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Report

GUIDANCE AND ACTUATION TECHNIQUES
FOR AN ADAPTIVELY CONTROLLED VEHICLE

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FINAL TECHNICAL REPORT

on

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FOR AN ADAPTIVELY CONTROLLED VEHICLE

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This technical report describes the work done on a fourteen month effort to support the overall DARPA Adaptive Suspension Vehicle program. The Battelle project consisted of two major tasks: vehicle guidance and actuation techniques. The objective of the first task was to derive a method for enabling an adaptive-suspension vehicle to traverse rough terrain. The second task was concerned with actuation techniques for the legs on the vehicle, specifically an in-depth analysis, design, and demonstration of the vehicle foot-lift hydraulic circuit.

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

This report describes the development of the vehicle guidance algorithms and the foot-lift hydraulic circuit design.

SUMMARY

This report describes the work performed on a project which was part of the DARPA Adaptive-Suspension Vehicle Program. The Battelle project consisted of two major tasks. The first task involved the guidance of the adaptive-suspension vehicle; the goal was to develop algorithms which would enable the vehicle to traverse rough terrain. The second task concerned the design and demonstration of a hydraulic circuit which would provide the actuation for vertical motion of the vehicle legs. These two tasks are discussed separately in this report.

Guidance Task

Work on the guidance problem involved the development of algorithms to determine vehicle trajectories and leg motions which would enable the vehicle to traverse rough terrain; the algorithms use the operator's requested vehicle velocities, information about the vehicle state, and data from a terrain-scanning system to determine those trajectories and leg motions. The guidance problem was approached by developing a graphic simulation with which to test the algorithms. The simulation provided a representation of both the terrain-scanning system and the vehicle itself, both of which are described in the main body of this report. In addition, the simulation display included information which allowed indicators (such as vehicle stability) of the algorithms' performance to be monitored.

Previous approaches to the problem of legged vehicle locomotion emphasized the concept of static stability, the condition in which the vertical projection of the vehicle's center convex of mass is within the convex polygon formed by the vertical projections of those of the vehicle's feet that are on the ground. Those earlier approaches used as their goal in vehicle control the maintenance of the vehicle in a state of static stability at all times. This method of control was

possible because the vehicle's speeds were quite low so vehicle dynamics were relatively unimportant.

However, for a vehicle operating at higher speeds, maintenance of static stability is not sufficient for vehicle security, since in many cases the vehicle's motion could carry its center of mass outside its support polygon. Thus, a faster vehicle requires some sort of dynamic control of vehicle stability to ensure its safety. (Of course, for those times when the vehicle is not moving, static stability is still sufficient.)

The approach that was developed in this work was to maintain at all times a plan by which the vehicle can be brought to a halt in a position of static stability. That is, at every point along its path, the vehicle has available a plan for a vehicle trajectory that will bring it to a halt, together with a sequence of leg motions that will allow it to follow that trajectory and will leave it in a state of static stability at the end of the trajectory.

The maintenance of such a trajectory clearly places limits on vehicle velocity over a given terrain. Thus, with this approach, operator requests for vehicle velocities are not always met, although the vehicle will follow those requests as closely as possible while still maintaining vehicle safety. It can be seen that this approach does not permit an inexperienced operator to get the vehicle in trouble, but it still allows an experienced operator to use the full capabilities of the vehicle.

Actuation Task

This task was initially intended to develop a Design Automation Package for analyzing the drive train of the Adaptive-Suspension Vehicle. However, at the initiation of this effort, a joint decision by DARPA, Ohio State University, and Battelle redirected this effort towards providing assistance to OSU with regard to the design and evaluation of the vehicle foot-lift (shank) circuit. This effort included a preliminary design of the hydraulic circuit, development of a computer model for analyzing this circuit, and providing assistance to Ohio State University in the hardware implementation of this design.

Based on the preliminary design requirement for the foot-lift circuit established by OSU, a number of system concepts were generated. The concepts were then evaluated in conjunction with OSU, and a relatively simple circuit was selected. However, early in November, the system design requirements were revised by OSU. The resulting change in the operational requirements of the foot-lift circuit necessitated a complete revision of the circuit design. New system concepts were generated and evaluated, again in conjunction with OSU. At the time of this report, the final selection of the new circuit design had not been made.

Modeling efforts on this task were primarily involved in analyzing the original circuit design. A preliminary model was developed to determine the hydraulic flow requirements associated with the system design requirements. A more refined model was later developed to analyze the circuit performance, including its energy efficiency. This effort was somewhat de-emphasized from its original scope in order to devote additional effort to revising the system circuit.

Battelle provided assistance to OSU in the hardware implementation of the circuit in three ways. First, the hydraulic components originally selected for the foot-lift circuit by OSU were assembled, demonstrated, and evaluated by Battelle. Second, the original foot-lift circuit designed by Battelle was assembled and operated prior to delivery to OSU. Finally, assistance was provided to the hardware implementation effort at OSU for the computer-controlled, foot-lift circuit demonstration through discussions and loan of some equipment.

FOREWORD

This research was supported by the Defense Advanced Research Projects Agency, Washington, DC, under Contract No. MDA903-82-C-0149. Work was conducted in the period January 1982 through March 1983 by personnel of the Battelle Columbus Laboratories. The principal investigators were Mark R. Patterson of the Digital Systems and Technology Section (guidance) and John J. Reidy of the Equipment Development Section (actuation). In addition, Robert Rudolph of the Equipment Development Section provided substantial input in the actuation task. Overall project management was provided by Barry J. Brownstein of the Digital Systems and Technology Section.

The project engineers would like to acknowledge the cooperation and assistance of the personnel at Ohio State University involved in the development of the ASV-84 vehicle, in particular Professors Robert McGhee and Kenneth Waldron. In addition, valuable guidance was provided by Dr. Clinton Kelly, the DARPA Project Officer for this program.

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1.0 INTRODUCTION

It has long been recognized that most man-made vehicles are greatly inferior to human beings and other terrestrial animals in off-road locomotion. The shortcomings of current vehicles are particularly noticeable in the area of mobility. On rough terrain, a vehicle with a passive suspension system must accommodate obstacles by gross body motions. On the other hand, a system with active suspension units, such as legs, can pick its way through rough terrain by selecting the most suitable footholds and stepping over obstacles and soft spots. In addition, a legged system can compensate for terrain irregularities on which it must step by actively adjusting leg heights, thus providing a much smoother ride. It is apparent that the mobility advantages of a legged system depend on its ability to select suitable body and leg motions and avoid unsuitable ones, as well as then to perform those actions. Work performed in this project on both of those problems is described in this report.

The first part of the report describes the development of the vehicle "guidance" algorithms, which use information from a terrain-sensing system to determine appropriate vehicle trajectories and leg motions. This portion of the report has three main sections. The first of those sections describes the vehicle and terrain-sensing system which the algorithms are intended to be used and describes the simulation that was used in developing the algorithms. The second section presents the performance requirements for the terrain-sensing system and describes the algorithms which convert the data from that system to a terrain elevation map for use by the guidance algorithms. The last major section of this portion of the report reviews earlier research in legged locomotion for both natural systems and man-made vehicles and describes the operation of the guidance algorithms developed in this project.

Once a foothold has been selected, the foot must be physically shifted to that spot, smoothly and accurately. With a target cruise speed of five miles per hour for the vehicle, the requirements of the leg actuation system--the drive, foot-lift, and adduction-abduction circuits--are demanding. The foot-lift circuit must also be capable of emergency extension to compensate for unexpected problems in foot placement, such as the foot slipping off a rock. The efforts towards designing and evaluating the foot-lift circuits are described in the second portion of this report. First, the design and evaluation of a preliminary foot-lift circuit is described, as is the development of a circuit model for analyzing the design. Second, the assistance provided to OSU in the hardware implementation of the foot-lift circuit is described. The final section describes the efforts directed toward developing a revised, more efficient circuit.

2.0 GUIDANCE RESEARCH

2.1 Introduction

This portion of this report discusses the development of computer algorithms that determine the body and leg motions required for a legged adaptive-suspension vehicle to walk over rough terrain. The first section describes the vehicle, its terrain-sensing system, and a computer graphic simulation of the vehicle and the terrain-sensing system that was used in developing the algorithms. The second secion describes the process of converting the information from the terrain-sensing system to a terrain elevation map for use in determining the body and leg motions required of the vehicle. The "guidance" algorithms which determine those required vehicle motions from the elevation map are described in the third section. The last section describes a tentative computer hardware design for implementing the guidance system of the vehicle. Finally, some conclusions from this research are presented at the end of this portion of the report.

2.2 Background

2.2.1 The Adaptive-Suspension Vehicle

The vehicle for which the guidance algorithms were developed is approximately 15 feet (4.6 meters) long and 4 feet (1.2 meters) wide. Its height can be varied between approximately 5 feet (1.5 meters) and 9 feet (2.7 meters) by changing the extension of its six three-degree-of-freedom legs. The legs are attached at the top of the vehicle, with one pair each near the front, middle, and rear of the vehicle.

The velocity of the vehicle is expected to be limited, at least on the rough terrain for which the guidance algorithms were developed, to a translational velocity of no more than 8 feet/second (2.4 meters/second) and a rotational velocity of no more than 30 degrees/second. Translational and rotational accelerations are expected to be limited to no more than 4 feet/second/second (1.2 meters/second/second) and 15 degrees/second/second, respectively.

2.2.2 The Terrain-Sensing System

The terrain-sensing system with which the algorithms described in this report are intended to work is a scanning system mounted at the front top of the vehicle. The system scans both in elevation and azimuth; so, for each scan, it provides information for a two-dimensional sector of terrain in front of the vehicle. For each point in its scan, the sending system measures the distance from the scanner to the terrain at its current elevation and azimuth angles.

2.2.3 Terrain-Vehicle Simulation

For testing the guidance algorithms developed in this project, a computer graphic simulation of the vehicle and terrain was developed. The simulation provides a display of the moving vehicle and of that portion of the terrain of which it is aware at any given time. The forward and yaw (turning) velocities of the vehicle are controlled interactively.

The simulation was intended to provide as realistic a test to the guidance system as possible. Toward that end, it provides terrain scan data to the system in the same way as the terrain-sensing system described above will provide the data to the hardware implementation of the guidance system. That is, the data are provided to the algorithms in the same order as the actual scanner will provide them, and the data are those range values that the actual scanner would return on the same terrain, corrupted by expected random noise values.

The simulation also provides a realistic representation of the vehicle described above, to the extent that it incorporates the dimensions of the vehicle body and legs, the limits of travel for the legs' degrees of freedom (and thus the reachable volumes of the legs), and the expected limits on the vehicle's velocity and acceleration. All of the parameters which define the characteristics of the vehicle and the terrain-sensing system can be easily changed to accommodate design changes in those systems.

In addition to the display of the vehicle and the terrain, the legs' support pattern (the convex polygon formed by the points representing the legs' horizontal positions) is also displayed with the center of mass of the vehicle. That display allows monitoring of the vehicle's changing stability margin (the distance from the vertical projection of the vehicle's center of mass to the nearest point of the support pattern). The numerical value of the stability margin is also displayed, as is information on the positions of the legs relative to their limits. All of this displayed information can be used in evaluating the performance of the guidance algorithms.

2.3 Terrain Elevation Map Algorithms

2.3.1 Purpose

As described above, the terrain-sensing system provides, for each of its scan points, information on the range to the terrain. Since the scanner is fixed to the vehicle, and since the vehicle is moving, each of the scan point range measurements is made from a different position. Thus, the input from the scanning system to the guidance system consists of scan point range data indexed by elevation and azimuth angles and measured with respect to the moving vehicle.

However, the form of terrain information most useful for the guidance algorithms is that of an elevation map indexed by horizontal positions. The implementation of that form chosen for this system is a terrain array divided into cells based on a horizontal plane; one scan point is stored per cell. The array, then, must have sufficient resolution and extent to allow accurate determination of appropriate body and leg motions for the vehicle.

Thus, the guidance algorithms introduce performance requirements for the scanning system and create a need for other algorithms to convert the scanning system output to an elevation map. These topics are discussed in the next two sections.

2.3.2 Terrain-Sensing System Specifications

The parameters that can be specified for the scanning system include the angular ranges of the scan in both the elevation and azimuth directions, the angular resolution of the scan in both of those directions, the frequency of the scans, and the accuracy of the range measurements. The requirements of the guidance algorithms are based on the need for information for selecting footholds and for determining body motions for the vehicle.

Effective selection of footholds requires that all those areas where the vehicle may have to step be scanned at least once with a linear resolution of about one-half the vehicle's foot size, or approximately 4 inches (10 centimeters). Given the vehicle's height, then, the angular resolution required is approximately one degree, and the expected maximum velocity for the vehicle requires that there be two scans per second. Since it is expected that terrain areas with slopes of up to 45 degrees may be usable as footholds, the lowest elevation angle for the scan should be no greater than -75 degrees. A range accuracy for the scan points of 1 to 2 inches (2 to 5 centimeters) should be sufficient, since that is less than the limit of terrain elevation uncertainty that can be accommodated by the vehicle's control system.

For smoothly accommodating (without large acceleration) the vehicle to large-scale terrain obstacles, scan information should be obtained to approximately two body lengths, or 30 feet (9 meters), in front of the vehicle; this requirement puts a lower limit of -15 degrees on the upper elevation angle at the scan. Finally, the required azimuth range can be determined from the desired turning radius of the vehicle. In order that all the terrain under the vehicle be scanned when the vehicle is turning (at maximum foward velocity) on a radius of approximately its length, the azimuth scan range should be at least $\pm$ 30 degrees.

In summary, the guidance algorithms and vehicle requirements produce the following approximate terrain-sensing system specifications: two scans per second, 1 degree angular resolution, a -75 to -15 degree elevation scan range, a $\pm$ 30 degree azimuth scan range, and a range accuracy of 1 to 2 inches (2 to 5 centimeters).

2.3.3 Elevation Map Algorithms

As described above, the input to the guidance system from the terrain-sensing system is in the form of scan point range data indexed by the scan elevation and azimuth angles. When one of those range values is received by the elevation map algorithms, the elevation and azimuth angles of that range value, together with the known elevation, pitch, and roll of the vehicle, are used to convert the range value to an elevation value. Then, using the position and yaw of the vehicle, the location of that elevation point is determined and the value is stored in the appropriate cell in the terrain array.

The terrain array storage is fixed in size, so, as the vehicle moves, data storage "wraps around" from one portion of the array to another. This approach results in the algorithms automatically "forgetting" areas of the terrain after the vehicle has passed some distance beyond those areas.

2.4 Guidance Algorithms

2.4.1 Purpose

The function of the guidance algorithms is to determine appropriate body and leg motion commands for the vehicle control system, based on current operator requests. Those operator requests can be for three components of vehicle velocity: forward, side (crab), and turning (yaw). The guidance algorithms attempt to match the vehicle velocity to the operator's requests as closely as possible; however, as discussed below, those requests are not always attainable, due to considerations of vehicle stability.

