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Mechanical Design Of A Stewart Platform-Based Crawling Vehicle

by

Paul A. Mele

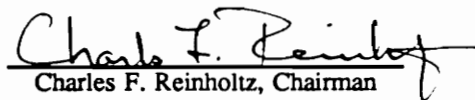
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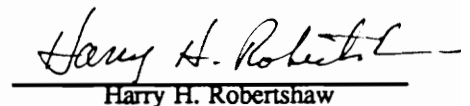
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Abstract

It is well established that a vast majority of the Earth's surface is inaccessible to conventional vehicles. Furthermore, projects alluding to the exploration of Mar's conclude that its surface is too rough for conventional wheeled vehicles. Man and cursorial animals, however, are capable of traversing virtually all types of terrain. These reasons, among others, have focused almost all development on walking vehicles having fixed torsos and articulated legs which emulate the locomotion of man and animals.

Insects such as the caterpillar move with fixed legs and an articulated torso. They too can traverse rough terrain but do so with greater stability than bipeds or quadrupeds. This thesis presents a design for a caterpillar-like crawling vehicle. An overview of the effort to develop walking vehicles is included to show the depth of interest in developing a vehicle capable of traversing rough terrain. A general overview of crawling vehicle objectives and the control problems hampering the realization of a crawling vehicle are then described. Finally, this thesis provides a detailed mechanical design with the kinematic and mechanical considerations governing that design.

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I must thank my parents whose love and support have provided the base without which I never would have reached the levels I have.

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CHAPTER 1

INTRODUCTION

1.1 Motivation For Vehicle Development

It has been well documented in papers such as those by McGhee (1985), Fichter, Fichter, and Allbright (1987), and Bares and Whittaker (1989), that about 50% of the Earth's surface is inaccessible by conventional wheeled or tracked vehicles (Stulce, Burgos, Dhande, and Reinholtz, 1990). Investigation has shown that the surface of Mars is also generally inaccessible to such conventional vehicles. Current long-range space programs are alluding to manned exploration of Mars, but manned missions currently lack financial and technical feasibility. Also, the half hour signal delay in round trip telemetry excludes teleoperated rovers from consideration. Perhaps the best alternative is to use a mobile, perceptive robot to explore the inaccessible areas on Earth and the surface of Mar's. This alternative motivates the design of a robot with unprecedented ability for autonomous, self-reliant exploration of rugged, barren terrains (Bares and Whittaker 1989). Man's primary systems for rough terrain locomotion have been wheeled or track laying-vehicles. These systems have very low terrain adaptability and are quite different than those used by cursorial animals and man himself. These species accomplish locomotion by a set of articulated mechanisms consisting of individual limbs capable of independently powered and flexible coordinated motion. Furthermore, off-road vehicle

speeds are usually restricted to a few miles per hour and power requirements are around ten horsepower per ton (McGhee, 1985). Under these conditions, man and cursorial animals are able to traverse the same rugged terrain with an order-of-magnitude more speed and much less energy expended than wheeled or tracked vehicles. In an attempt to emulate man and cursorial animals, walking vehicles have traditionally been designed as fixed torso, articulated leg devices. A vehicle with articulated legs possesses increased terrain adaptability because it can actively choose supporting leg points (Hirose, Morishima, 1990). Unfortunately the articulated, high degree-of-freedom leg requires a heavy leg driving system which reduces the payload capability of the vehicle (Hirose, Morishima, 1990). McGhee summarized the benefits of legged locomotion as increased speed, improved fuel economy, and greater mobility. It is for these reasons that there has been much research to date on the development of walking machines or rovers that utilize legged locomotion principles (McGhee, 1985).

To further show the need for an alternative type of vehicle, the disadvantages of wheeled locomotion (the current predominant form of locomotion) will be detailed here. First, traversability is limited because the need for continuous wheel contact limits the ability of a wheeled vehicle in discontinuous terrain. Secondly, there is much energy lost in rugged or soft terrains due to slippage, shear, and bulldozing resulting from the wheels sinking into the terrain surface. Furthermore, the required mechanical complexity of a wheeled vehicle to achieve three-dimensional motion is excessive. Likewise, teleoperation of wheeled vehicles becomes very complicated in rugged, large scale terrains

because terrain features become much larger than wheel diameter. Furthermore, the frequency of confronting obstacles in a rugged environment complicate teleoperation. It is also not possible to accurately predict and assess wheeled vehicle motion because detailed terrain-interaction models are unmanageable for wheel contact through rugged terrain. The models are difficult because of surface contact constraints, wheel compliance, and the three-dimensionality of rugged terrain (Bares and Whittaker, 1989). Lastly, because most wheeled vehicle motions are not quasi-static, incremental, reversible motions are unlikely (Bares and Whittaker, 1989). For instance, imagine the front wheels of a car rolling into a large crevice causing the car to "bottom out". Depending on the configuration (rear or front wheel drive) and the power of the car it may or may not be able to reverse its motion (back out).

The following sentence taken from a paper on insect walking by Wilson (1966) leads us to consider an alternative to wheeled of articulated leg locomotion. "The mechanics of various types of animal locomotion fall into two major categories; locomotion involving appendages and locomotion involving only movements of the trunk." In this thesis vehicles modelled after the first category will be referred to as walking vehicles and vehicles modelled after the second will be referred to as crawling vehicles. Most animals and vehicles fall into one of these two categories. Within each category further distinctions can be made when describing man-made vehicles. The most common type of vehicle has a rigid body and is driven by wheels or tracks. The vehicle may be single body, such as a car or tank, or multibodied, such as a train. Another vehicle type is that

with a rigid body or bodies and articulated legs. Most research in the area of walking vehicles has been aimed at the realization of this type of vehicle. The last category of vehicles is that with rigid (non articulated) legs and multiple bodies. Locomotion in this class is achieved by relative controlled motion of the multiple bodies (Stulce, et al., 1990). This thesis presents a mechanical design for such a crawling vehicle.

A crawling vehicle appears to not only have excellent terrain adaptability but also a greater payload capability and smaller minimum passable area than conventional walking vehicles (Hirose, Morishima, 1990). These merits are attributed to the articulated body of the crawling vehicle. Furthermore, an articulated body lends itself to easy maintenance, transportation, and modification (Hirose, Morishima, 1990). Similar to its natural counterpart the caterpillar, an articulated body vehicle can maintain static stability throughout all phases of movement (Stulce, et al., 1990).

Although the motivation for walking and crawling machine development arises from rugged terrain navigation, the envisioned uses of such a machine are not limited to this utilization. A walking vehicle could be used in any task determined unsafe for human labor e.g. nuclear facility maintenance (Carton and Bartholet, 1987) or in a wide range of military, construction, or mining applications.

1.2 REVIEW OF PREVIOUS WALKING VEHICLE PROJECTS

Efforts to create an artificial walking vehicle have spanned roughly the past twenty years. The study of walking vehicles has evolved through three phases within these

twenty years according to Hirose (1984). Phase one centered on the study of walking vehicles with mechanical coordination. Phase two involved generalized walking vehicles with multiple degrees-of-freedom (DOF's). The investigation of computer controlled legs also began in this phase. The last phase, phase three, is characterized by the systematic consideration of both energy efficiency and control of the walking vehicle.

Three notable projects related to the current work have been described in detail in the paper entitled "Conceptual Design Of A Multibody Passive-Legged Crawling Vehicle" (Stulce, et al. 1990)

These include a project headed by Waldron and McGhee at Ohio State University to design and build a six legged adaptive suspension vehicle (Waldron and McGhee, 1986).

A second walking vehicle project lead by Raibert of Carnegie-Mellon University built and experimented with a one-legged, dynamic hopping machine. His research of a three DOF, one-legged hopping machine focused on stability and control of dynamic vehicles (Raibert, Brown, and Chepponis, 1984).

Hirose of the Tokyo Institute of Technology constructed a four-legged walking vehicle. His design of a quadruped vehicle concentrated on energy efficiency and developed elegant control strategies (Hirose, 1984).

A project not investigated by the Stulce et al. paper is the AMBLER project at Carnegie-Mellon University. Bares and Whittaker are currently developing an autonomous walking robot for the exploration of the surface of Mars. The Autonomous, Mobile, Exploration, Robot (AMBLER), shown in Figure 1, is a six-legged robot that

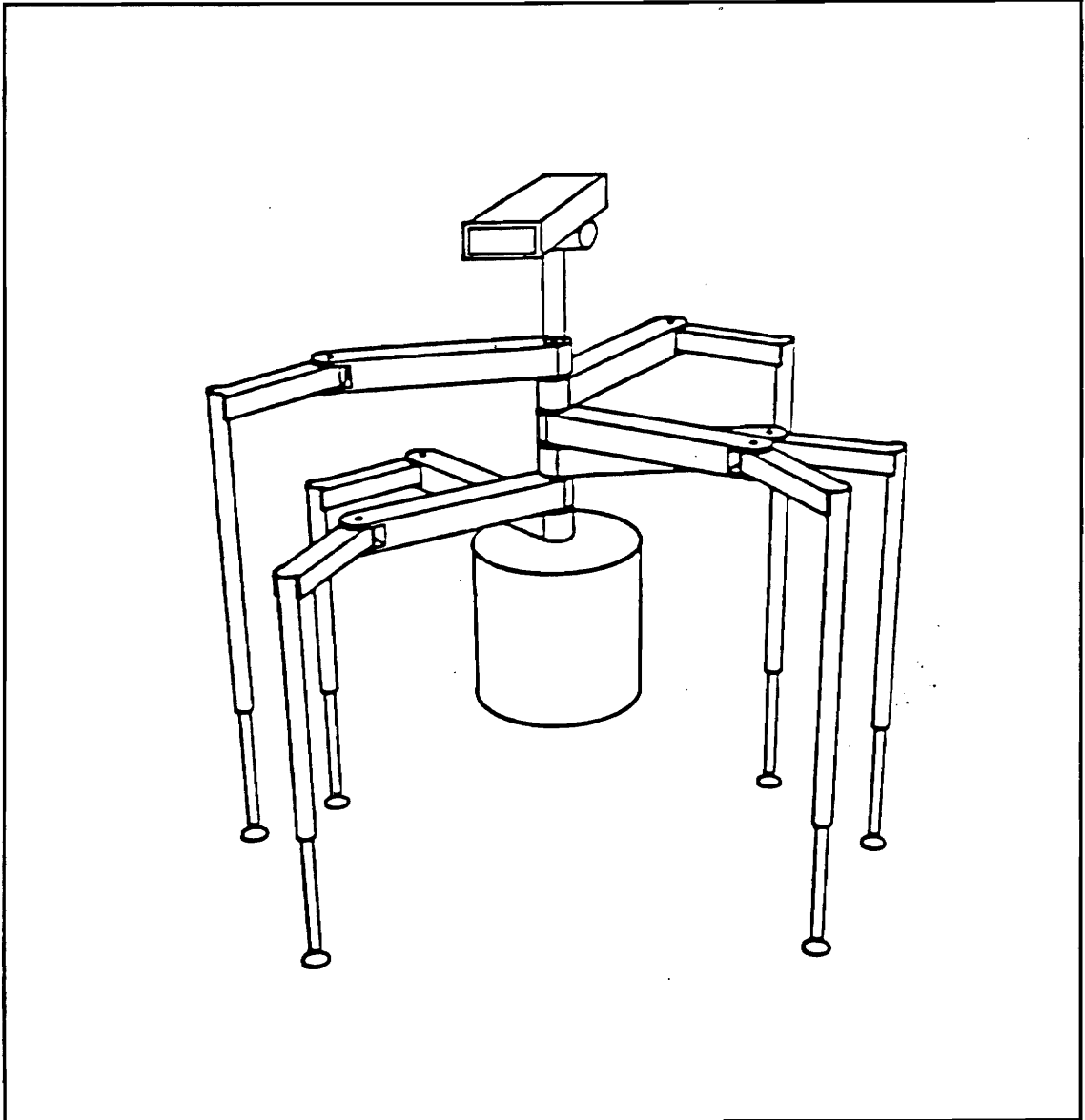


Figure 1: The AMBLER (Bares and Whittaker, 1989)

perceives and models terrain, and plans and executes tasks and motions. It is unmanned, self contained, and power efficient. The AMBLER was designed with extreme self-reliance as a primary goal. In an attempt to achieve this goal highly predictable mechanisms were used and there was conservatism at all levels of planning.

The AMBLER is approximately 11.5 ft (3.5m) tall and 10 ft (3m) wide. Each of its six legs has two revolute motions in the horizontal plane that position the leg above the terrain and a vertical telescoping action that extends the foot until contact with the ground. Each of these legs is mounted at a different vertical position along the central axis of the body. Furthermore, each leg can rotate fully around the one meter diameter body located below the leg stack. The body encloses the power generation equipment, as well as computation, scientific, and sampling instrumentation. The authors describe the propulsion of the AMBLER as analogous to the poling of a raft. As legs reach the limit of their stroke, they are replaced ahead of the walker much like the pole is replaced ahead of the raft.

The AMBLER uses a laser range scanner to gather local terrain data. The data is processed to generate both an elevation map and derived attributes such as slope and curvature. Updated AMBLER models include world position, position with respect to the elevation map, joint positions, and sensor orientations.

Several different planners control various aspects of behavior of the AMBLER. One set of planners determines "task" plans while a second set plans "motion sequences". At every stage of planning, competing objectives must be considered including energy

expenditure, stability, and rate of progress. The derived course of action for the AMBLER is then analyzed for the conditions that are expected to occur. This analysis is then used to accept or modify the course of action. The primary reason for course analysis is to prevent tipover; the fragile sensors and antennas could not survive the results of such a fall (Bares and Whittaker, 1989).

1.2.1 Articulated Body Vehicles

The effort to develop a caterpillar like or articulated body vehicle has not been as prevalent as that to develop a walking vehicle. However, there are two notable studies worth mentioning.

The vast majority of the research on articulated body vehicles has been done by Hirose of the Tokyo Institute of Technology. His studies of articulated bodies began in 1971 from a curiosity of why snakes, which have no appendages, can move so effortlessly across all types of terrain. He has built many articulated body vehicles with the most caterpillar-like being the ACM IV (see Fig 2). This vehicle has twenty body segments and propels itself with only wave motion (Hirose, Morishima 1990).

Chirikjian and Burdick (1990) of the California Institute of Technology have done a study of hyper-redundant robotic locomotion. Just as Hirose and Stulce, et al. have concluded they feel that rough-terrain locomotion should emulate caterpillars or slugs rather than man. Similar to the concepts and designs later proposed in this thesis, they tender the concept of using a variable geometry truss as the active mechanism for relative

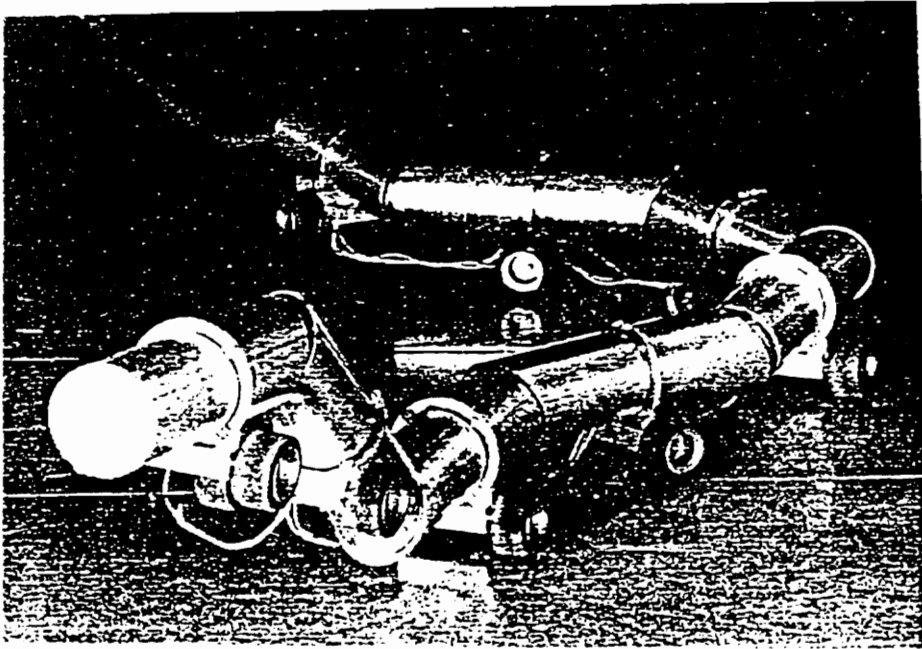


Figure 2: The ACM IV (Hirose and Morishima, 1990)

body control. Although they have investigated the kinematics of several motion gaits, they have neither built or designed a vehicle.

1.3 CATERPILLAR LOCOMOTION

To better understand the principles of crawling locomotion the authors of the Stulce, et al. paper studied two crawling arthropods, the millipede and the tent caterpillar (larva of *Malacosoma americanum*). They determined "that wave-like foot movements begin at the rear of the body and propagate forward in the direction of travel." The paper goes on to present the following passages.

Wilson (1966) claims that this scheme of locomotion is true for almost all animals with appendages. As mentioned earlier, the caterpillar is especially interesting because of its ability to easily overcome obstacles that are large relative to its size, despite having very limited sense organs (Dierl, 1972). Caterpillars have twelve body segments: eight of these have pairs of short legs. The two feet on either side of each body segment move in-phase, as a unit. This is in contrast of most other animals in which the motion of any lateral pair of legs is exactly a half cycle out of phase (McGhee, 1985). By varying the frequency of their locomotion cycle the caterpillars can achieve different speeds. Based on their observations and their previous work in the area of variable geometry truss (VGT) technology, the authors concluded that a robotic system incorporating features of a VGT could closely emulate the crawling motion of caterpillars.

A preliminary study (Burgos, May, et al., 1989) supported this conclusion by indicating that variable geometry trusses would be feasible actuation mechanisms for the trunk motion of a multibody crawling vehicle (Stulce, et al., 1990).

1.4 VARIABLE GEOMETRY TRUSS TECHNOLOGY

A variable geometry truss is obtained when one or more of the members of a statically

determinate truss is made extendible. A VGT has several advantageous characteristics. By making some of the members variable in length the basic geometry of the structure does not change. Therefore, VGT's retain the excellent stiffness-to-weight ratio of static trusses. Also, in a VGT-based vehicle the failure of one or even several variable-length links will not disable the entire vehicle.

VGT's have previously been used for collapsible, adaptive, and actively damped structures, as well as for robotic manipulators and antenna controllers (Reinholtz and Gokhale, 1987). Most useful applications of the VGT concept have been based on the octahedral geometry (Miura and Furuya, 1985). The fundamental "cell" of the octahedral VGT geometry has twelve links forming eight triangular faces. Some are variable in length. By connecting these cells along their longitudinal axes a repetitive chain is formed. The number of degrees of freedom of this chain is equal to its total number of variable-length links.

It is difficult to create a VGT based structure that behaves as an ideal truss. To do so every connection between links (variable and fixed length) must be such that no bending moments or torques can be transmitted from one link to another. Spheric joints prove useful for modeling but their restricted range of motion often excludes them from actual implementation. Combinations of revolute joints have been used to emulate the spheric joints, but some configurations can transmit moments and torques between links. Thus the resulting structure is not an ideal truss but it still has much better stiffness characteristics than comparable open loop serial devices (Stulce, et al., 1990).

1.5 CURRENT CONCEPTUAL DESIGN FOR A CRAWLING VEHICLE

The following passage was taken directly from the Stulce, et al. conceptual design paper. The subsequent mechanical design for a crawling vehicle is based on this conceptual design.

The conceptual design was divided into the lateral leg pair structures and the vehicle body actuation system. To imitate the caterpillar's excellent mobility, the lateral leg pairs must be distributed along the length of the vehicle, and each leg pair must be able to move independently of the leg pairs ahead of and behind it. This is achieved by having a lengthwise alternating configuration of a leg pair, then an actuated unit, then another leg pair, and so forth. Such an arrangement is shown in Figure 3.

The legs of the crawling vehicle are not actuated. Connected to each leg is a foot having two relative, passive rotational DOF's to accommodate surface irregularities. Also, it may be desirable to have elastomer pads on the feet for softening impact forces with ground. The conceptual design includes storage capacity to carry on-board drive systems, control systems, sensors, and payload. Rigid frames or box structures hold and protect the cargo. Since the lateral leg pairs need to be mounted on rigid sections of the vehicle, it is advantageous to divide the storage capacity into several payload boxes, and rigidly attach one pair of legs to each payload box. Thus, the payload boxes also serve as the leg pair support structures. In addition, this has the desirable effect of distributing the vehicle's mass evenly. This assembly of payload box, two legs and feet is rigid and moves as a unit; for convenience, it will be referred to simply as a "leg pair".

The vehicle body actuation system can be driven by either hydraulic or electric motors. Each actuated unit is a variable geometry truss connected between two leg pairs. Individual actuated units have their own dedicated drive system.

The proposed actuation scheme for the crawling vehicle is based on a "Stewart's platform" type VGT. As can be seen in Figure 3, the two lateral faces, each with three pairs of nearly coincident joint locations, act as rigid planes and facilitate attachment to the adjacent leg pair assemblies. Each of the six longitudinal, variable-length members contains a linear actuator. This

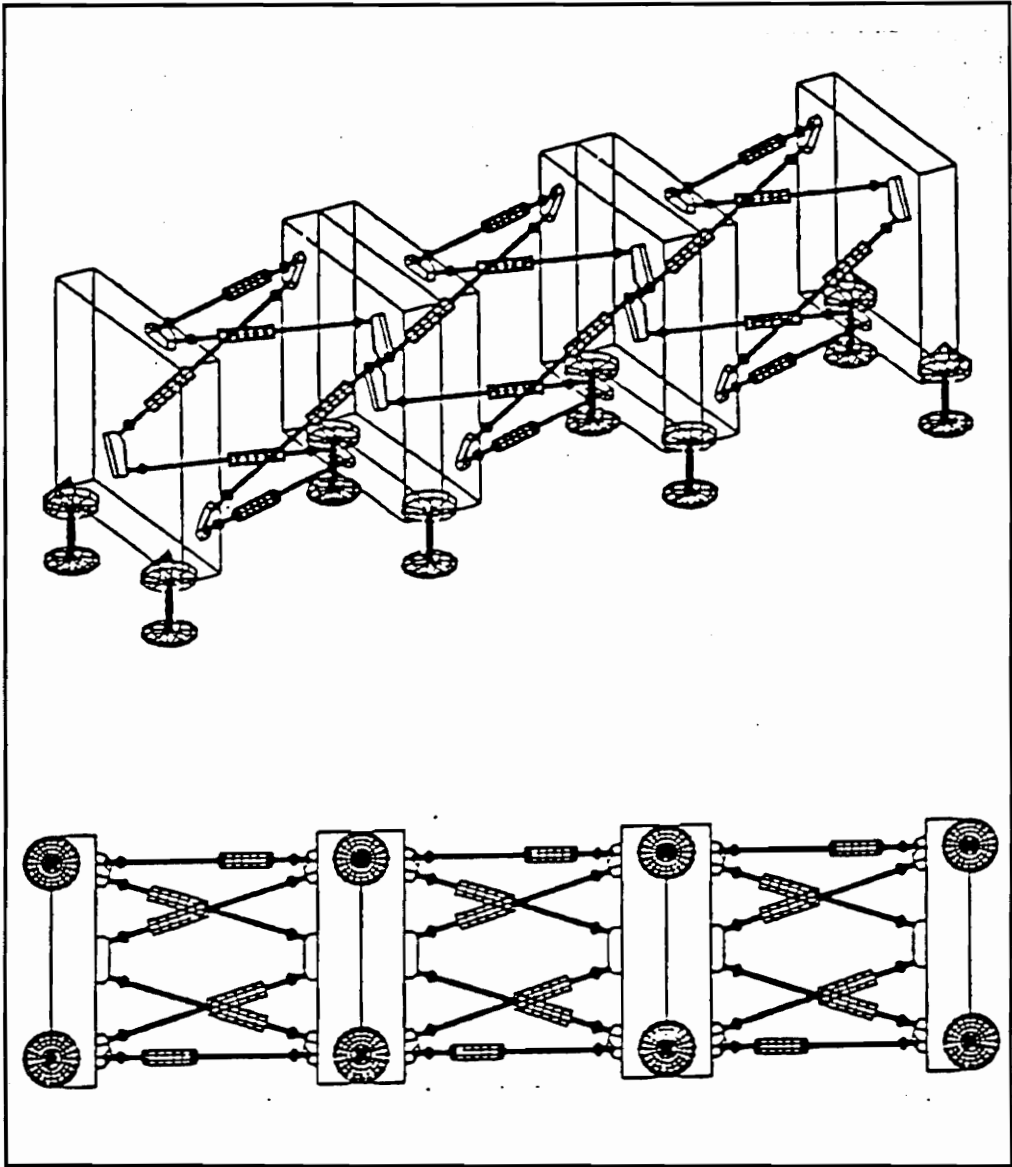


Figure 3: VGT-Bases Crawling Vehicle
(Stulce, et al., 1990)

configuration produces six-parallel DOF's contained within a single octahedral cell. Within a limited workspace, each leg pair of the machine can take any arbitrary position and orientation relative to its adjacent leg pairs, thus giving the crawling vehicle increased flexibility and terrain adaptability. Moreover, the solution of the inverse-kinematic problem of Stewart's platform is available in closed form (Stewart, 1965-66). It should be noted, however, that other actuation schemes are possible for use in a multibody passive-legged vehicle.

