius of 4", Figures 16 illustrate quantitatively the loading characteristics of the arm in different positions. The arm will probably be mounted to the submarine as shown in Figure 17.

For a given loading state of the manipulator to be realizable the criteria which must be met are $0 < T_i < 240$ lb, $i = 1, 2, ..., 6$. Physically this means that no cable can support a compression load, and that the actuators saturate at 240 lb tension.\footnote{At $P_{\text{supply}} = 3000$ psi and spool diameter = 1.75".}

In general, to maintain tension in all cables the minimum preload $P^*$ corresponding to the load $D$ and manipulator configuration $X$ must be

\footnote{At $P_{\text{supply}} = 3000$ psi and spool diameter = 1.75".}
FIGURE 16
LOADING EXAMPLES

- LIFTING 1000 LB

- TORQUEING 70 FT LB

- 70 LB
Supporting 30 lb
at 60° inclination,
5' reach
FIGURE 17
MANIPULATOR MOUNT ON SUBMARINE
maintained: with this preload $T_{\text{min}} = 0$. However, in certain instances this gives rise to $T_{\text{max}} > 240$ lb. In this event, the load magnitude has exceeded the capacity $D_{\text{limit}}$ of the manipulator. The examples do not necessarily depict the limit load (which may be calculated as in the previous section) but do illustrate the high load capacity of the arm, even at a 5° reach. Figure 16(a) shows the manipulator at full extension lifting 1,000 lb. This load is far beyond the capacity of any single actuator, but is possible because the six motors act in tandem. The manipulator could push with equal force by increasing the spine preload. Any radial load adds directly to the spine preload (with sign accounted for) and may be compensated thereby. Perpendicular and moment loads, however, must be sustained through differences in opposing cable tensions. Figures 16(b)-(f) illustrate this type of loading. Note that in its upright position the perpendicular load capacity increases from 50 lb at 5 feet to 120 lb at 2 feet (fully retracted). Figures 16(c),(d) show the arm subject to moment loading, which occurs when a force is exerted whose line of action does not pass through the upper plate gimbal, e.g. leverage from an extended weight, or in applying torque to a valve or bolt. Maximum torques on the order of 100 ft-lb can be developed with actuators 1/5 that capacity. This is a tremendous advantage of the configuration over the serial linkage with rotary actuators whose individual capacities cannot be exceeded. Figures 16(e),(f) show the positional dependence of load capacity. When fully extended and inclined at 30° latitude, (weakest configuration) the force capacity is 30 lb; the moment capacity remains high. The parallel configuration behaves in many respects like a beam of high area in-
ertia about any axis, sustaining perpendicular, axial, and moment loads through tension in its widely spaced fibers and compression in the spine. Note that while able to support a load of 30 lb in position (f), its capacity in position (a) exceeds 1000 lb!

A final observation concerning loading analysis is that the inverse map $S^{-1}$ (eq. (vii)) allows rapid calculation of the applied load $D$ given the spine preload $P$ and cable tensions $T$. This calculation would be of use for static force sensing; such feedback permits a human operator some feel for the task at hand.

6.3 DYNAMIC SIMULATION

Because of the highly coupled nature of the configuration, wherein each cable-actuator is subject to a load $T_i$ which depends on the other $T_j$, the spine force $P$, and the plate load $D$ (which includes inertial forces), the issue of stability required investigation. Before purchasing expensive motors and constructing the hardware, some indication that the completed system would not tremble uncontrollably was necessary. Homayoon Kazerooni synthesized a simulation program for the manipulator, using as tools the Fortran code for the forward and inverse kinematics and static load-analysis which he had written.

The simulation was based on a simple, linearized model of the servo-valve - hydraulic motor dynamics which is detailed in chapter 7. Note that in themselves the kinematics form a "dry" description of the manipulator, containing no information about stability of dynamic response of the system. They are merely a translation of linkage geometry into a set of mathematical relations. However, given this description one can determine how forces and translations in actuator coordinates (e.g. $T$
are resolved into inertial-frame motion and forces \((X, D)\) and vice-versa. This is precisely the use of mappings \(F, G, S\) and \(S^{-1}\) which have been described. Using these mappings as linking tools, one can decompose a complex system into components which can be modeled according to their individual inertial, stiffness and dissipative characteristics. The response of the components to their resolved loads/displacements can then be calculated, and the subsystems in their new states then linked back together to yield the response of the unified system. This process is iterated over a suitable period using time increments \(dT\) several times smaller than the smallest time constant of the various components.

In the simulation of the manipulator, the linkage was decomposed into the six cable-motor-valve assemblies, Figure 18. No load mass was included as this would have complicated matters considerably, and for masses on the order of one kilogram, the inertial loads \((-m\ddot{x})\) would be an order of magnitude lower than the cable forces due to the system preload \(P\).12

The simulation revealed that the manipulator, as modeled, was a stable system. Design and construction of the hardware could now begin: this is the topic of the next chapter.

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12 For the actual system, accelerations of 100 \(\text{ft/sec}^2\) were measured. The induced inertial load for a 1 kg mass (e.g., a drill) is under 5 lb — about 3% of the nominal cable load of 150 lbs.
Six Coupled "Second Order" Systems

Cable, Motor

Mappings F, G, S implemented on PDP/11 by Homayoon Kazerooni

System is stable, as modelled
Chapter 7

HARDWARE DESIGN

This chapter describes the construction of the arm: the design problems and their solutions. The manipulator has been constructed not as a prototype, but as a functional piece of hardware to perform work under the water. This is because in terms of both cost and effort, there were little savings to be realized by building a fully operative toy: the final design is so simple that essentially all components present on the full-scale arm would need to have been included in a prototype.

7.1 POWER MEDIUM

The first design decision was the choice of an energy transfer medium. Both electric and hydraulic power are feasible means, which have been used to drive other undersea manipulators and submersibles. The choice of hydraulics was made for three reasons:

(i) The Recon V ("Seagrant I") submersible is powered by hydraulic motors, so that a 10 gpm, 3000 psi power supply is available — including extra ports used to power an earlier manipulator.

(ii) High-pressure hydraulic actuators have the highest power density (horsepower/in\(^3\)) of any drive available. The 0.5 in\(^3\), 3 hp motors selected have a higher power output than electric motors three times their size. Hydraulic actuators are also better suited for low speed/high torque applications than are electric motors. Given the 100 – 200 lb cable loads anticipated, reasonably sized electric motors would require some form of gear reduction, while a hydraulic actuator can be
used as a direct drive for cable spools. Also, hydraulic actuators do not themselves dissipate energy when stalled, so cooling is no problem.

(iii) In contrast with an electric motor, most pressure-compensated (i.e. return pressure = ambient + 7 psi) hydraulic actuators can be directly immersed in water without sealing problems. In fact, some actuators are powered by sea water.

7.2 LINEAR vs ROTARY ACTUATORS

The next issue was linear vs rotary hydraulic actuators. Because of the small hydraulic pump on the submersible (10 gpm @ 3000 psi), it was imperative that leakage across the manipulator actuators be kept to a minimum: preferably less than 20 in³/min per actuator. This specification was easily met by linear cylinders, but proved a stringent requirement for the rotary motors. The rotary actuators are commonly available for heavy industrial continuous-duty use, and tolerances are such that leakage across the pistons or rotors is generally excessive. The cost of a suitable hydraulic motor was on the order of $1,000 — approximately ten times that of a linear actuator.

For the following reasons was felt that Planet Products' ball piston motors were worth the expense:

(i) Given the specification for working reach, at least 30" - 40" of cable travel would be necessary. Clearly, the use of a linear actuator to provide this stroke would be awkward, involving gears or pulley tackles to provide suitable cable travel from a reasonably sized cylinder (six to ten inch length). Such transmissions add complexity, increase friction and introduce backlash.