Since the three vehicle velocity components mentioned above are usually the only ones which are of direct interest to the operator, the guidance algorithms automatically control the vehicle elevation, pitch, and roll based on the terrain over which the vehicle is passing. Then, once all six vehicle velocity components are specified, the guidance algorithms determine the leg motions required to attain those velocities while maintaining vehicle stability. The information required by the guidance algorithms to provide these body and leg motion commands includes, in addition to the operator's requests and the terrain elevation map described in the previous section, information concerning the current vehicle body state (position and velocity) and the positions and support states of the legs, as well as knowledge of the limitations on vehicle velocity and acceleration and on leg motions.

The use of that information in the operation of the guidance algorithms is discussed in detail below. First, though, previous work in the area of rules for coordinating walking, both in vehicles and in natural systems, is discussed in the next section.

2.4.2 Previous Walking Research

In the past, little work has been done on the behavior of walking systems on rough terrain, probably because studies of walking on smooth ground have proven complex enough that the added complications introduced by rough terrain have often not been considered. Recently, though, the particular problems introduced to walking systems by rough terrain, and the systems' responses to those problems, have begun to be studied more extensively. Some of this research, concerning both natural systems and man-made vehicles, is reviewed in the following paragraphs.

2.4.2.1 Natural Legged Systems

Until very recently, the work on walking insects on rough terrain has addressed only very specific problems in the area. For example, Cruse

(1976) investigated the control of the body position of stick insects walking on uneven surfaces. His work indicated that these insects hold their bodies so their slope is approximately equal to that of the terrain (he only investigated walking over obstacles perpendicular to the path of the insect). In the area of foothold selection, Cruse (1979) found that walking stick insects place their rear legs at footholds immediately behind and outside the footholds of their middle legs (he also found some evidence that the placement locations of the middle legs are based on those of the front legs). Thus, his work indicated that, at least in some situations, the insects use a "follow-the-leader" method of selecting footholds.

However, more recent work, conducted by Pearson and Franklin in association with the present adaptive-suspension vehicle program (Pearson 1982), found little evidence indicating use of the "follow-the-leader" method in locusts. In fact, the locusts' individual legs appeared to use no information from the other legs in selecting their footholds. Instead, the behavior of individual legs in attempting to find support on rough terrain had three independent aspects: rhythmic searching movements by legs when they failed to find support (similar searching was also mentioned in Cruse 1979), lifting reflexes initiated when legs contacted objects in their transfer phases, and local searches for specific support sites following contact with the surface.

In addition to the investigation of foothold selection in locusts, Pearson also reported an examination of the insects' gaits, which proved to be quite variable, especially on rough terrain. The locusts did exhibit general tendencies for posterior-to-anterior stepping sequences in the legs on each side of the insect and for coordinations of legs on the two sides of the insect which were either 180 degrees out of phase or almost exactly in phase, but in many instances, especially on rough terrain, the insects' behavior contradicted these tendencies.

To account for the wide variety in the insects' stepping patterns, Pearson conjectured that a necessary condition for lifting a leg is that the leg be unloaded beyond a critical value. Since the loading on a leg depends on the insect's body position, the positions of all its legs, and the forces being generated in the legs, the unloading condition could introduce very complex leg coordination behavior that is highly dependent on the terrain over which the insect is walking. (An interesting characteristic of the unloading condition is that, in using it, a leg would not require additional information from the other legs, since their positions and forces would already be reflected in its loading.)

Finally, Pearson reported that the locusts used vision to detect obstacles that they were approaching, but that they did not appear to use it in selecting individual footholds, using, instead, the foothold location methods discussed above. This is in direct contrast to the work on walking vehicles discussed below, which in each case assumes that the vehicle has knowledge of the terrain over which it is walking.

2.4.2.2 Legged Vehicles

One of the earliest reports of work done on automatic guidance of a legged vehicle over rough terrain was that of Okhotsimski and Platonov (1973). They considered obstacles that could be overcome by recalculating foothold locations without changing a periodic gait. For example, the problem of traversing a trench located where one of the vehicle's feet would normally have been placed was solved by moving that foothold to one of the edges of the trench and adjusting the nearby footholds to provide a smooth transition.

It is apparent that, when obstacles are large or numerous, the vehicle's gait must be modified and the method of Okhotsimski and Platonov cannot be used. One of the first reports of work addressing this problem was that of Kugushev and Jaroshevskij (1975). They introduced the "free gait," a gait in which leg motions are determined not from a periodic sequence but from knowledge of the current state of the vehicle, its kinematic limits, and the terrain over which it is walking.

Central to their free gait algorithm is the concept of a foothold's existence segment, the portion of the vehicle's path of motion over which a particular leg can reach that foothold. At each iteration of the algorithm, the determination is made of which leg will reach the end of its existence segment soonest. That leg is then lifted, if that can be done without making the vehicle unstable; if lifting the leg would make the vehicle unstable, another leg (or legs) is moved first to allow it to be lifted. Footholds for leg placement are apparently found by positing a vehicle location farther along its motion path and determining if a foothold can be found there which the leg can reach; if one cannot be found, the posited vehicle location is moved farther back toward the vehicle's actual location until a foothold can be found on which to place the leg.

Kugushev and Jaroshevskij's work was extended by McGhee and Iswandhi (1979). They introduced the concept of the kinematic margin of a foothold, which is the distance along the vehicle's path of motion from its present position to the point at which the leg at that foothold would reach its kinematic limit. They then used that concept to implement a free gait algorithm with which, at each iteration, the leg with the smallest kinematic margin is lifted. If necessary, another leg is placed in a position that allows the first leg to be lifted. Leg placement is accomplished by examining all available footholds for a particular leg and then placing the leg at the foothold with the greatest kinematic margin. It can be seen that McGhee and Iswandhi's algorithm lifts legs whenever it can and only places them when required for stability; the algorithm thus sacrifices stability for adaptability.

Both of the free gait algorithms discussed above assume a flat terrain containing areas that cannot be used for footholds. Klein and Patterson (1982) considered the application of a free gait algorithm on a cylindrical surface, which presents several problems not present with strictly two-dimensional terrain. One of these problems is that, as the relative orientation and distance between the vehicle and the terrain change, the location, size, and shape of the terrain area which a leg can reach changes. That area, which for many legs on flat terrain is a sector of an annulus, becomes very irregular in rough terrain; the solution used in this work is to determine a minimum reachable area based on the regular cylindrical terrain. Klein and Patterson also modified the kinematic margin concept, basing it on the instantaneous velocity of the base of the leg instead of the predicted path of the vehicle; the kinematic margin can then be calculated even for vehicle motions such as turning in place. In addition, they modified the free gait algorithm in such a way that legs are lifted only when their kinematic margins fall below a threshold margin. Different settings of that threshold can then be used to optimize the algorithm for either stability or adaptability.

In summary, each of the research papers discussed above introduced techniques which would enable walking vehicles to traverse some specific kinds of rough ground. However, none of them described an approach that would allow the locomotion of such vehicles over fully three-dimensional terrain. The guidance algorithms presented in the next section are intended to address that problem.

2.4.3 ASV Guidance Algorithms

The walking algorithms described in the previous section all operate by maintaining the vehicle in a condition of static stability (that is, with the vertical projection of the vehicle's center of mass within the support pattern of its legs). Actually, though, only if the vehicle is motionless (static) does static stability ensure that it will not fall over, since if it is moving, its motion could carry its center of mass outside the boundaries of its support pattern. Ensuring the stability of a moving vehicle by using the static stability criterion requires that at all times there exist for the vehicle a sequence of support patterns through which the vehicle can be safely brought to a halt at a position satisfying the static stability criterion.

Thus, the overall task of the adaptive-suspension vehicle guidance algorithms is to continually generate those support pattern sequences with their associated trajectories for halting the vehicle. Such trajectory-support pattern sequence combinations, which will be referred to as deceleration plans, provide the body velocity and leg motion commands that the guidance system sends to the vehicle's control system. For use in calculating the deceleration plans, the guidance system receives from the vehicle control system the operator-requested vehicle velocities, the actual vehicle position and velocity, and the actual leg positions and support states. That information, together with the terrain elevation map and knowledge of the vehicle's configuration and capabilities, is then used to generate the deceleration plans as described in the following paragraphs.

2.4.3.1 Vehicle Velocity Determination

As discussed above, ensuring the stability of a walking vehicle requires that a deceleration plan be maintained at all times for the vehicle. Since this requirement places a limit on the vehicle's velocity for any given position on the terrain, the ensuring of the vehicle's stability may at times conflict with the implementation of the operator's requested velocities. Thus, the guidance algorithms have the responsibility for controlling the vehicle velocity in such a way as to make it follow as closely as possible the operator's requests while never allowing the vehicle to enter a state from which it could become potentially unstable.

The approach which the algorithms use to perform that task is shown in Figure 1. As can be seen there, the algorithm first tries to generate (as described below) a deceleration plan for the operator's requested vehicle velocities. If a plan can be generated for those velocities, that plan is then used to determine vehicle velocity and leg motion commands until a new plan is generated. Thus, in that case, the vehicle will accelerate (or decelerate) as the operator requested.

However, if no deceleration plan can be generated for the operator-requested velocities, the algorithm attempts to generate plans for, first, the operator-requested vehicle turning radius but the current vehicle translational velocities (thus following the operator's requested path) and, finally, the current vehicle velocities. If a plan can be generated for either of those conditions, that plan is then used to provide the vehicle body and leg commands. If neither of those conditions allows a plan to be generated (that is, if no new plan can be generated), the most recently generated plan (from an earlier iteration of the algorithm) is used to determine the vehicle commands. In that case, then, the vehicle

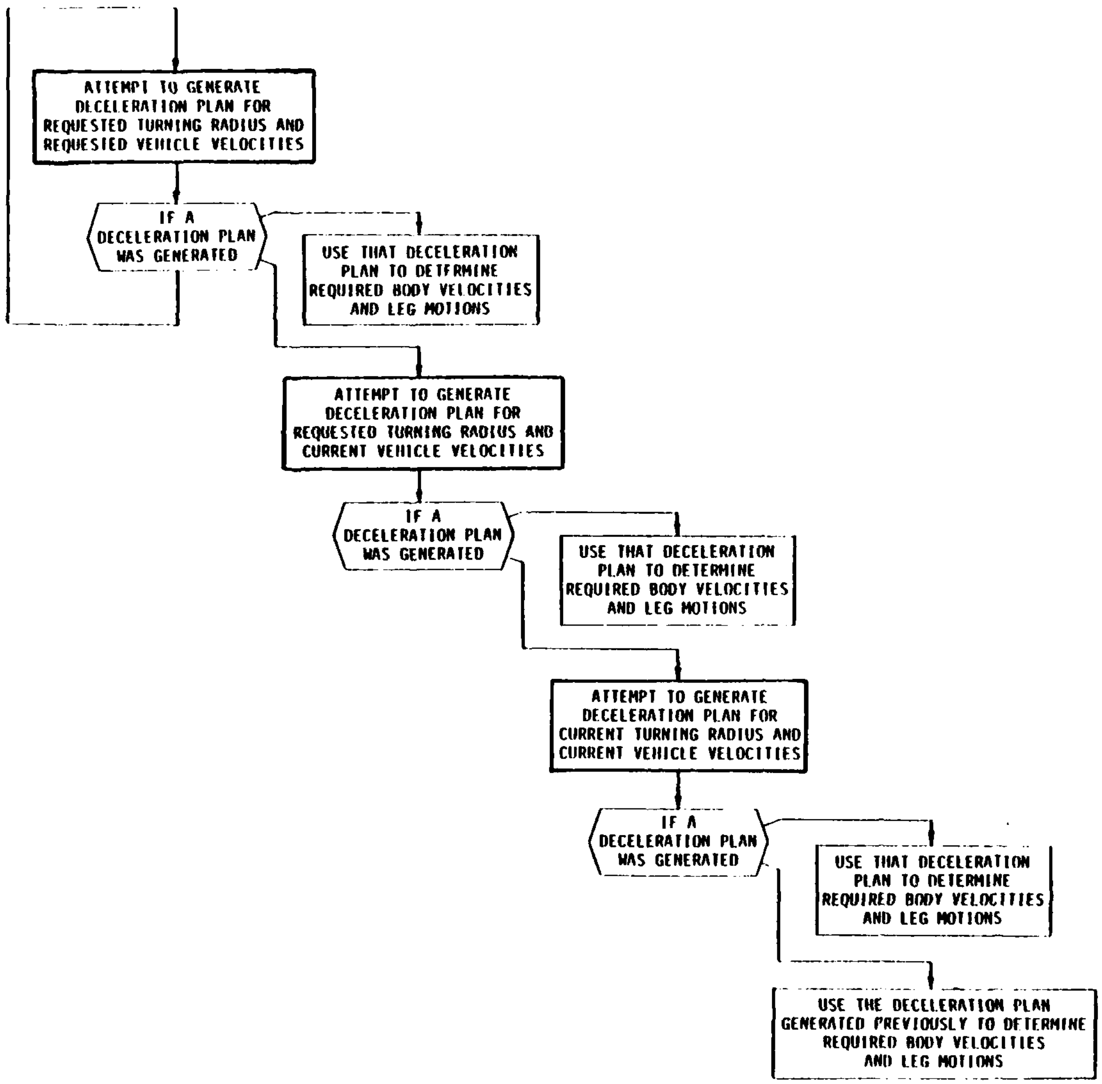


FIGURE 1. GUIDANCE ALGORITHM FLOW CHART

(FLOW CHART CONVENTIONS ARE DESCRIBED IN APPENDIX A)

will begin to decelerate according to the trajectory stored in the deceleration plan.

In summary, the algorithm tries to provide vehicle velocities as close as possible to the operator's requests. It does that by attempting to generate deceleration plans for, first, the operator's requested velocities and, then, for successive fallback positions incorporating fewer of the operator's requests. Only if no plan can be generated in any of those attempts does the algorithm actually begin to bring the vehicle to a halt using a previously generated deceleration plan's vehicle trajectory and support pattern sequence. The way in which such trajectories and support pattern sequences are generated is described in the following paragraphs.

2.4.3.2 Deceleration Plan Generation

The overall approach to the generation of deceleration plans is shown in Figure 2. The algorithm uses an iterative approach that proceeds until either a plan is completed for bringing the vehicle to a complete halt or a point is reached at which no acceptable continuation of the deceleration plan can be found. The iterations of the algorithm take place at successive distance increments along the vehicle path (which is known, since a constant turning radius is assumed for the vehicle).