The proposed VGT configuration produces a repetitive, multibody, longitudinally segmented structure which is similar to the body of several arthropods, such as the caterpillar. Previous designs of walking machines used the legs as the active elements. As a result, complicated power transmission system between the prime mover and the legs were required. A VGT crawling vehicle is based on attaching non-actuated legs to the lateral planes of the truss body and using the variable links of the truss body as active power elements for mobility as well as power transmission. Thus, an advantage of this machine may be a simplified power transmission design.

Several other characteristics of a VGT crawling vehicle may prove to be advantageous. The multibodied, multiple degree of freedom structure allows the vehicle to control the location of its center of gravity relative to its base of support. This feature coupled with having several pairs of legs makes the machine highly stable and will provide excellent terrain adaptability and maneuverability. In addition, this stability allows the vehicle to move in a quasi-static manner, reducing the controller complexity. Since all of the leg pairs and actuation units are essentially identical, it may be possible to use a single basic control strategy for all the leg pairs, rather than having a separate control scheme for each of them. Finally a multibody system has the ability to send multiple "waves" of foot steps along the body for increased speed, while still maintaining static stability.

A vehicle of the configuration proposed above presents some interesting control problems not envisioned until the mechanical design of the concept presented in this thesis was undertaken. The problem arises from the attempt to control the six degrees-of-freedom of each leg pair with twelve actuators (six to the front and six to the rear). In the work presented in this thesis and in parallel work by Brennan this problem has been

labeled "coupled redundant control". The coupled redundant control problem is addressed in depth in Chapter 3.

CHAPTER 2

OVERVIEW OF PROJECT OBJECTIVES

2.1 Motion Programming

Assuming the coupled redundant control problem can be solved attention must be directed towards efficient motion programming of the crawling vehicle. Motion programming involves path planning and gait (sequence plans for lifting and placing legs) selection. There has been some investigation on the possible schemes of locomotion for the crawling vehicle. The following is the motion programming section of the paper entitled "Conceptual Design Of A Multibody Passive-Legged Crawling Vehicle" (Stulce, et al., 90).

An important aspect of any autonomous, electro-mechanical system is the concept of motion programming. In the case of a crawling vehicle, it has been found that many gaits (sequence plans for lifting and placing legs) are possible. In order to control the machine in these various modes it is necessary to generate data about the extendible links as a function of the cycle time. This is referred to as motion programming.

For control purposes, a hierarchy of motion was conceived for the crawling machine. The hierarchy descends from a gait, through a machine step, through a leg step, and down to an individual movement. An individual movement is an incremental displacement of a leg pair. A leg step is the completion of a sequence of individual movements which moves a leg pair one step forward. A machine step, or locomotion cycle, is when all of the machine's legs have each completed a step. A gait is one or more machine steps which use the same distinct leg step sequence.

The VGT based crawling vehicle is geometrically capable of many types of gaits. Two of the classes of gaits, wave gaits and sidestepping gaits, are introduced via several examples. For the sake of simplicity, the example leg steps are broken down into a sequence of three rudimentary movements: lift a leg pair, move it forward, and lower it to the ground (see Figure 4).

Wave gaits are used by many animals, including the caterpillar. McGhee and Song (McGhee, 1985) determined that wave gaits have optimal longitudinal stability for hexapod vehicles. The following three examples demonstrate three basic maneuvers using a very simple wave-type gait.

Forward motion, shown in Figure 5 is initiated by lifting the rearmost leg pair, moving it forward a specified step length, and then lowering it. As the feet of the rearmost leg pair touch the ground, the second leg pair from the rear is lifted and goes through the movements of its foot step. Then the third leg pair moves, and so on. To an observer, this motion looks like an actual wave traveling up the body of the vehicle. Reverse motion can be implemented in a similar manner, only the leg steps start with the frontmost leg pair and progress toward the rear of the vehicle. To achieve higher speeds, the controller could initiate new machine steps before the first machine step is completed, causing several "waves" to progress along the vehicle simultaneously.

Turning motion, shown in Figure 6, is produced by lifting a leg pair, rotating it through an angle, translating it along a chordal direction, and then lowering it. The turning radius and total angle are pre-specified and the vehicle follows a circular arc until it completes the turn.

Simple climbing motion, shown in Figure 7, is produced by bringing the rear leg pairs forward, lifting the front leg pair and extending it over the obstacle, and repeating this pattern until all the leg pairs have cleared the obstacle. These steps are fundamentally the same as the forward motion sequence except when the lift movement must be increased to clear the obstacle.

A VGT crawling vehicle would also be capable of performing sidestepping gaits. The authors did not observe the millipede or caterpillars using sidestepping. However, the preliminary study (Burgos, May, et. al, 1989) indicated that sidestepping gaits might result in faster vehicle speeds than wave gaits.

Sidestepping motion, shown in Figure 8, is produced by simultaneously lifting the two inner leg pairs, swinging them to the side, lowering them, then lifting the two outer leg pairs, swinging them to the side, lowering them, and then repeating the sequence.

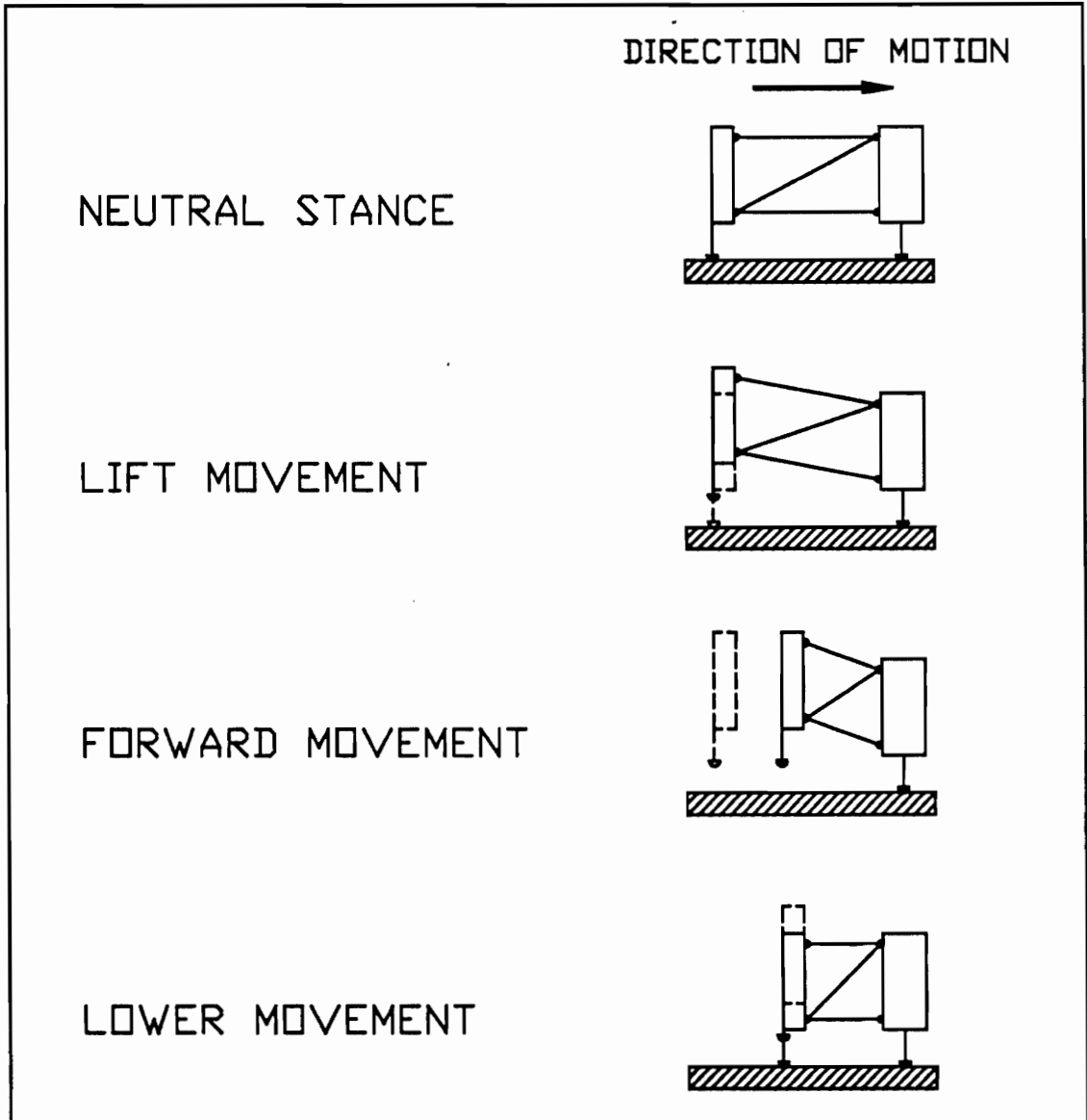


Figure 4: A Leg Step (Stulce, et al.)

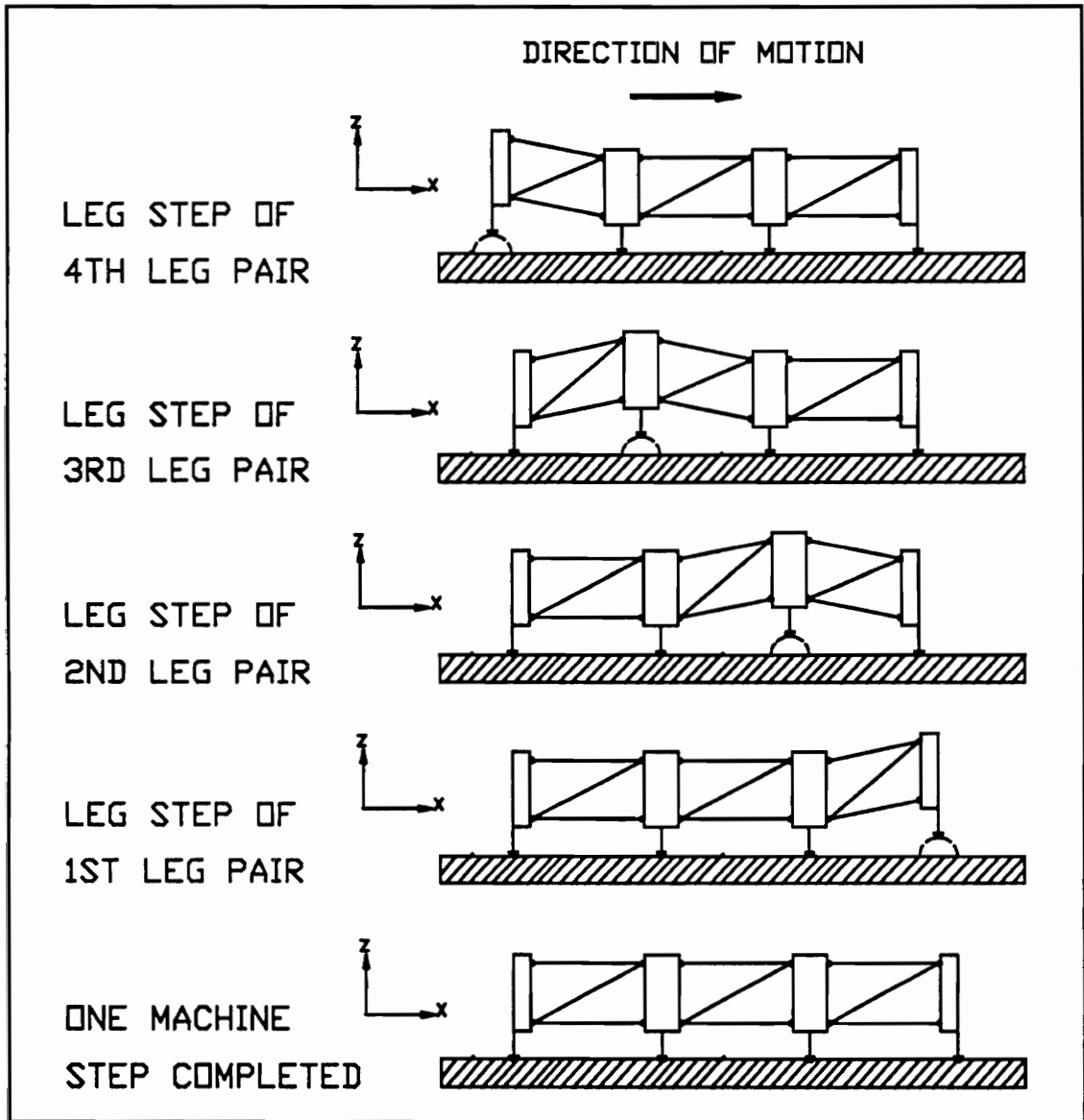


Figure 5: A Machine Step (Stulce, et al.)

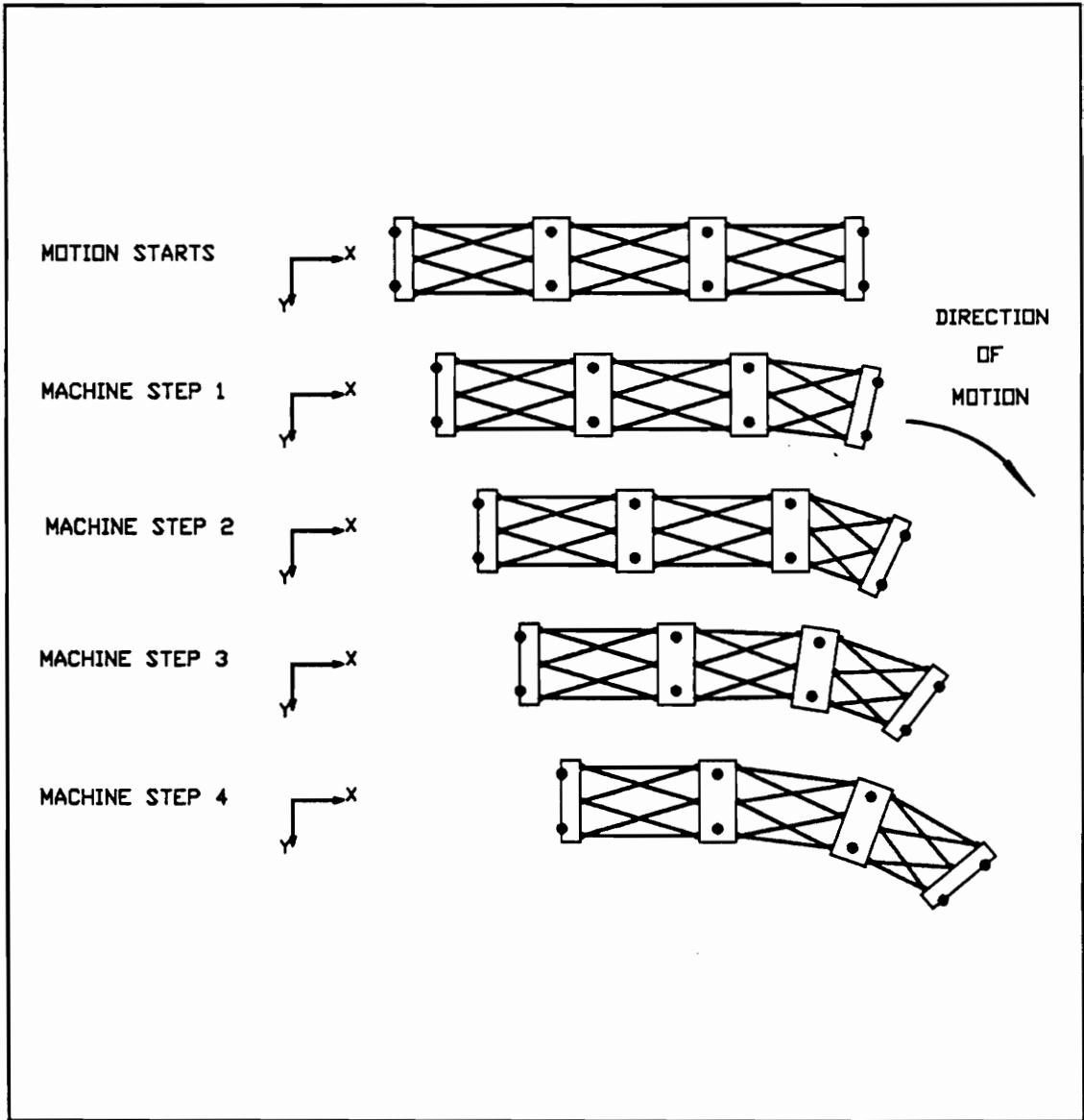


Figure 6: Turning Motion (Stulce, et al.)

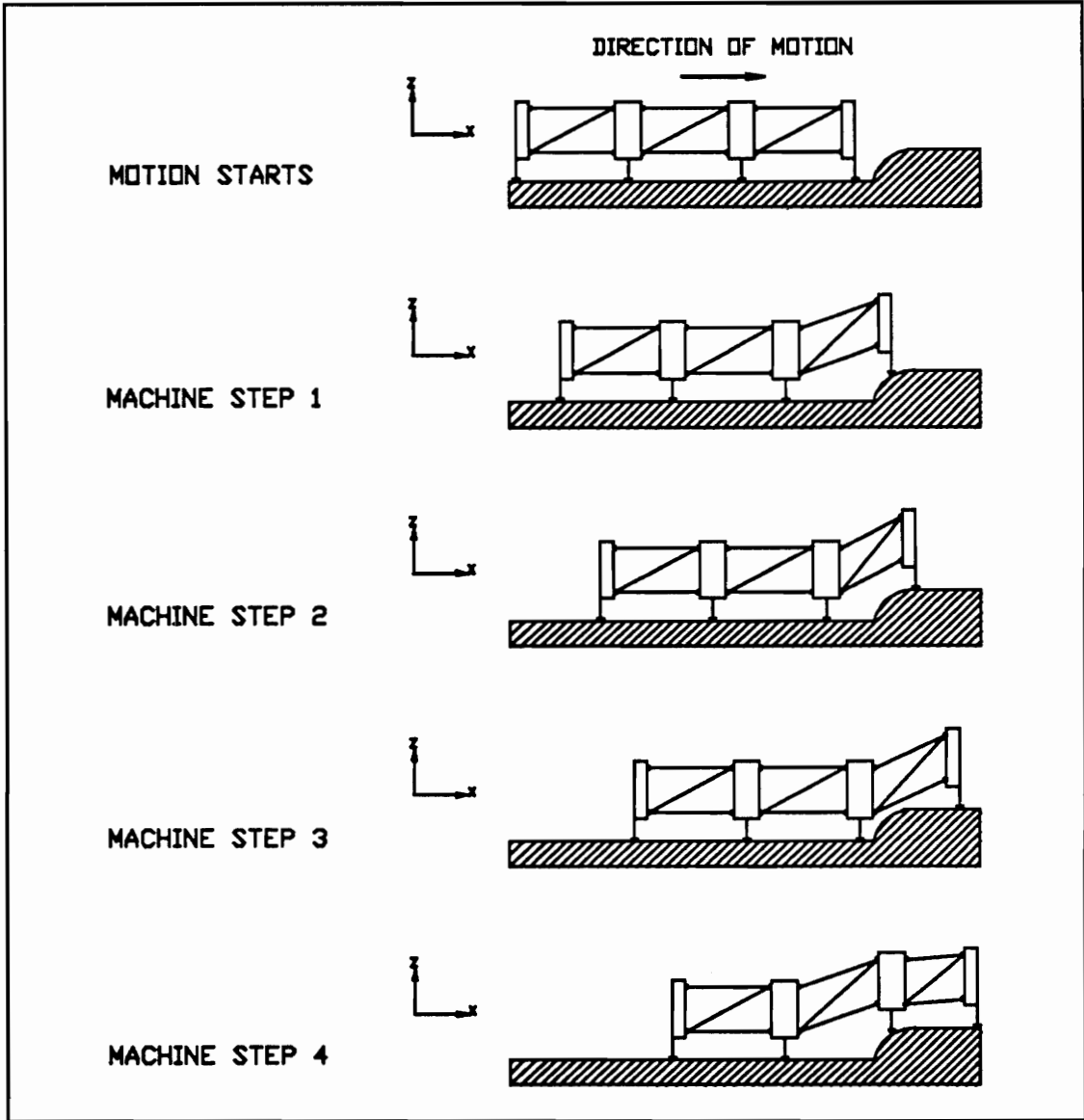


Figure 7: Climbing Sequence (Stulce, et al.)

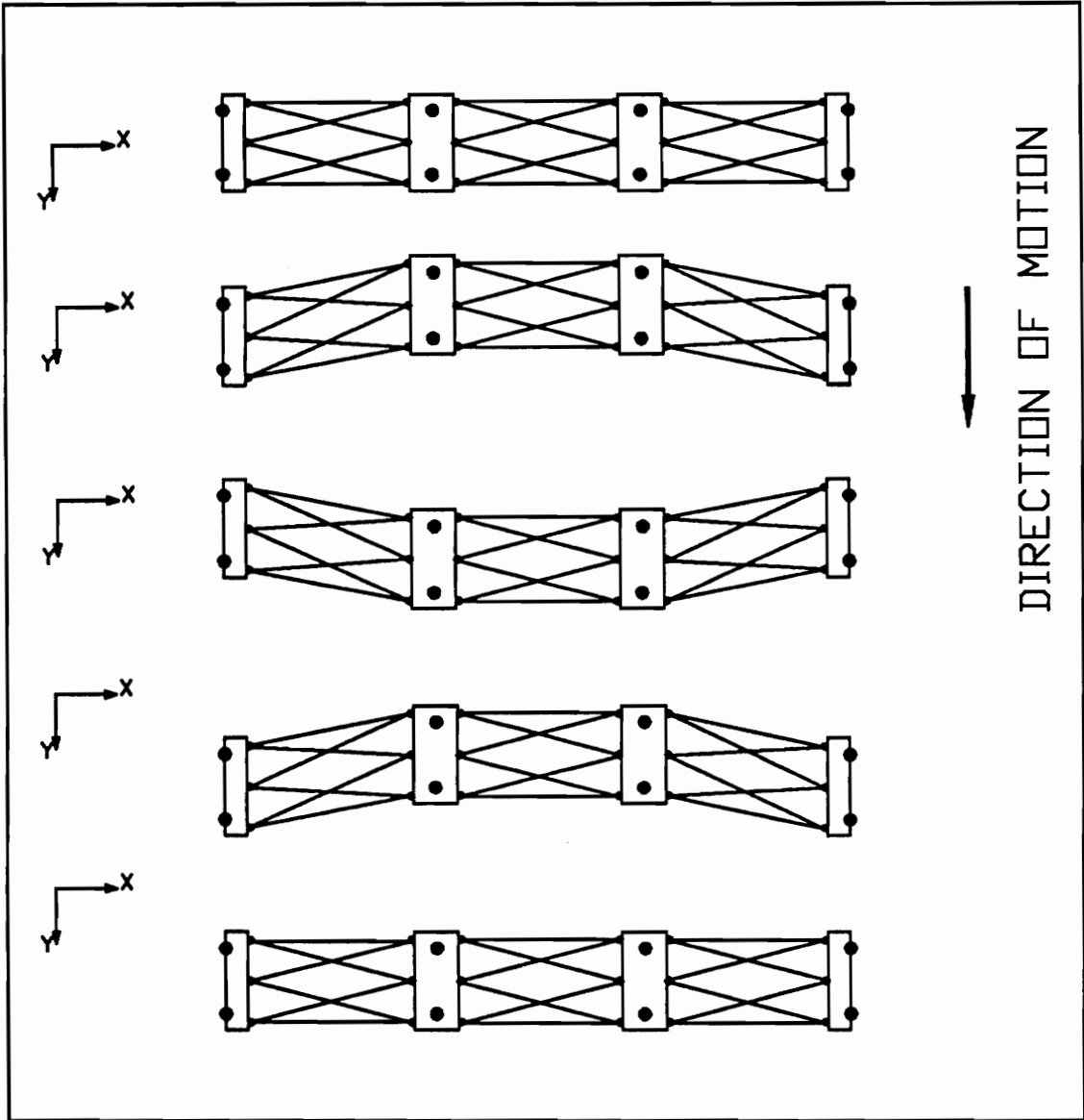


Figure 8: Sidestepping Sequence (Stulce, et al.)