(ii) Continuous rotation motors are compact, and strong enough to
FIGURE 19
PLANET MOTOR, INCLUDING SERVOVALVE AND POTENTIOMETER
provide a direct cable drive. The 0.5 in³ Planet motors, Figure 19, provide 140 in lbs of torque at a 2000 psi drop.

(iii) There is no high-pressure seal stiction in the ball-piston motors, which can rotate smoothly below 1/4 rpm. Figure 20 is a cross-sectional view of the motors. The close-tolerance ball pistons result in high efficiency: less than 10 in³/min leakage. The through-shafts are fitted with heavy-duty radial bearings to support high cable loads, and permit direct mounting of the feedback transducers at the base of the motor. The servomotors form an integral package including manifold, servovalve and feedback potentiometer.

Protection: The cast-aluminum motors will run in pressure-compensated oil, and hence can operate at virtually any depth. The shaft seal prevents excessive "weeping" of oil to the surrounding environment. Cross-port relief valves are included as a safety feature in case of sudden shock loading to the motor.

7.3 SERVOVALVES, TRANSDUCERS AND AMPLIFIERS

Controlling the flow of fluid to these motors are Cadillac Gage servovalves mounted on the motor manifold. This arrangement leads to a bulkier motor, but improves response by minimizing the dynamics of fluid inertance and line capacitance: the system is lighter and stiffer. The servovalves were removed from 20-year-old Polaris missiles. Their bandwidth of 60 Hz is half that of a Moog A076 valve, and are physically twice the size: but at $79 apiece they were about 1/20 the cost. As demonstrated in performance tests (chapter 9) the valves are entirely satisfactory. Their 2 gpm rated flow and low threshold current permit the actuators to meet both the high and low speed requirements for the
FIGURE 20
BALL-PISTON MOTOR CUTAWAY VIEW

Pintle shown partially withdrawn from rotor. Normal position of pintle is inside rotor.

Eccentricity between anti-friction bearing and rotor.

- Pressurized Fluid
- Case Drain Fluid
manipulator. The valve flow gain, pressure gain and droop have been measured directly.

Protection: The valves are protected by a valve cover, and will operate in a pressure-compensated oil bath. The driving current to the valve torque motor coils (-8ma < i < +8ma) is provided through a sealed, locking Cannon 6-pin connector mounted to the cover.

To provide position feedback, 10-turn, film type CIC potentiometers are fitted to the rear of each motor shaft. These 0.05% linear, low-noise analog transducers are a good match for the simple analog valve driving cards purchased from Moog, Inc. These cards consist of two operational amplifiers: a voltage summer at the input and current drive on the output stage. For the time being they provide proportional position control for the actuators; a closer look at control is given in the next chapter.

Protection: The feedback pots are protected by a cover at the rear of the motor, and will be immersed in compensated oil. Electrical connections are through sealed Cannon connectors. The drive cards will be housed inside the submersible's sealed pressure-cannisters.

7.4 CABLE

The cable selected is 1/8" diameter 7x19-strand stainless, coated with nylon for abrasion resistance. The cable's breaking strength of 2000 lb yields a nominal factor of safety of ten over the maximal working load of 200 lb tension. This figure is misleading, however, because the fatigue life of the cable will be under $10^6$ cycles owing to the use of cable pulleys and spools of relatively small radius. The cables should therefore be replaced periodically.\(^{13}\) The use of two

\(^{13}\) Sava Industries, the cable manufacturer, recommends a minimum bending radius of 25 cable diameters for maximum fatigue life. Consultation with the factory revealed that this is a very conservative figure, and that our ratio of 16:1 should provide good service.
smaller-guage cables in tandem was considered, as this would permit smaller pulleys. A third pulley to balance tension would then be necessary, however; this increase in complexity seemed unjustified at so early a stage of the manipulator's development.

In spite of its flexibility, the stiffness of the cable is impressive: total cable stretch under a 200 lb load is less than 0.1" at a 50" extension.\textsuperscript{14} A discussion of the inaccuracies thereby introduced and their compensation is given in Section 8.3.

**Protection:** The stainless should provide good service, but periodic inspection is necessary as the cable will tend to corrode from inside to out. Stainless stops swaged to the cable will be used at both ends: currently zinc has been used to allow manual crimping with a special tool.

7.5 MATERIALS

Before further discussion of hardware, a brief note about material selection is in order. Three materials were used primarily in the construction of the manipulator: aluminum 6061-T6, 316 stainless steel, and delrin. The aluminum and steel alloys both have good corrosion resistance, but due to the cathodic effect in salt water, must be kept as

\textsuperscript{14} Note: Total stretch = stretch due to relative motion of strands (non-linear) + elongation of strands. It shows some hysteresis.
isolated from one another as possible. These metals were used only where necessary, in components of high stress and restricted size: examples are pulley shafts and gimbal forks. Wherever possible, delrin has been used because of its high strength, low weight, and machinability. Delrin is an acetal thermoplastic whose tensile yield strength of about 10,000 psi is comparable to that of aluminum, while its specific gravity of 1.5 is half that of the metal. It is ductile (20% elongation at break) with water absorption of 0.20%/24hrs — much lower than nylon’s 1.5%. Delrin has good abrasion resistance — it is used commercially in the manufacture of light gears — while having self-lubricating properties which make it suitable as a bearing surface. It is free machining; without nylon’s tendency to burr. Delrin pieces can be manufactured in half the time of aluminum 6061-T6 (which itself is very machinable) and one quarter the time (and wear) of 316 stainless.

Note: Some care must be taken when using delrin in applications requiring a clearance or interference fit with metals. The dimensional instability is due both to its hygroscopic property and relatively large coefficient of thermal expansion (K = 4.5x10⁻⁵/°F): the manipulator may be assembled at 70°F but used in 35°F sea water. In fact, during assembly of the swivel pulleys and mounting of the cable spools a good snug-fit was obtained by first boiling the delrin parts (or heating them

15 Specifications from Commercial Plastics and Supply Co. Superior properties are obtainable in glass-fibre reinforced polycarbonates, but these are currently available on a commercial basis only.
with a hair-dryer) and then slipping them over refrigerated metal shafts.

7.6 SPOOL S

The cable spools are machined of delrin, and keyed directly to the motor shafts, Figure 21. Ten 1/4" pitch, 3/16" deep threads (cable O.D. including coating) are cut into the cylinders, the thread diameter being 1.75": at this curvature, the cables have approximately 90% of their full fatigue life. This yields a spooling length of 55". Given the motor output torque of 175 in-lb at a 2500 psi drop, this transforms to a 200 lb tension capacity. This allows the cable drive to meet speed, reach and load capacity specifications. The use of delrin will minimize cable abrasion. Note that due to the cable height variation for different displacements of the spool, some error is introduced by assuming a fixed (centerline) length between spool and pulley. The maximum error is 0.07", and if necessary can be compensated for by more precise inverse kinematics. Note that the variation has no effect on the repeatability of the machine.

**Protection:** Close-fitting spool covers will be added to keep out debris and prevent fouling of the cables.

7.7 SWIVEL PULLEYS

To allow the cable spools to wrap correctly, while enabling the top plate to float freely, an intermediary idler pulley was necessary. Ideally, one could pass the cable through a small, frictionless eyelet in its transit to the end effector. Practically, this was impossible as:
(i) An eyelet would be far from frictionless at 200 lb cable tension.  