For each iteration of the algorithm, the first step is to determine the vehicle's position and orientation at that point along its path. That determination is accomplished by setting the vehicle's pitch and roll to make it parallel to the average terrain slope at that point (as calculated from the terrain elevation map) and by setting the vehicle's elevation relative to the terrain based on the preset vehicle altitude. (The vehicle horizontal position and yaw angle are determined by its position along the path of motion.) The calculation is then made of the time at which the vehicle reaches that point along its path using its maximum possible deceleration from the previous point. During that calculation, which is also performed iteratively, the legs' support states

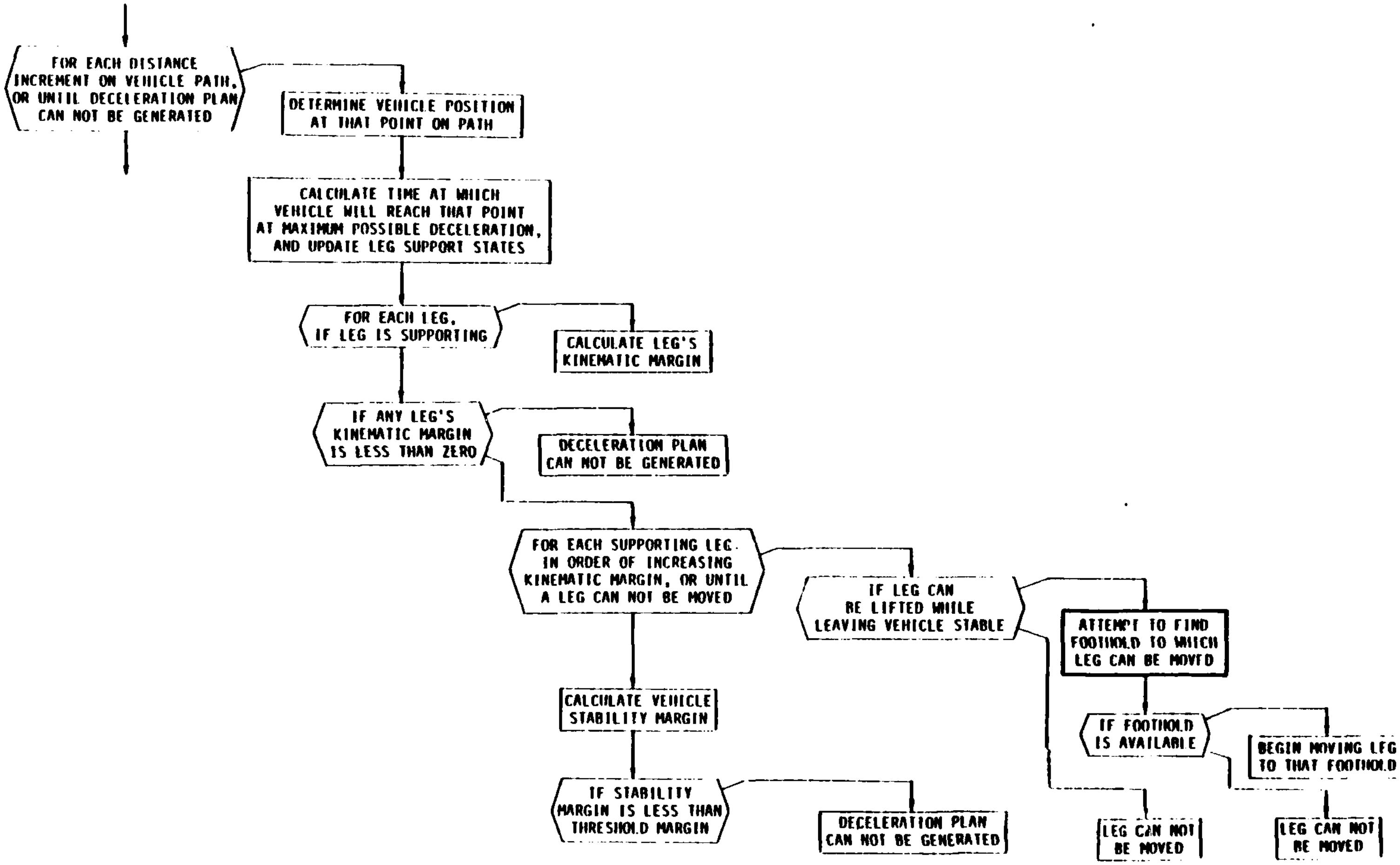


FIGURE 2. DECELERATION PLAN GENERATION FLOW CHART
(FLOW CHART CONVENTIONS ARE DESCRIBED IN APPENDIX A)

are also updated, if necessary, using the knowledge of their transfer times from foothold to foothold.

Next, the algorithm calculates the kinematic margin for each leg supporting the vehicle. If any of the legs has a kinematic margin of less than zero (in other words, if any of the legs would be out of its limits when the vehicle is at that position), then a deceleration plan cannot be generated for the given vehicle velocity. In that case, the algorithm exits from the deceleration plan generation routine.

Otherwise, the algorithm, beginning with the leg with the lowest kinematic margin and continuing to that with the highest, determines if any of the supporting legs can be moved. It does that by first evaluating whether the leg can be lifted while leaving the vehicle stable. If it can, the algorithm then attempts (as described below) to find a foothold to which the leg can be moved; if it finds an acceptable foothold, it begins to move the leg to that foothold. If no foothold is found, or if the leg cannot be lifted, the algorithm terminates the attempt to move a leg.

Finally, the last step in an iteration of the deceleration plan generation is the evaluation of vehicle stability. If the vehicle is stable at its current position, the algorithm begins another iteration; if it is not stable, a deceleration plan cannot be generated for the given velocities.

2.4.3.3 Foothold Search

The foothold search procedure which was mentioned above is outlined in Figure 3. The first step is to determine a nominal position at which it would be desirable to place the foot at the end of its transfer phase. Terrain elevation map cells near that nominal position are then evaluated as possible footholds until either a foothold is found or all cells near the nominal position have been examined unsuccessfully.

The evaluation of each cell proceeds by first checking whether the map has information for that cell. If no terrain information is available, no (known) foothold exists there; if information is available, that information is used to calculate the slope at that foothold. If the

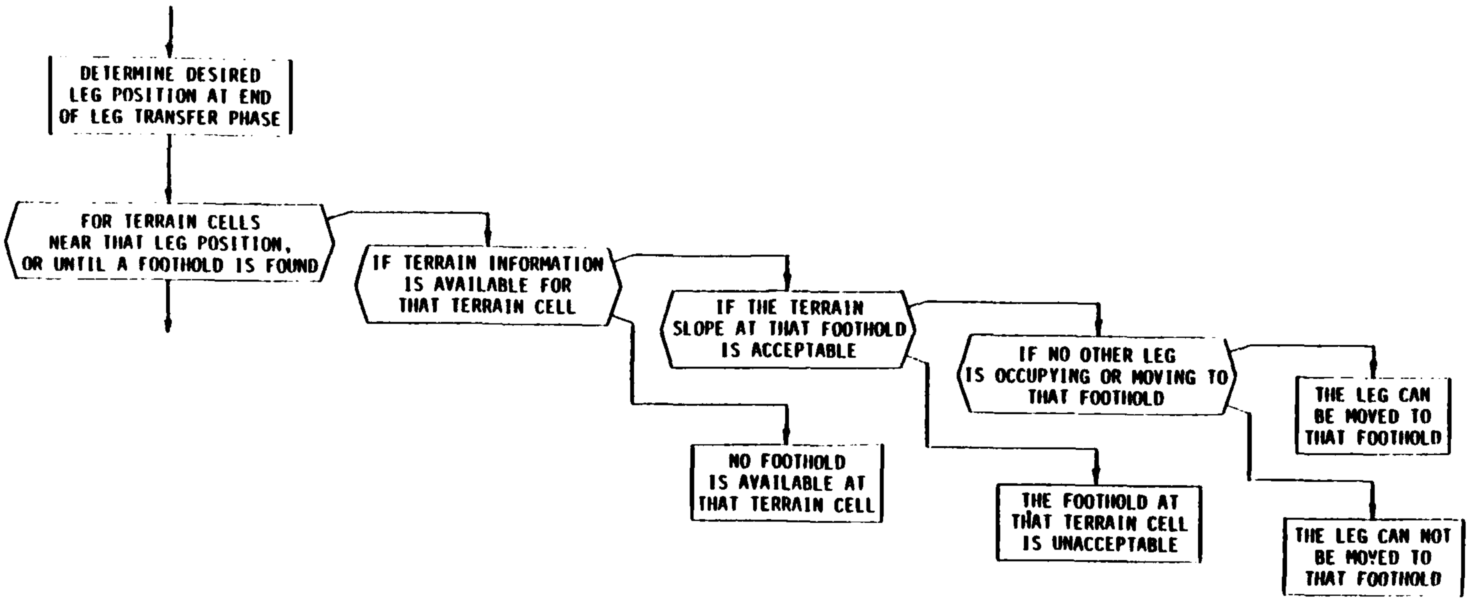


FIGURE 3. LEG FOOTHOLD SEARCH FLOW CHART
(FLOW CHART CONVENTIONS ARE DESCRIBED IN APPENDIX A)

slope is too high, the foothold is unacceptable; if the slope is sufficiently low, a final check is made for any other leg already occupying or moving to that foothold. If none is, the leg can be moved to that foothold; otherwise, the foothold is unavailable.

2.4.4 Algorithm Performance

The algorithms presented above were tested extensively against the terrain-vehicle simulation described earlier. The simulated vehicle which was used had body dimensions and leg configurations and kinematic limits that corresponded to those expected for the physical vehicle. Similarly, the expected velocity and acceleration capabilities of the physical vehicle were used in the simulation. Typical examples of terrains used in the tests are shown in Figures 4 and 5.

The simulated vehicle performed quite well, having little trouble with terrains such as those shown in the figures. For example, in walking up the hill shown in Figure 5, although the vehicle's velocity varied up and down depending on the vehicle's situation, the lowest value the velocity reached was approximately 4 feet per second. Considering the severity of that terrain, such performance seems quite good.

2.5 Tentative Hardware Structure

A tentative approach to the overall structure of a hardware system to implement the algorithms presented above is described in this section. The description is extremely general because, first, the goal of this research project was to develop algorithms, not hardware, and, second, no effort was made to optimize the algorithms or to determine how much computational power they will require to operate in real time.

However, regardless of the computation that will be required by the algorithms, a reasonable approach to the structure of a hardware system to implement them is likely to be to separate the functions of the elevation map algorithms and the guidance algorithms into two independent processing systems. The elevation map system (consisting of one or more

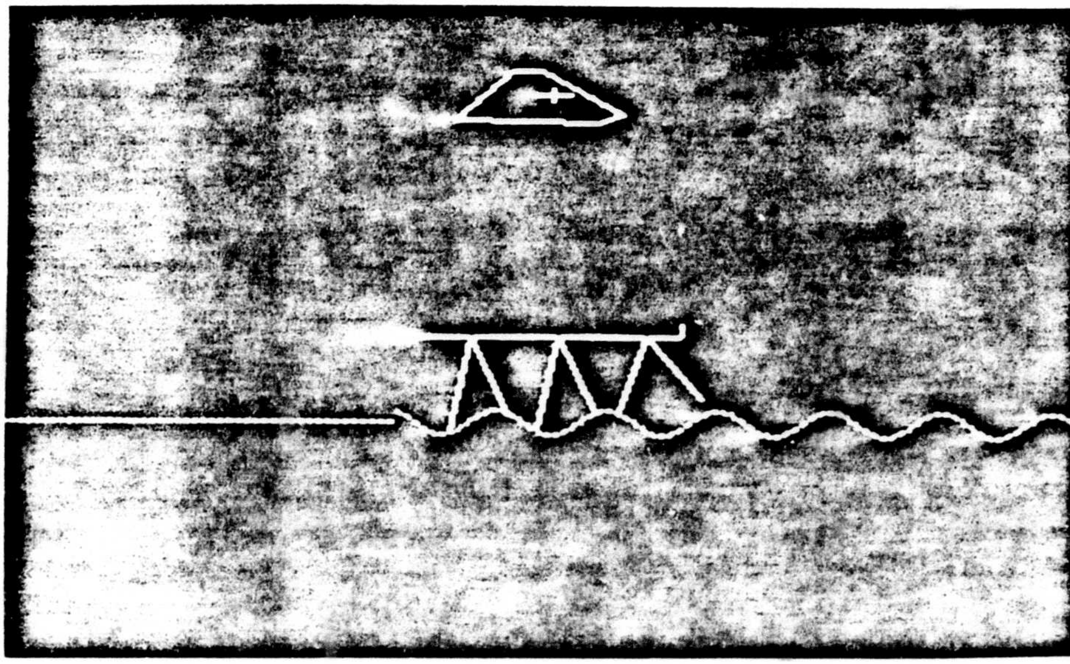


FIGURE 4. VEHICLE WALKING OVER TERRAIN
(PEAK-TO-VALLEY VERTICAL DISTANCE - 2 FEET, OR 0.6 METER)

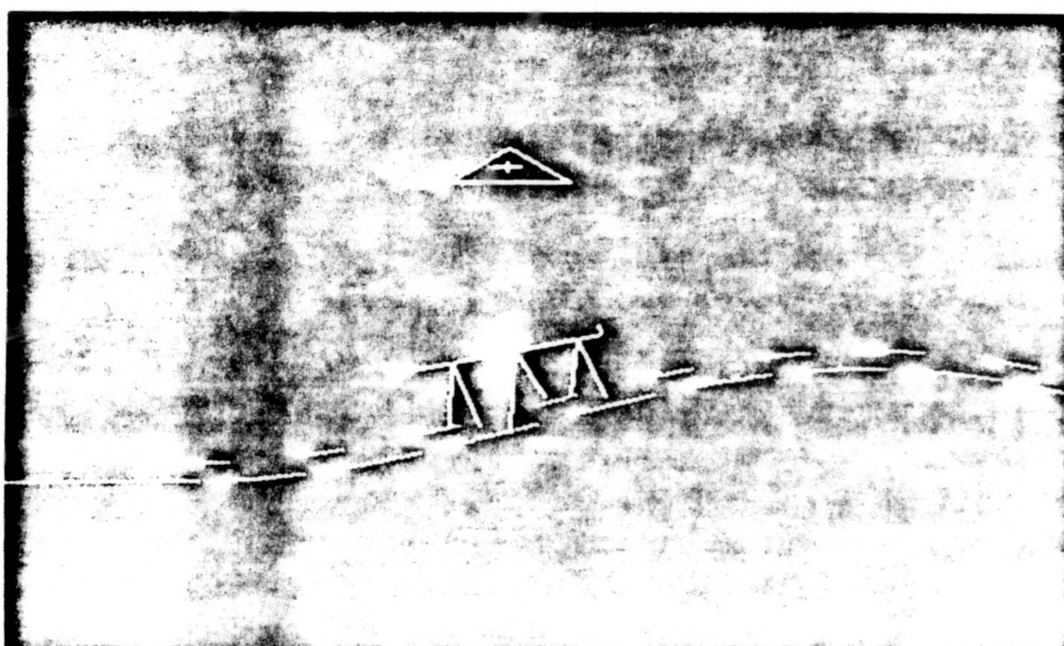


FIGURE 5. VEHICLE WALKING OVER TERRAIN
(HILL HEIGHT - 12 FEET, OR 3.6 METERS; OBSTACLE HEIGHTS - 2 FEET, OR 0.6 METER)

processors) will then receive input data from the terrain scanning system and produce the elevation map, which it will store in memory dedicated to that purpose. The system for the guidance algorithms (which will also consist of one or more processors) will use the stored elevation map together with information from the vehicle's control system to determine appropriate body and leg commands, which it will then send to the control system.