The mechanical design submitted in this thesis introduces no major changes to the proposed locomotion schemes. The design will enable the vehicle to achieve all configurations necessary for the motion patterns proposed in the conceptual design paper (Stulce, et al. 1990). However, the coupled redundant control problem must be solved for efficient, realization of the aforementioned motion patterns. This problem will be described in great detail in the following chapter.

2.2 POSSIBLE VEHICLE APPLICATIONS

There are many envisioned uses for the crawling vehicle. The vehicle can be used in any circumstance where a teleoperated, robust, highly flexible vehicle is needed. Several specific envisioned uses will be described in detail here.

The proposed crawling vehicle concept has been submitted to the Space Exploration Outreach Program as a potential rover for the exploration of Mar's surface. In boulder-strewn soft-sand terrains such as Mars, maximum flexibility and survivability are critical to the success of the mission. The crawling vehicle's extremely high stability and large range of motion may offer superior terrain traversability, making it able to access regions that could not be explored by other types of rovers.

Minor variations on the baseline vehicle can be used to perform space station maintenance tasks. With the addition of appropriate securing mechanisms, e.g. griping feet, this vehicle can traverse orbiting trusses such as those envisioned for the space station, just as caterpillars traverse twigs and branches.

The crawling vehicle could also be used in areas too hazardous for human occupation. One such area is nuclear facility maintenance. Fitted with the proper instruments or tools, the crawling vehicle could enter high radiation areas to repair facilities or gather data while the controller remains safe. The same concept could be used in many military applications such as the clearing of mine fields.

Other proposed uses include pipeline maintenance, construction applications, and mining.

2.3 FURTHER USES OF CRAWLING VEHICLE TECHNOLOGY

Technology either developed or advanced as a result of the crawling vehicle project may be of use in other areas. For instance, the coupled redundant control experiments would not only help advance the crawling vehicle project, but would also be a significant advance in applied control theory. This technology could be used in a variety of other new robotic systems. Since biological studies show that humans and animals use redundant control for their high accuracy movements, it is believed that similar techniques would improve robot performance.

The crawling vehicle technology can also be adapted to space-based manipulators. By fixing one end of the crawling vehicle, the machine can be used as a high-degree-of-freedom robotic arm. Possible applications of such a robotic arm include use aboard the space shuttle or as a flexible construction crane for assembly of the space station.

CHAPTER 3

COORDINATED REDUNDANT CONTROL

The coupled redundant control problem described in Chapter 1 must be solved if the crawling machine is to become a feasible, efficient rover. Patrick Brennan currently in the masters program at Virginia Polytechnic Institute, is working to develop an appropriate control scheme to solve this problem. Many passages in the following sections are taken directly from his proposed "Coupled Redundant Control" thesis. These passages are set off in single-spaced blocks.

3.1 PROBLEM STATEMENT

Brennan's proposed thesis offers the following description of the control problem.

The proposed design of the crawling machine implies that there are points on the chain where we wish to control six degrees of freedom with twelve actuators. We believe this can lead to a situation where actuators may enter contention, or a "tug of war", particularly since the kinematic relationships between the actuated variables are nonlinear. We expect this situation may give rise to undue stresses and vibration, or even the failure of a link or an actuator in the machine. The redesign of the machine to eliminate the redundant degrees of freedom is not possible; the machine could not fulfill its function without giving rise to redundancy of this type.

The purpose of this investigation is to determine methods which might be applied to the control of the crawling machine to mitigate or eliminate the effects of such contention.

3.2 MOTIVATION FOR THE WORK

Also from Brennan's proposed thesis is the following passage describing the motivation for the coupled redundant control work.

The problem of actuator contention arose out of work on the application of a variable-geometry truss (VGT) to a crawling machine. In order for the machine to crawl, each leg pair must in succession be picked up, moved in the desired direction, and placed back down. This presents no special problem for the front and rear leg pairs, since they each only have six actuators controlling them. It's a different story for the middle leg pairs, however, since the crawling machine's design implies that we are trying to control each middle leg pair's six degrees of freedom with a total of twelve actuators.

In order to see why this is a problem, imagine a mass positioned between two linear actuators. The actuators are collinear, and each has its own controller with a different control law governing it. When these controllers each receive a command to move the mass to a new position (the same command is given to each controller), they will each attempt to move the mass in a fashion particular to the individual control law governing it; for example, one controller may be slightly overdamped while the other is critically damped. If this is the case, then there will be points in time where the actuators are trying to push or pull the mass in different directions; even when they are each trying to move the mass in the same direction, they may be trying to do so at different speeds. The inevitable result will be stress upon the system, and the possibility of excess vibration, mechanical failure or amplifier or motor burnout.

Brennan proposes four strategies to deal with the problem of actuator contention. In his words they are:

- (1) precision kinematic solutions (open-loop control);
- (2) building passive compliance into the system;
- (3) force feedback control of the actuators (closed-loop control); and
- (4) passive mechanisms (backdriving).

The first category involves using the inverse kinematic solutions for the machine to generate actuation commands for the actuators. This pure open loop control strategy is

probably unacceptable because it provides no feedback on the state of the system e.g. stress levels in links.

The second category assumes that the deflection in all machine components due to actuator contention will be small. The solution suggests using a material with high elasticity to absorb the stresses rather than introducing the stresses to the components. This solution is simple and inexpensive but would reduce the positional accuracy of the machine.

The next category offers the most likely solution. This category is best described by the following passage from Brennan's proposed thesis.

The third category of control, an active feedback-driven system, is the primary candidate technology for this project, although it is realized that this technique will probably have to be augmented by one or more of the remaining alternatives. An active feedback controller will attempt at all times to satisfy the position commands while minimizing the stress on the system. The notion of the current system is one in which the controller generates position commands for "master" actuators. The master actuators, in the course of adjusting the configuration of the machine, will generate stress which the "slave" actuators will respond to, following the master actuators and effectively removing contention, most particularly if the slave controls are faster than the master controls.

The fourth category depends on the ability of the actuator mechanisms to be backdriven. For the envisioned hardware of the machine this is not a viable solution. Furthermore, using only six actuators to position the leg pair while allowing the other six to passively follow is very inefficient.

3.3 MECHANICAL DESIGN OF THE TEST RIG

In order to study the coupled redundant control problem an apparatus that could create the problem was needed. A test rig was designed and constructed for this purpose. Although the problem as it pertains to the crawling vehicle involves twelve actuators and six degrees-of-freedom, the apparatus can effectively model the problem with two actuators and one degree-of-freedom.

Originally it was believed that a simple apparatus, with two linear actuators and the mounting arrangement shown in Figure 9 would suffice to fully study the problem. By placing a mass between the two actuators a simple linear case of the control problem as described in the previous section could be created. Further thought revealed that the simple rig was unable to create the nonlinear relations that existed between the actuators on the proposed crawling vehicle. Therefore, several other test rig configurations were investigated with the final choice being the V frame test rig shown in Figure 10. The V (the V is formed by the two followers and the coupler connecting them) frame replaces the mass in the simple collinear case but since the actuators are no longer collinear, the relations between them are now nonlinear.

The mechanical design of the test rig began at the follower pivot point (see Figure 10). The requirements for this pivot were that the axes of rotation for the two followers and their associated potentiometers be collinear. One-turn potentiometers are used to determine the position of each follower. Furthermore, the followers must rotate

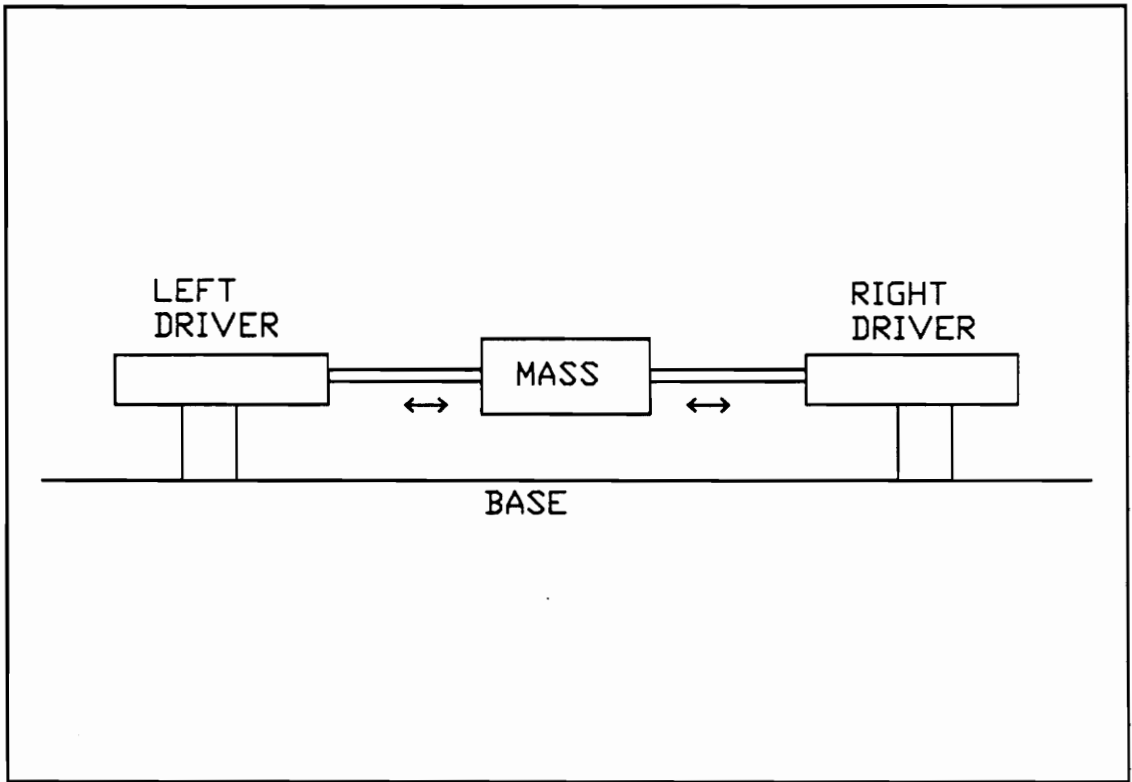


Figure 9: Simple Test Rig

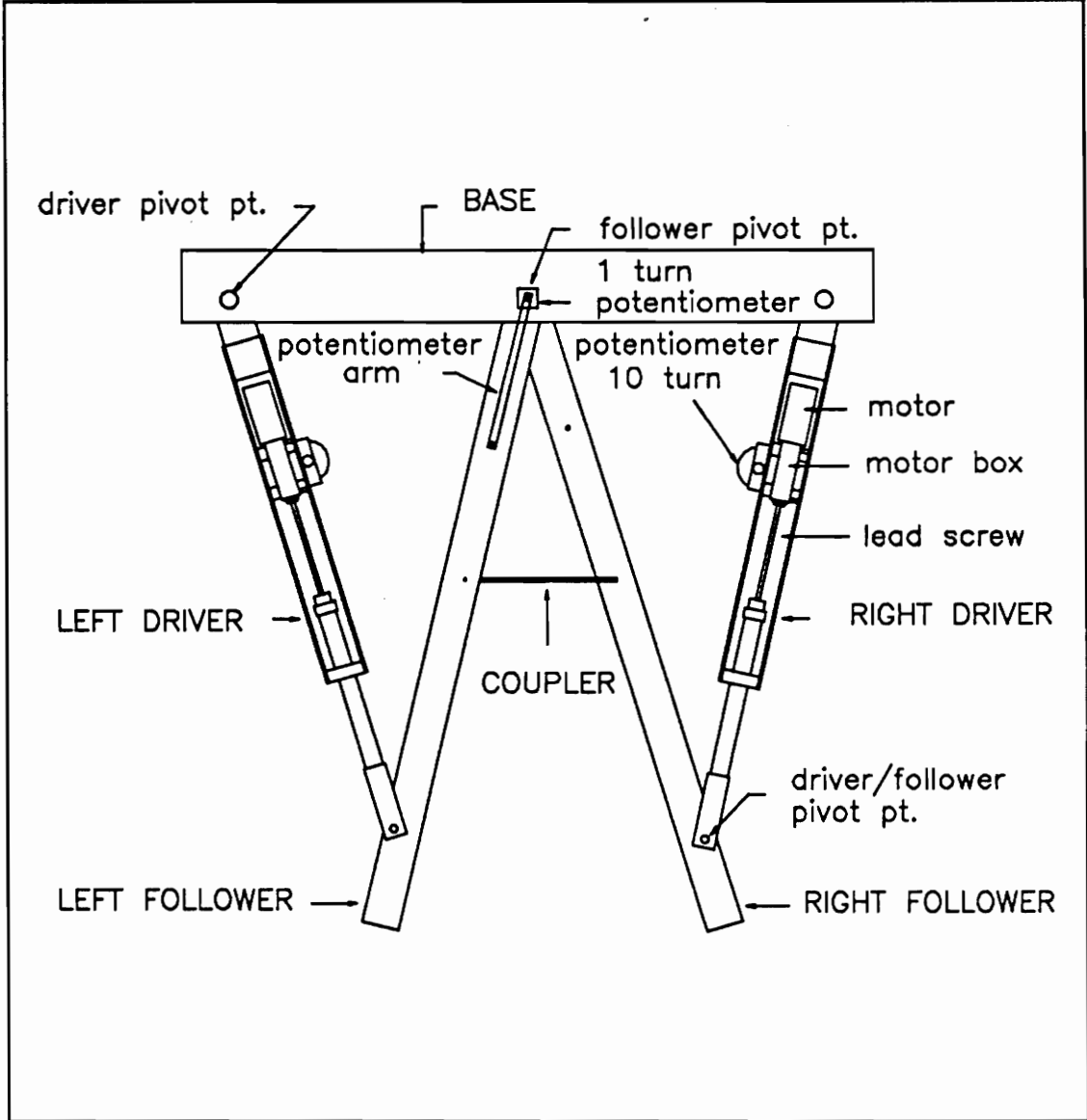


Figure 10: V Test Rig

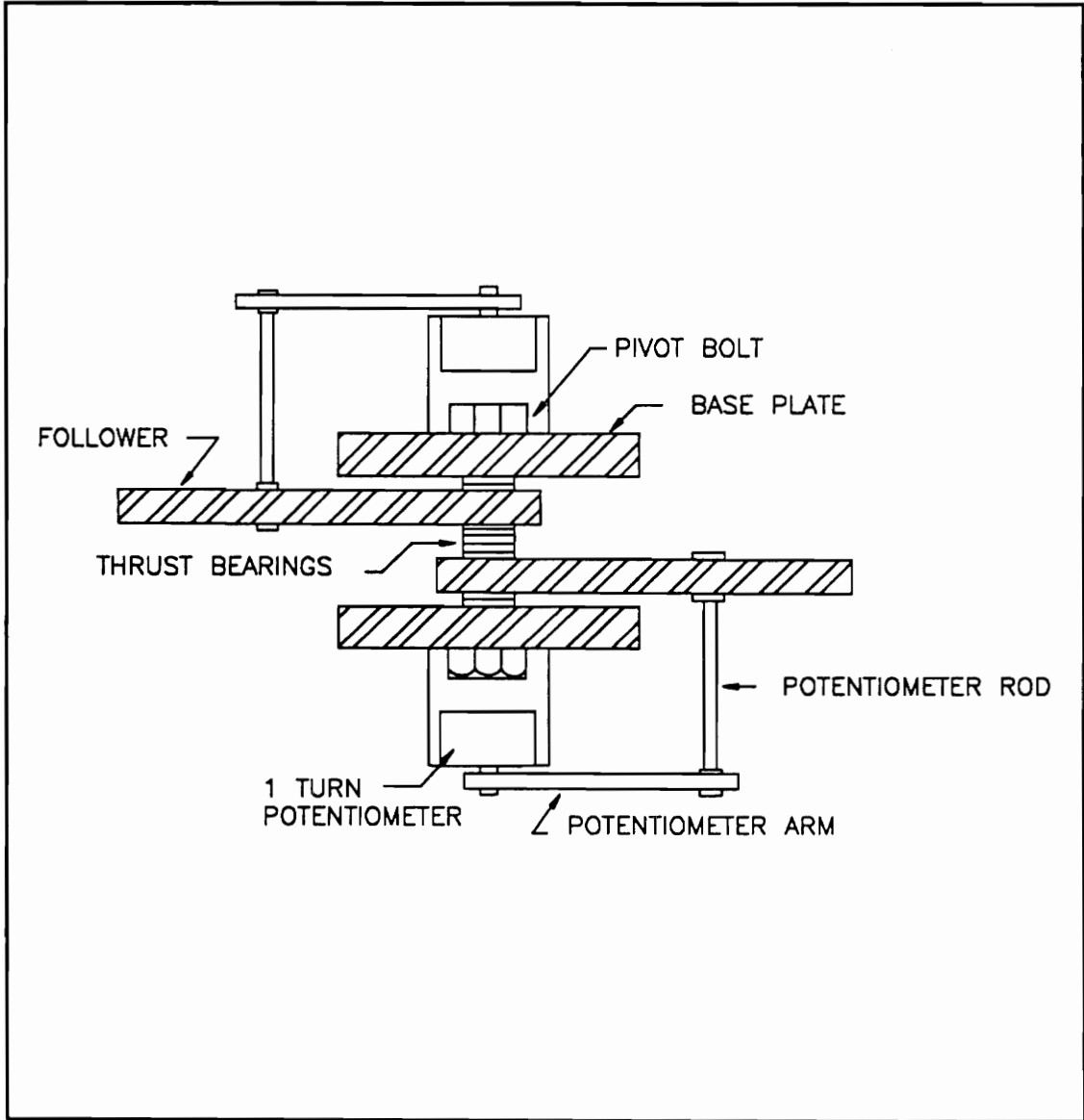


Figure 11: Follower Pivot

independent of each other. The follower pivot point design shown in Figure 11 resulted from satisfying these mechanical requirements and utilizing materials on hand. The pivot was created by mounting a 5/8 in. bolt through an upper and lower 0.375 in. thick by 4 in. wide (9.5mm X 102mm) base plate. At first the bolt was to be supported by only a lower base plate but an upper plate was added to alleviate the twisting of the base plate and bending of the bolt under the weight of the cantilevered followers. The pivot holes in the followers were fitted with 5/8 in. cindered bronze, oil-impregnated bushings which were on hand (thus, the reason for the 5/8 in. pivot bolt). The followers themselves are 0.375 in. by 2 in. (9.5mm X 51mm) aluminum pieces (this material too was on hand). They are separated from each other and from the base plates with 5/8 in. by 1/8 in. by 1 in. thrust bearings of the same material as the bushings. The bracket that holds the one turn potentiometers is a two inch length of 2 in. by 2 in. (51mm X 51mm) aluminum tube. Concentric mounting holes, one for attachment to the base plate by the 5/8 in. pivot bolt and one for the shaft of the potentiometer, were drilled. The one-turn potentiometers were chosen for their high resolution while measuring small rotations. Each potentiometer was attached to its respective follower by a potentiometer arm (extending parallel to the follower) and potentiometer rod (extending between the pot arm and follower).

The actuator mechanisms (left and right driver) are a combination of motor, motor box, and lead screw assembly. The box absorbs all thrust loads through a series of bearings and clamps on the screw, thus protecting the motor. The lead screw extends and retracts

a tube fit with a lead screw nut at one end and a bracket for attachment to the follower at the other. The motor, motor box, and lead screw assembly are braced by two (one on each side) 1/8 in. by 2 in. by 26.5 in. (3.2mm X 51mm X 673mm) aluminum pieces. These braces serve to prevent bending in the lead screw. A piece of bakelite fit with a hole for the tube to slide through is attached to the end of the braces. The hole is collinear with the lead screw so that the block of bakelite serves as a guide for the tube.

A piece was required at the end of the braces that would serve as a pivot for the entire driver assembly. The followers are not centered between the base plates (one is above center, one is below, refer to Figure 11). Each driver must be coplanar with its respective follower, therefore, one driver is above center and one is below. The pivot piece must allow for the coplanar alignment. This was accomplished by fashioning the section of the pivot piece that contains the pivot hole so that it was much thinner than the distance between the base plates. The space between the pivot piece and base plates allows for movement of the driver assembly along the longitudinal axis of the pivot bolt for correct alignment with the follower. The space between the pivot piece and each base plate was then filled with an appropriate number of the cinkered bronze thrust bearings. The pivot piece itself is fit with a cinkered bronze bushing and is attached to the braces which run the longitudinal length of the driver assembly.

The coupler between the followers serves two purposes. By connecting the followers, the test rig is left with one degree-of-freedom. The two actuator, one degree-of-freedom case is now established. The nonlinearity of the system and the natural variation from

drive to drive will cause the followers to rotate at different rates when the drivers are activated. The coupler will therefore undergo strain as the angle between the followers fluctuates. The measured strain in this coupler is the primary feedback in the control scheme. The coupler is a length of aluminum channel beam. This beam is thin enough to experience measurable strains under a light load (resulting from angle incompatibility between the followers) and will also resist buckling under compressive loads. The placement of the coupler between the followers has no bearing on the system in general. However, by placing it lower (nearer the pivot) in the V increases the strain in the coupler.

The overall scale of the test rig was determined by the size of some preexisting components and the space available to operate the rig. The first dimension specified was the base (refer to Figure 10) length; this was picked to fit lab space available. With the base length established several iterations of geometrical solutions were executed to determine the optimal distance (distance d_3 in Fig. 12) between the follower pivot point (centered on the base) and the driver pivot points (DPR and DPL). The goal of these iterations was to determine the location for the driver pivot points that allowed the greatest angle of rotation for the entire V . It was concluded that within the range available (the length of the base) changes in d_3 have little effect on the angle of rotation. Therefore, the driver pivot points were placed at the maximum d_3 possible (ends of the base) where they would have maximum mechanical advantage while rotating the followers.

The last dimension to be determined was the distance between the follower pivot point and the driver/follower pivot point (distance d_4). Again a geometrical solution was used to determine the distance that would allow for the maximum angle of rotation of the V.

The following is a description of the geometrical solution used to solve for d_3 and d_4 . All figures must be to scale. The solution began with the base line (line DPL, F, DPR in Figure 12). On this line the follower pivot point (F, center of base line) and two driver pivot points (DPL, DPR). Note that DPL and DPR must be equidistant from F if the V is to have an equal amount of rotation to each side. Centered on point DPL circle A with a radius equal to the maximum length of the left driver was drawn. Likewise, centered on point DPR circle B with a radius equal to the minimum length of the right driver was drawn. The uppermost point of intersection of these two circles was labeled point a. Line 1 representing the bisector of the V was then drawn from point F through point a. The angle α between the base line and line 1 is the maximum rotation the V may travel to one side of the test rig. As mentioned above, varying the length d_3 had an insignificant effect on angle α .