(ii) A bend radius of at least 3/4" is necessary to prevent early fatigue failure of the cable.  
The design solution was a swivelling pulley whose axis of rotation is tangential to both the idler pulley and cable spool so that the centerline separation is constant, regardless of pulley orientation, Figure 22. The idler pulley and swivel bracket are machined from delrin, the idler pulley shaft 316 stainless, the swivel shaft is 316 stainless pipe, and the Dodge pillow block housing gray cast iron.  
The idler pulley is fitted with a light-duty radial ball bearing, while the Dodge block houses two opposed, heavy-duty Timken tapered roller bearings, to take the high moment and thrust loads. Several alternate configurations for two-point support of the swivel shaft were considered, but two constraints made simple, readily-purchased pillow blocks the final choice:  

(i) The spacing between adjacent cable tie-points (the point on the base to which a cable is hypothetically attached) was to be minimized, in order to maximize the load-handling capacity. For the same reason, these points were to be as close to the base perimeter as possible.  

(ii) Because of the variation in cable height at the spool, clearance was necessary which dictated a minimum distance of 6" between the spool and pillow block. The use of a large diameter inboard bearing was feasible, but more awkward than the single unit design. To reduce weight and plate stresses, however, large radial conrad-type bearings in delrin housings are a modification which may be made in the future. Note that the variable wrap angle about the idler pulley introduces an uncertainty
FIGURE 22
SWIVEL PULLEYS

LUBRICATION PORTS
in the tie-point positions. The resulting error in cable length is about 0.75" maximum, but may be precisely accounted for, if necessary, by more precise kinematics. As with the spool errors, the variable wrap angle has no effect on the manipulator's repeatability.

**Protection:** Both the swivel and idler pulley bearings will be lubricated via a pressure compensated oil feed. The Dodge bearings come standard with a lubricating nipple and oil seals, while the idler pulley shaft was provided with a special bore to allow oil feed to the interior of the bearing, which is fitted with low-friction shaft seals. Corrosion should be no problem save for the cast iron blocks, which will be epoxy-coated.

7.8 COMPRESSION MEMBER

The seventh actuator of the six-degree-of-freedom manipulator plays a passive but vital role in determining the position of the manipulator. Its job is to maintain a preload in the system sufficient to keep all cables in tension, thereby providing rigidity. At present this "backbone" is a single-stage 9" stroke Clippard cylinder of brass wall, stainless rod construction, pressure rated @ 2000 psi (hydraulic). The Buna-N piston seal is low-friction, with a breakaway pressure of 4 psi: a small fraction of the nominal 200 - 1500 psi operating pressures. Minimal stiction has been observed, and this may later be killed by the injection of a small-amplitude, high frequency dither signal to a pressure regulating servovalve. The cylinder is fitted with an aluminum sleeve attached to the end of the rod, with a delrin bushing at the other end which slides on the smooth O.D. of the cylinder, Figure 23. This sleeve supports the small moment loads imposed on the spine by
frictional torques in the gimbal bearings, insuring correct alignment and preventing bending of the 1/4" cylinder rod. The cylinder is mounted to the gimbal via a 1/4" stainless rod to provide clearance between the spine and inner motor spools (not yet mounted). The rest length of the spine, from gimbal pivot to tripod tip, is 21". Hydraulic supply to the actuator is through 3/8" Synflex flexible tubing which passes through a smooth-edged hole in the delrin base. This 3000 psi tubing has a bend radius of only 0.75". Currently the pressure to the spine is controlled in open-loop fashion via a manually set pressure regulating valve, and monitored with a pressure gage. Chapter 8 (Control) discusses the advantages of replacing this by a pressure-control servo valve: system response will be improved, and programmable preload will lead to a more versatile system, capable of safely performing work in an unfamiliar environment where the manipulator might injure itself or the workpiece. The spine will ultimately be replaced by a telescopic actuator to increase the stroke from 9" to 3", yielding an extended reach of about 5°. The telescopic action can be achieved either by one multi-stage cylinder, special-ordered at $8000, or by staging clusters of single-stroke cylinders.16 This arrangement would be a bundle less than 3" in diameter when collapsed.

Protection: The brass cylinder, aluminum sleeve and stainless rod should prove durable. The brass and aluminum are isolated via the delrin bearing. The stainless rod will have some cathodic interaction with the heavy brass base, but in turn, it will be protected from corro-

16 Telescopic cylinders come standard in dump truck size only.
sion.

7.9 SPINE GIMBALS

In order that the spine pivot freely through its full working cone (up to 60° from the vertical) a gimbal was constructed and mounted to the manipulator base; when an end-effector plate is added to replace the tripod tip, an identical unit will be mounted there. The standard universal joint was inadequate because of its solid working angle of only 60°. The problem was to design a universal pivot with intersecting axes of rotation which was of light, very compact and seaworthy construction, while able to support compression loads of up to 1000 lb through a solid arc of 120°. The entire assembly had to fit within approximately a 1" x 5" region bounded by the cables and drive spools. The gimbal consists of a one-piece aluminum fork supported by stainless shafts which rotate in 1/2" Nyliner bearings. These are inserted in two delrin support blocks bolted to the base. The fork supports a 5/8" stainless shaft on which rides another Nyliner bearing. This bearing is pressed into the central pivot, a stainless disk which is threaded to receive the 1/4" stainless rod from the spine, Figure 24. Note that the delrin pillow blocks serve a dual role as thrust bearings for the aluminum fork. Note further that the Nyliner bearings provide adequate tolerances, as it is not the spine which determines the position of the end effector, but the cables. In fact, there is hardly detectable free-play in the unit. The length of the gimbal, including pillow blocks, is 6", and the width 17/16".

Protection: The gimbal will be enclosed by an oil-filled rubber boot with a collar clamp about the spine shaft: much like a stick-shift
FIGURE 24
SPINE GIMBAL
cover. This protection is not crucial, as the components are designed to brave the seawater, but would be helpful in excluding all grit and preventing electrolytic interaction between the stainless and aluminum members. 17

7.10 TRIPOD TIP / UPPER PLATE

In its present form the manipulator is a three-degree-of-freedom (3 D.O.F.) mechanism, consisting of three control cables kept taut by the spine, Figure 25. The three cables and spine meet at the tripod tip, which the manipulator positions in space without regard to orientation. Note that this implies a restriction of load capacity for this subsystem to forces whose lines of action pass through the intersection point of the spine and three cables: excessive moments about that point may cause bending of the spine, which is designed primarily for compression loading.

The tripod tip is made of delrin, and has three ports to receive the cables, a steel thread insert where it joins the spine, and threaded bore at the top for attaching small masses, an end effector, or instrumentation such as an accelerometer. The ports are parabolic funnels designed to maximize the cable bending radius and so increase their fatigue life.

On the complete six-degree-of-freedom manipulator the tripod tip is replaced by an upper plate, supported by the spine via a gimbal identical to the one at its base. The six cables will attach to this plate through ports similar to those on the tripod tip, arranged in pairs of

17 Aluminum was used in the fork to facilitate machining and reduce weight.
FIGURE 25
3 D.O.F. MANIPULATOR
two to form an equilateral triangle, Figure 26. As discussed in Section 6.2, the size of the upper plate and arrangement of the ports affect the loading characteristics of the manipulator: different configurations allow for higher force or moment capacities, or load bias. If the manipulator is to loosen rusted valves, a large plate is preferable; for lifting heavy boulders, a smaller port radius is best. Nominally, a 4" radius port circle will be used. Strength-wise this is a good compromise, and allows a relatively small profile for the end effector.