2.6 Conclusion

In summary, this portion of this report describes a set of algorithms developed to enable a legged vehicle to walk over rough terrain. The algorithms use data from a terrain-scanning system, information from the vehicle's control system, and knowledge of the vehicle's capabilities and limitations to determine the body and leg motions required for the vehicle's locomotion over the terrain. The algorithms have been successfully tested with a simulation of a vehicle that is currently under construction, and a hardware implementation of the algorithms will soon be built and tested with that vehicle.

3.0 ACTUATION RESEARCH

3.1 Foot-Lift Circuit Design

3.1.1 System Requirements

The design requirements for the foot-lift circuit of the Adaptive-Suspension Vehicle were initially established through discussions with Professor Ken Waldron of O.S.U. These requirements were based on the original O.S.U. leg design and consisted of the following:

Cruise Mode

(1) Rapid retraction - lift foot 2 to 8 inches in 0.05-0.10 seconds (10-15 percent of return cycle).
(2) Slow extension - allow foot to coast down, or extend the foot under pressure, in approximately 0.30-0.40 seconds.
(3) Lock on - no extension of foot under load.

Turn or Crab

(1) Small displacements at full load.

Emergency

(1) Rapid extension under no load.

Additional system considerations included:

- Total cylinder stroke - 48 inches.
- Stride frequency (cruise mode) - 1 Hz.
- Estimate weight of lower leg - 50 pounds.

At Prof. Waldron's suggestion, a maximum load of 1800 pounds per foot was assumed. The original leg design upon which these requirements are based is shown in Figure 6. A schematic of the hydraulic circuit developed by O.S.U. for this original leg configuration is shown in Figure 7. The total stroke of the leg was established at 48 inches and the weight of the lower leg was estimated to be 50 pounds.

3.1.2 Preliminary Analysis

The original O.S.U. circuit design was analyzed with regard to the system requirements. First, the relationship between cylinder size and system pressure was investigated using the definition of pressure, $P = \frac{F}{A}$, and a maximum load of 1800 pounds (see Table 1). In these calculations, area represents the net area of the cylinder over which the pressure is acting. To further illustrate this relationship, it was assumed that the system pressure acted on the entire piston area with zero back pressure, and the required system pressure was calculated based on standard diameters. This relationship between cylinder area and pressure is graphically illustrated in Figure 8.

This relationship can be further developed into a relationship that can be used in selecting a system pressure, using the pressure and the extension rate of the cylinder. As a first approximation,

$$\frac{Q}{A} \times \frac{231\frac{in^3}{gal}}{60\,\frac{sec}{min}} = \dot{X}$$

where Q = flow (gpm)
A = area (in^2)
$\dot{X}$ = cylinder extension rate (in/sec).

If the previous relation is substituted.

$$P = F/A$$

$$A = F/P = 1800\text{ lb}/P$$

$$\frac{Q \times P}{1800} \times \frac{231}{60} = \dot{X}$$

$$\frac{Q \times P}{467.35} = \dot{X}$$

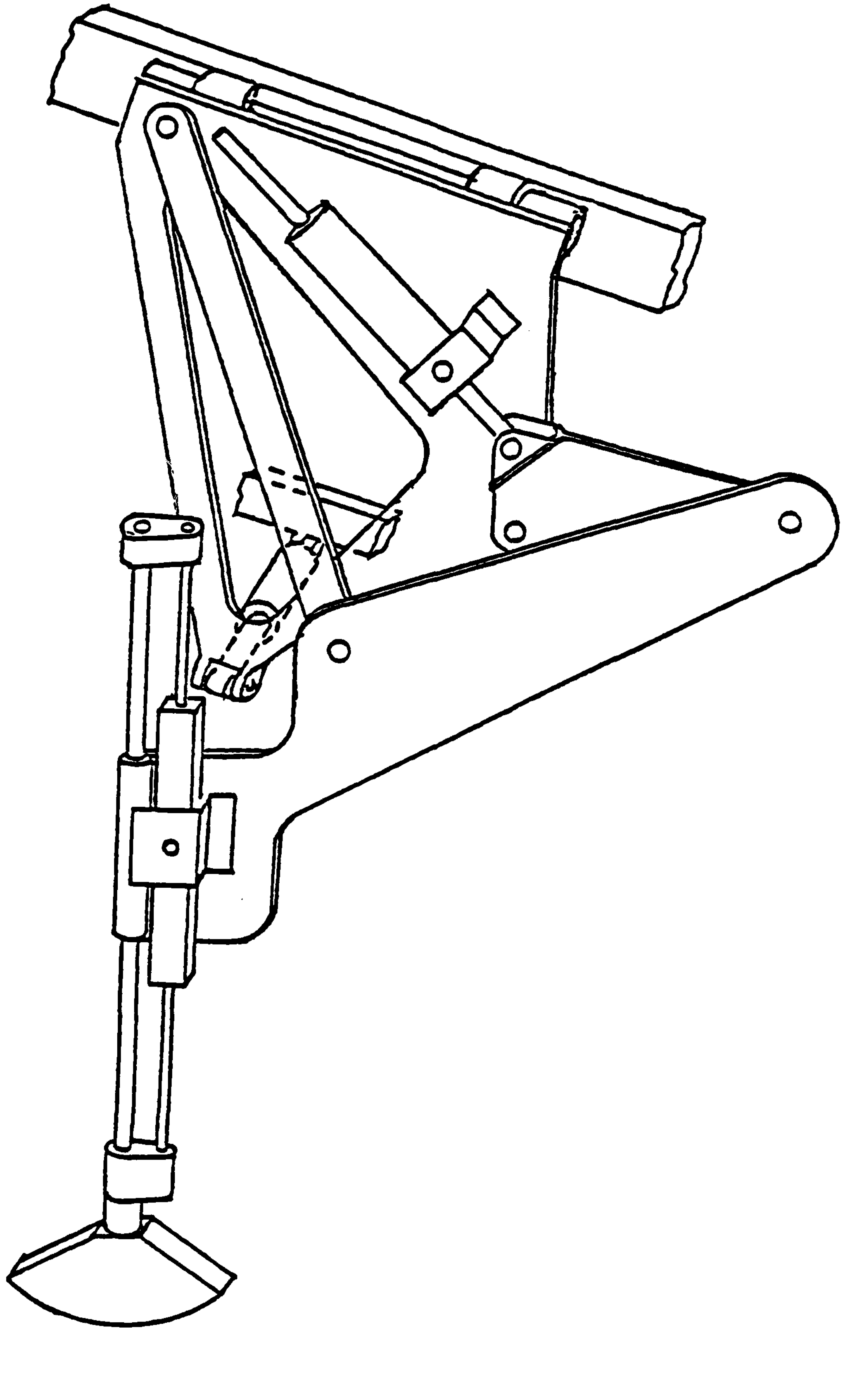


FIGURE 6. ORIGINAL O.S.U. LEG DESIGN

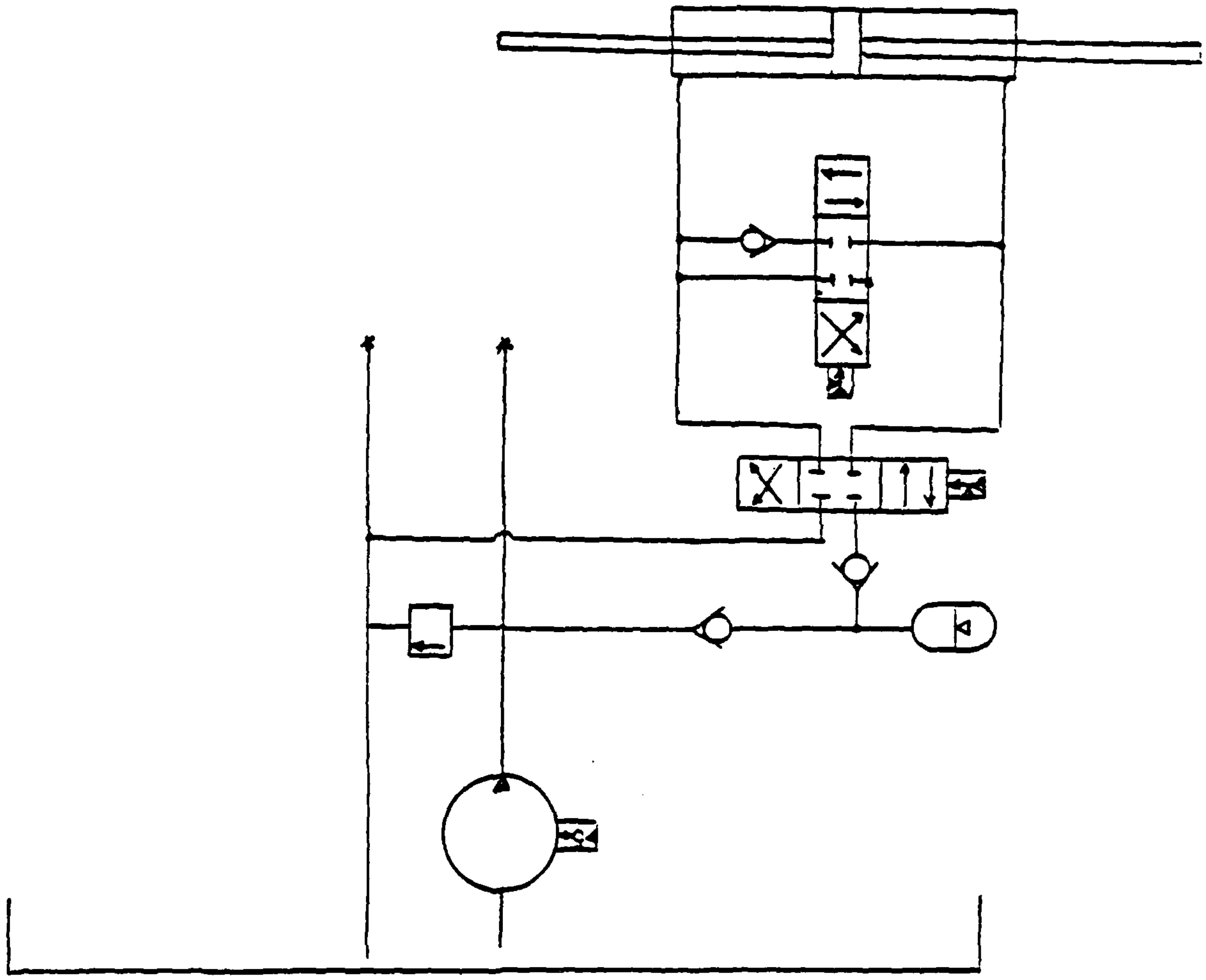


FIGURE 7. FOOT-LIFT CIRCUIT DESIGN, ORIGINAL O.S.U. LEG CONFIGURATION

TABLE 1. FOOT-LIFT CIRCUIT PRELIMINARY HYDRAULIC FLOW CALCULATIONS

Pressure (psi)	Area (in^2)	Diameter[1] (in)
1000	1.8	
1500	1.0	
2000	0.9	
2500	0.72	
3000	0.6	
4074	0.442	3/4
2292	0.785	1
1811	0.994	1 1/8
1467	1.227	1 1/4
1212	1.485	1 3/8
1018	1.767	1 1/2

[1]Values represent cylinder bore diameters with zero pressure on the rod end of the cylinder. Maximum load = 1800 lb.

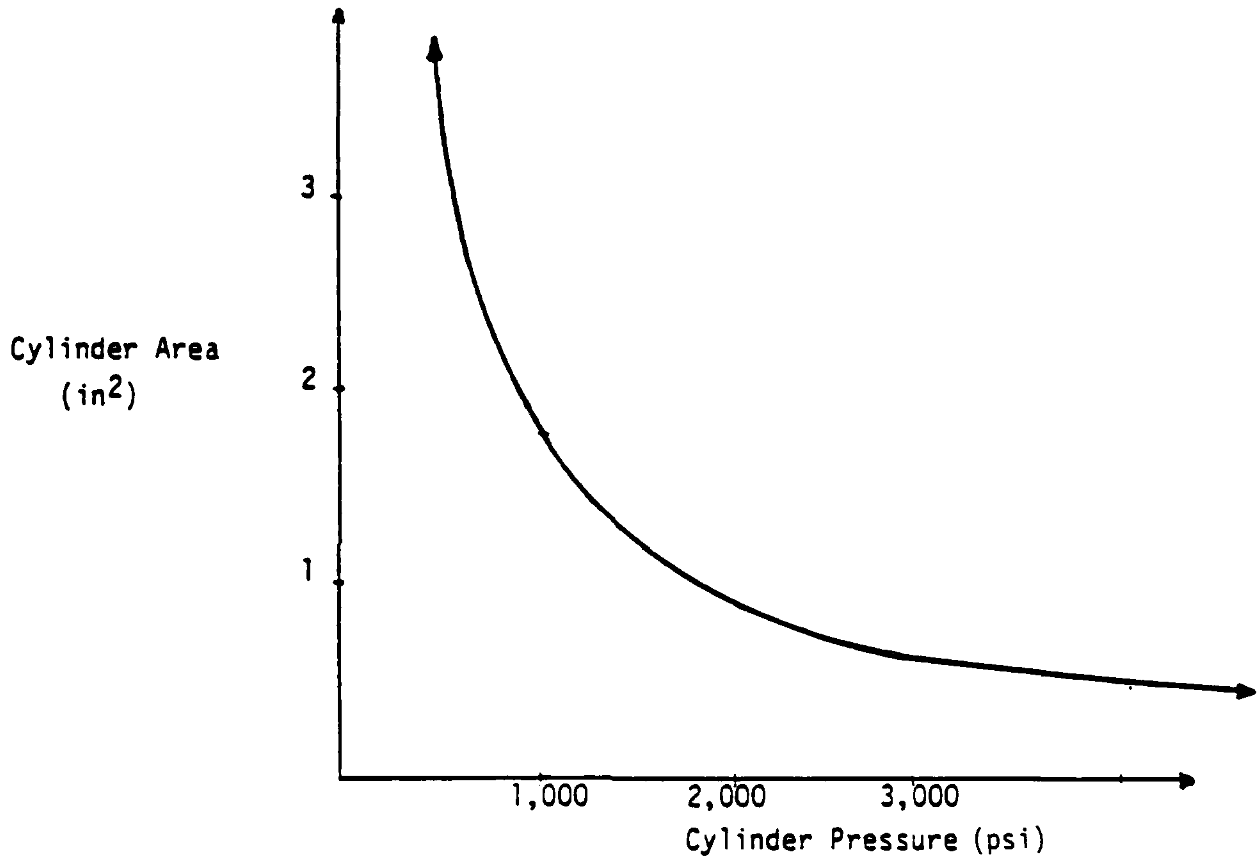


FIGURE 8. CYLINDER AREA VS PRESSURE RELATIONSHIP FOR 1800 POUND LOAD

Some representative values have been calculated and are included in Table 2. The relationship is graphically illustrated in Figure 9.