With d_3 established the last dimension to be determined was d_4 . The distance d_4 need not be the same for each follower in order for the test rig to operate. A different d_4 on the left follower than that on the right will simply yield different angles of maximum V rotation to each side. For simplicity, we desire a symmetric test rig, therefore distance d_4 was made the same for both followers. Furthermore, for all cases (symmetric or not) the endpoint of distance d_4 (measured from the driver pivot points) must lie inside circle

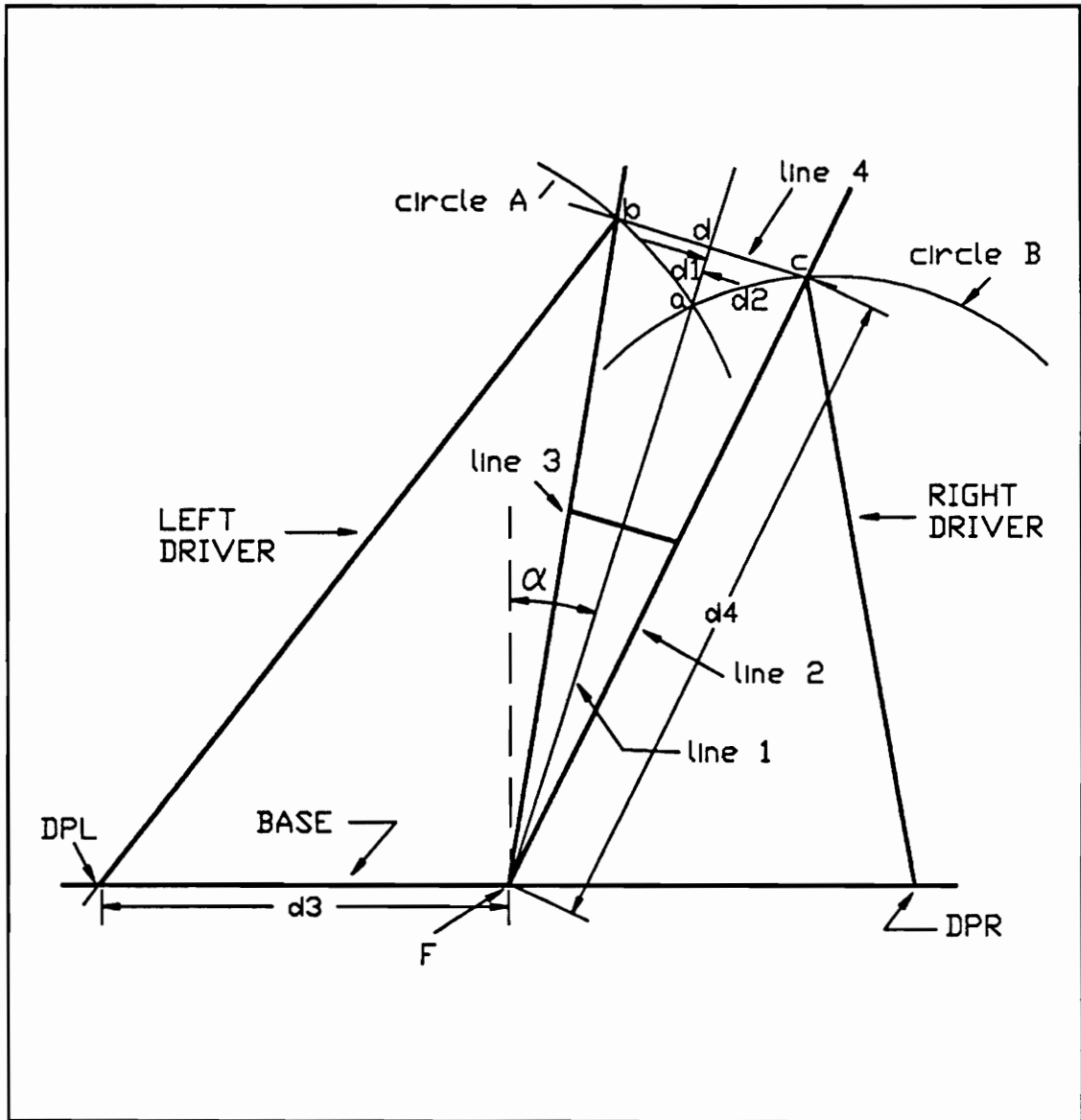


Figure 12: Graphical Solution Reference

A and outside circle B, since circles A and B represent the maximum and minimum length of the drivers. The above requirements were satisfied by finding point d on line 1 where d_1 was equal to d_2 (d_1 and d_2 are the perpendicular distances from line 1 to circles A and B respectively). Line 4 was then drawn perpendicular to line 1 through point d. Points b and c are the points of intersection of line 4 with circles A and B respectively. Points b and c mark the driver/follower pivot points. The distance from point F to point c was then measured to determine distance d_4 .

With all necessary dimensions determined a set of shop drawings was completed and the test rig was built.

3.4 OBJECTIVE OF CONTROL EXPERIMENT

To develop the proper control laws that mitigate the coupled redundant control problem, a set of experiments will be carried out. The experiments will use the test rig described in earlier sections to simulate the coupled redundant control problem. First level testing will be carried out using a computer simulation of the test rig while second level testing will use the actual test rig. The coupled redundant control experiments will evolve through two phases. The first phase of testing will use a naive control algorithm while the second will use a master-slave algorithm.

In the first phase of testing the followers of the test rig will be coupled but the control laws governing each driver will not be aware of this coupling. To move the V, the same position command will be given to each driver. The only feedback the drivers receive

is the position data from their ten turn potentiometers. Although the V will move to the desired position differences in control laws for each driver may induce stress on components within the system, particularly the coupler. The system is not aware of the stress on the coupler, and therefore no measures are taken to alleviate it. The buildup of this stress may lead to component failure.

The second phase of testing will use a master-slave algorithm that not only monitors the stress in the coupler but acts to alleviate this stress. To move the V the same position command is given to each driver. Though each driver receives the same position command they each receive different amounts of system feedback. The master drive will receive only position feedback from the ten turn potentiometer mounted to it. The slave drive will likewise receive position data from its ten turn potentiometer but will also receive data from the strain gauge on the coupler. The slave will not only attempt to position the V, just as the master does, but also to keep the level of stress in the coupler below some predetermined level. It will do so by adjusting itself (i.e. slow down, speed up, reverse direction) to alleviate the created stress.

The first phase of testing will allow for familiarization with both the system and possible magnitudes of stress in the coupler. The master-slave system of second phase testing is the arrangement that must be realized for efficient motion programming of the crawling vehicle. Once the correct control laws have been written for the master-slave algorithm, a maximum master drive actuator speed must be determined. Simply put, how fast can the master go with the slave still able to keep the stress levels in the coupler

below the desired level.

The techniques that were used to develop the control laws for the test rig drivers may later be applied to the actuation mechanisms on the crawling vehicle.

CHAPTER 4

KINEMATIC DESIGN

CONSIDERATIONS OF THE

CRAWLING VEHICLE

The material in the previous chapter dealt with the design of a test apparatus to be used in the coordinated redundant control experiments. This background is essential to the ultimate success of the crawling vehicle. Assuming this problem can be solved, the next objective is to construct and test a fully operational crawling vehicle.

There are a number of kinematic design considerations that affect the mobility and efficiency of the crawling vehicle. These considerations involve the initial configuration of the vehicle. Joint triangle configuration, joint triangle offset, and number of leg pairs are the primary kinematic design considerations. These considerations and the design choices made concerning them will be described in detail in the following sections. An analysis of the workspace and stability of the crawling vehicle will also be presented.

4.1 DESIGN CONSIDERATIONS

The current conceptual design presented in Chapter 1 utilizes the Stewart Platform as the actuation mechanism between leg pairs. The conceptual design recognizes that "perfect truss" joint triangles (one in which joint pairs meet at a single point) for the Stewart Platform would be difficult to build. The design therefore incorporates an offset

between each branch of a joint pair to allow space for mechanical components. The resulting symmetric joint triangle developed by Stulce, et al. (Stulce, et al., 1990) shown in Figure 13, is functional and visually pleasing but it is very inefficient. Starting with the arrangement proposed in the conceptual design (Stulce, et al., 1990) several revisions were executed to develop the optimum design for the desired motion.

The original joint arrangement shown in Figure 13 provides no mechanical advantage to the top and bottom links (actuators) for the initial lifting of a leg pair. This requires the diagonal links (actuators) to provide all initial lifting force in a vehicle movement. This is an inefficient use of the available actuator thrust in the top and bottom actuators.

Also, the arrangement shown in Figure 13 requires different link lengths in the neutral stance (leg pairs as close as possible on level terrain). This is an inefficient use of the available actuator stroke. Imagine the top and bottom links of Figure 13 at their minimum length ($L_{\min} + 0$ actuator stroke). The diagonal links are now required to be some length greater than their minimum length ($L_{\min} + X$ actuator stroke). This arrangement inefficiently uses the available extension of the actuator by requiring X stroke of the actuator when the vehicle is in the neutral stance. Simply put, some stroke of the actuator is spent but no leg pair movement is gained. Similarly, when the diagonal link is fully extended the top and bottom links will still have stroke available. This again is an inefficient use of the available stroke of the actuators.

To assist the diagonal links in lifting the leg pairs the joint triangles were reoriented so that the top and bottom links also had mechanical advantage with respect to lifting the

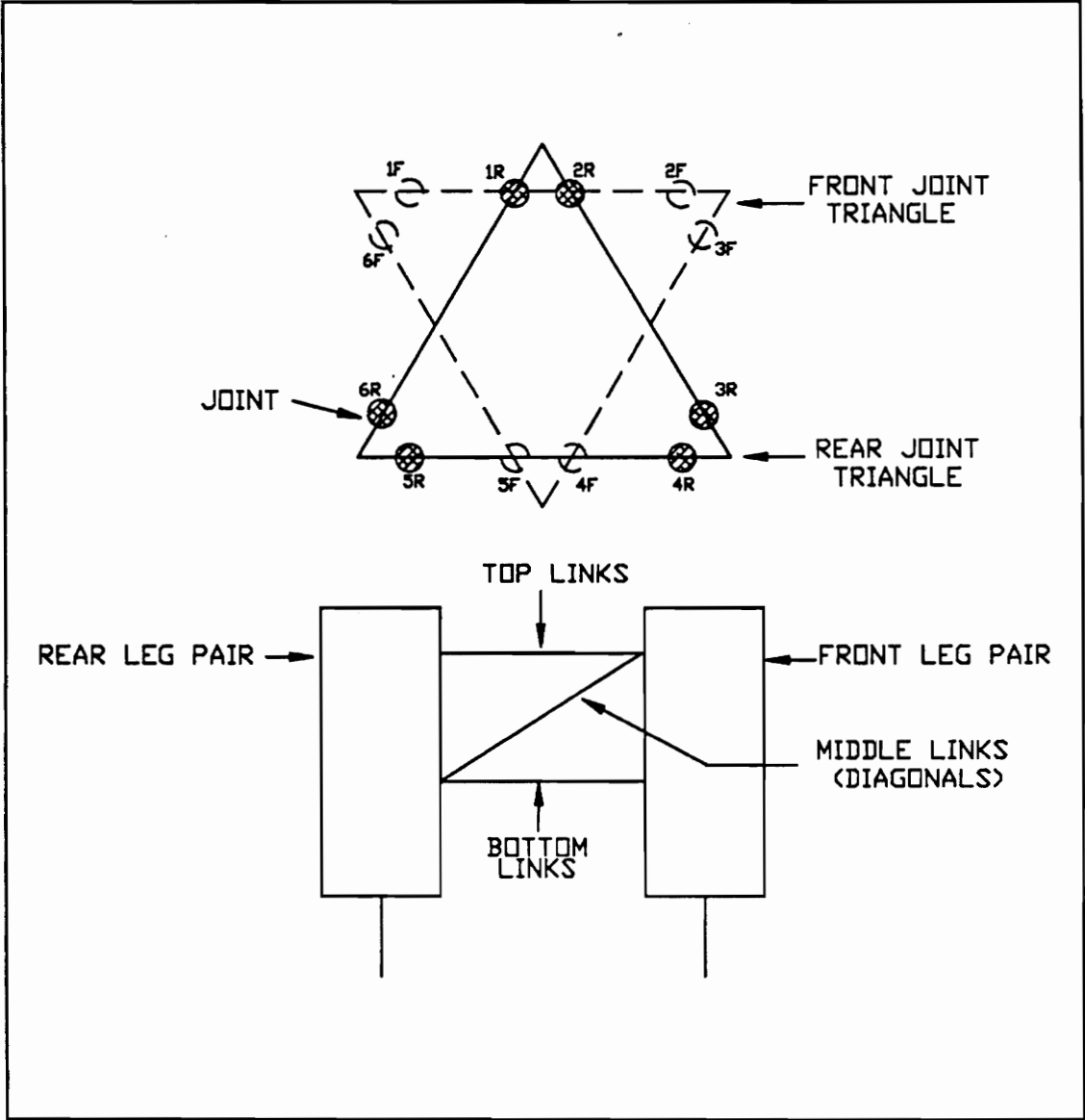


Figure 13: Original Joint Triangle Configuration (Stulce, et al., 1990)

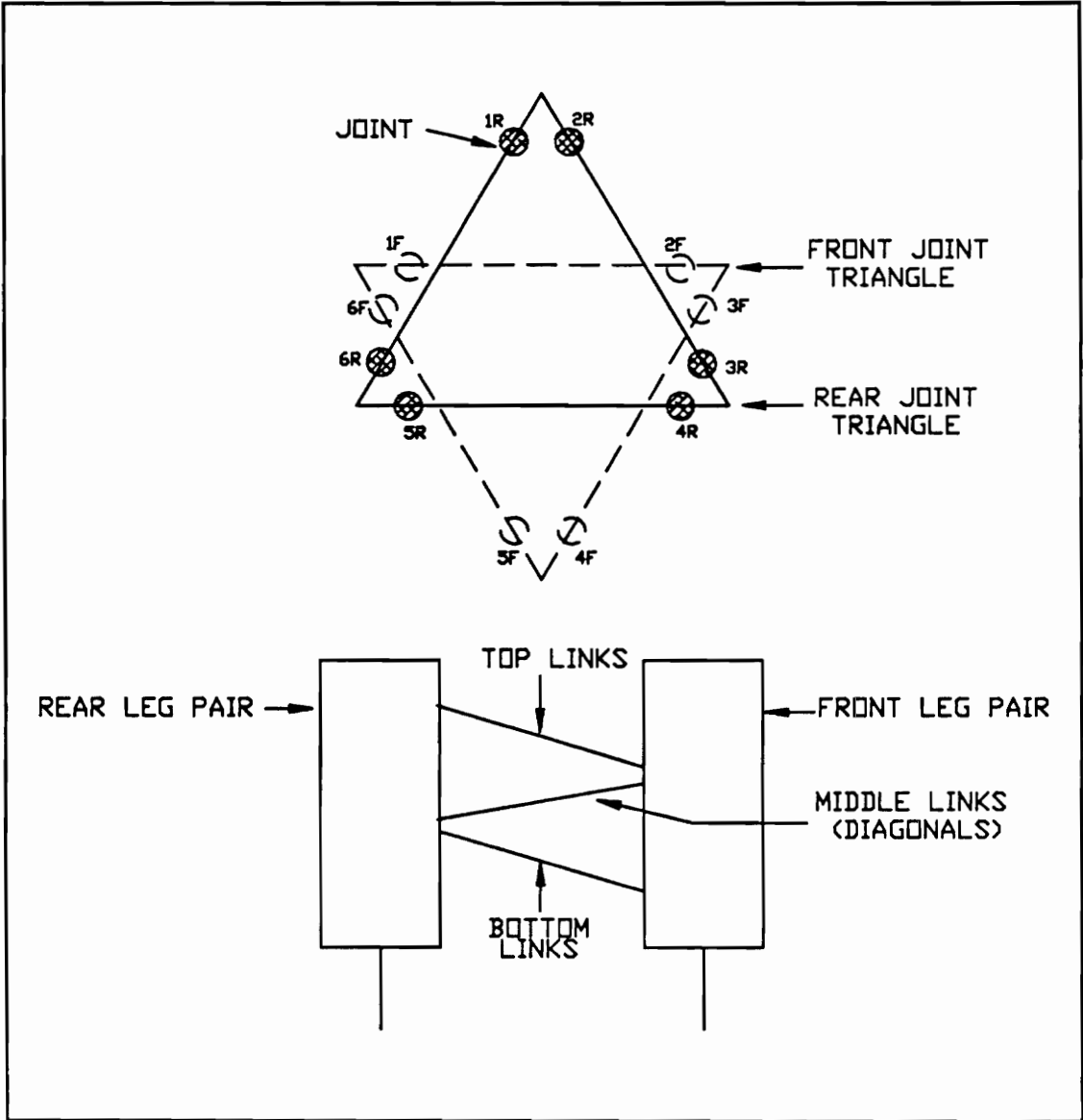


Figure 14: Revised Joint Triangle Configuration

leg pairs. The new joint triangle arrangement, shown in Figure 14, allows all links to contribute to the lifting of a leg pair. However the middle links (links that connect 3_F to 3_R and 6_F to 6_R , F designates triangle on front leg pair, R the rear triangle) are still a different length than the top and bottom links (links that connect 1_F , 2_F , 4_F , and 5_F to 1_R , 2_R , 4_R , and 5_R respectively) when in the neutral stance. This arrangement still wastes available actuator stroke.

The final joint triangle arrangement was acquired by modifying the previous arrangement. By repositioning joint locations 6_F , 6_R , 3_F , and 3_R the desired arrangement was produced. This arrangement, shown in Figure 15, allows all links to contribute to lifting and has all links of the same length ($L_{min} + 0$ stroke) when the vehicle is in the neutral stance. Furthermore, as a leg pair moves through a simple step (no pitching) on level terrain the actuators remain a common length. The maximum length leg pair step results in all actuators being at full extension. The use of 100% of available actuator stroke allows the vehicle to take the largest leg pair step possible with a given actuator.

Another consideration involved in designing the vehicle is the number of leg pairs necessary to accomplish the desired crawling motion. Specifically, the minimum number of leg pairs capable of crawling had to be determined. Simple stability analysis proved the minimum number to be four leg pairs. Imagine a vehicle with only three leg pairs. The center of gravity for this vehicle would be in the center of the middle leg pair. Now if the front or rear leg pair is lifted, the stability margin is extremely small. With one of the end leg pairs lifted the smallest external force or vehicle motion could cause the

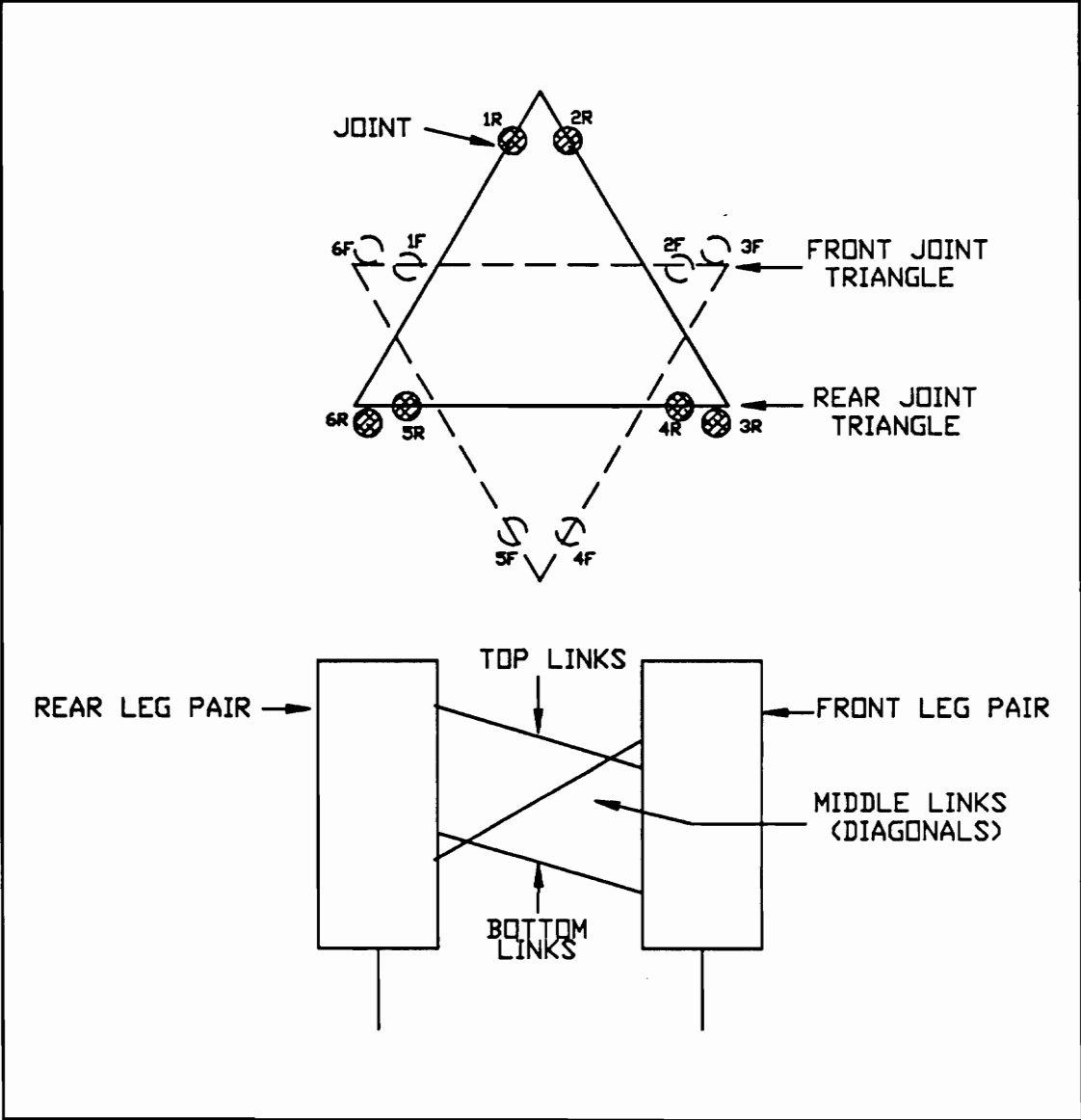


Figure 15: Final Joint Triangle Orientation

center of gravity to shift towards the lifted end. If this were to happen the vehicle would "rock" back onto to the lifted legs. This phenomena could become cyclic causing the vehicle to teeter like a seesaw with the middle legs as the fulcrum. Because of this the minimum number of leg pairs needed to effectively achieve crawling motion is four.

4.2 WORKSPACE ANALYSIS

This section will present a general overview of the workspace attainable by the crawling vehicle described in previous sections. An in-depth analysis of all possible motions within this workspace was not performed. Instead, limits on various types of movement (side travel, up and down travel, maximum roll, pitch, and yaw for a single actuation bay) were determined. Figure 16 depicts the various types of rotation that the vehicle is capable of. The limits were determined using a computer simulation generated by a program written by John Stulce currently in the master's program at Virginia Polytechnic Institute. One leg pair was fixed while the second was moved about to find its workspace limits. In some cases the limits of travel of the actuators constrained the motion while in other cases, the angular limits of the universal joints constrained the motion. A description of the workspace boundary determination process follows.