Although initially to be equipped with a hydraulic gripper, the plate will ultimately serve as a tool socket complete with electro-hydraulic quick-release connectors. Tools for cleaning, welding, cutting and gripping will be stored in a tool kit within reach of the manipulator. The plate will be constructed of delrin.

Protection: A rubber boot for the upper gimbal.

7.11 MANIPULATOR BASE

The design of the manipulator base addressed two problems:

(i) Locating the six motors, six pulleys, six pillow blocks, spine and spine gimbal to allow adequate clearance while realizing maximum economy of space.

(ii) Creating a structure of adequate strength and rigidity to bear the high forces and bending moments caused by cable tensions and spine compression, while keeping mass and bulk to a minimum. The servomotors, though stationary, are quite bulky due to the integrally mounted valves. The swivel pulleys and blocks require clearance for free rotation, but must be positioned in three closely-spaced pairs. Furthermore, a spacing of at least 6" is necessary between the spools and swivel blocks to
FIGURE 26

TRIPOD TIP

UPPER PLATE
accommodate the variation in cable height on the spools. Finally, the six spool-pulley drive units must not interfere with one another: cables must not cross.

A satisfactory configuration was synthesized by extensive use of C.A.D. (Cut-out Aided Design) whereby cardboard figures representing each component were arranged and re-arranged until a suitable plan was formulated, Figure 27. An a priori constraint that everything had to fit within a 24" diameter circle was imposed to encourage creative conservation of space. In fact, the final arrangement is optimal in that the motors are packed as densely as is physically possible (spacing of 1/16" apart) while just touching the perimeter of the 22" diameter base. The swivel pulleys are positioned at the circumference of the base to maximize load capacity, while the spacing is so close between adjacent units that they must rotate in tandem within 45° to avoid interference. This condition is automatically satisfied because the cables all track the end-effector plate.

The motors, pulley blocks and spine gimbal are mounted to the 1/2" thick delrin plate with 1/4" through bolts. The plate is reinforced by 1" x 3" and 1/2" x 3" delrin bars, and a 3" diameter centerpiece, as shown in Figure 28. The 1/2 x 3 bars dovetail with the centerpiece and extend radially, interlocking with the 1 x 3 beams. The centerpiece and 1/2 x 3 beams serve primarily to support the center of the plate, which may be subject to loads as high as 1000 lb from the spine. They also add torsional rigidity to the 1 x 3 bars. The 1 x 3 beams form an equilateral triangle under the base plate, and are positioned to support the bending moments induced by the six pillow blocks and cable spools, which
FIGURE 28
CONSTRUCTION OF MANIPULATOR BASE

1/2 x 3 BEAMS

CENTERPIECE

1 x 3 BEAMS
may be as high as 500 in-lb. The 1 x 3 bars and centerpiece are attached to the plate with 1/4" screws, 1/2 x 3 beams by 8-32 screws. The screws are spaced so as to support the high transverse shear stresses which arise at the plate-beam interface.

The vertices of the 1 x 3 triangle are reinforced by 1/2" thick aluminum gusset plates, each attached by four 1/4" through bolts to the base plate. These three blocks serve as structural reinforcements by maintaining the vertical orientation of the beams, and also furnish mounts for attaching the manipulator to the sub. The centerpiece, at the center of the beam "web", may also be used as a single-point mount.

The 3-fold symmetry in the arrangement of motors and swivel pulleys on the base allowed use of a simple and strong triangular support geometry, as well as facilitating the machining task considerably: the mounting holes in the plate were drilled on a mill using a rotary table, enabling three holes to be determined through one measurement.

Owing to its delrin construction, the manipulator base is both strong and surprisingly light: the underwater weight of the entire structure is under 4 lb. It will have no electrolytic interaction with either the motors or other hardware mounted to it, nor with the submersible's steel frame to which it mounts.

7.12 HYDRAULICS

With the exception of Synflex tubing which serves the spine, all hydraulic line is 316 stainless tubing with Parker "CPI" stainless compression fittings. The Synflex is remarkably flexible — a bend radius of only 3/4" — yet has a 3000 psi pressure capacity, and is not affected by the seawater. Serving the motors, the 3/8" and 1/2" tubing of
wall thickness 0.035" and 0.065", respectively, are sized to provide a factor of safety of eight against burst at 3000 psi. With a nominal flow of 2 gpm to each servomotor (which is the rated flow of the Cadillac Gage valves) the head loss in the 1.5" sections of 3/8" tubing will be under 20 psi. The power thereby expended (in the form of heat from viscous friction) is about 15 watts per tube. It is interesting to note that the temperature difference dT necessary to dissipate this power is about 50F under water, yet more than 1000 in the air, due to its low convective coefficient. In fact, this should not be a problem above surface as the servomotors are normally stationary. A Moog aluminum-housed filter provides 2 micron filtration to keep the valves clean, while a pilot-operated solenoid shut-off valve is located upstream for safety. One does not kill the system by shutting off the control current to the servovalves: the actuating pressure itself must be nulled.

7.13 LUBRICATION

A supply of oil at about 8 psi above ambient will keep the swivel pulley bearings, pillow blocks and spine gimbals (optional protection) operating smoothly at virtually any depth. This will be provided by a spring loaded double-acting hydraulic cylinder with sea water at one port and lubricating vegetable oil on the pressurized side of the piston. The submarine's 3000 psi hydraulic supply is already compensated via bladders; through internal porting, it is this fluid which will protect the servovalves and potentiometers.
Chapter 8
CONTROL

Three topics have been included under the broad heading of "Control". Section 8.1 is an overview of the control sequence: command input, input processing and manipulator response / feedback. Various command strategies are considered, together with a discussion of the supervisory / subordinate roles to be played by microprocessors. Section 8.2 investigates the dynamics of the individual valve-motor subsystems, to arrive at some estimate for the system time response and behavior under load. Suitable block diagrams and transfer functions are derived, including the effects of disturbance loads, for the simple linear model considered. Section 8.3 is a brief discussion of future control strategies to increase system performance and capabilities.

8.1 CONTROL OVERVIEW

At the time of writing, the 3 D.O.F. manipulator operates under direct joystick command, without aid of a computer. Three command potentiometers specify the angular displacement of the spool motors and hence the three cable lengths L. The operator must furnish the inverse kinematics by "feel", and through practise develop dexterity through this "hard-wired" master-slave mode of control.18

18 A human operator (indeed - any creature with articulated limbs) is capable of on-line "solution" to inverse kinematics orders of magnitude more complex than the mapping G described in Section 5.2: this solution is called "sense of feel" which is developed through many hours of practice, e.g. tennis.
Simplicity and reliability are advantages of this mode of operation. Why not make use of the operator's physical intuition as a valuable resource? For the majority of practical tasks to be completed, this scheme is viable.

In performing tasks of a repetitive nature, or those requiring superhuman patience, accuracy or ranges of speed (high and low) the operator's skill may either be enhanced or replaced by a computer. This mode of either shared or delegated control over the manipulator introduces a hierarchical structure to command synthesis and execution known as Supervisory Control. 19

It is not the intent of this section to exhaust the topic; however, a helpful analogy might be drawn between Supervisory Control and our nation's structure of Law and Order. The Congress devises and ratifies new laws, but has neither the capacity nor the inclination to enforce them: this task is delegated to much larger and in most respects more capable forces the police and military. In rare emergencies, or delicate situations, our commander-in-chief the president may directly intervene and take the reins.

The "larger, more capable force" in this case is a computer, probably of microprocessor capacity, which would execute the wishes of the operator: either short-term commands or long-term strategy. The microprocessor may serve several functions:

(i) "translator" in a master-slave mode of control: position or velocity commands \( \overset{\ddot{}}{X} \) or \( \overset{\ddot{}}{X} \) specified by the operator are converted to actuator commands \( \overset{\ddot{}}{L} \) or \( \overset{\ddot{}}{L} \) via the inverse kinematics \( G \) or its derivative. The input device may be an instrumented arm, joystick or even keyboard.