TABLE 2. CYLINDER EXTENSION RATE
AS A FUNCTION OF PRESSURE AND FLOW

Cylinder Pressure (psi)	Flow (gpm)				
	5	10	15	20	30
	Extension Rate (in/sec)				
1,000	10.69	21.39	32.08	42.78	64.17
1,500	16.04	32.08	48.13	64.17	96.15
2,000	21.39	42.78	64.17	85.56	128.33
2,500	26.74	53.47	80.21	106.94	160.42
3,000	32.08	64.17	96.25	128.33	192.50

At this point, some basic assumptions regarding the system were made. First, it was assumed that the foot-lift circuit had a flow of 15 gpm available. Second, it was assumed that, in cruise mode, three legs would be operated simultaneously; therefore each cylinder would have 5 gpm available to it. Third, flow rates could be increased by a factor of 2 by incorporating accumulators into the system. Based on these assumptions, each leg would be provided between 5 and 10 gpm to extend 18 inches in 0.05-0.10 seconds ($\dot{X}$ = 180 - 360 in/sec). As can be seen in Figure 9, these operational requirements are not within the capabilities of the system under consideration.

After further discussions with Prof. Waldron at O.S.U., the system requirements were revised to the following:

<u>Cruise Mode</u>

(1) Retraction - lift foot a maximum of 12 inches in 0.25 seconds.

(2) Extension - must be powered down to extend the foot 12 inches in approximately 0.25 seconds

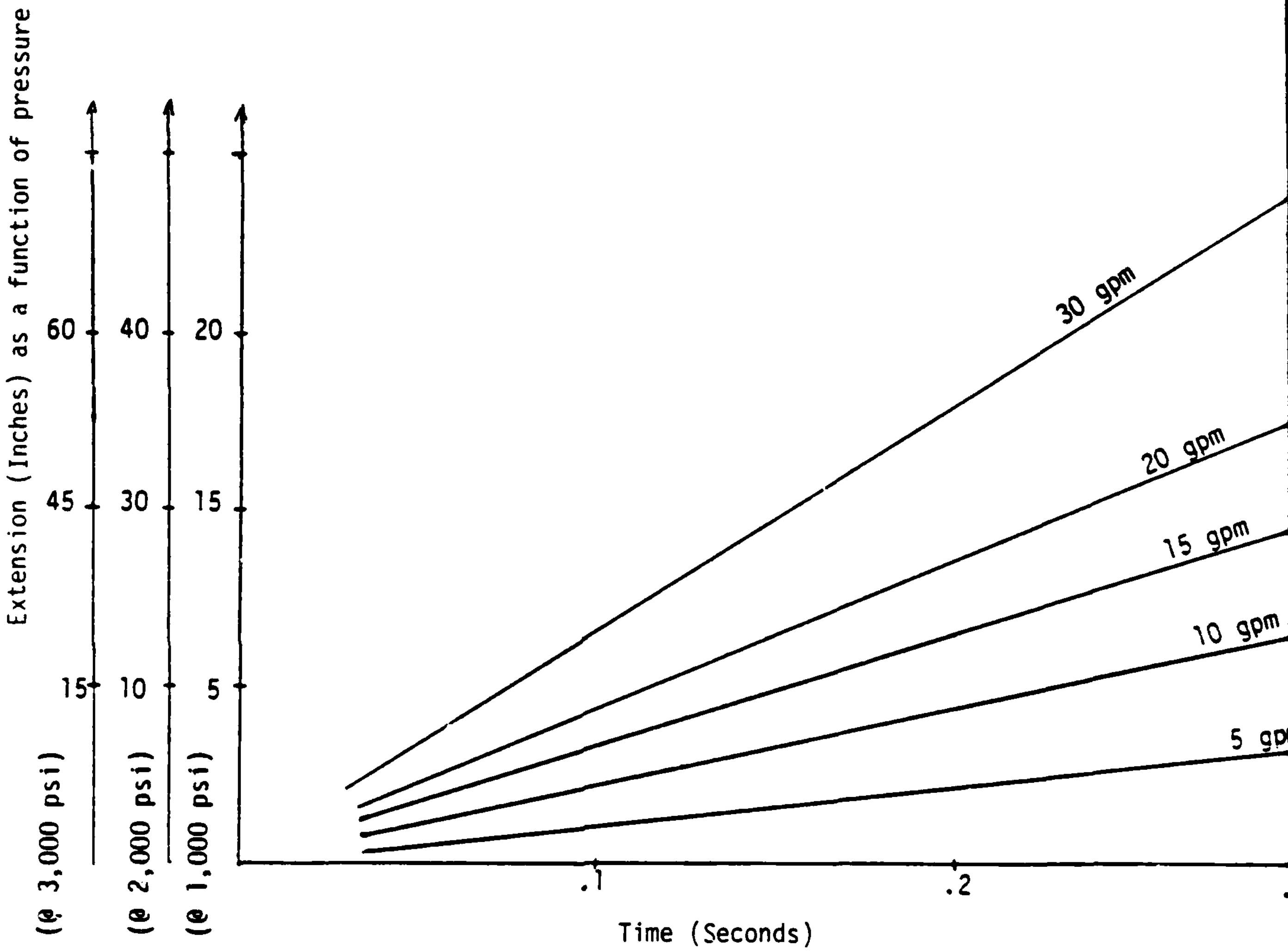


FIGURE 9. CYLINDER EXTENSION VS TIME

(3) Lock on - no extension of the foot under load for approximately 0.5 seconds.

All other requirements remained unchanged.

With 15 gpm available to the circuit, and in light of these revised requirements, the following conclusions can be drawn from Figure 9:

(1) A system pressure of 1000 psi is inadequate to meet the revised system requirements under the original leg configuration.

(2) A system pressure of 2,000 psi, with accumulators, might be marginally adequate to meet system requirements, within the accuracy of these assumptions.

(3) A system pressure of 3,000 psi, with accumulators, would be capable of meeting system requirements under the original leg configuration.

As a result of this analysis, it was agreed with Prof. Waldron at O.S.U. that the system pressure for the foot-lift circuit should be 3,000 psi.

A second analysis, based on the revised system requirements, was then developed to further evaluate this circuit design. This second approximation incorporated the inertial effects of the leg and the pressure losses due to the valve, hoses, and fittings. An iterative model was developed to calculate the maximum flow rate possible under given conditions.

First, according to Newton's law,

$$F = M \frac{d^2x}{dt^2}$$

where

F = force on piston rod

M = mass of lower leg

x = cylinder extension (ft)

t = time (sec).

$$\text{Area} \times (P_{pump} - \Delta P_{losses}) = M \frac{d^2x}{dt^2} \quad .$$

From the definition of the C_v factor,

$$Q = C_v \left(\frac{\Delta P}{S.G.}\right)^{1/2}$$

where Q' = flow (in^3/sec) and S. G. = specific gravity of fluid.

$$\Delta P_{losses} = \frac{S.G. \times Q'^2}{C_v^2}$$

$$\Delta P_{pump} - \frac{S.G. \times Q'^2}{C_v^2} = \frac{M}{Area} \frac{d^2x}{d^2t} .$$

$$\text{Also, } Q' (\text{in}^3/\text{sec}) = \text{Area} \times \frac{dx}{dt} .$$

$$\text{Therefore, } P_{pump} - \frac{S.G. \times Area^2 \times \left(\frac{dx}{dt}\right)^2}{C_v^2} = \frac{M}{Area} \frac{d^2x}{dt^2} .$$

$$\frac{d^2x}{dt^2} = a = \frac{A}{M} \quad P_{pump} - \left[\frac{S.G. \times Area^2 \times \left(\frac{dx}{dt}\right)^2}{C_v^2} \right] .$$

This equation can be solved iteratively for acceleration by substituting into the equation a value for the velocity $\frac{dx}{dt}$ that is either an assumed initial value or one previously calculated. Once the acceleration for a given time step has been calculated in this fashion, the velocity and the displacement x for that time step can be calculated:

$$v = v + at$$
$$x = x + vt.$$

This iteration is carried out until the values of acceleration approach zero, indicating that the system has reached steady state.

An assumed system, consisting of the following hydraulic components, was analysed:

(1) Moog 15 gpm servovalve (A076-104)

(2) Schroeder NF30 filter with N10 element

(3) Forty feet of 1/2-in schedule 80 pipe (convenient and conservative model for 1/2 in tubing)

(4) Four 1/2-inch elbows

(5) Ten feet of 1/2-in hydraulic hose

It was conservatively assumed that 15 gpm flowed through all components in calculating their valve coefficients. A C_v factor of 1.6 was calculated for the overall system (see Appendix B).

In these equations, a pump pressure was assumed and a piston area was calculated based on the pressure. Initial values of zero were assumed for both the velocity and displacement. Some preliminary calculations were carried out for pump pressures of 1500, 2500, and 3200 psi. The results of these calculations indicate that the system will reach steady state rapidly and, therefore, the inertial effects of the leg will not have a significant impact on the design of the hydraulic system.

A computerized analysis is shown in Tables 3 and 4. The input data, shown in Table 3, is for a pump pressure of 3,000 psi with a 2-inch cylinder bore and a leg weight of 50 pounds. This cylinder was oversized in response to buckling considerations. The results of this analysis, shown in Table 4, demonstrate that the cylinder will achieve steady state in less than 0.004 seconds. Since the total projected time-of-extension is 0.25 seconds, the cylinder will be accelerating for only 2 percent of its extension phase. Therefore, it would be reasonable to assume that the cylinder extends at a constant velocity.

The simulation results also indicate that the 2-inch diameter cylinder will extend at a rate of 19 inches/second at a pump pressure of 3,000 psi. Therefore, the cylinder would extend 5 inches in approximately 0.25 seconds. The flow rate corresponding to this extension can be easily calculated as approximately 60 in^3/sec, or 15.4 gpm.

This simulation was a preliminary approximation. It modeled the foot-lift mechanism as a vertical mechanism with a 50-pound lower leg (based on original O.S.U. estimates). The line losses were approximated by rough valve-coefficient calculations. However, it did serve to indicate that the acceleration of the foot-lift mechanism would likely play a relatively minor role in the mechanics of leg extension. It also demonstrated that, for the original linear extension model, the anticipated flow rates would be inadequate to meet the system flow requirements.

TABLE 3. INPUT DATA FOR PRELIMINARY FOOT-LIFT SIMULATION FOR 3000 PSI CIRCUIT

PRELIMINARY DARPA LEG SIMULATION

INPUT DATA

PRESSURE	SPECIFIC GRAVITY	CYLINDER DIAMETER	CYLINDER AREA
3000.0000	0.8500	2.0000	3.1416

VALVE COEFFICIENT	LEG WEIGHT	TIME INCREMENT	TOLERANCE
1.5930	50.0000	0.0001	1.0000

TABLE 4. RESULTS OF PRELIMINARY FOOT-LIFT SIMULATION FOR 3000 PSI CIRCUIT

TIME	ACCELERATION	VELOCITY	DISPLACEMENT
0.00010	28671.9316	2.8672	0.0003
0.00020	28012.8047	5.6685	0.0009
0.00030	26095.6914	8.2780	0.0017
0.00040	23177.6680	10.5958	0.0027
0.00050	19670.2715	12.5628	0.0040
0.00060	16017.8750	14.1646	0.0054
0.00070	12585.3232	15.4232	0.0070
0.00080	9599.7266	16.3831	0.0086
0.00090	7151.6445	17.0983	0.0103
0.00100	5231.8086	17.6215	0.0121
0.00110	3775.4038	17.9990	0.0139
0.00120	2697.1565	18.2687	0.0157
0.00130	1912.8574	18.4600	0.0175
0.00140	1349.5502	18.5950	0.0194
0.00150	948.6028	18.6898	0.0213
0.00160	665.0264	18.7563	0.0231
0.00170	465.3612	18.8029	0.0250
0.00180	325.2216	18.8354	0.0269
0.00190	227.0771	18.8581	0.0288
0.00200	158.4516	18.8739	0.0307
0.00210	110.5181	18.8850	0.0326
0.00220	77.0628	18.8927	0.0345
0.00230	53.7225	18.8981	0.0363
0.00240	37.4429	18.9018	0.0382
0.00250	26.0936	18.9044	0.0401
0.00260	18.1860	18.9062	0.0420
0.00270	12.6746	18.9075	0.0439
0.00280	8.8293	18.9084	0.0458
0.00290	6.1530	18.9090	0.0477
0.00300	4.2840	18.9094	0.0496
0.00310	2.9843	18.9097	0.0515
0.00320	2.0813	18.9099	0.0534
0.00330	1.4513	18.9101	0.0553
0.00340	1.0103	18.9102	0.0571
0.00350	0.7047	18.9103	0.0590

At this point in the program, Battelle was directed by O.S.U. to assume that a pantograph leg design had been selected for the ASV-84 vehicle. As shown in Figure 10, this design eliminates the need for a cylinder with a 48-inch stroke. Due to the nature of a pantograph mechanism, vertical motion of the cylinder rod results in a proportional vertical motion of the foot. The ratio selected for this mechanism was 4:1, which translates into an extension rate design requirement of 3 inches in 0.25 seconds. At the same time, due to revised estimates of the vehicle weight, the load-carrying capacity per leg was increased to 2,000 pounds, which, with the 4:1 ratio of the pantograph leg, resulted in a cylinder load carrying capacity of 8,000 pounds. The weight of the lower leg was estimated to be 150 pounds for the pantograph leg design.

In October, the design requirements were further revised by O.S.U. The foot extension rate for three legs was specified as 6 inches in 0.25 seconds for each foot. This translates to a cylinder extension rate of 1.5 inches in 0.25 seconds. It was further specified that the system be capable of lifting one foot 12 inches in 0.25 seconds while the other two feet were raised only 1 to 2 inches in the same time span. This results in a cylinder extension rate of 3 inches in 0.25 seconds for one cylinder and 0.25 to 0.50 inches in 0.25 seconds for the other two cylinders. The normal foot lift was assumed to be 1 to 1.5 inches.

In response to these revised requirements, the conceptual system was modified and reanalyzed using the preliminary leg simulation program. A 2-inch diameter cylinder was used in the modified system to support the 8,000 pound load. A weight of 150 pounds was assumed for the lower leg. The remaining input parameters were unchanged (Table 5). The results of this analysis, shown in Appendix C, indicate that the system takes 0.01 seconds to reach steady state (acceleration <1 ft/sec^2). However, it reaches 99% of its steady state velocity in 0.005 seconds. Therefore, it is still reasonable to assume a constant velocity.