Let us begin by defining (S) to be the distance between adjacent leg pairs when all actuators are at their minimum, L_{\min} . The leg pairs were tested at $S+0.05m$, $S+0.15m$, $S+0.30m$, and $S+0.40m$ to find their mobility limits. In all four cases a limit for side

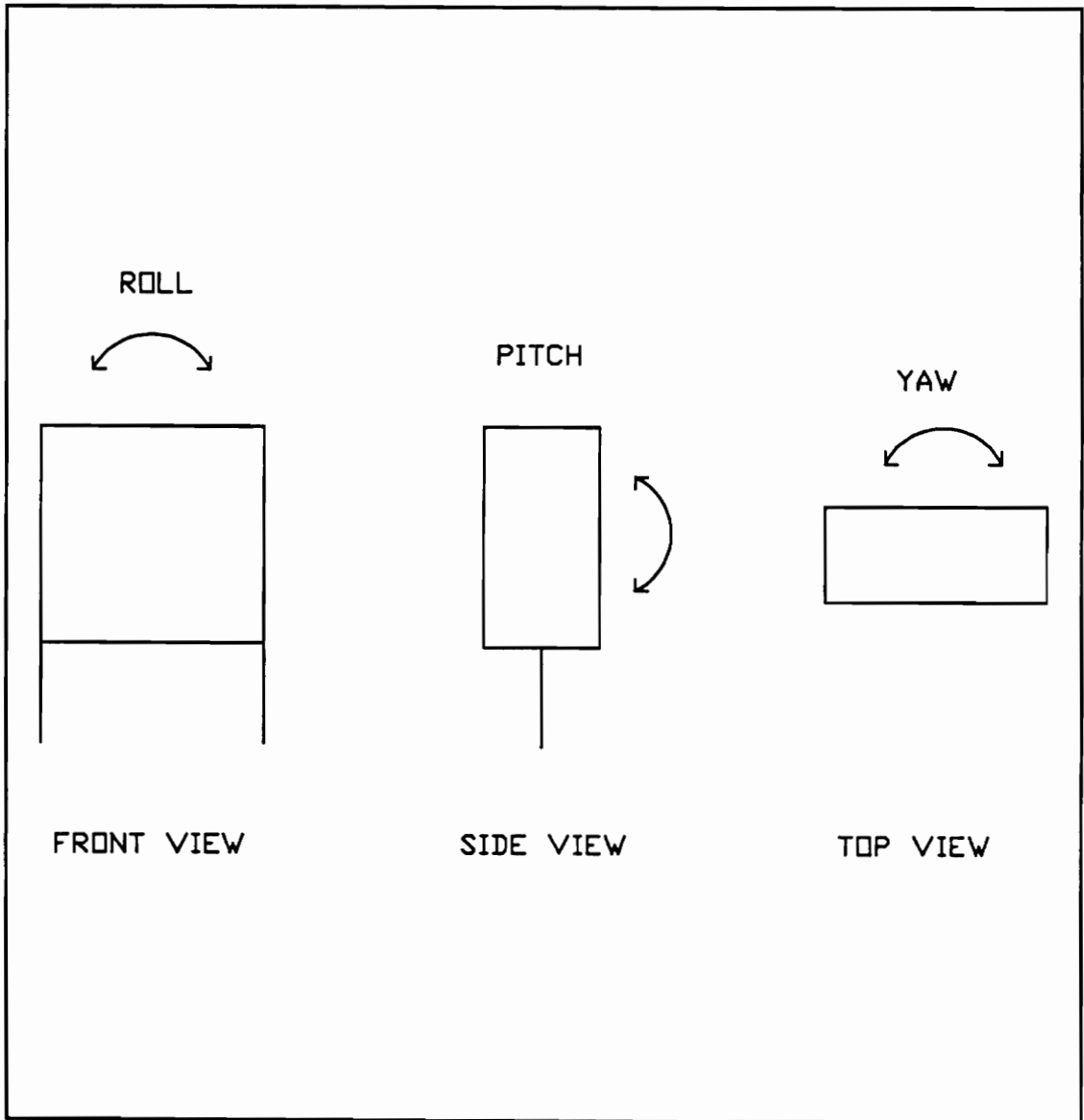


Figure 16: Various types of leg pair rotation

travel of one leg pair relative to an adjacent leg pair was determined, as well as a similar limit on up or down travel. Note, the limits for travel to one side of the stationary leg pair are the same as those for travel to the opposite side. Similarly, the limits for upward travel of the moving leg pair are the same as those for downward travel. Table 1 presents the limits determined for the four cases tested and their limiting factors.

Again using the computer simulated model several cases of leg pair spacing were investigated to determine the maximum amounts of yaw and pitch possible between adjacent leg pairs. The leg pair stances used were $S+0.15M$, $S+0.23M$, $S+0.30M$, and $S+0.40M$. Table 2 presents the limits determined for the four cases tested.

To determine the maximum angle of roll between adjacent leg pairs testing was conducted using a one-half scale model of the two leg pair vehicle. The computer model does not take into account the interference of vehicle components and was therefore not suitable for determining a roll limit. The model was tested at the one-half scale equivalent of $S+0.05M$, $S+0.15M$, $S+0.30M$, and $S+0.40M$. The approximate resulting maximum degrees of roll are given in table 2.

The limits presented are only a general overview of the possible workspace of the crawling vehicle. They do not guaranty nor preclude any particular motion patterns. To do so would require an analysis of all positions between the initial and final stance. Again, this analysis presents only the extreme positions, it makes no reference to the transitional positions leading to that extreme.

Table 1: Workspace Limits (distances in meters)

Stance	Maximum ° Side Travel	Side Travel ° Restraint	Upwards ° Travel	Upwards ° Restraint
S+0.05	0.46	U-Joint	0.38	U-Joint
S+0.15	0.53	U-Joint	0.46	U-joint
S+0.30	0.48	Actuators	0.46	Actuators
S+0.40	0.30	Actuators	0.25	Actuators

Table 2: Pitch, Yaw, and Roll Limits (angles in degrees)

Stance	Maximum Pitch °	Maximum Yaw °	Maximum Roll ^m
S+0.15	50	50	80
S+0.23	37	37	80
S+0.30	27	27	80
S+0.40	10	10	80

c - denotes data generated by computer simulation (Stulce, 1990)

m - denotes approximate values determined from experimental mock-up

4.3 STABILITY ANALYSIS

This section will present a general overview of the stability of the crawling vehicle. Specifically, two subjects will be detailed. These are the static stability limits and the righting of a toppled (overturned) vehicle. To determine the limits on stability and the righting procedure both a computer simulation and a one-half scale model were used.

To determine the maximum slope the crawling vehicle could stand on the computer simulation was used. The conditions imposed on the vehicle were that the vehicle was in-line (all leg pairs on the same longitudinal axis) and that the vehicle stood perpendicular to the slope. Under these conditions the maximum slope the vehicle can stand on is thirty degrees. This suggests that, for travel across slopes of greater than twenty-five degrees sidestepping or another motion pattern is needed. One solution may be to travel in the usual in-line manner but traverse the slope with a curved path (never placing the longitudinal axis perpendicular to the slope).

Perhaps the most important stability aspect of the crawling vehicle is its ability to right itself after toppling. This ability is essential if the crawler is to be a robust, autonomous, vehicle as it is proposed.

One proposed method of righting a vehicle is the twist method. Imagine the entire vehicle overturned and resting on its top (the face opposite the one that contains the legs). Now the vehicle rolls each leg pair to its maximum angle of roll. Since the maximum angle of roll per actuation bay is 80 degrees, it will require a minimum of three bays to

enable the vehicle to twist 180 degrees. For instance a vehicle with four leg pairs could have one end on its top and the other on its legs. This vehicle, however, would not be capable of righting itself because one leg pair on its feet does not provide enough support to twist the others into the righted position. It is believed that a vehicle with twice the minimum required number (to twist 180 degrees) could twist several of the leg pairs into the upright position and then use these as a base for lifting and twisting the remaining leg pairs into the righted position. Hence, we believe a vehicle with a minimum of six bays (seven leg pairs) could right itself. Testing may prove that fewer bays are actually required for righting the vehicle using the twist method.

The second proposed method of righting a toppled vehicle is the fall method. This method has not received the same detailed attention that the twist method has, but a general overview will be presented here. Again, imagine the vehicle resting on its top. Now some number of leg pairs on one end are lifted into the air forming the vehicle into an "L" (imagine a scorpion lifting its tail). If these raised leg pairs are now tilted to one side of the vehicle their weight will cause them to fall, thus turning the vehicle onto its side. Straightening the vehicle and repeating the process will stand the vehicle upright. Note that this is a dynamic method of righting a toppled vehicle. The required number of raised leg pairs and the ability of the vehicle to lift this number is still undetermined.

CHAPTER 5

MECHANICAL DESIGN OF THE CRAWLING VEHICLE

The following chapter describes in detail the mechanical design proposed for the crawling vehicle. For some components, strength requirements mandated the design, while for others, simple geometric requirements governed the design. These requirements as well as the final design for each component will be presented.

5.1 JOINT PANELS

The joint triangle orientation detailed in chapter 4 (refer to Figure 15) was used as a starting point for the mechanical design. The leg pairs were designed to provide rigid support at each of the joints. A panel at each corner of the joint triangle will provide the rigid support required. Each of the three panels will support two joints (see Figure 17).

The material chosen for the joint panels is aluminum, even though the panels will be subjected to repetitive reversed loading. The resulting stress is not severe enough to cause fatigue failure (within a reasonable number of cycles, $50(10)^7$). The aluminum panels will be 0.5 in. (12.7mm) thick. The top panel will be 3.25 in. (82.6mm) wide while the base panels will be 4 in. (102mm) wide. These panels are designed to withstand the full thrust of the actuators. In other words, if two consecutive leg pairs were held in place while the actuators between them were powered the panels would not

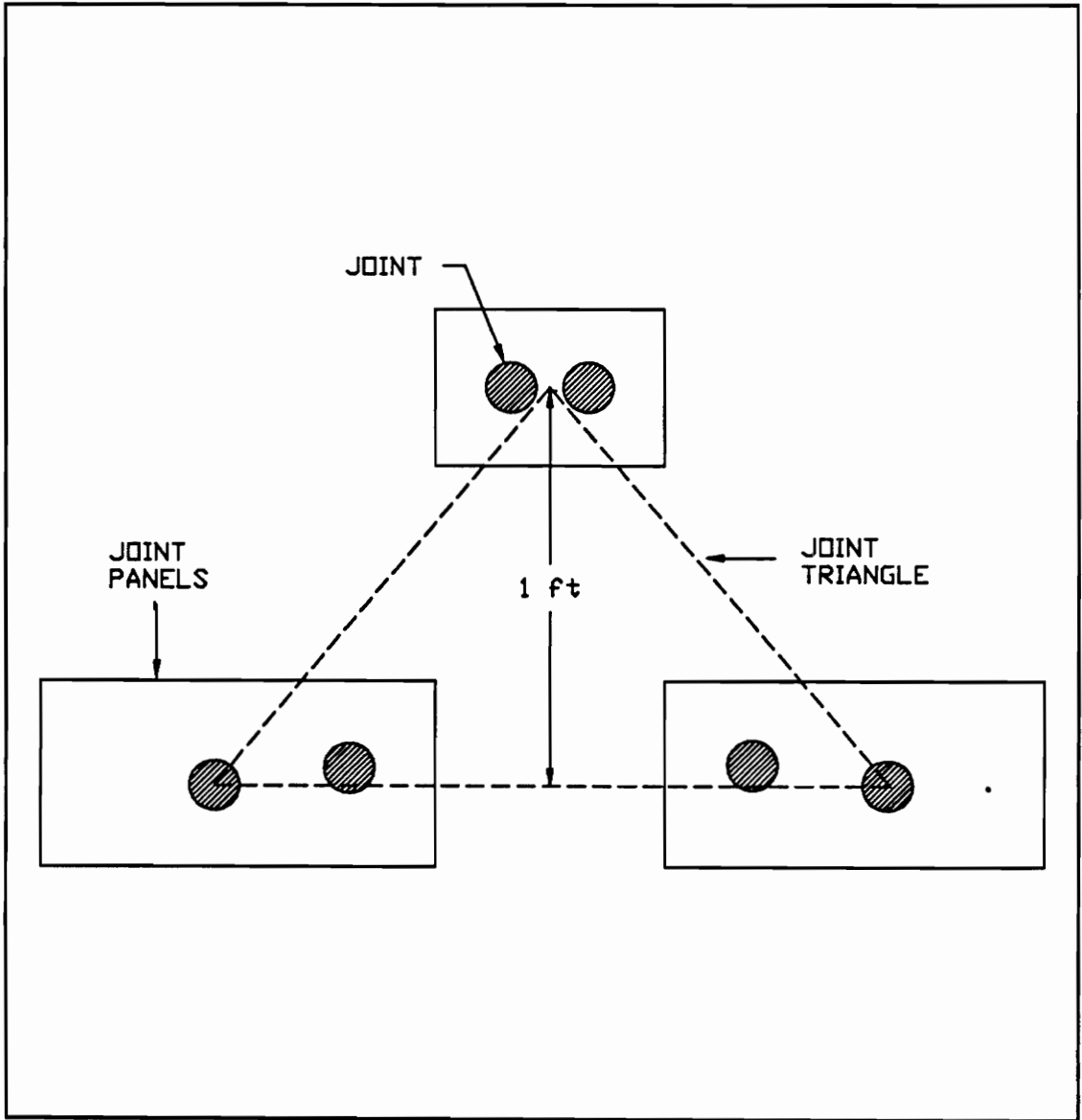


Figure 17: Joint Panel Layout

yield, of the three panels the base panels experience the highest stress. The resulting maximum stress in these panels, under the full thrust condition described above, (5.3 kpsi, 36.6 MPa), is below the fatigue strength of most aluminum alloys. The panels will be bolted to members of the leg pair frame. Appendix A gives specific data on the joint panels and their attachment to the leg pair frame.

5.2 LEG PAIR FRAME

The next component designed was the leg pair frame (see Figure 18). The leg pair not only contains the rigid lateral legs but it must also support a joint triangle on each lateral face. The leg pair frame supports the joint panels described above to which the joints are mounted. The joint triangle described in detail in chapter 4 is 1 ft (0.3m) high. The vertical offset between a joint triangle on the front face of a leg pair and one on the rear face of a leg pair is .5 ft (0.15m) Thus, the leg pair must be at a minimum 1.5 ft by 1.5 ft (0.45m X 0.45m) if it is to support both joint triangles. To allow room for mechanical components and to ensure no joint or actuator components protrude outside the frame of the leg pair the frame was expanded to 2.5 ft by 2.5 ft (0.75m X 0.75m). This oversized frame protects the actuator components by creating buffer space between them and any harsh terrain features they may traverse.

The distance between the front and rear faces of the leg pair was arbitrarily chosen to be 1 ft (0.3m). The space between faces will allow for the addition of payload or necessary on-board control and power components. The legs were also arbitrarily chosen

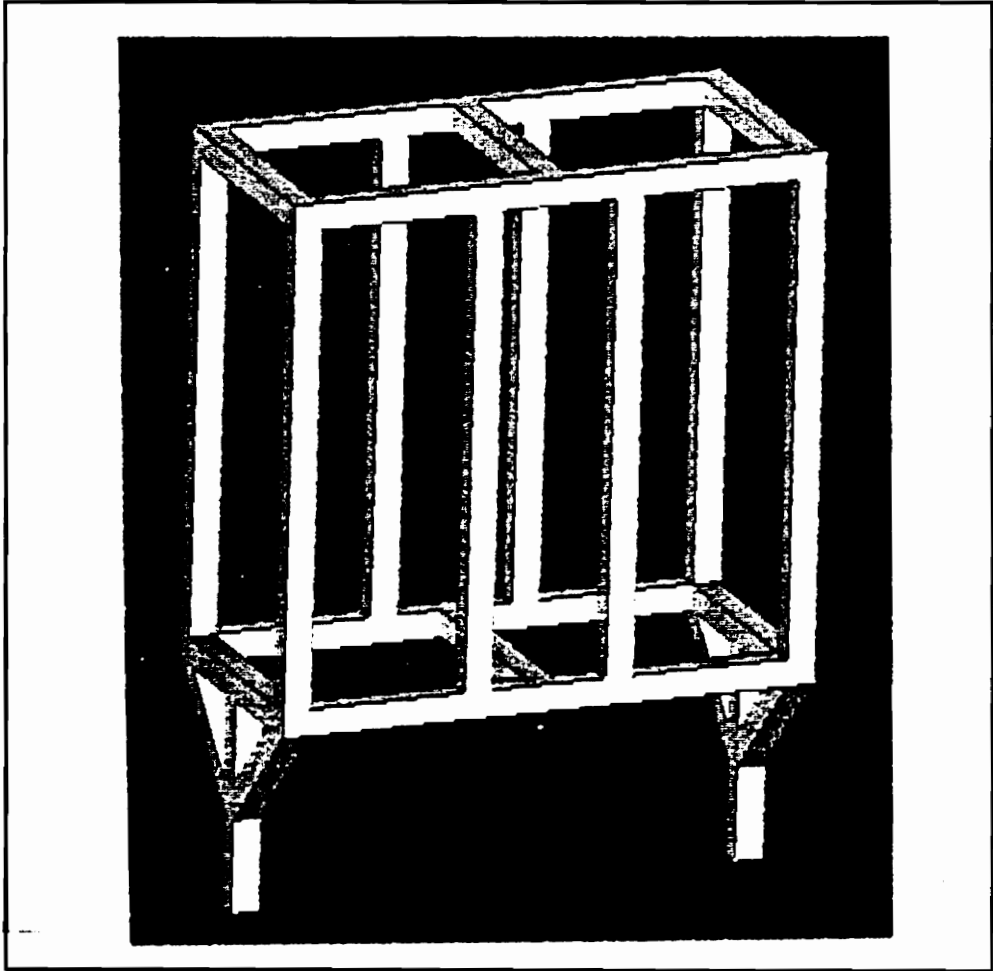


Figure 18: Leg Pair Frame

to be 1 ft (0.3m). A 1 ft leg will further protect the actuator components by elevating them above many of the potentially damaging terrain features. It will also aid in the realization of efficient motion programming by enhancing the "pitching" in several gaits of motion (see chapter 2).

With the above dimensions satisfying all geometric requirements a material for the above leg pair frame design was chosen. The leg pair frame will be constructed from 1.5 in. (38mm) square aluminum tube . This tubing is sufficient for supporting the joint panels and is light weight. Similar to the joint panels the leg pair frame is subjected to repetitive reversed loading but the resulting stress is not large enough to exclude the use of aluminum.

5.3 ACTUATORS

The linear actuator that will control the positioning of the rigid leg pair was the next component to be specified. The primary considerations in actuator specification were motor arrangement, actuator speed, thrust, and the availability of a linear position measuring device such as a potentiometer (previous work had determined that add on, self-fabricated potentiometers were mechanically very sloppy and cumbersome). There are three types of motor arrangements available on commercial linear actuators. The in-line type, where the motor and cylinder are along the same longitudinal axis, (see Figure 19a) requires little operating space. There were no actuators of this arrangement that met the primary considerations, specifically thrust, therefore other types of arrangement were

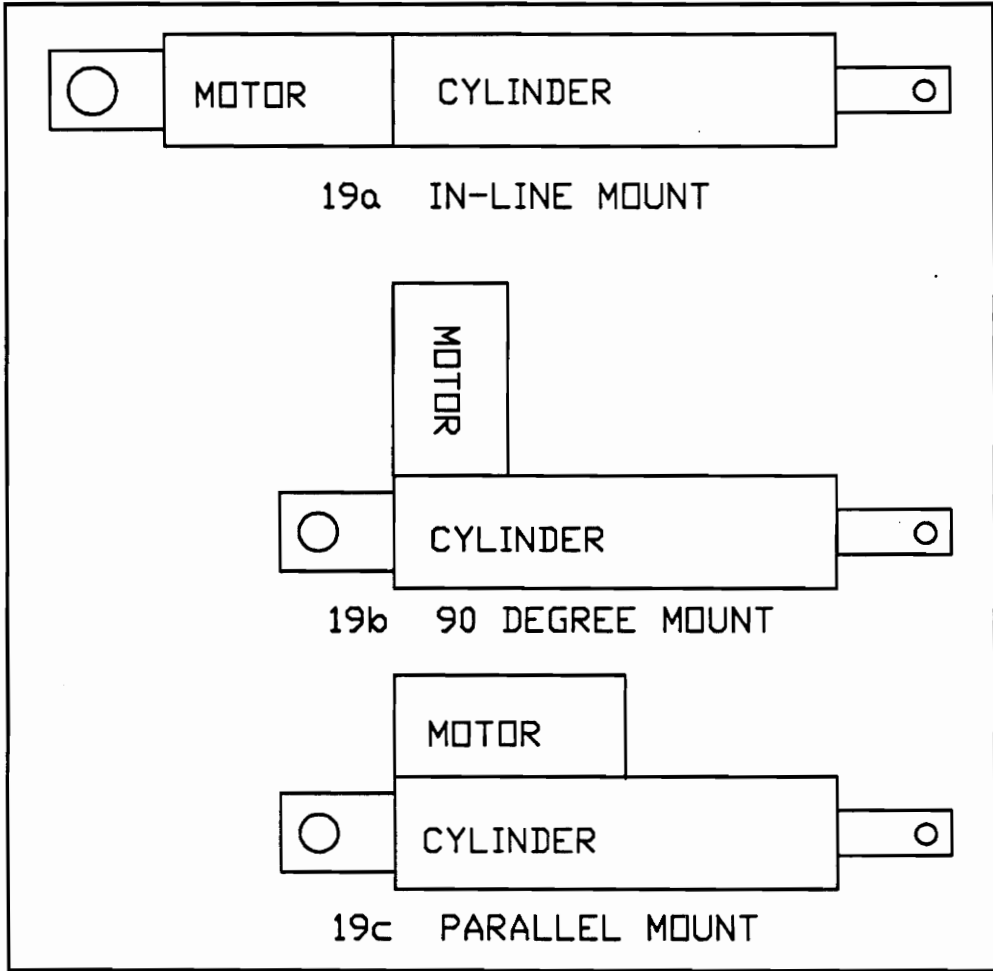


Figure 19: Motor-Cylinder Arrangements

investigated. This arrangement also limits workspace, because the ratio of L_{max}/L_{min} (L measured from pivot to pivot) is relatively small. The second arrangement shown in Figure 19b is the ninety degree mount. This arrangement is not suitable for the crawling vehicle because the motor is mounted at a ninety degree angle to the cylinder, thus, it requires a great deal of operating space. In the third arrangement the motor and cylinder are mounted side by side, thus requiring very little operating space (see Figure 19c). This arrangement was found to be suitable for the crawling vehicle application.

To determine the required thrust of the actuator, the crawling vehicle was examined using some worst-case situations (on level terrain only). Using an eighteen-inch actuator stroke, two leg pairs were placed at their maximum distance from each other. In this position the actuators have the worst possible mechanical advantage for further moving the leg pairs. Using the weight of the leg pair (approximately 58 lbs (32kg) including the joints and joint bases to be described later) and an assumed cargo weight of 100 lbs (45kg) the required thrust of the actuators was determined. For the case described above the required thrust is 108 lbs (480N) per actuator. The actuator chosen has a 210 lb (934N) maximum thrust. The high thrust of this actuator will allow the vehicle to continue moving even after several of the actuators have failed.

The actuator chosen to meet the thrust requirements has a linear actuator speed of 20 in. per second at no load. In the simplest locomotion gait where a single wave propagates along the vehicle, this actuator speed will allow a four leg pair vehicle to move at approximately .284 mph (.45 km/h) or 25 feet per minute.

There were many options available for the actuator chosen. The most important of these was the linear potentiometer package. This potentiometer is necessary for position feedback in the controls scheme. With this option, the actuator satisfies all the primary considerations. There are several more options which make the actuator very suitable for the crawling vehicle.

The actuator was equipped with a brake to prevent backdriving. The actuator chosen is a ball screw model (ball screws have less friction than acme screws) that would backdrive under the force of a suspended leg pair without the installed brake. The brake is a ninety volt on-off type device.

The actuator was further specified to include free rotation of the cylinder about its longitudinal axis. This allows for the rotation (roll) of one leg pair relative to any other. This option was especially attractive because it alleviated any need for modifications to the universal joints (in order to create the necessary rotation) at each end of the actuator. The two degree of freedom universal joints plus the longitudinal rotation provided by the actuator will allow the leg pairs to behave as if they were joined by spheric joints.

Although there were many mounting options available for the motor end of the actuator, none could be found that was suitable for attachment to the universal joint. The mounting piece or motor end cap was therefore custom designed. This cap will be described in subsequent sections. The option chosen for the rod end of the actuator is simply a fitting on the end of the rod with a 0.5 in. (12.7mm) lateral hole. This was chosen because it could be easily attached to the universal joint. This connection will

also be described in subsequent sections.