(ii) The micro may also assume the role of foreman, either reciting a predetermined sequence of commands as in following a given trajectory during seam welding, or even synthesizing its own commands, e.g. inspection work.

(iii) The micro may manage subordinate microprocessors, dedicated to tasks of information processing and storage from environmental probes in addition to on-line feedback control of the manipulator actuators, including the spine.

(iv) A very important service of the micro would be as a software stop: policing input commands, however derived, to ensure that the physical locus of the manipulator is not exceeded.

8.2 SERVOMOTOR DYNAMICS

We now trace the flow of signals by which an actuator command becomes the flow of hydraulic fluid through a servovalve, as a preliminary to studying the valve-motor dynamics. A given state \( \{X, \overset{\dot{}}{X}\} \) of our manipulator will be determined by the linkage lengths \( \{L, \overset{\dot{}}{L}\} \) which are directly proportional to the 6-tuple of motor angular displacements:

\[
\overset{\dot{}}{L} = R\theta
\]

\[
\overset{\dot{}}{\overset{\ddot{}}{L}} = R\overset{\dot{}}{\theta}
\]

where \( R \) is the spool radius.

Now \( \overset{\dot{}}{\theta}_i, i=1,2,\ldots,6 \) are themselves directly proportional to the
flow rate \( Q_i \) of hydraulic fluid to the motor:

\[
\dot{Q}_i = K_w Q_i
\]

where \( K_w \) is determined by the (fixed) motor volumetric displacement. For our 0.5 in\(^3\) motors, \( K_w = 12.5 \text{ rad/in}^3 \).

The flow \( Q_i \) is provided by the servo valve, mounted directly to the motor, according to the orifice equation:

\[
Q_L = \left( \frac{i}{i_0}\right) Q_0 \sqrt{1 - \frac{P_L}{P_S}}
\]

Figure 29 is a graph of this relation, also including the effects of valve spool leakage. Normally this quadratic is linearized about \( P_{\text{load}} = 0 \), and valve leakage accounted for, by the relation

\[
Q_L = K_q i - K_c P_L
\]

where \( i \) is the current to valve (-8mA < \( i < 8\)mA) and \( P_L \) is the pressure drop across the motor. \( K_q \) and \( K_c \) are "constants" of the valve known as the flow gain and pressure droop, respectively.\(^{20}\)

To see why this linear relation is a good approximation for small variations in load \( P_L \), note that (neglecting leakage):

\[
Q_L = \left( \frac{i}{i_0}\right) Q_0 \sqrt{1 - \frac{P_L}{P_S}}
= \left[ K_q (P_S + i) \right] \sqrt{1 - \frac{P_L}{P_S}}
\]

Hence \( \partial Q / \partial P_L = -\left(1/(2P_S)\right) [K_q(P_S + i)] \left(1 - \frac{P_L}{P_S}\right)^{-1/2} \)

\[
= -5 \times 10^{-5} (1 - \frac{P_L}{P_S})^{-1/2} \text{ cis/psi}
\]

\(^{20}\) These constants are linear coefficients which, in general, depend on the operating point (\( i, \) \( P_{\text{Lo}} \)) and the total pressure drop \( P_{\text{supply}} - P_{\text{return}} \) (nominally (0,0) and 1000 psi supply). In fact, \( K_c \) is directly proportional to \( P_S \). The parameters were determined for our valves by direct measurement.
FIGURE 29
SUPPLY PRESSURE vs FLOW

FIGURE 30: BLOCK DIAGRAM
using nominal values $i = 2$ ma, $P_S = 3000$ psi. The slope is flat within 0.5\% until $P_L = 2999.7$ psi. Thus for small deviations about a nominal load $P_L$ we can assume the flow $Q_L$ to be a function of current only, with a factor $-K_C P_L$ to correct for valve spool leakage. In fact this leakage is reported to enhance the linear characteristics of the servovalve.

For constant angular velocity, neglecting viscous effects and seal friction\textsuperscript{21}, the output torque of the motor is in direct proportion to the back pressure $P_L$:

$$\text{Torque} = T_m \times P_L.$$  

Note: $P_L + P_{\text{valve}} = P_S - P_{\text{return}}$. The current $i$ is supplied to each valve through its driving card, which consists of two operational amplifiers: the first sums input command/feedback voltages, while the second provides a proportionate driving current.

Presently, the cards serve as simple proportional position controllers for the motors, as illustrated in the block diagram of Figure 30. The inner (dashed) loop represents the force balance on the motor:

$$P_L \times T = C_1 \theta + C_2 \dot{\theta} + T_{\text{disturbance}}$$

$T$ is actuator torque gain, units {in-lb/psi}

Again, upon the manufacturer's recommendation, the viscous term $C_2$ is neglected. The transfer function $G_1(s) = \theta / \theta_c$ for our simple, linear model is thus:

$$G_1(s) = K \times 1.17 \times 10^5 / s^2 + 7.78 \times 10^3 s + 1.17 \times 10^5$$

\textsuperscript{21} "Breakaway" pressure for the motor is less than 20 psi: typically, the operating pressure drop across the motor exceeds 2,000 psi.
where the controller gain $K$ has units $\text{ma/radian}$. Note that the supply pressure $P_s$ acts as a gain because $K_q$, the valve flow gain, is in proportion to it. This will be demonstrated in the performance results, Section 9.5. By experiment, the maximum stable gain was found to be 4.8 $\text{ma/rad}$. At higher gain resonance was observed, as high-frequency modes in our system were excited — producing "hum". This parameter choice results in the following characteristics of our servomotor (as modelled):

(i) Two real poles at

\[-7.7 \times 10^3 \text{ sec}^{-1}\]
\[-73 \text{ sec}^{-1}\]

Clearly the pole at $-73 \text{ sec}^{-1}$ dominates. For our system operating without load, we expect approximately a first order step response with time constant $t_s = 0.01 \text{ sec}$ and a $95\%$ rise time of about 0.03 sec. This is quite close to the measured response, which varied from 0.02 - 0.04 sec in tests under varying operating pressures: for details see Performance Section 9.2.

(ii) A natural frequency $\omega_n = 120 \text{ Hz}$. In fact, this frequency is just under the natural frequency of the spring-mass system consisting of the motor shaft and 30" of the cable, assuming one end fixed at the end effector. Based on the stiffness of the cable, spool diameter and shaft inertia, $\omega_n = 160 \text{ Hz}$ for an extension-relaxation mode of vibration. The fundamental frequency for a standing wave with half-wavelength 30" at 100 lb tension in the cable is:

\[
\left(\frac{1}{2L}\right)\sqrt{\frac{T}{\rho}} = 5 \text{ Hz}
\]
Hence we are operating well above the fundamental resonant frequency of this mode. Excitation of higher modes $n \neq f_0$ has not been a problem; $n$ is order(30). Note that the motor loading, considered a disturbance torque, as well as valve hysteresis enter as system disturbances before our "free integrator" 1/s term:

$$\dot{\theta} \rightarrow 1/s \rightarrow \theta.$$  

Hence when either is present, we expect some steady-state position error in our system.