This simulation also indicates that the cylinder will extend at a rate of 19 inches/second. Since the revised system design requirements provide for a 3-inch extension in 0.25 seconds, the modified system

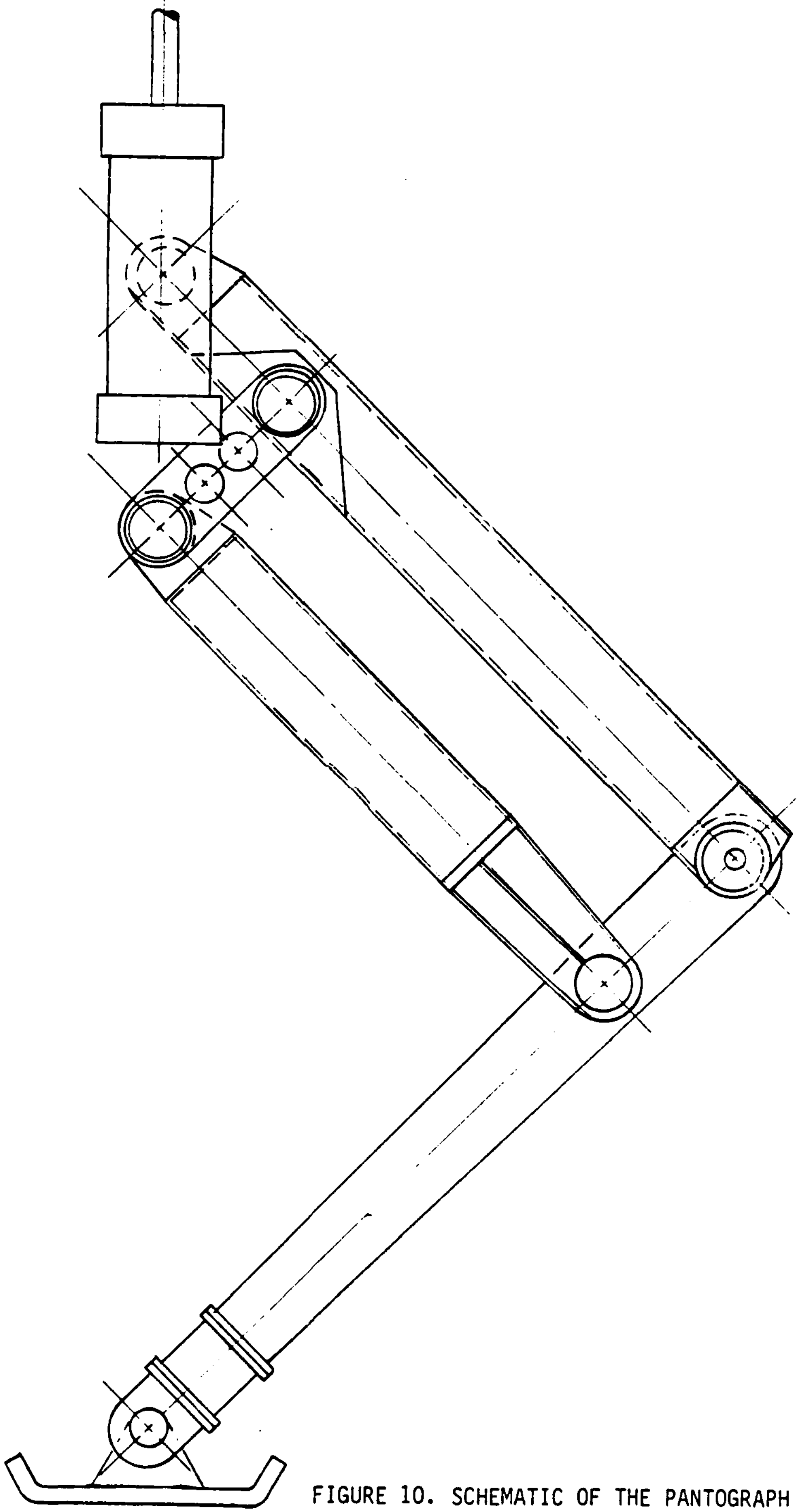


FIGURE 10. SCHEMATIC OF THE PANTOGRAPH LEG DESIGN

concept appears to be operationally acceptable. The system requires a flow rate of 15.3 gpm, which is within the projected pump capacity. Since the maximum total extension for three cylinders is 4.5 inches, this should not pose any significant problems.

3.1.3 Component Review

In conjunction with the preliminary analysis of the foot-lift circuit, manufacturers of several of the components were contacted to determine representative performance specifications. Information was initially requested for representative response times of servovalves and accumulators. According to Moog (the servovalve manufacturer), the 15 gpm Moog servovalve will either open fully or shut fully in 50 milliseconds. (A 5 gpm valve is available that requires even less time.) This represents 20% of the time available for leg extension (or retraction). However, it is anticipated that a reasonably fast operating time for the valve, such as 50 milliseconds, can be compensated for to a certain degree by computer control of the valve actuation.

Greer Hydraulics was also contacted for information regarding hydraulic accumulators. No information was readily available concerning the time required to discharge. Some limited testing may be required to obtain this information. Information was received on both the adiabatic and isothermal relationships between pressure and volume for a one gallon Greer accumulator that may be useful in future modeling efforts for the hydraulic circuit.

At the request of O.S.U., manufacturers of hydraulic pumps were also contacted. A Vickers PVB 10 hydraulic pump had been tentatively selected by the University of Wisconsin for the initial testing of the foot-lift circuit. (Specifications for this pump have been included in Appendix D.) The specifications were reviewed, and information on other, comparable pumps was accumulated. These pump characteristics are shown in Table 6. After reviewing the relative performance specifications for these pumps, it was concluded that while pumps were available

TABLE 6. PUMP CHARACTERISTICS

Pump Model	Pressure (psi)	Flow (gpm)	Speed (rpm)	Price (dollars)	Delivery (weeks)	Weight (lb)
Sunstrand 22-2059 Scott Equipment (Mr. Trent)	320	53	3200	2346	7-10	150
Denison PAV07-01C-041-3R-01-1A04 Hyd. & Air Controls (Ruth)	5000	44	2500	3570	1	110
Vickers AVB-10-FR-SY-CC (Pabco)	3000	18	3200	575	2	31
Rexroth A7V-28DR-2.0-RPF000 (Hydrotech - Harold Fraser)	5070	22	3000	1343	16	79
Rivett PV 2024-2569	2000	24	2400			56
Denison P46V-01-D-103-1R-03-1-04	5000	53	2500			

with significant improvements in performance over the Vickers, these pumps had substantially higher costs and/or weights relative to the Vickers. Since the Vickers appeared to meet the preliminary requirements for the foot-lift circuit, and since the selection of the final leg configuration had not yet been made, it was concluded that from the standpoint of the hydraulic design of the foot-lift circuit that the Vickers pump was a reasonable unit to purchase.

While reviewing the Vickers pump, it became apparent that a supercharge pressure would be necessary at the suction inlet of the pump. A simple design for the supercharge circuit, shown in Figure 11, was developed. The supercharge circuit involves joining the return oil from the foot-lift circuit and the return oil from the charge pumps for the hydrostatic transmission. By joining the circuits above a check valve, supercharge pressure can be supplied to the foot-lift circuit pump. However, care must be taken to make sure the cooler is large enough that the supercharge oil does not exceed the maximum temperature allowed for the pump.

3.1.4 Foot-Lift Circuit Concepts

Based on the revised system requirements, a number of concepts for the hydraulic design of the foot-lift circuit were generated and evaluated. These concepts, presented in Appendix E, were reviewed with personnel from O.S.U. A summary of major system characteristics is shown in Table 7.

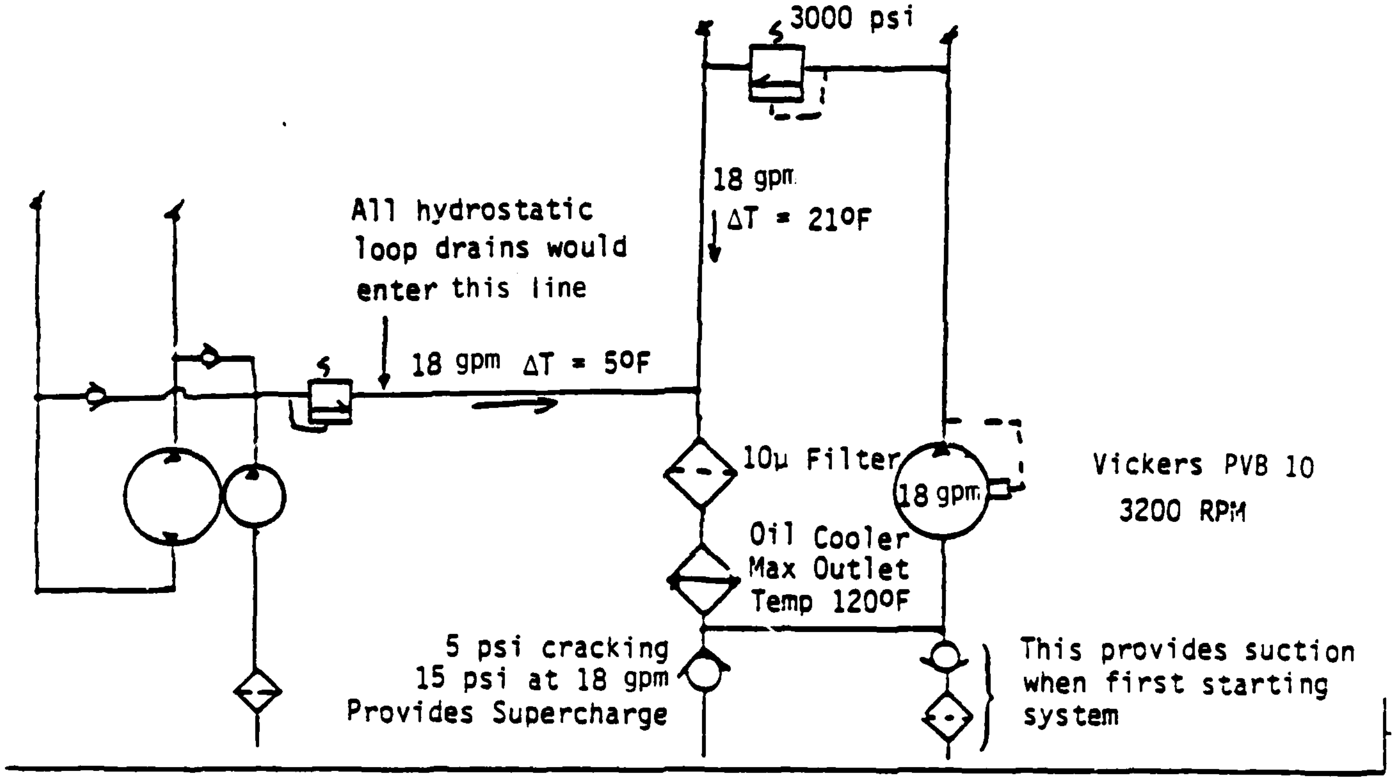


FIGURE 11. SUPERCHARGE CIRCUIT FOR VICKERS PVB 10 PUMP

TABLE 7. FOOT-LIFT CIRCUIT CONCEPTS

Concept	Average Flow	Estimated Horsepower
Rod End vs Weight		
Regeneration	5.0 gpm	3.66
No Regeneration	8.6 gpm	4.6
Piston End vs Weight		
Regeneration	3.1 gpm	1.0
No Regeneration	4.3 gpm	1.72
Double-Ended Cylinder	7.4 gpm	3.33

The system selected for the ASV-84 foot-lift circuit was one which supported the weight of the vehicle with the piston end of a single-acting cylinder. This minimized the size of the cylinder and the corresponding flow rate requirements. A regenerative circuit was not incorporated into the design. A schematic of the recommended foot-lift hydraulic circuit is shown in Figure 12. The associated system components are listed in Table 8.

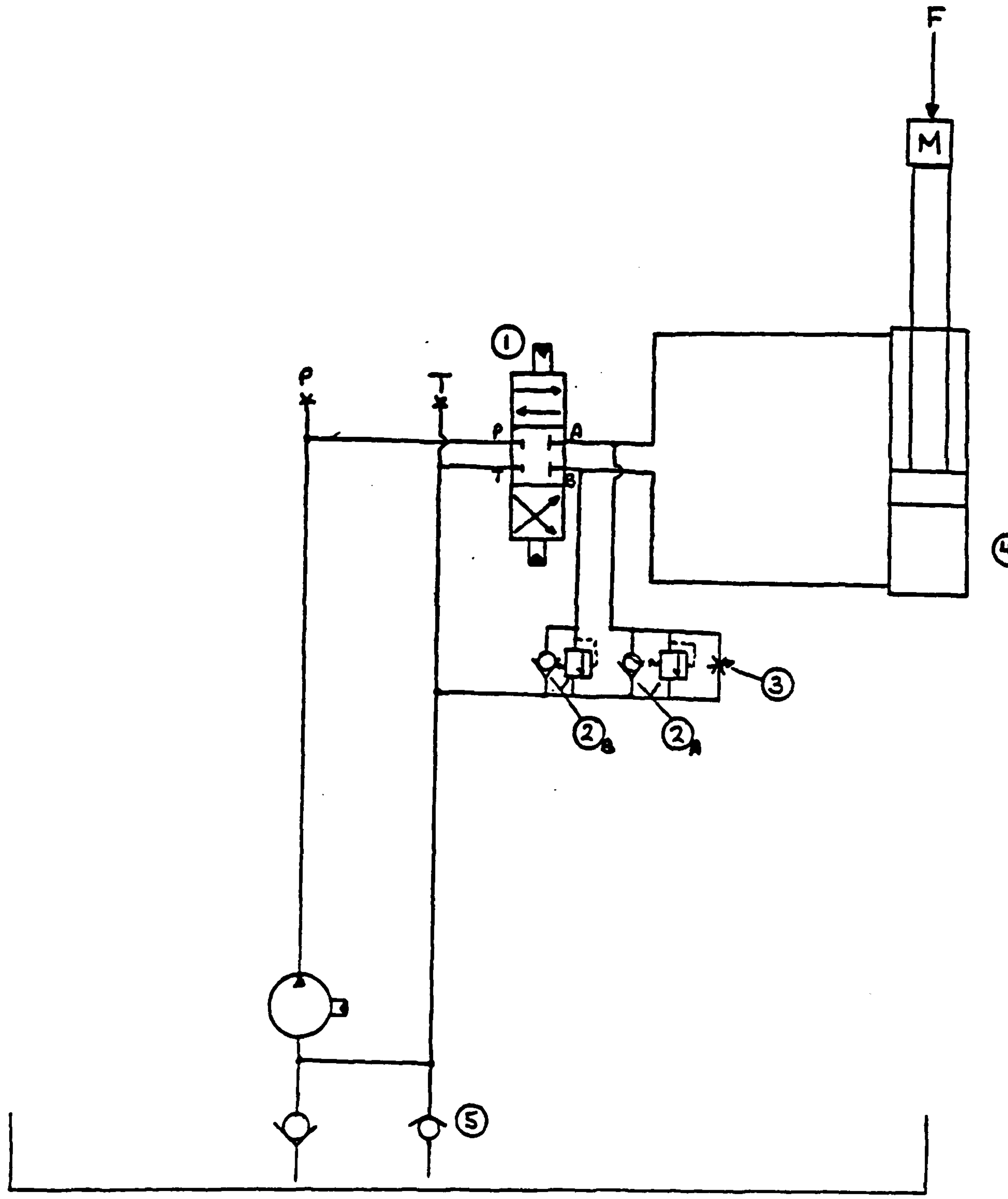


FIGURE 12. RECOMMENDED FOOT-LIFT HYDRAULIC CIRCUIT

TABLE 8. COMPONENTS FOR RECOMMENDED FOOT-LIFT CIRCUIT

Description
Moog servovalve 760-104
Sun cartridge relief valve
Bleed orifice
Milwaukee cylinder: 2-in bore, 1-in rod, cushions both ends, MF_2 mounting style, 13-in stroke
UCC check valve UC-CV-8P-04

3.1.5 Hydraulic Circuit Modeling

The emphasis on developing an analytical model of the foot-lift hydraulic circuit was de-emphasized to devote additional effort to the development of an energy-efficient leg design. A preliminary model of the original model was developed, however, which can be modified and updated for analysis of the final system. The model is modular in design, with components such as pumps, valves, and tubing included in the program. Since the earlier simulation efforts indicated that the acceleration of the cylinder rod was relatively insignificant, these effects were not included in this model. The entire circuit is modeled from pump to reservoir, with the flow rate modified at the cylinder to simulate a single-acting cylinder. All constants are defined and given values in the main program. The program user interactively provides pump pressure and rotational velocity in the input module.