The actuator described above is an Industrial Devices Corporation actuator, model **H152B-18-000-FE2-B-L**. Appendix A contains specific actuator construction, performance, and cost data.

5.4 JOINTS

The universal joint to connect each end of the actuator to its respective joint triangle is a Gray and Prior Machine Company universal joint. The Gray and Prior U-joints have the largest off-the-shelf range of motion (90 degree cone) available. The selected U-joint is 1 in. (25.4mm) in diameter and 3.375 in. (86mm) long. It weighs 0.61 lbs (.28kg) and costs \$18.20. The U-joints will have a 0.26 in. (6.6mm) diameter lateral hole in each shaft of the U-joint. The hole will be centered 0.5 in. (13mm) from the end of each shaft.

Now that the leg pair, actuator, and joints have been specified they must be connected to each other. Specifically three mating problems must be solved. These are U-joint to joint panel, U-joint to motor end cap, and U-joint to rod end. All connections utilize the same principle, namely, the pinning of the U-joint shaft inside a cylindrical cup. This method allows for easy attachment and detachment of components. The ease of component assembly and breakdown will aid in transporting the vehicle and will simplify vehicle modification (number of leg pairs etc.). The specifics of each connection are detailed in the following sections.

5.5 U-JOINT BASE

The U-joint Base, depicted in Figure 20, will connect the U-joint to the joint panels. The U-joint Base must provide enough clearance for the U-joint to travel to its maximum rotation of 45 degrees. Without the proper base dimensions (specifically L min. depicted in Figure 21) the motor interferes with the joint panel before the U-joint has reached its maximum rotation. Using the clearance diagram (Figure 21) it was determined that the minimum distance between the joint panel and the axis of rotation in the U-Joint must be 1.67 in. (42.4mm) if the joint is to reach full rotation.

The base is a 1.25 in. (31.8mm) long piece of 1.375 in. (35mm) diameter aluminum rod. Centered 0.5 in. (12.7mm) from one end of the rod is 0.26 in. (6.7mm) diameter lateral hole for the pin which holds the U-joint in the base. The other end of the rod piece is welded to a 0.375 in. (9.5mm) thick, 3.25 in. (82.6mm) diameter plate. The bore for the joint is drilled through both rod and plate. The plate has 4 holes for attachment to the joint panel by 1/4 in. bolts. Appendix A has complete specifications for the U-joint base.

5.6 MOTOR END CAP

The motor end cap, shown in Figure 22, will connect the motor end of the actuator to the U-Joint. The end cap is 1.25 in. (32mm) long piece of 1.275 in. (35mm) diameter aluminum rod with the same lateral hole at one end as the U-Joint base. The rod section

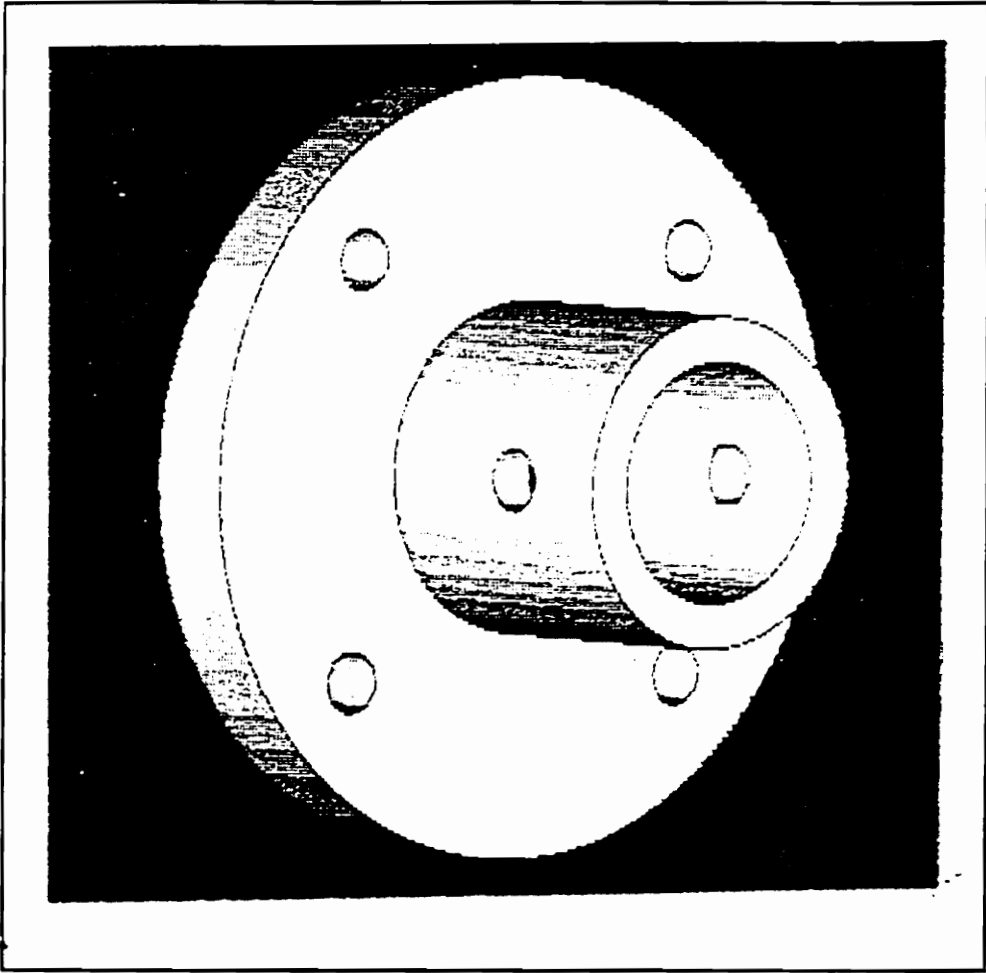


Figure 20: U-Joint Base

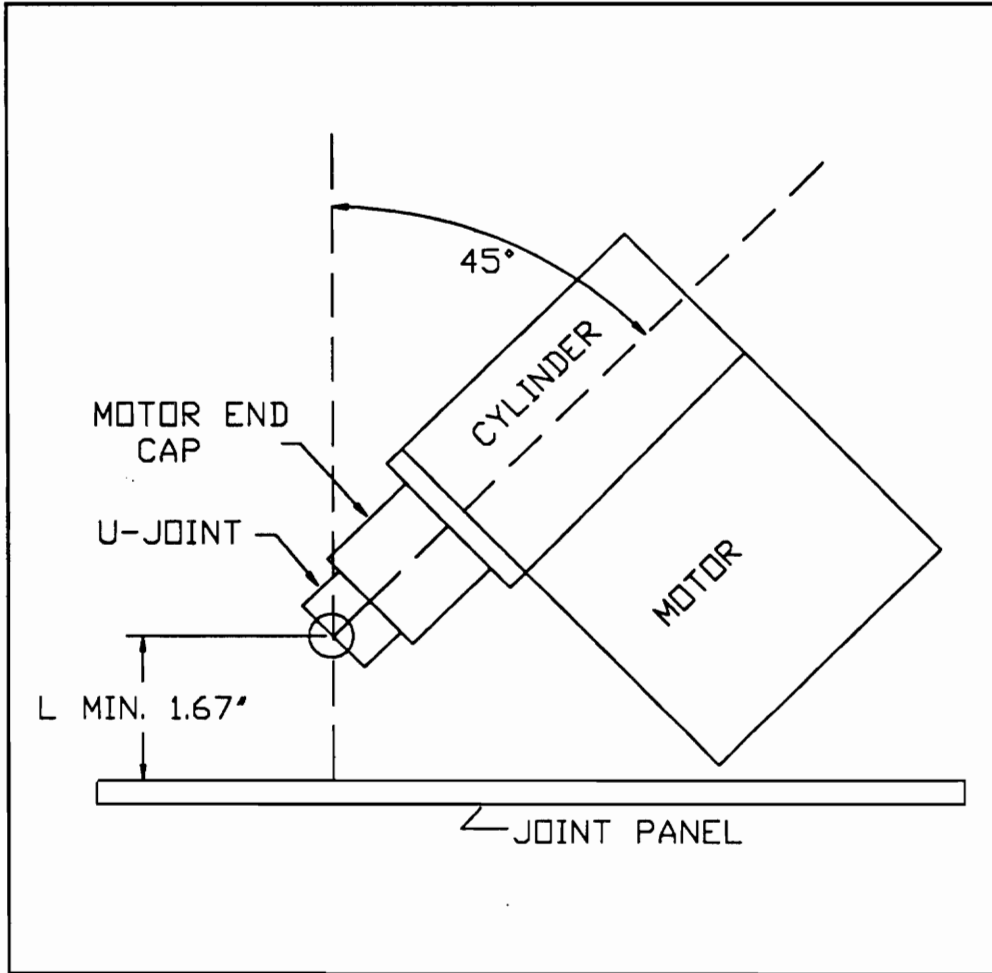


Figure 21: Clearance Model

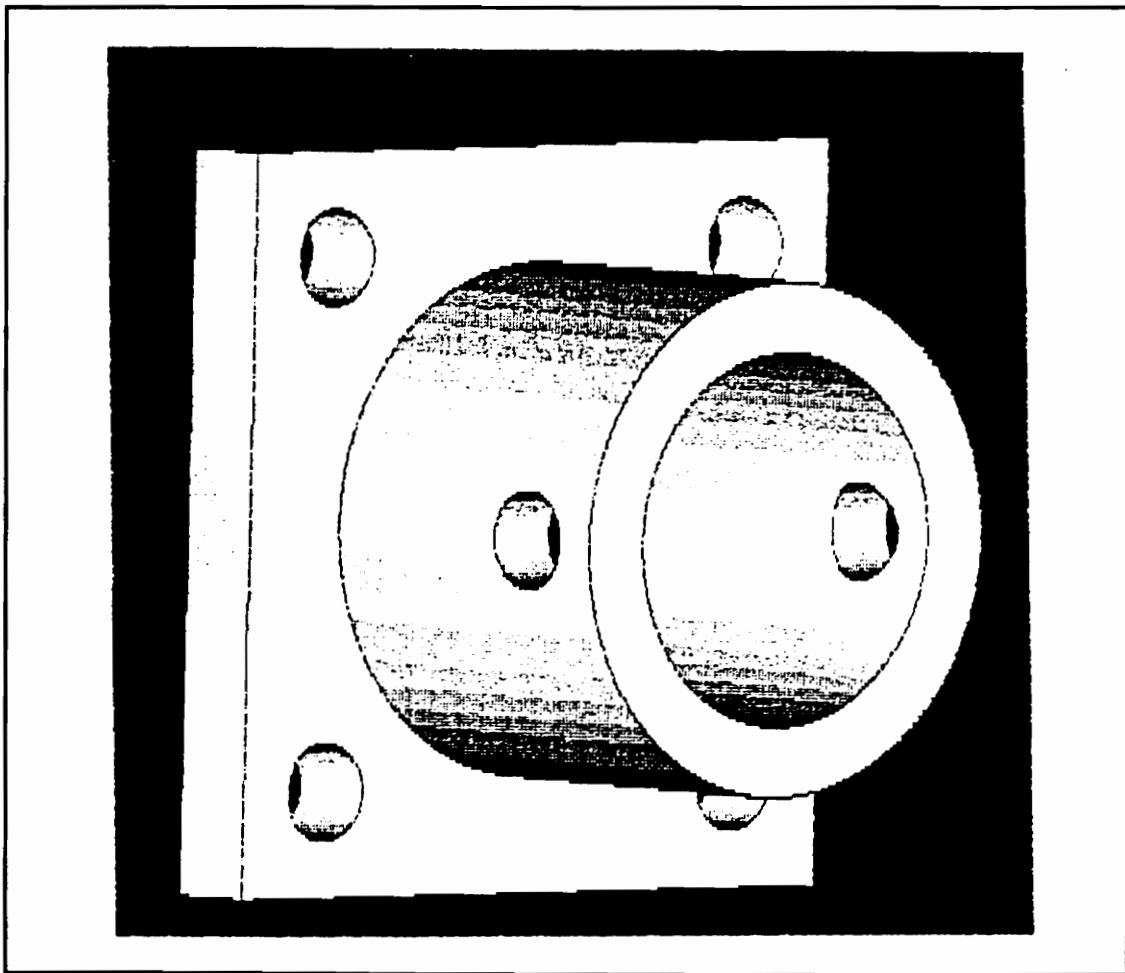


Figure 22: Motor End Cap

is welded to a 2 in. by 2 in. (51mm by 51mm), 0.25 in. (6.4mm) thick aluminum plate. Once attached, a collinear (with the rod) 1.0015 in. (25.5mm) bore is drilled through both rod and plate. The plate has 4 holes for attachment to the motor end of the actuator by 0.25 in. bolts. Appendix A has complete motor end cap data.

5.7 ROD END CAP

The rod end cap will connect the rod end of the actuator to the U-joint. The rod end, shown in Figure 23, is also fabricated from the 1.375 inch diameter aluminum rod. This piece is 2.5 in. (63.5mm) long with the collinear 1.0015 in. (25.5mm) diameter bore all the way through the piece. One end of the Rod End Cap has a 0.5 in. (12.7mm) lateral hole for mating with the rod end while the other end has a 0.26 in. (6.6mm) lateral hole for the pinning of the U-joint shaft. Appendix A has complete rod end cap data.

Figure 24 shows the assembly of the components described above. The components themselves are detailed in complete drawings in Appendix A. Appendix A also contains information such as suppliers, costs, and assembly instructions.

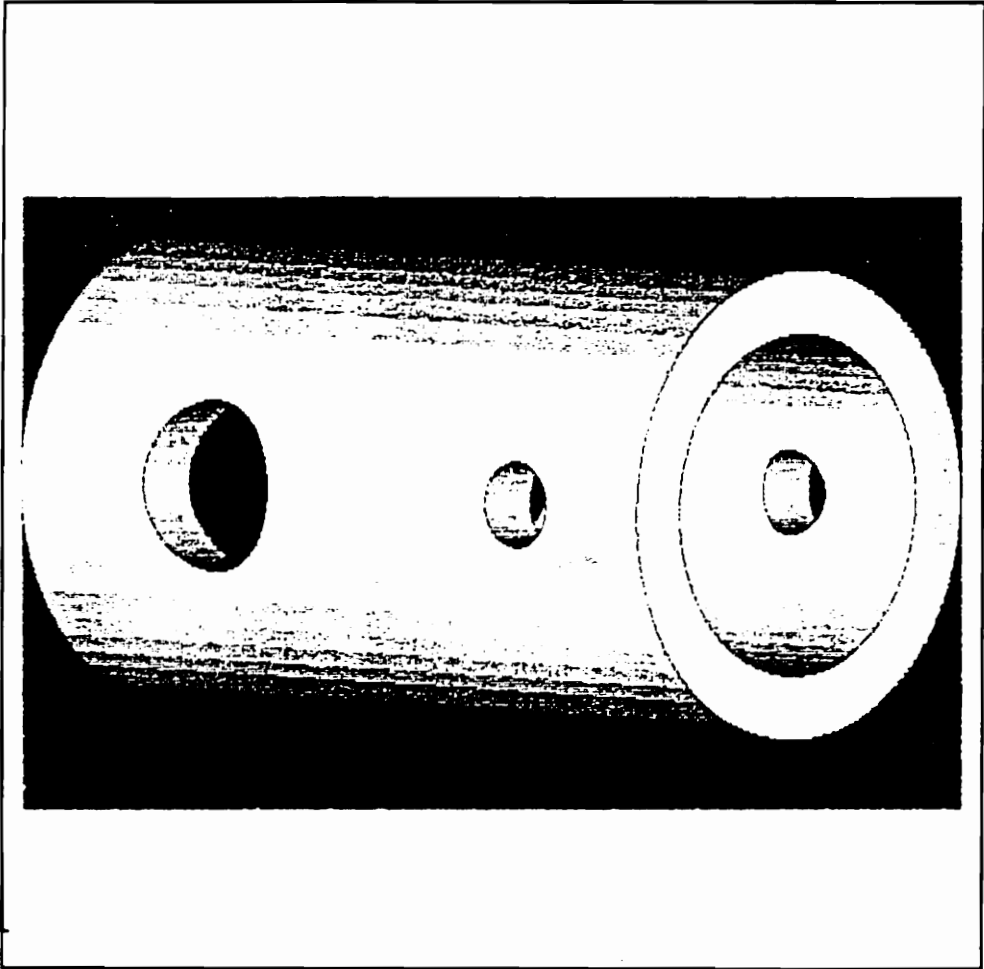


Figure 23: Rod End Cap

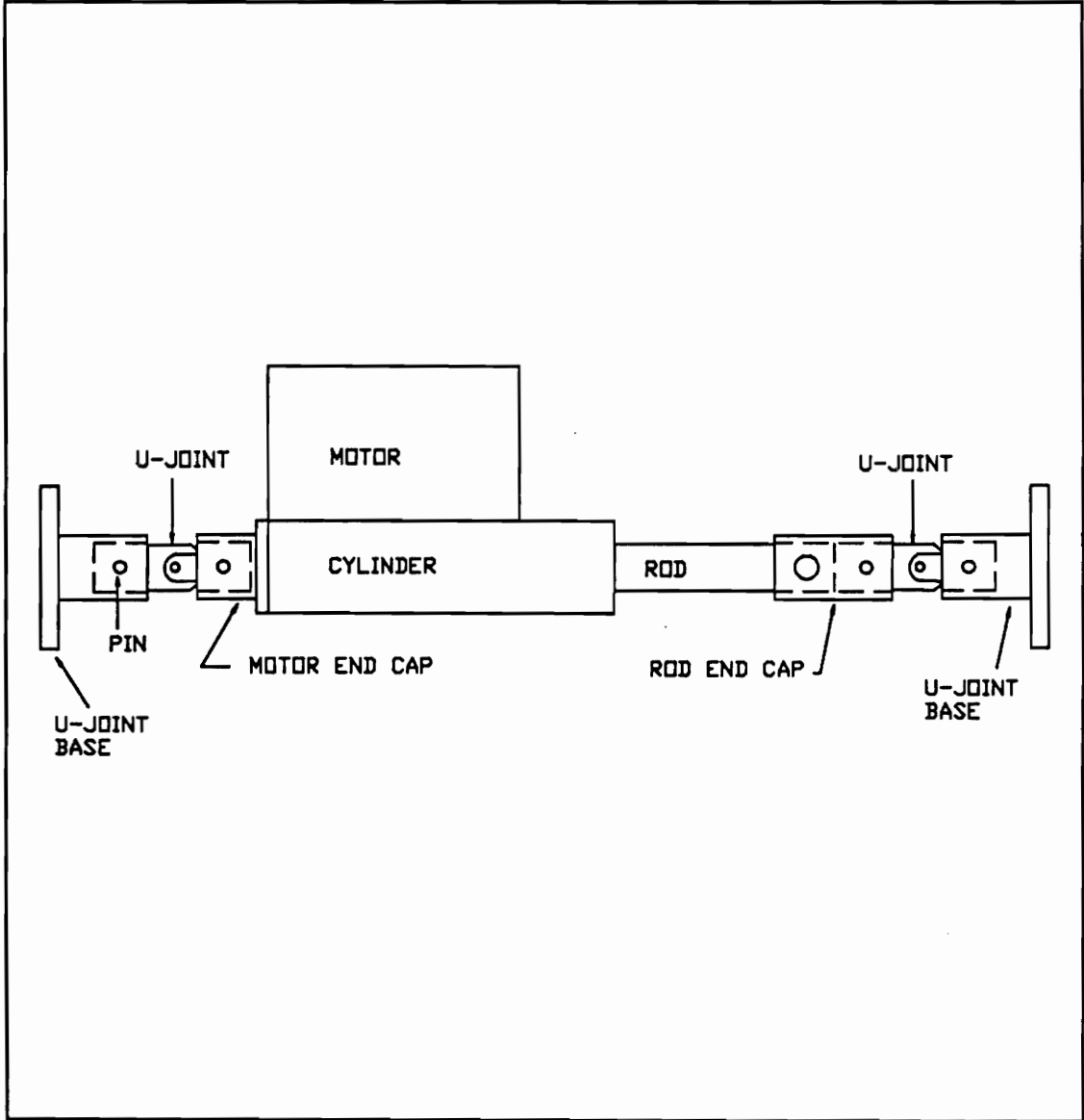


Figure 24: Line Assembly of Components

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

It has been noted that a vast majority of the Earth's surface as well as the surface of Mars is inaccessible to conventional vehicles. During the effort to develop a walking vehicle that could traverse these currently inaccessible areas, an intriguing observation was made. Insects such as the caterpillar can also traverse rough terrain but do so with greater stability than bipeds or quadrupeds. A design for a caterpillar-like vehicle has been presented here.

To show the depth of interest in developing a vehicle capable of traversing any terrain, an overview of walking vehicle research was provided. An overview of the crawling vehicle objectives as well as the control problems hampering the realization of an efficient crawling vehicle were then presented. A detailed mechanical design including the governing kinematic and mechanical considerations was also described. Finally, a workspace and stability analysis of the mechanical design was presented.

Future efforts to develop the crawling vehicle should be aimed at building a working prototype and investigating areas where such a vehicle would be suitable. One must not overlook areas where the crawling vehicle as a whole may not be suitable but the enabling technology may be useful.

Before any prototype is built, the computer simulation program should be enhanced.

Currently the program does not monitor joint angles or component interference. Maximum joint angles and actuator interference are severely limiting factors and must be incorporated if the simulation is to be a complete design tool. A mobility analysis based only on actuator link lengths will often yield incorrect workspace limits. Once enhanced, the computer simulation program should exhaustively examine all required motions for a specific application before any hardware is built. As with the Hirose and McGhee projects, all requirements (mobility, strength, etc.) should be investigated in detail before a design is begun. Because a specific application had not been specified, this project has proceeded in a somewhat reverse order. The design was generated with only very rough mobility and strength requirements and then analyzed to determine its specific limits. Now any specific requirements not met, necessitate a redesign.

When a prototype is built it should be instrumented with strain gauges to assist in the removal of material for weight reduction. A finite element analysis on the design would also assist locating excess material and the strengthening or restructuring of components as needed.

Serious attention must also be given to solving the coupled redundant control problem. Control laws that mitigate this problem are essential to crawling vehicle success. The development of needed software and hardware to implement these control laws also demands attention.

Though not crucial to proving the crawling vehicle concept will work, the vehicle may be improved later by using actuation devices other than electric drives. An investigation

of hydraulic or pneumatic actuators may uncover benefits these possess that will enhance the crawling vehicle.

Finally, a deeper investigation of the motion sequences used to right an overturned vehicle is needed. The self-righting ability is a strong point of the proposed crawling vehicle, thus it must be certain that the vehicle can do so in any condition.

APPENDIX

Table A1: COMPONENT LISTING FOR FOUR LEG PAIR VEHICLE

COMPONENT & # per vehicle	MATERIAL per component	COST per component
Leg Pair Frame (4)	42 feet of 1.5" x 1.5" Al tube	\$147.00
Joint Panels	1/2" Al plate (6) 3.25" X 9.5"	\$6.00
	(12) 4" X 11.75"	\$9.50
U-Joint Base (36)	3/8" Al plate 3.25" dia. section	\$2.10
	1 3/8" dia. Al rod 1.25" section	
Motor End Cap (18)	1/4" Al plate 2" x 2" section	\$0.90
	1 3/8" dia. Al rod 1.25" section	
Rod End Cap (18)	1 3/8" dia. Al rod 2.5" section	\$.80
Actuators (18)	H152B-18-000-FE2-B-L Ind. Devices Corp.	\$1600.00
Pins	(18) 1/2" X 2"	\$1.50
	(54) 1/4" X 2"	\$1.00
U-Joints (36)	Dimension A = 1.00" Standard length	\$18.20

TOTAL VEHICLE COST* \$30,380.00

* mechanical components only

SUPPLIERS

ACTUATORS - C. Arthur Weaver Co. Inc.
7562 Hitech Road
Roanoke, VA 24019
phone - (703) 563-9761
contact: Jim Loving
distributor for Industrial Devices Corp.