The transfer function $G_2(s) = \theta / T_D$ where $T_D$ is the load torque, is:

$$G_2(s) = \frac{416}{s^2 + 7.78 \times 10^3 s + 5.62 \times 10^5}$$

for the controller gain $K = 4.8 \text{ ma/rad}$. Hence we expect a steady state position error due to loading:

$$e_{ss} = 7.4 \times 10^{-4} \text{ rad/in-lb}$$

Given a nominal load of 100 in-lb torque (= 115 lb tension) our error in cable length is about 0.065". Physically, this is the position error necessary to compensate for leakage across the valve (which is desirable as it improves the linearity and smoothness of the valve). Either integral compensation or feedforward compensation based on load sensing is necessary to eliminate this offset.

Valve hysteresis and null current phenomena will also lead to a steady-state static position error under the simple proportional controller:

$$e_{ss} = 1/4.8 \text{ ma/rad} = 0.208 \text{ rad/ma}$$

This is in effect the "play" or current "backlash" in the valve, which was determined experimentaly to be a maximum of 0.08 ma. Hence we
expect a maximum position error of about 0.015" due to this effect. Here again, some integral control action (in conjunction with derivative action to stabilize and speed the response) would eliminate the error, but care must be taken to avoid "hunting" or limit cycle behavior.

One must beware of the limitations of this simple linear model, particularly in predicting the dynamic response of the system under appreciable loading. As mentioned, the flow gain $K_q$, usually given under the no-load condition, in fact depends on the motor load $P_L$:

$$K_q(\text{load}) = K_q(\text{no load}) \times \sqrt{1 - \frac{P_L}{P_s}}$$

Hence as:

$$P_L \rightarrow P_s, \quad K_q \rightarrow 0,$$

which means that our effective gain:

$$K \times K_q$$

decreases with a high load on the motor. Conversely, for:

$$P_L \rightarrow -P_s, \quad K_q \text{ increases (load helps the motor).}$$

Furthermore, the valve saturation current of ±8 ma imposes a physical limitation on the system which is not embodied in the linearized equation. Given the actual controller gain, the saturating displacement command is 2". These nonlinearities will result in an asymmetrical response of the servomotor when given step commands opposing vs in the direction of a load. The compression member constantly provides such loading, so that retract-extend responses will not be identical. When a spool motor is commanded to reel in cable, the response will be slower than our linear model predicts, and vice-versa for an extension command. For example, with a supply pressure of 3000 psi, the effective flow gain $K_q$ will be halved at $P_L = 2200$ psi, corresponding to the motor spo-
oling in cable at 150 lb tension. Conversely, $K_q$ is increased by a factor of 1.3 at $P_L = -2200$ psi when the motor releases cable at this tension. This behavior is detailed further in the next chapter, describing system performance.

Finally, note that the loading $T_D$ on each motor is not state-independent, but highly correlated to the length of each cable: this correlation is described by the mapping $S$ of chapter 5. Hence $T_D$ is not a "disturbance" load in the conventional sense, but feedback from within the system. Thus the above analysis, predicated upon a dynamically uncoupled system of actuators, furnishes but a rough estimate of system performance.

The simple strategy of independent proportional control for each spool motor and a preset force in the spine is effective, and the dynamic response respectable, as will be quantified in the following chapter. The analog control is smooth, fast and reliable: it is expected to be completely adequate for the intended undersea use of the manipulator. Undoubtedly, however, it may not be optimal for high-speed tasks for which this low-inertia, powerful manipulator may otherwise be well-suited. Clearly there is room for exploration of a more intelligent, centralized control regime providing coordination of, if not communication between, the seven actuators. Specifically, there would be three advantages to "smart" on-line control over the spine preload, $P$, via a pressure-feedback servovalve:

(i) As the spine serves to maintain rigidity in this "floating" configuration, providing tension in each cable, its bandwidth is a limit on the entire system bandwidth. Currently, for cost reasons, the spine
pressure is manually set via a simple spring-loaded pressure-relief valve. Its bandwidth is under 1 Hz, compared with about 40 Hz for our servomotors. Physically, the effect of this is that when the motors spool in cable, a high back pressure develops in the spine [increasing $T_D$], while a quick spooling out of cables, or combination whereby the end-effector's radial displacement $r$ from the lower gimbal is increased, tends to leave cables momentarily slack. This non-linear phenomenon can be seen in the loaded response of the manipulator given in the next chapter. Control over the spine pressure using servo-valves with open-loop bandwidth of at least 60 Hz (even our 20 year-old valves are this fast) would eliminate this sluggish behavior.

(ii) As the spine maintains tension in each cable, and rigidity in the system as a whole, it could serve as an arbiter between the six spool actuators: when relaxed, the actuators cease to "fight", and when preload is increased, the actuators all are subject to disturbance loads tending to draw out more cable. This potential role as arbiter enables the spine to improve response times by exerting greater force when the command $X_C$ extends the spine, while reducing $P$ when the cables act to contract the spine. Such control over the spine preload would serve to compensate for the non-linearity inherent in the coupling of our actuators, which gives rise to the disturbance loads on each.

(iii) As discussed in Section 6.2, a variable preload $P$ increases the flexibility of the system by allowing on-line control over the load capacity of the manipulator. In delicate handling tasks, the preload would be reduced to a safe level, while for heavy loading, as in lifting a rock (or even the submersible itself) the preload would be maximized.
pressure is manually set via a simple spring-loaded pressure-relief valve. Its bandwidth is under 1 Hz, compared with about 40 Hz for our servomotors. Physically, the effect of this is that when the motors spool in cable, a high back pressure develops in the spine \([T_D]\), while a quick spooling out of cables, or combination whereby the end-effector's radial displacement \(r\) from the lower gimbal is increased, tends to leave cables momentarily slack. This non-linear phenomenon can be seen in the loaded response of the manipulator given in the next chapter. Control over the spine pressure using servo-valves with open-loop bandwidth of at least 60 Hz (even our 20 year-old valves are this fast) would eliminate this sluggish behavior.

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(iii) As discussed in Section 6.2, a variable preload \(P\) increases the flexibility of the system by allowing on-line control over the load capacity of the manipulator. In delicate handling tasks, the preload would be reduced to a safe level, while for heavy loading, as in lifting a rock (or even the submersible itself) the preload would be maximized.
The range of load capacity would be from a few ounces to several hundred pounds, accomplished without actuator gain adjustments.

8.3 FORCE SENSING AND ERROR COMPENSATION

Finally, microprocessor control over the actuators would allow open-loop compensation for the known compliance in the cables and servovalves. This compensation is based on static force-sensing in each servo-motor through the error current \( i_e \) necessary to maintain a pressure \( P_L \) across the motor, according to:

\[
0 = K_q i_e - K_c P_L, \quad \text{or,}
\]

\[
P_L = (K_q/K_c) i_e = K_p i_e
\]

where \( K_p \) is known as the pressure gain for the valve. \( K_p \) was measured as 4,000 psi/ma for the Cadillac Gage valves used. Given the tension in the each cable, its length and stiffness, the stretch \( \Delta L \) (as well as \( \Delta \theta \) for each motor, if there is no integral control) can be estimated and a compensating command:

\[
\Delta_{\text{comp}} = -(\Delta L/R_{\text{spool}} + \Delta \theta_i)
\]

given to the actuator by the microprocessor. This correction would be performed once only for a given position command \( \bar{X} \) and not as a continuous feedback (which could cause instability).