A calculation module in the main program performs several unit conversions and calls the subroutines. Subroutine PUMP calculates the flow rate at the pump outlet. Subroutine TUBE calculates the pressure drop in the tubing. Subroutine VALVE calculates the pressure drop over the valve.

The results of a representative simulation of the original circuit are shown in Table 9. A description of the model is found in Appendix F.

3.2 Hardware Implementation

As part of the effort to design and evaluate the foot-lift circuit for the DARPA Adaptive-Suspension Vehicle, Battelle was to provide assistance to O.S.U. in the hardware implementation of this circuit. This task initially involved the assembly and demonstration of a preliminary circuit, using equipment supplied by O.S.U. Following the development of the original foot-lift circuit design, this circuit was assembled and demonstrated at Battelle. A preliminary experimental evaluation of this system was also carried out at this time. Finally, assistance was provided to O.S.U., on an as-needed basis, in installing and operating the foot-lift circuit on a prototype leg at O.S.U.

Originally, a small effort was projected to providing assistance to O.S.U. for a demonstration involving the operation of three legs. However, this demonstration did not take place during this program.

3.2.1 Preliminary Demonstration

O.S.U. supplied Battelle with a 15 gpm Hydura pump, with its associated servo control and electronic controller. This system was assembled, with a hydraulic reservoir and hydraulic motor, into a preliminary foot-lift circuit demonstration unit. The purpose of this demonstration was to verify the performance of the pump and controller and to illustrate the basic control concept. This system was demonstrated to Dr. Clinton Kelly from DARPA at the program review in May. At this time, the pump appeared to operate as expected with one exception: the make-up pump (a Viking gear pump) generated significantly more heat

TABLE 9. DARPA EFFICIENCY MODELING PROGRAM

```
FOR THE INPUTS:

   PUMP PRESSURE.........3000.0 PSI
   PUMP ROT. VELOCITY....3200.0 RPM

THE FOLLOWING DATA RESULTS:

   SUPPLY FLOW RATE......68.6 CU.IN./SEC  (17.8 GPM)

   TUBE NO. 1
      FLOW RATE..........68.6 CU.IN./SEC
      PRESSURE IN........3000.0 PSI
      PRESSURE OUT.......2982.9 PSI
      DELTA P............17.1 PSI  (0.1784 HP)

   TUBE NO. 2
      FLOW RATE..........22.9 CU.IN./SEC
      PRESSURE IN........2982.9 PSI
      PRESSURE OUT.......2981.5 PSI
      DELTA P............1.4 PSI  (0.0149 HP)

   VALVE, SUPPLY (CV=1.74)
      FLOW RATE..........22.9 CU.IN./SEC
      PRESSURE IN........2981.5 PSI
      PRESSURE OUT.......2832.2 PSI
      DELTA P............149.2 PSI  (1.5558 HP)

   RETURN FLOW RATE......17.1 CU.IN./SEC  (4.5 GPM)

   VALVE, RETURN (CV=1.74)
      FLOW RATE..........17.1 CU.IN./SEC
      PRESSURE IN........2832.2 PSI
      PRESSURE OUT.......2748.2 PSI
      DELTA P............84.0 PSI  (0.8754 HP)

   TUBE NO. 3
      FLOW RATE..........17.1 CU.IN./SEC
      PRESSURE IN........2748.2 PSI
      PRESURE OUT........2747.2 PSI
      DELTA P............1.1 PSI  (0.0113 HP)

   TUBE NO. 4
      FLOW RATE..........51.4 CU.IN./SEC
      PRESSURE IN........2747.2 PSI
      PRESSURE OUT.......2734.3 PSI
      DELTA P............12.8 PSI  (0.1338 HP)

   TOTAL HORSEPOWER LOSS 2.7695 HP
```

than was expected. It appeared to be attributable to a slight misalignment of the pump rotor with the housing: in effect, the rotor rubbed against the housing, generating heat.

In June, high pressure filters and a Moog servovalve were incorporated into the demonstration system for the original O.S.U. design, as shown in the following hydraulic circuit drawing (see Figure 13). The system was then operated with the Hydura pump stroked in only one direction so that flow was always supplied to the pressure port of the Moog servovalve. The servo was driven by a dc power supply at voltages varying between zero and 3.2 volts, with the servovalve coils hooked up in parallel.

As power was applied to the Moog servo, the hydraulic motor would start turning and increase in speed with increasing voltage. By reversing the voltage to the servo coils, the motor could be run in the opposite direction. The Moog servovalve performed well in these tests.

The gear pump on the Hydura pump continued to run warm as before. Unlike before, it took about 10 minutes to warm up to 140^{o}F, which was much slower than before. This suggests that the pump was probably still "milling" its pressure plate, but not as much as before. The return line filter showed considerable back pressure, indicating that a lot of dirt (possibly milling chips) has been collected.

At the same time, a demonstration model of a regenerative hydraulic circuit was assembled. A regenerative circuit is one in which the rod end of the cylinder is connected to the blind end in a way that allows rapid extension of the piston. This may have some application to the foot-lift circuit, particularly in the case of an emergency extension.

3.2.2 Foot-Lift Circuit Demonstration

The hydraulic foot-lift circuit was simulated so the basic responses of the hydraulic system could be checked. The primary concern

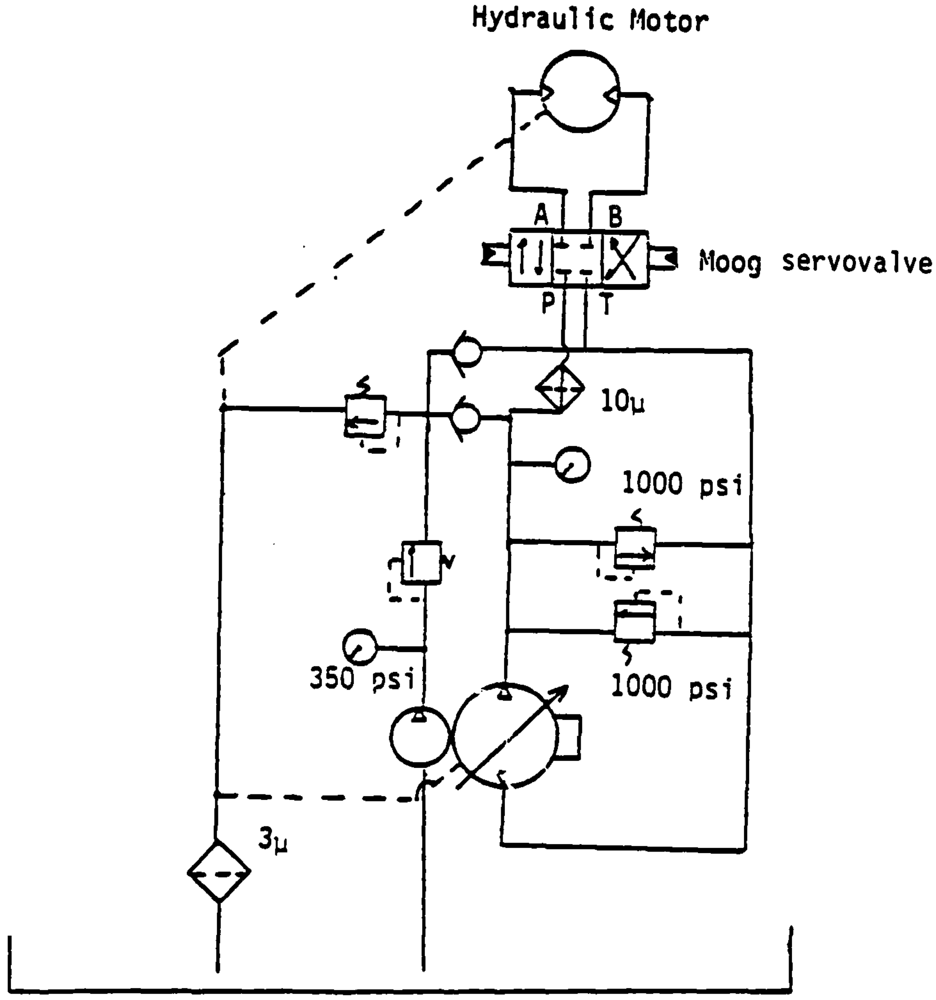


FIGURE 13. HYDRAULIC CIRCUIT OF DEMONSTRATION UNIT

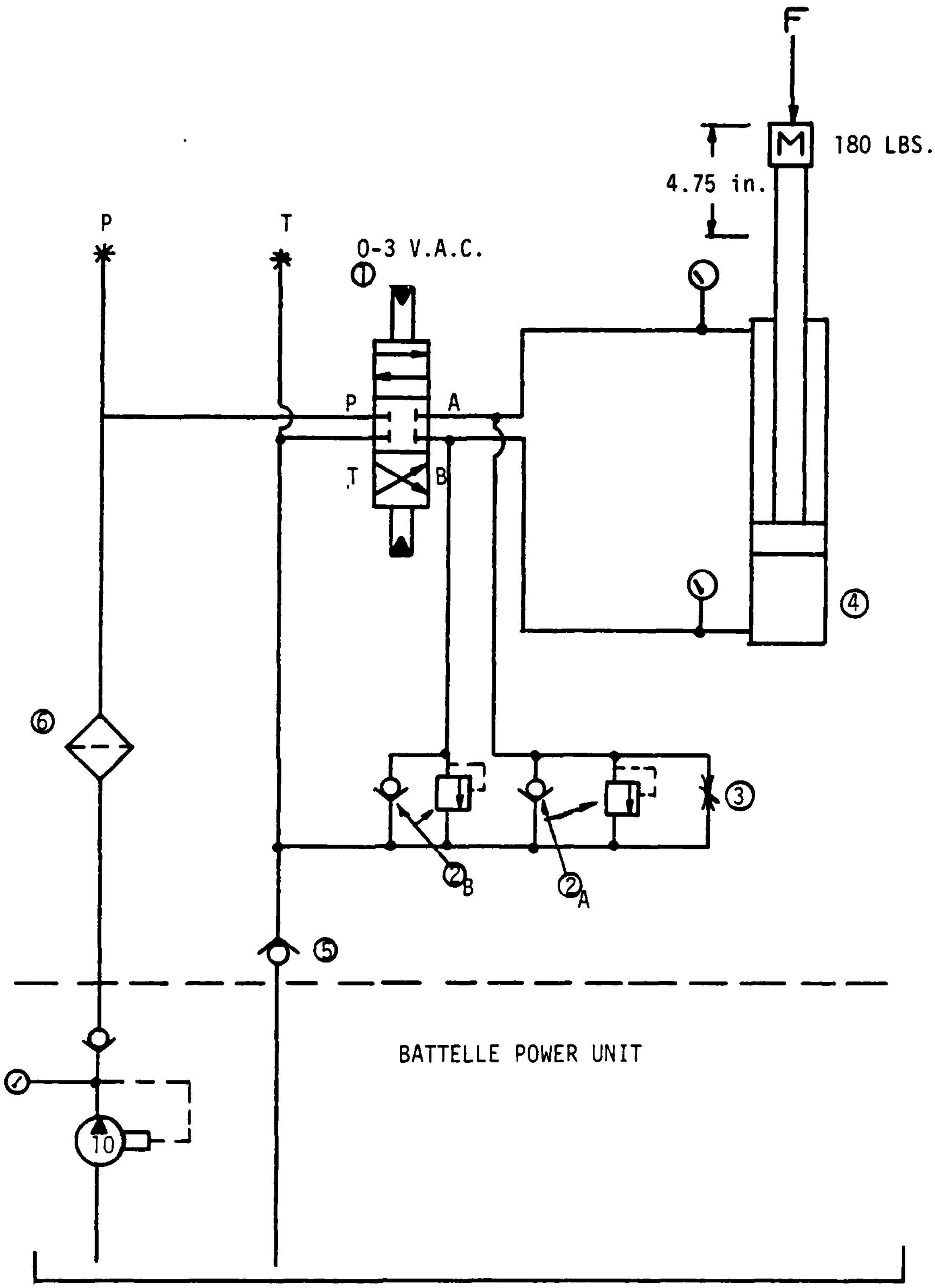
was whether or not the foot-lift circuit would respond quickly enough to meet specifications. In the simulated-circuit shown in Figure 14, a weight of 180 lb to simulate the foot was placed on top of a cylinder (2-in bore, 1-in rod, 13-in stroke) and attached to the cylinder rod. To reduce friction as much as possible, the weight was guided along the entire stroke length and the fixture was lubricated. Since the cylinder was cushioned at the ends of the stroke, the cycling portion of the test was run outside the cushioned portion of the stroke.

The circuit was cycled by means of limit switches and relays. At the top end of the desired stroke, the top limit switch tripped a relay, which reversed the voltage to the Moog servovalve and moved the cylinder in the opposite direction. The same sequence occurred at the other end of the stroke; thus, the cylinder was continually cycled up and down until the cycling was stopped by shutting off the power to the servovalve. The servovalve was driven by a voltage between 0 and 3 Vdc. During cycling, the voltage from the dc power supply was not varied.

The hydraulic power supply included a Vickers PVB-10 pump that delivered a nominal 10 gpm at 0 psi. The pressure compensator on this pump was set at 900 psi during all runs. At this pressure, the output from the pump was approximately 9.6 gpm.

The displacement of the weight (foot) was measured using a variable resistor attached to a rotary wheel by means of a wire and a spring. The spring was stiff enough to prevent the wire from going slack during haul-in (down stroke). The variable resistor, in series with a 100 ohm resistor, was powered by a 6-volt battery. The voltage output over the 100 ohm resistor was recorded on a strip chart, which allowed the time taken for each cycle to be determined.

The total stroke distance was also measured. The distance from limit switch to limit switch trip point was 4.5 in; but during cycling the measured stroke was 4.75 in, indicating an overshoot at either end of the stroke. The overshoot is caused by the delay times of the relay and the servovalve. The measurements made during operation of the simulated circuit are tabulated below:



RECOMMENDED FOOT-LIFT HYDRAULIC CIRCUIT

FIGURE 14. CIRCUIT SCHEMATIC OF BATTELLE DEMONSTRATION UNIT

Stroke Distance (in)	Cycle Time (sec)	Valve Voltage (Vdc)	Compensated Cycle Time for 3-in Stroke (sec)
4.75	1.12	1	0.71
4.75	0.90	3	0.57

As can be seen from the tabulation above, the valve voltage (valve shift) does affect the quickness of the overall system. This effect can be explained as follows: At full valve voltage (3 Vdc), 900 psi is enough pump pressure to drive the system without compensating the pump; at the lower valve voltage, however, the pressure drop at the valve is large enough to shift the pump to a lower displacement, causing the system to respond more slowly.