U-JOINTS - Gray and Prior Machine Co.
95 Granby Street
Bloomfield, Conn. 06002
phone - (203) 243-8381

PINS - Medalist Rein Leitzke
P.O. Box 305
Hustiford, WI 53034
phone - 800-558-9535

COMPONENT DESIGNATIONS

ACTUATORS - IDC Model # H152B-18-000-FE2-B-L
with rotation of cylinder about its longitudinal axis

U-JOINTS - Dimension A= 1.00", Standard Length

PINTS - 1/4" Medalist Code 30-08
1/2" Medalist Code 30-47

**ALL OTHER COMPONENTS AND THE MATERIALS FOR THOSE COMPONENTS
WILL BE SUPPLIED BY THE MECHANICAL ENGINEERING MACHINE SHOP**

ASSEMBLY ORDER

1. **Attach U-Joint Bases to Joint Panels** with 1/4" bolts, see fig A1 for placement
2. **Attach Joint Panels to Leg Pair Frame** with 1/4" bolts, see fig A2 and A3 for placement
3. **Mount Motor End Cap** to motor end of **Actuator** with 1/4"-28 UNF X 1.75" long screws
4. **Pin Rod End Cap** to end of actuator cylinder
5. **Pin U-Joints into U-Joint Bases**
6. **Pin U-Joints into Motor End Cap**
7. **Pin U-Joints into Rod End Cap**
8. Repeat 1-7 for all leg pairs

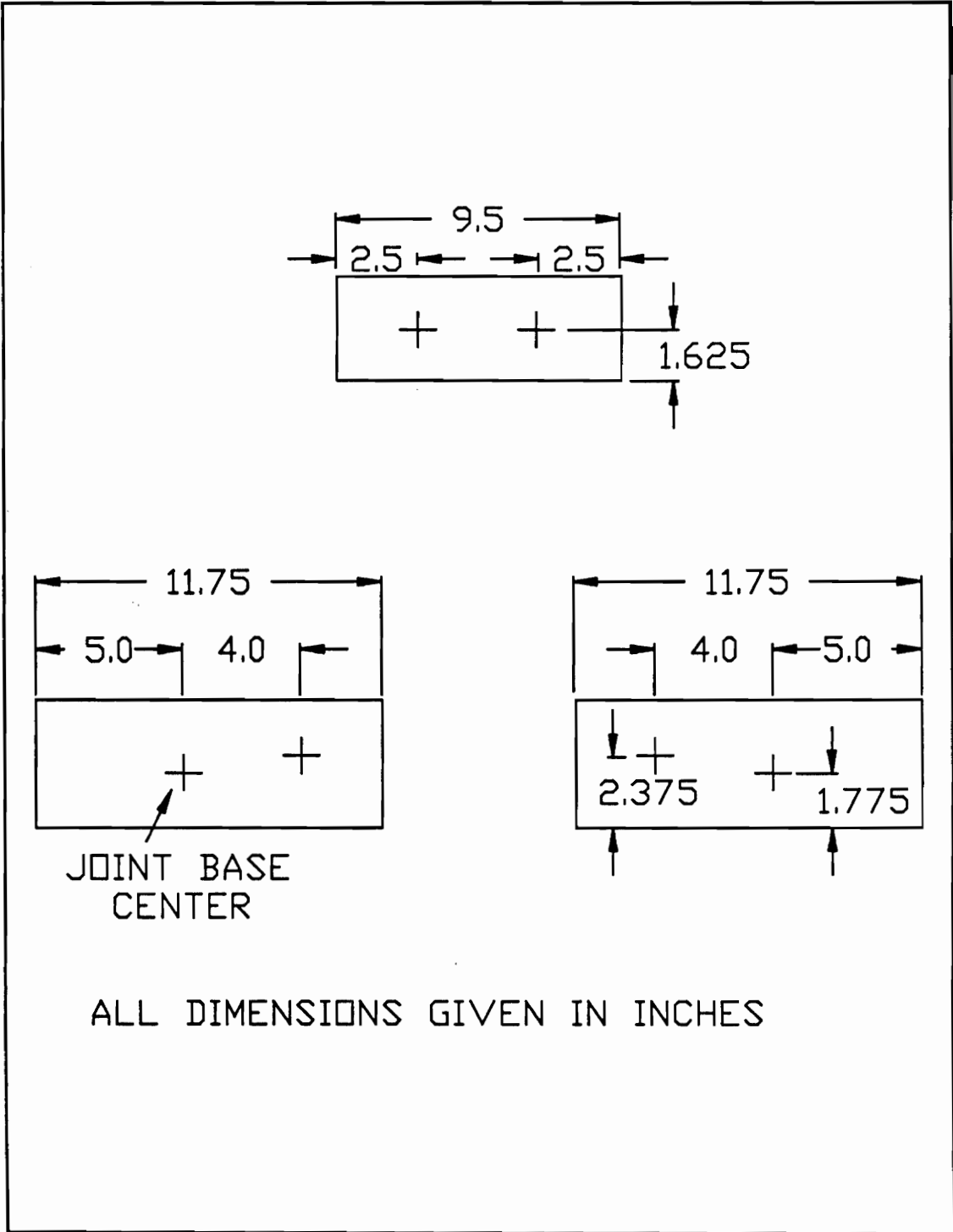


Figure A1: U-Joint-Base Placement on Joint Panels

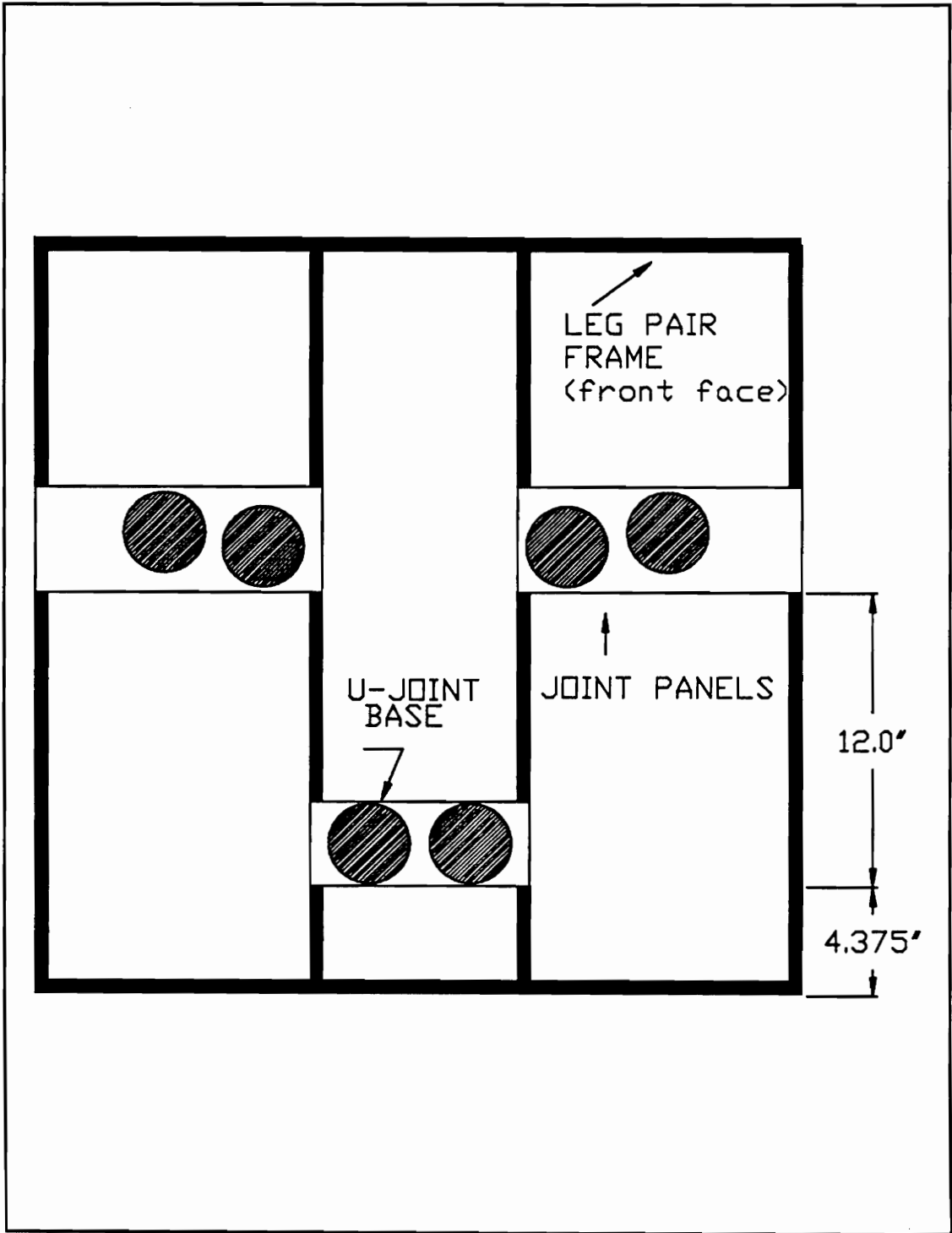


Figure A2: Panel Placement for Front Face of Leg Pair

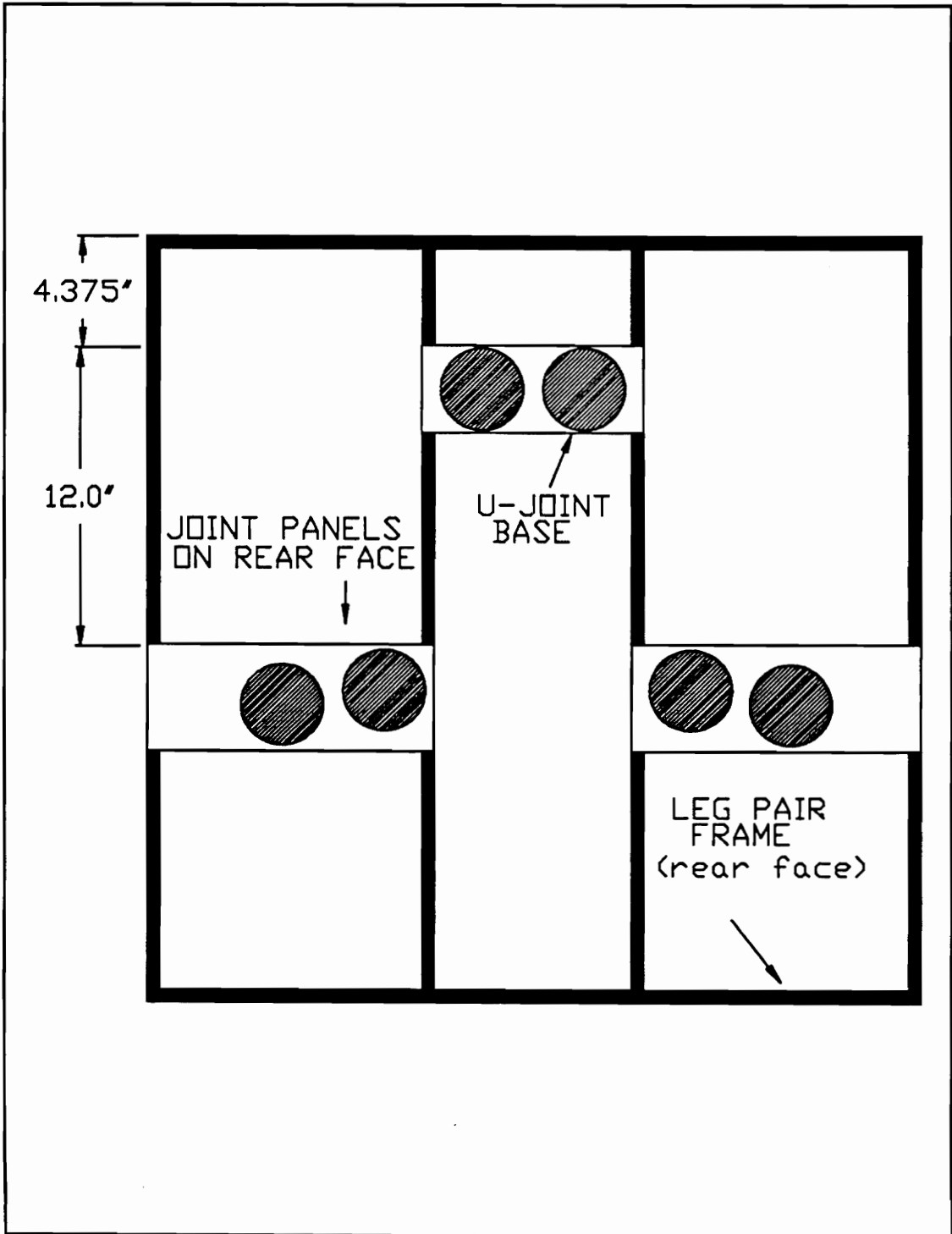


Figure A3: Joint Panel Placement on Rear Face of Leg Pair

H SERIES CYLINDERS

Common Specifications

Thrust Load	800 pounds max
End Play	0.010 max
Side/Torque Load	20 in-lbs max
Stroke Lengths	2, 4, 6, 8, 12, and 18 inches
Weight	9 to 12 lbs. depending on stroke length

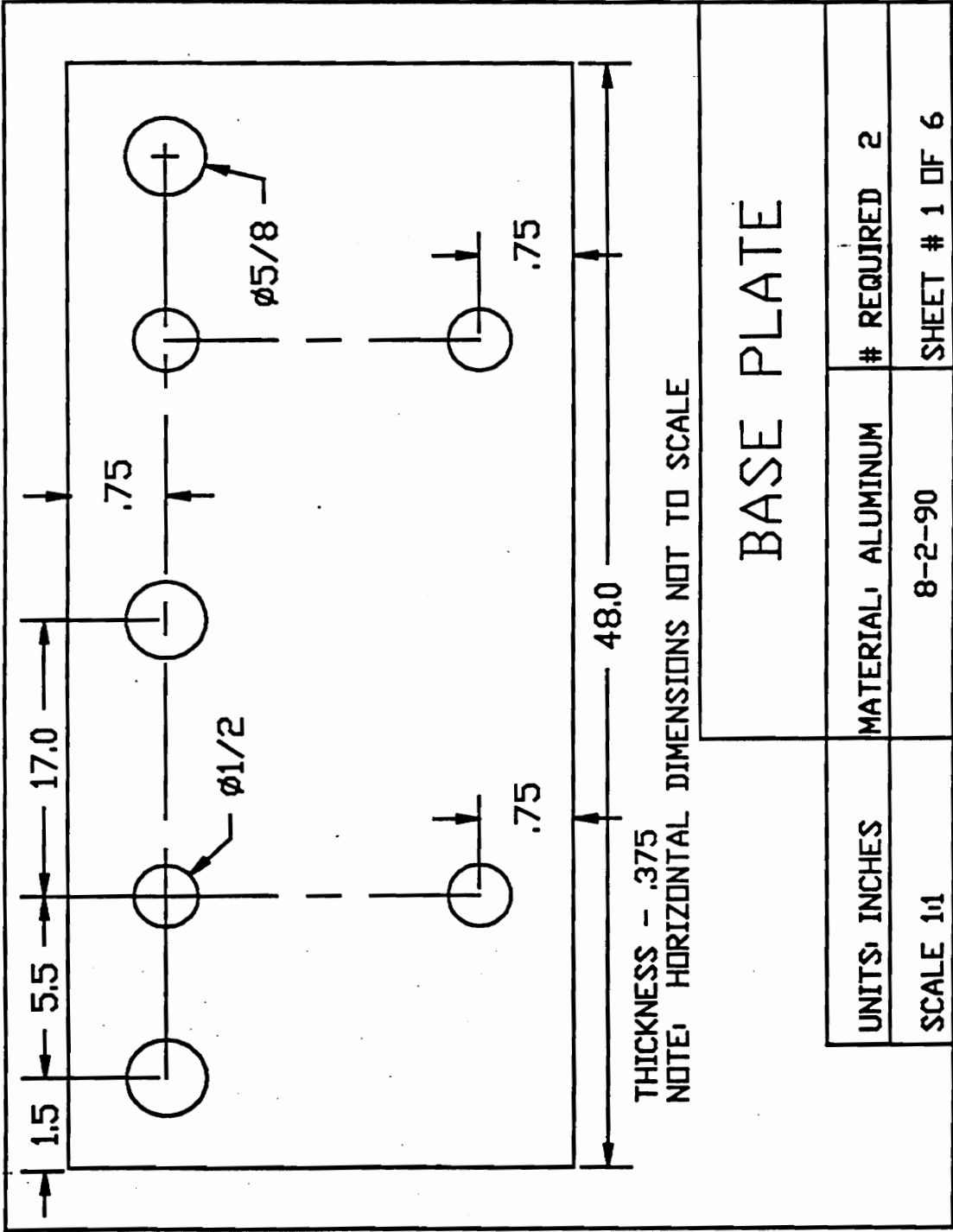
Construction Materials:

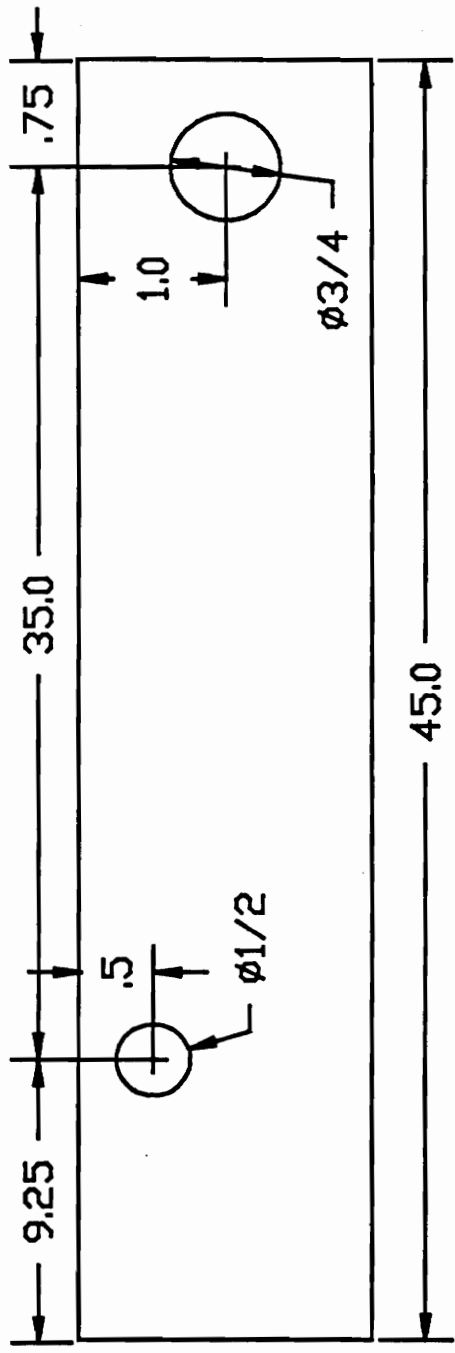
Housing	Type 384 die cast aluminum, epoxy coated
Guide Cylinder	6061 T-6 aluminum hard coated black anodized and Teflon impregnated
Thrust Tube	Type 304 stainless steel; ground and polished
Tie Rods	Type 304 stainless steel
Bearings	Deep groove ball bearings
Wiper Seal	Teflon
Lead Screw/Drive Nut	300 Series stainless steel: Acme-Lubricated Bronze Ball Screw-high carbon steel

Motor:

Type	Permanent magnet 4-pole DC motor; replaceable brushes
Input Voltage	150V DC
Current	No load: 0.5 Amps Rated Load: 2.0 Amps Peak in-rush: 5.0 Amps Current limited to 5.0 Amps by IDC Control
Motor Leads	Quantity-2 Length-6" inside conduit box
Anticipated Cycle Life of Brushes	5,000,000 cycles/5,000 hours
Case Temperature	Not to exceed 180 F (82 C)
Environmental	
Temperature	-20 to 140F (Below 32F, use -F option)

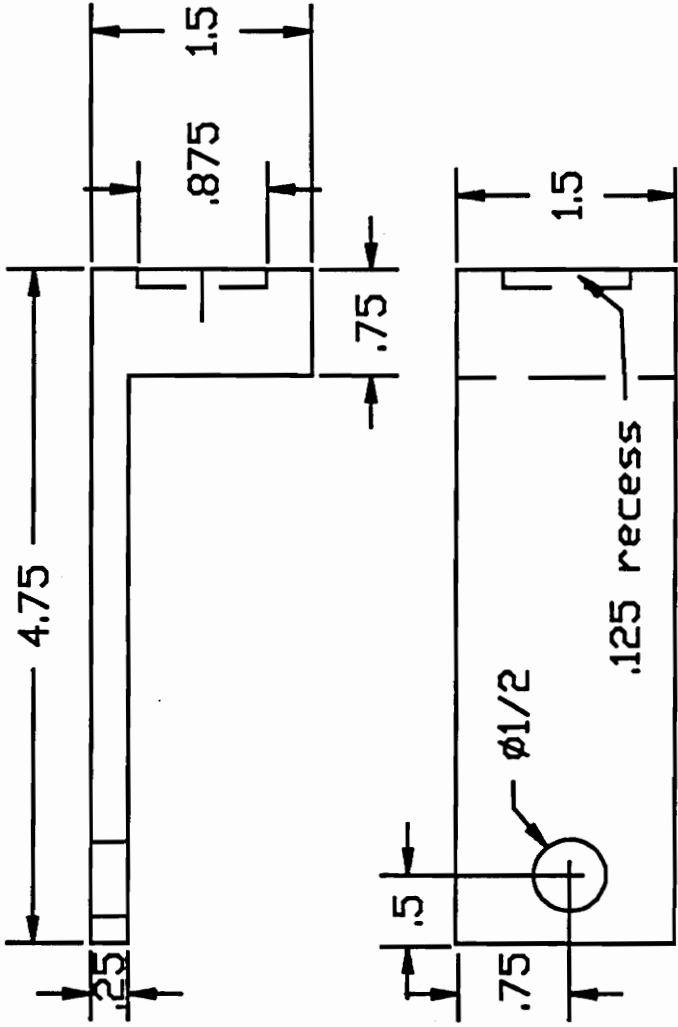
CONTENTION EXPERIMENT TEST RIG SHOP DRAWINGS





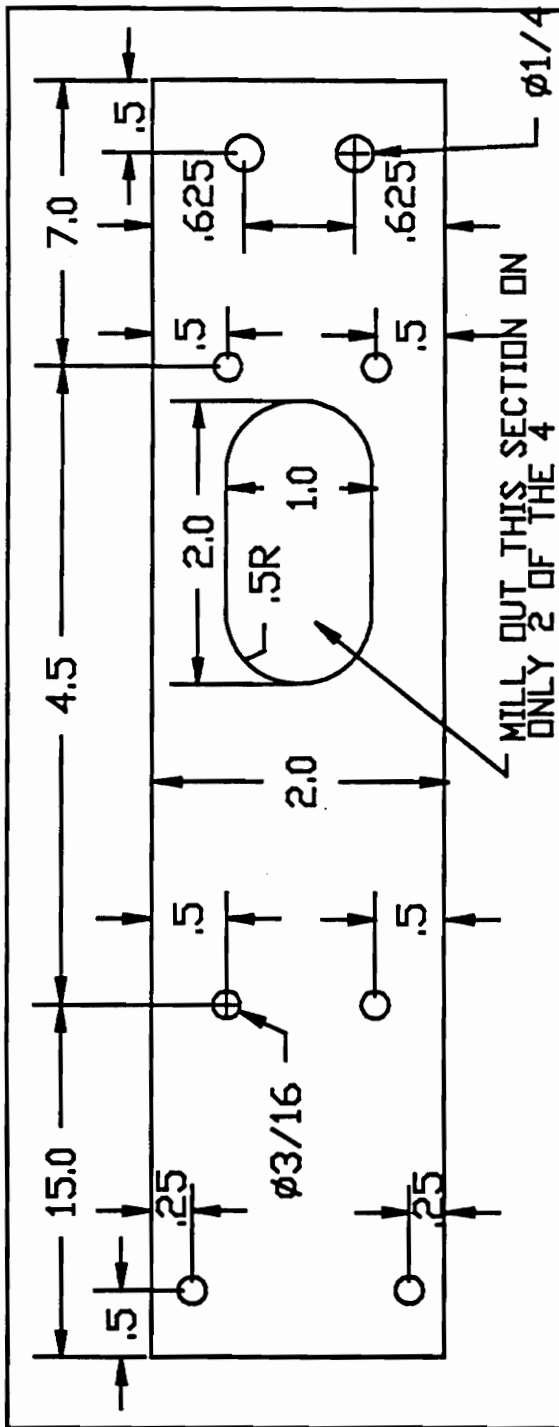
THICKNESS - .375
 NOTE: HORIZONTAL DIMENSIONS NOT TO SCALE

SWING ARM		MATERIAL: ALUMINUM	# REQUIRED 2
		8-2-90	SHEET # 2 OF 7
UNITS: INCHES			
SCALE 1:1			



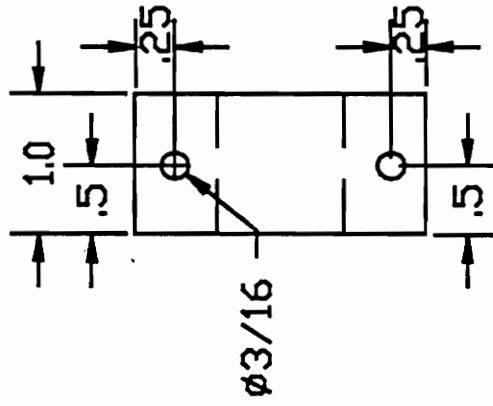
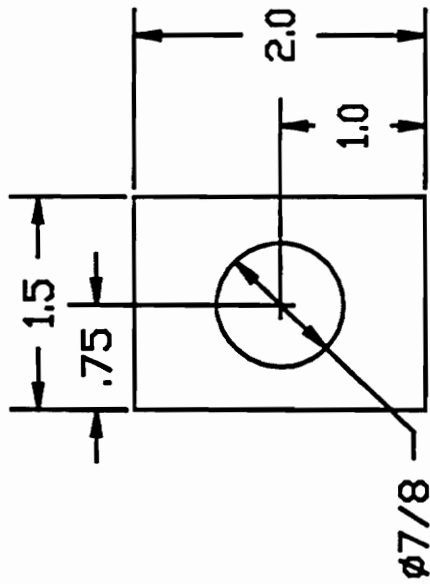
TUBE PIVOT