The ability to continually monitor the cable tensions \( T \) provided by the measurement of \( i_e \) and known \( K_p \) allows rapid calculation of the external force applied by the manipulator, given the low internal friction of the motors and pulleys. This is done simply through \( S^{-1} \) as discussed in Section 6.2, for the quasi-static case where inertial loading is negligible. However, given that in our system the endplate constitutes the only significant inertia (and viscous term in the under-
sea environment), the dynamic loading due to this plate can readily be calculated and subtracted, yielding the externally applied forces and moments. Thus current sensing provides a force sensing capability. This information, suitably furnished to the operator or supervisory computer in the form of displays or back-driven joysticks, can be just as important as visual feedback -- which may be of very poor quality. The high-resolution optical-fibre tactile sensor developed in our laboratory by Schneiter and Sheridan would further refine this force-sensing capability.
Chapter 9
PERFORMANCE

Extensive performance testing has been carried out on the system at this stage of its development. The focus has been primarily on investigation of the dynamic response of the individual actuators, both free and under load, as these characteristics will hold true for the complete manipulator. Some preliminary evaluations have been made of system accuracy, deflection under load, the effect of loading on dynamic performance, and the system's maximum slew rate.

The results are described below, in the same order as the initial performance specifications determined in Chapter 2.

9.1 STRENGTH

The original strength requirement of 20 lb force, 20 ft-lb torque should be easily met by the completed manipulator. This is demonstrated by the results of the static analysis of Section 6.2 wherein the maximal capacity of the arm is found to be 1100 lb force, 70 ft-lb moment, with nominal capacity of one-half these figures for most of the manipulator workspace. These calculations are more convincing when substantiated by the performance of the 3 D.O.F. subsystem. Although the tripod tip was not designed to support external loading, a twenty-five pound weight was suspended via a pulley from its neck, Figure 31. The effect of this loading was slight both in terms of deflection and dynamic response of the arm, as quantified in Sections 9.3 and 9.4. Furthermore, spine preloads of up to 400 lb have been sustained by the mechanism without difficulty. On the other hand, with spine preloads on the order of 50 lb, the mani-
FIGURE 31
MANIPULATOR UNDER LOAD
pulator is easily deflected by the hand, even with full 3000 psi supply pressure to the actuators: as explained in Section 6.2, the preload serves as a loading governor.

9.2 REACH

The prescribed reach for the manipulator was nominally 3 ft. The complete manipulator, equipped with a multi-stage spine, will have a workspace as depicted in Figure 8. This is a cone of solid angle 120°, inner radius approximately 2", extended reach of about 5". Note that the servomotors are at all times well-protected beneath the base plate, while during periods of inactivity the upper plate may be folded down via the spine gimbals so as to rest flat on the base, protruding less than 8" from the surface. The 2" base diameter seems reasonable in proportion to the 5" reach; a standard two-limbed serial arm normally has a rest length (not including mount) of two thirds its workspace radius. The present system, with single-stage spine, has a rest length of 21" and reach of 30".

9.3 ACCURACY

The measurement of absolute positioning accuracy will have to wait for microprocessor - based trajectory control, as well as the completion of the manipulator: it has no orientation capabilities in its present form. The repeatability and load-deflection characteristics have been measured with the aid of a dial indicator. To determine repeatability, the manipulator was given a 10" step command and then instructed to return to its original position. The average return position error was less than 0.005": well within the specification of 0.1".
Two tests were made to determine the manipulator's deflection under load. The first was to measure the vertical deflection of the tripod tip when the spine preload was increased from 60 to 300 lb force. The average maximum deflection was 0.3". The second test, more characteristic of an actual loading situation, was to measure the perpendicular and in-line deflection of the tip when subjected to an external 25 lb load (barbell suspended from the tripod tip). The deflections were 0.003" and 0.07" maximum, respectively.

The use of compensation, via force sensing, for the predictable actuator-linkage compliance is a simple, viable means of increasing the apparent stiffness of the loaded manipulator. Even without such compensation, the manipulator's stiffness should be easily adequate for its undersea missions: with working loads on the order of ten pounds, 0.05" deflections are no problem, and when lifting a 300 lb rock, or the submersible, 0.2" deflections will not be quibbled over.

9.4 SPEED

The two specifications for operating speed were:

(i) Top speed of 80 in/sec (double the sub's speed).

(ii) Threshold minimum speed of 0.1 in/sec.

With the driver cards saturating at $\pm 4$ ma, the average slew speed over a step of 10" was 100 in/sec. However, as the valves saturate at $\pm 8$ ma current, beefing up the current amplifier section of the controller cards should result in a top-speed slew rate of 200 in/sec, or 5 times the speed of the sub at full throttle. Hence such maneuvers as bottom sampling, simple cleaning or inspection "on the fly" should be possible.

To determine the low-speed threshold of the manipulator small am-
plitude, low frequency oscillatory commands were given the manipulator, while measuring (continuously) the deflection of the tripod tip with a dial indicator. The manipulator was able to smoothly track a 0.004" amplitude, 0.1 Hz signal (measured at a 27" extension): this is an average speed of under 0.002"/sec: about 7"/hour. Such a low speed is attributable to the smoothness of the planetary hydraulic servomotors and low friction in the cable pulleys and compression member.

Another test was made to determine the smallest step increment to which the manipulator would respond faithfully: this step size was found to be approximately 0.0005".

9.5 DYNAMIC RESPONSE

The above specifications are of practical interest for underwater work, but say little regarding the dynamic response of the system or its actuators. Given the novelty of the configuration, and predicted nonlinearities as described in Section 8.2, some dynamic investigation seemed appropriate. Time and frequency response measurements were made for both the unloaded servomotors and complete 3 D.O.F. manipulator, in free-running mode and under inertial and external loading. A knowledge of these dynamic characteristics serves not only a descriptive function, but provides concrete plant information which will be of use in formulating more sophisticated control regimes for the manipulator. Both step and closed-loop frequency response tests were conducted on the loaded and free (cables disconnected) actuators. Figures 32 - 40 illustrate the response under the varying conditions. They are photographs of the display of a storage oscilloscope.

The bandwidth of the unloaded servomotors operating at 3000 psi,
FIGURE 32
FREQUENCY RESPONSE

32(a) UNLOADED
(40 Hz)

32(b) LOADED
(40 Hz)

32(c) LOADED
(30 Hz)
FIGURE 33
AMPLITUDE ATTENUATION

33(a) 60 Hz

33(b) 90 Hz
measured as the 3db down frequency, was 40 Hz. When the cables were as-
sembled the expected non-linearity became apparent: the sinusoidal re-
response was distorted, with a "retract the spine" amplitude ratio of 0.5
and "extend" AR of about 1.3 at 30 Hz. See Figure 32 for a comparison
of the loaded and unloaded response. As discussed in Section 8.2, the
asymmetry results both from high back pressure in the spine, and the
non-linear behavior of the loaded servovalve. Figure 33 shows the steep
(-40 db/dec) amplitude attenuation past the 40 Hz corner frequency.

The loaded and unloaded step response for a step size of one inch
(2 ma maximum current to the valve) further demonstrates the effect of
coupling within our system, as well as the dependence of response on
supply pressure to the actuators. To illustrate the effect of supply
pressure, Test 1 (Figures 34) shows the step response of the unloaded
servomotor operating at 500 - 3000 psi. Two features are significant:

(i) The direct proportionality between rise time
and supply pressure.

(ii) The departure from the "1st order" exponential response, above
pressures of about 1000 psi.

The rise time of 0.03 sec predicted in Section 8.2 occurs at about 1500
psi (the pressure at which the valve constants were measured), while
significant overshoot appears at pressures above 2000 psi. The supply
pressure behaves as a gain in that $K_q$, the valve flow gain, increases
in proportion to $\sqrt{P_s}$. Hence we expect greater overshoot and the ap-
pearance of higher order dynamics at the 3000 psi operating point:
indeed the 1000 psi response appears of much lower order than that @
3000 psi. As the higher order poles become significant, the simple sec-
FIGURE 34
STEP RESPONSE

34(a) 500 psi supply pressure

34(b) 1000 psi

34(c) 1500 psi
ond order model of Section 8.2 becomes a weaker approximation to reality.

The asymmetry in response under cable loading, already illustrated in the response to a sinusoidal input, is even more apparent in the step response of the free-running manipulator, Figure 35. In this test the manipulator is given first a command which retracts the spine (curve 1), and then the inverse command (2) which returns the manipulator to its original position. As discussed in Section 8.2, the loading induces a gain variation due to the sensitivity of \( K_q \) to back (load) pressure, \( P_L \). Indeed, the rise time of the system while opposing the load \( P \) of the compression member is about 100 ms, vs 20 ms when the spine "helps" the motor to release cable (\( P_L < 0 \)). It is important to note that although we are attempting to describe the response of the entire manipulator, in particular the behavior of the end effector, the measurements shown are taken from one servomotor. This to some degree hides the effects of cable dynamics: although at 1000 psi the loaded actuator shows no overshoot, a certain amount of oscillation (on the order of 0.06" amplitude) is apparent at the tripod tip. A tip-mounted transducer such as an accelerometer would have been a useful addition, but time and resources impose limitations. This will have to wait for complete testing of the six-degree-of-freedom manipulator.

When a 5 kg mass is added to the tripod tip and the step test repeated, the inertial loading introduces overshoot during extension of the spine which is apparent at the actuator shaft, Figure 36. Even with the inertial load, the "retract" rise time is under 40 ms, while the "extend" settling time is about 30 ms. This is to be compared with the un-
loaded servomotor response in Figure 34(d), also exhibiting overshoot, with a settling time of 30ms. The addition of the mass has little degrading effect on the response times.

To further investigate the effect of loading on the manipulator response, a 25 lb horizontal force was applied to the neck of the tripod by use of the barbell and pulley configuration shown in Figure 37. The manipulator, even in its infantile state, shows little sensitivity to this loading (which nearly overturned the table to which the arm is mounted). Figure 38 shows the difference in response between pure inertial and (inertial + force) loading: here the step causes the spine to retract and weight to lower. Note the steady-state position error $e_{ss}$ induced by the higher cable load, as predicted in Section 8.2: this error is directly attributable to valve spool leakage.

Figures 39 further illustrate the effect of force + mass loading on the response. Figure 39(a) is the extend/retract response where the applied load is perpendicular to the step; in 39(b) the two are parallel. In 39(c), which is the response to a square-wave input, both the oscillatory behavior and steady-state error in extension are apparent. This behavior results from three factors:

(i) High system gain at 3000 psi operating pressure, the oscillation being present in the unloaded response Figure 34(f);

(ii) Effect of the non-linear history of the force (barbell is suddenly decelerated as it "hits the stops");

(iii) Sluggish spine must "catch up" with the quickly-released cable, which causes further variation in the disturbance loading of the motor.

As discussed in Section 8.2, more sophisticated feedback control over
FIGURE 39: RESPONSE WITH MASS & 25 lb LOAD

39(a) LOAD PERPENDICULAR TO STEP

39(b) LOAD PARALLEL TO STEP

39(c) SQUARE WAVE INPUT

RETRACT

EXTEND

RETRACT

EXTEND

20 ms

100 ms
the actuators (at least some derivative action to increase damping) and, in particular, smart control over a high-bandwidth, pressure-servoed spine would yield superior performance — although in its present simple form the manipulator easily meets the undersea performance specifications.

Figure 40 shows the response of the manipulator to a round-trip excursion of two 10" steps. The response is suprisingly linear, with the return velocity of 140 in/sec (during which the spine extends) about double the velocity while cable is spooled in. The uniform velocity (\( \ddot{\theta} = 0 \)) is in the presence of variable "disturbance" loads on the motor from the other actuators as the manipulator changes position. This stability is attributable to the insensitivity of the flow gain constant \( K_q \) to small changes in load pressure \( P_L \) for \( P_L/P_s \ll 1 \), as explained in 8.2. Because the driving card operational amplifiers and power transistors are saturating at 4 ma, whereas the valves saturate at 8 ma, the measured full-throttle flow (and proportional speed) should be one-half the actual capacity of our actuators. Thus we expect a top-speed slew rate of approximately 200 in/sec.

9.6 WEIGHT

The immersed weight of the manipulator will be about 95 lb; 88 lb are due to the six motors and swivel-pulley pillow blocks. Through use of delrin pillow blocks and lighter motor/value cover construction, the submerged weight can be trimmed to under 70 lbs. Except for the 2 lb delrin end effector (+ tool) all weight is stationary, and so may be statically compensated by a few cubic feet of syntactic foam. Given a working reach of five feet and load capacity of 500 lb, the
FIGURE 40
10" STEP RESPONSE
(SLEW SPEED)
(reach) x (strength) / weight ratio seems reasonable.

9.7 DEPTH RATING / SEAWORTHINESS

The manipulator should operate smoothly at virtually any depth. The seven actuators will be powered by pressure-compensated fluid, and all pulley bearings, servovalves and potentiometers will be immersed in oil at a pressure slightly above ambient. Through the extensive use of delrin, and isolation of the 6061-T6 aluminum and 316 stainless components, corrosion should be minimal. The manipulator should prove rugged and easy to maintain.

9.8 SAFETY

A pilot-operated solenoid emergency cut-off valve has been installed to shut down supply pressure to the manipulator immediately in the event of power loss to the controller cards, or other mishap. The normally-off valve is placed on the same electrical circuit as the driving cards, and is also equipped with a manual kill switch.
9.9 PERFORMANCE SUMMARY

STRENGTH -  30 - 1100 lb force, 20 - 70 ft-lb torque

REACH -  2° (retract) - 5° (extend)

ACCURACY -  ABSOLUTE: to be determined
            REPEATABILITY:  0.005"

SPEED -  MAX:  200 in/sec
            THRESHOLD:  0.002 in/sec
            STEP INCREMENT:  0.0005"

BANDWIDTH (-3 db):  30 Hz

RISE TIME (95%):  30 ms

DEPTH RATING - 5000 ft

WEIGHT (immersed) - 95 lb
Chapter 10

APPLICATIONS

At this stage of completion, the manipulator has demonstrated the advantages inherent in the parallel configuration, while proving the novel actuating principle on which the system is based. Still in its infancy, the arm has nonetheless exhibited the power, speed, accuracy and smooth action which can be realized by the floating linkage in which no rigid member exists between base and end effector. These characteristics of hydraulic servo-drives are enhanced by the simple, low-friction, low-inertia, backlash-free linkages which operate in concert, rather than series, to determine the position of the end effector. The final load capacity will exceed 1000 lb, slew speed over 200 in/sec, incremental step size less than .0005", and repeatability within .005".

The arm will thus be suitable for a wide range of applications, including:

(i) High-speed precision assembly, e.g. P.C. Board manufacture. In addition its to speed and accuracy, the arm's configuration lends itself readily to force sensing and control. The dynamics and inverse kinematics are readily calculated on-line with minimal computing power.

(ii) Welding and painting, where rapid yet smooth motion is required.

(iii) Large payload tasks, e.g. loading and manipulation of engine blocks or castings. The manipulator's load capacity even makes it suitable for use as a submarine grabber arm, to secure the vehicle in the presence of currents.
(iv) The manipulator proves to be an exciting vehicle for implementing more sophisticated control regimes, including so-called impedance control and various supervisory strategies for efficient task execution in a remote environment.

The cost of the manipulator will be relatively low, estimated less than $20,000, it is simple and rugged, and the necessary computer support lies within the capacity of a microprocessor or personal computer.
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