UNITS: INCHES	MATERIAL: ALUMINUM	# REQUIRED 2
SCALE 1:1	8-2-90	SHEET # 3 OF 6



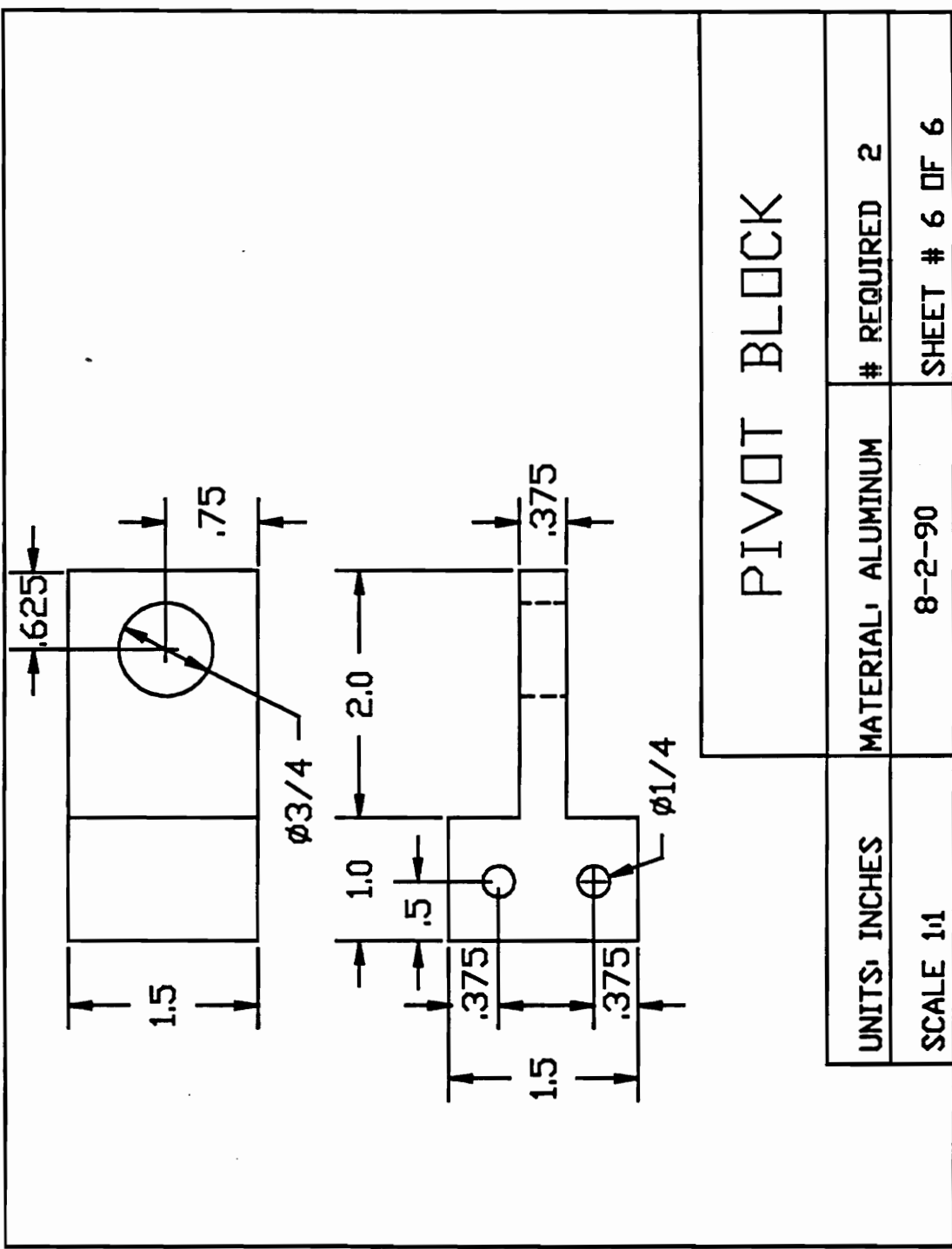
THICKNESS - .125
 NOTE: HORIZONTAL DIMENSIONS NOT TO SCALE

SCREW BRACES		
UNITS: INCHES	MATERIAL: ALUMINUM	# REQUIRED 4
SCALE 1:1	8-2-90	SHEET # 4 OF 7



TUBE GUIDE

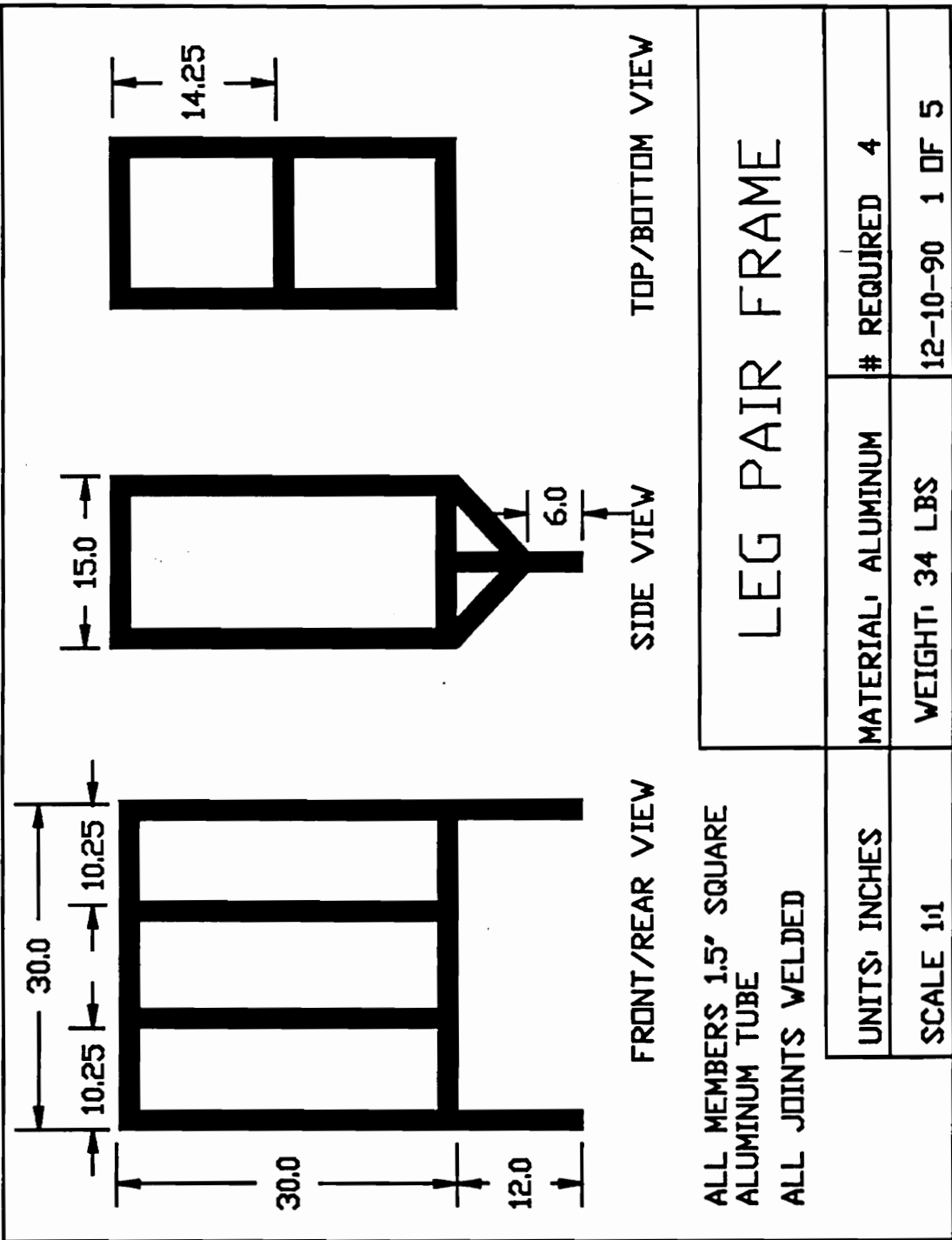
UNITS: INCHES	MATERIAL: BAKELITE	# REQUIRED 2
SCALE 1/1	8-2-90	SHEET # 5 OF 6



PIVOT BLOCK

UNITS: INCHES	MATERIAL: ALUMINUM	# REQUIRED 2
SCALE 1/1	8-2-90	SHEET # 6 OF 6

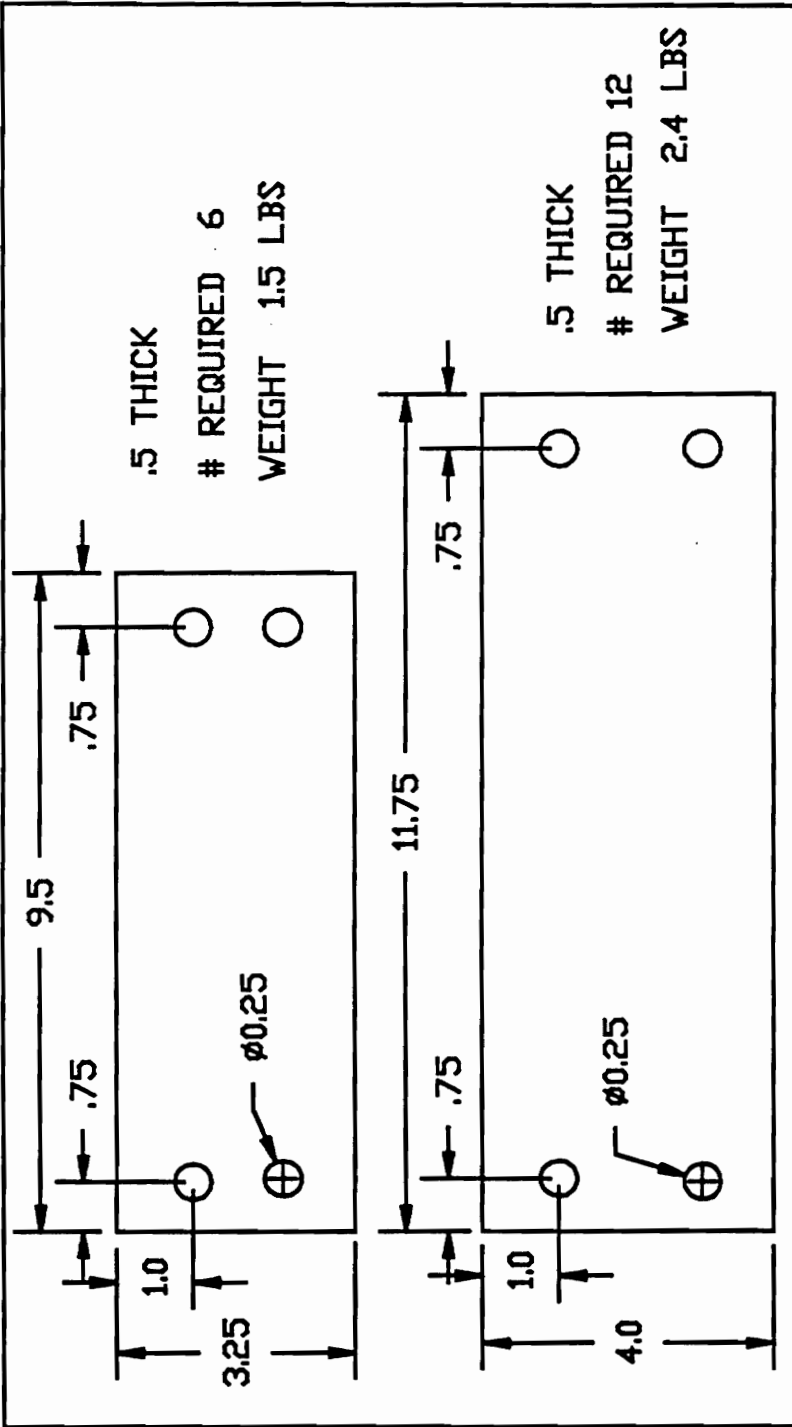
**CRAWLING VEHICLE
SHOP DRAWINGS**



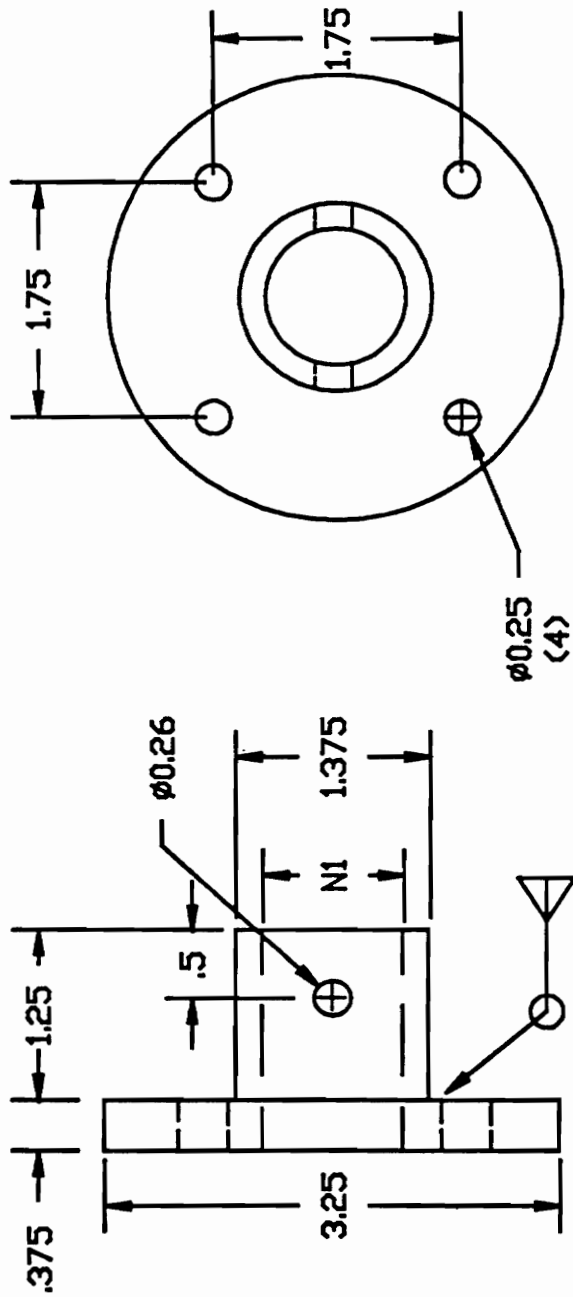
ALL MEMBERS 1.5' SQUARE
ALUMINUM TUBE
ALL JOINTS WELDED

LEG PAIR FRAME

UNITS: INCHES	MATERIAL: ALUMINUM	# REQUIRED 4
SCALE 1/1	WEIGHT: 34 LBS	12-10-90 1 DF 5



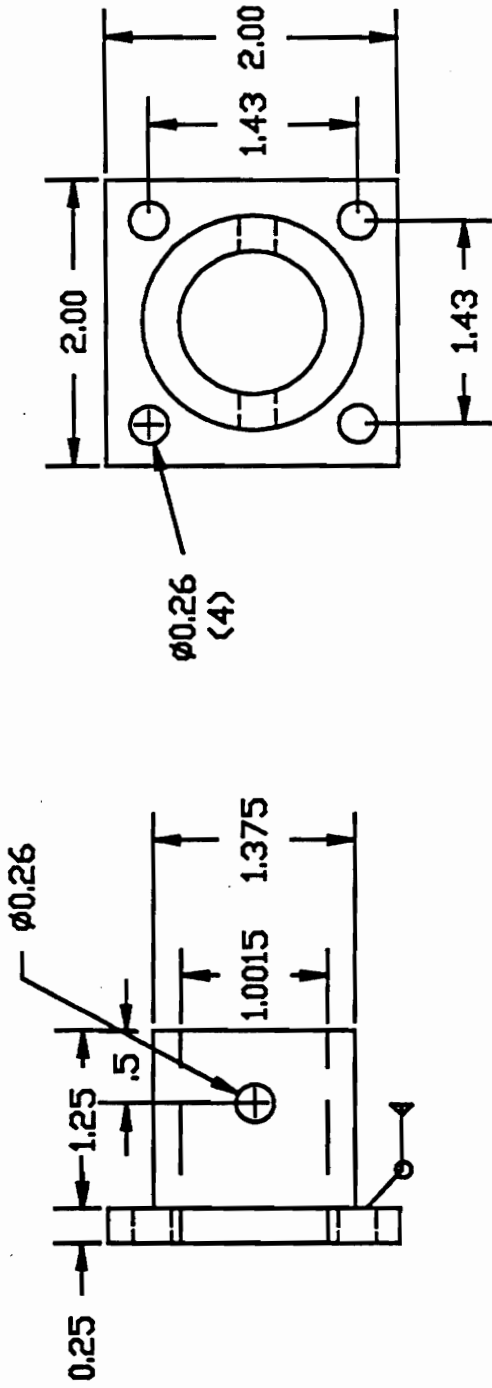
JOINT PANELS		
UNITS: INCHES	MATERIAL: ALUMINUM	#REQUIRED SEE ABOVE
SCALE 1:1	12-11-90	SHEET # 2 OF 5



N1-BORE SIZED FOR SNUG FIT OF 1.00" DIAMETER SHAFT

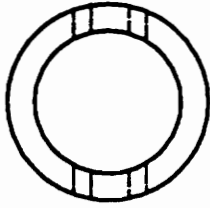
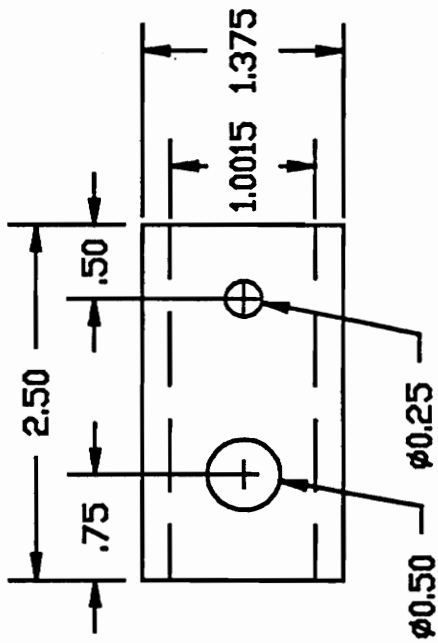
U-JOINT BASE

UNITS: INCHES	MATERIAL: ALUMINUM	# REQUIRED	36
SCALE: 1-1	WEIGHT: .369 LB	12-11-90	3 OF 5



MOTOR END CAP

UNITS: INCHES	MATERIAL: ALUMINUM	# REQUIRED 18
SCALE 1:1	WEIGHT: .18 LBS	10-11-90 4 DF 5



ROD END CAP

UNITS: INCHES	MATERIAL: ALUMINUM	# REQUIRED 18
SCALE 1:1	WEIGHT: .167 LB	10-11-90 5 OF 5

References

1. Bares J.E. and W.L. Whittaker, "Configuration of an Autonomous Robot for Mars Exploration," *Proceedings of the 1989 World Conference on Robotics Research: The Next Five Years and Beyond*, Society of Manufacturing Engineers, 1989, pp. 1-37 to 1-52.
2. Burgos, W., C. May, R. Harris, and B. McKimmens, "Mechanical Design of a VGT Crawling Machine," Senior Design Project Report, submitted to the Mechanical Engineering Department, Virginia Polytechnic Institute and State University, May 1989.
3. Carlton, R.E. and S.J. Bartholet, "The Evolution of the Application of Mobile Robotics to Nuclear Facility Operations and Maintenance," *Proceedings of the 1987 IEEE Conference on Robotics, and Automation*, Vol. 2, 1987.
4. Chirikjian, G.S. and J.W. Burdick, "Hyper-Redundant Robotic Locomotion: Locomotion without Wheels, Tracks, or Legs," Robotics and Mechanical Systems Report, School of Engineering and Applied Science, California Institute of Technology, March 1990.
5. Fichter, E.F., B.L. Fichter, and S.L. Allbright, "Arthropods: 350 Million Years of Successful Walking Machine Design," *Proceedings of the 7th World Congress on the Theory of Machines and Mechanisms*, Vol. 3, 1987, pp. 1887-1880.

6. Hirose, S., "A Study of Design and Control of a Quadruped Walking Vehicle," *International Journal of Robotics Research*, Vol. 3, 1984, pp. 113-133.
7. Hirose, S., and A. Morishima, "Design and Control of a Mobile Robot with an Articulated Body," *International Journal of Robotics Research*, Vol. 9, 1990, pp. 99-114.
8. McGhee, R.B., "Vehicular Legged Locomotion," *Advances in Automation and Robotics*, Vol. 1, JAI Press Inc., 1985, pp. 259-284.
9. Miura, K., and H. Furuya, "An Adaptive Structure Concept For Future Space Applications," *Proceedings of the 36th Congress of the International Astronautical Federation*, Pergamon Press, 1985.
10. Raibert, M.H., H.B. Brown Jr., and M. Chepponis, "Experiments in Balance with a 3D One-Legged Hopping Machine," *International Journal of Robotics Research*, Vol. 3, no. 2, Summer 1984, pp. 75-92.
11. Stulce, J.R., W.D. Burgos, S.G. Dhande, and C.F. Reinholtz, "Conceptual Design of A Multibody Passive-Legged Crawling Vehicle," *Proceedings of the 1990 ASME Mechanisms Conference*.
12. Waldron, K.J., and R.B. McGhee, "The Adaptive Suspension Vehicle," *IEEE Control Systems Magazine*, Vol. 6, no. 6, December 1986, pp. 7-12.
13. Wilson, D.M., "Insect Walking," *Annual Review of Entomology*, Vol. 11, 1966, pp. 103-121.

VITA

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Paul A. Mele