The response, theoretically, should have been

$$t_{cycle} = \frac{A_c l + A_r l}{Q}$$

where t_{cycle} = cycle time

A_c = area of the cap end of the cylinder (in^2)

A_r = area of the rod end of the cylinder (in^2)

l = compensation length (in)

Q = pump flow (gpm)

Thus, from the experimental run with the pump putting out 9.6 gpm, the response time should have been:

$$t_{cycle} = \frac{(3.14\ in^2)\ 3\ in + (2.36\ in^2)\ 3\ in}{231\ in^3/gal} \times \frac{60\ sec/min}{9.6\ gpm}$$

$$t_{cycle} = 0.45\ sec.$$

The actual cycle time of 0.57 sec (after compensation for 3-in travel) resulted from the fact that all of the 9.6 gpm did not reach the cylinder. At 900 psi, the internal leakage of the Moog 760-104 servovalve is 0.35 gpm. In addition, there was leakage across the bleed orifice at port A of the servovalve during one-half of the cycle. The gauge pressure measured at this point during cycling was 750 psi on a 1/16-in diameter orifice. The flow loss for this orifice is:

$$Q = \sqrt{P}\ (A \times 23.5) = 1.98 \text{ gpm.}$$

For the overall cycle, $Q = \frac{1.98}{2}$ gpm = 0.99 gpm $\approx$ 1.0 gpm. Therefore, the actual flow to the cylinder would be

$$9.6 \text{ gpm} - (1.0 \text{ gpm} + 0.35 \text{ gpm}) = 8.25 \text{ gpm.}$$

With this flow, the modified theoretical cycle time would be 0.52 sec. The difference of 0.05 sec between the theoretical cycle time of 0.52 sec and the actual cycle time of 0.57 sec can be accounted for by leakage at the cylinder piston, acceleration and deceleration of the weight, and lower pump output.

3.2.3 Computer Controlled Preliminary Foot-Lift Circuit Demonstration.

After the foot-lift circuits had been demonstrated under simple relay control at Battelle, the equipment was delivered to O.S.U. for incorporation into the single-leg test fixture under computer control. The power supply used in the demonstration at Battelle was also loaned to O.S.U. to facilitate the early stages of testing. The actual installation of the equipment on the test rig was carried out by personnel at O.S.U.; no assistance by Battelle personnel was required. After this assembly was completed, the operation of the leg was observed. At the time of the completion of this contract, no major experimental evaluations of the foot-lift circuit under computer control had been conducted. No major modifications of the system design had been undertaken.

3.3 Energy-Efficient Designs

The initial system design requirements for the ASV foot-lift circuit were generated by O.S.U. in May, 1982. These requirements specified that, in cruise mode, there would be no extension of the foot under load. (The energy efficiency of the circuit is of major importance in cruise mode.) A preliminary design for the foot-lift circuit was generated by Battelle based on these initial requirements. By proper sizing of the system's components, it was apparent that the pump pressure required to extend the foot under no load was relatively low, minimizing system horsepower requirements. However, after further consideration, it was decided by O.S.U. that this isolation of the individual foot-lift cylinders under load in cruise mode was impractical. Therefore, in November, the design requirements for the foot-lift circuit were revised, eliminating the isolation of the cylinders under load.

The practical effect of this modification was to require system operating pressure to remain at 3000 psi at all times. Since the original design for the foot-lift circuit involved relatively high flow rates, a high system operating pressure resulted in a significant increase in the required horsepower for the circuit. At the same time, a goal of 7 horsepower (hydraulic) was established by O.S.U. for the foot-lift circuit. Under the conditions, the original design for the foot-lift circuit was inadequate.

Since this revision occurred late in the program, it was agreed that Battelle would assist O.S.U. in developing new, energy-efficient designs for the foot-lift circuit. A variety of concepts have been generated and jointly evaluated with O.S.U. As a result of these efforts, four circuit designs were recommended for further evaluation for solving the efficiency problem: a hydrostatic transmission, in which each leg has a separate hydrostatic pump; a dual-pump system, which incorporates a high-volume, low-pressure pump and a low-volume, high-pressure pump; a hydraulic brake unit, in which the original circuit is modified by using a small-bore, short-stroke cylinder in series with the present cylinder; and a dual-cylinder circuit, which incorporates a small bore cylinder and a large bore cylinder, both using low-volume, high-pressure fluid.

3.3.1 Hydrostatic Transmission Circuit

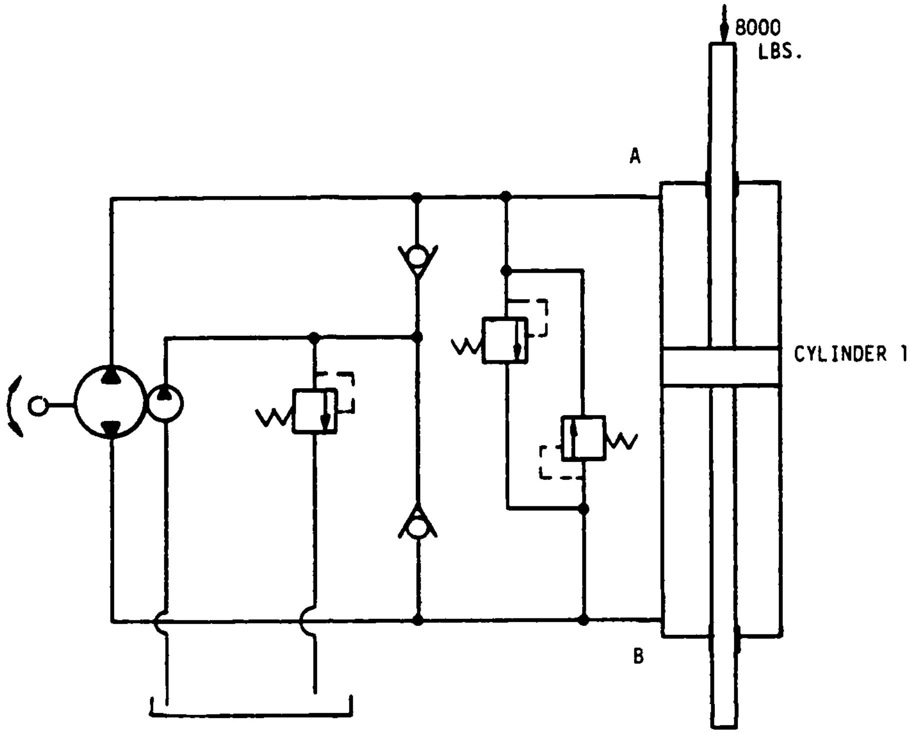
With the hydrostatic transmission circuit shown in Figure 15, each leg has a hydrostatic pump, and the flow from the pump can be varied from zero to maximum in both directions. Each pump has a charge pump for replenishment, with a capacity approximately one-fourth that of the main loop flow. The time required by the pump to respond to the computer signal is unknown, but it must be kept short, compared with 500 milliseconds, or the flow rate of the overall system would have to be higher.

The advantage of this design is the absence of a directional valve, which would produce a large pressure drop. Additionally, using separate pumps allows each leg to draw its own required pressure, thereby reducing the horsepower losses that result from sharing a common pressure source.

The horsepower needed to lift one foot is 1.07 hp, and to place three feet, 3.4 hp; the total horsepower required is 6.7 hp. Horsepower calculations are in Appendix G.

3.3.2 Dual-Pump Circuit

The dual-pump circuit shown in Figure 16 is commonly known as a high-low hydraulic curcuit. A high-volume pump is used for the actual foot-lift operation at low pressure. When the foot is on the ground, a low-volume, high-pressure pump provides the fluid to move the foot under load. An additional servovalve per leg is needed to take care of the high-pressure system.



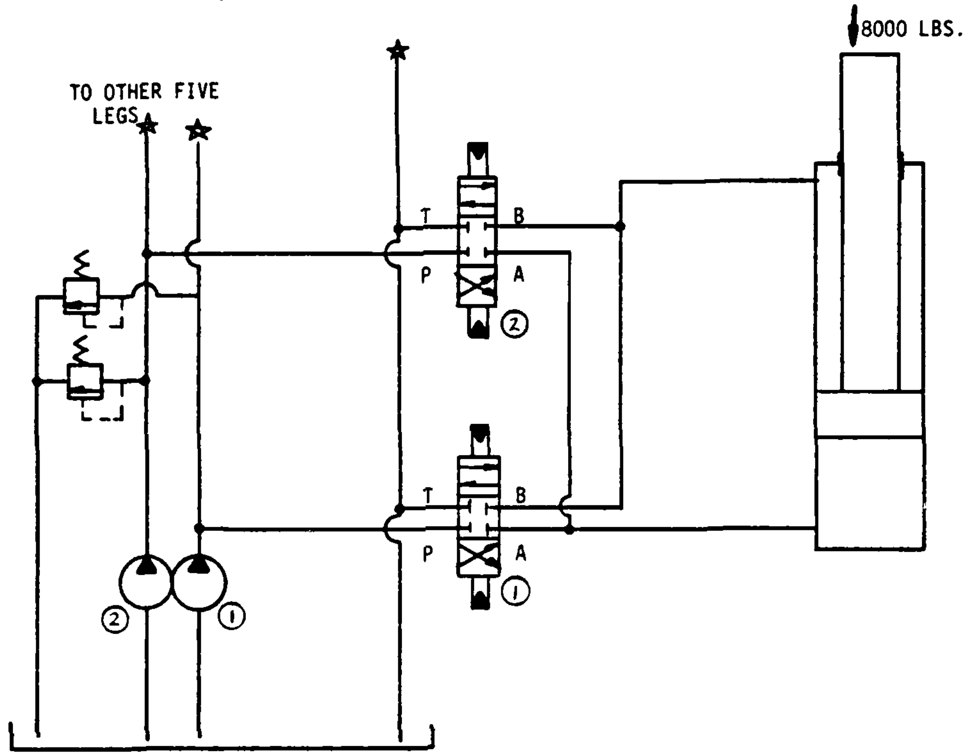
Holding Position: The hydraulic pump is shifted to side B of cylinder 1, allowing the foot to be moved under load. The amount of pump shift will be low.

Foot Lift: Shift pump output to side A of cylinder 1 so large flow is available to raise foot.

Foot Lower: Shift pump to side B of cylinder 1 to drive down foot quickly.

Hold and Adjust: When the foot hits the ground, shift the pump back to a low displacement to allow for foot movement under load.

FIGURE 15. HYDROSTATIC TRANSMISSION CIRCUIT



Holding Position:	Valve 1 is shifted to the B side, providing high-pressure, low-volume flow from pump to adjust the foot under load.
Foot Lift:	Valve 1 closes and valve 2 opens to A side, porting oil to cylinder rod end and driving foot up. Flow through valve 2 from pump 2 is low-pressure and high-volume.
Foot Lower:	Shift valve 2 to the B side to drive the foot down.
Hold and Adjust:	When foot reaches the ground, close valve 2 and shift valve 1 to the B side so oil from pump 1 can be used to adjust under load.

FIGURE 16. DUAL-PUMP CIRCUIT

The horsepower needed to lift one foot is 2.3 hp; to place three feet, 5.1 hp. A total of 12 hp is needed. The horsepower calculations are in Appendix G.

3.3.3 Hydraulic Brake Circuit

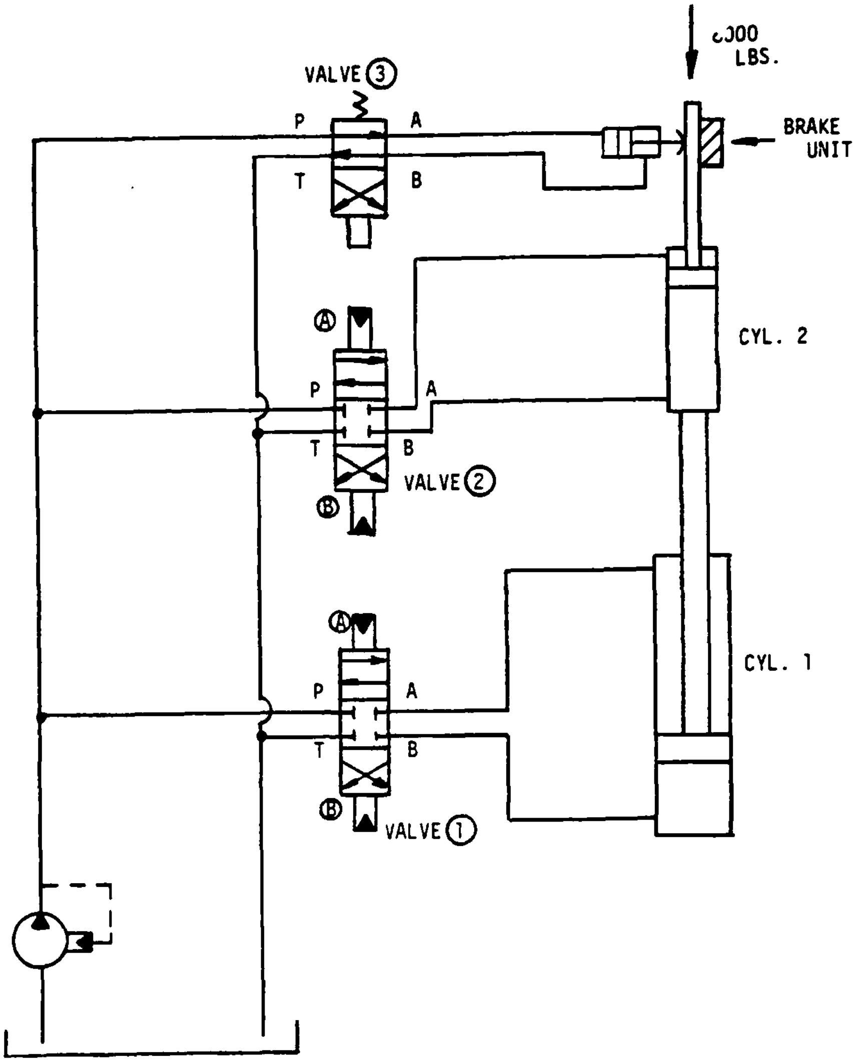
The hydraulic brake circuit shown in Figure 17 is a modification of the existing circuit. A small-bore, short-stroke cylinder is used in series with the present cylinder. The advantage of this concept is that the small-bore cylinder can use low flow at high pressure to produce quick movement during the lift phase. The large cylinder can be used to hold the load when the foot is on the ground. Because the small cylinder must be in the extended position when the foot is on the ground, a brake must be applied to remove the load from the hydraulics of the small cylinder, otherwise the small-bore cylinder would develop very high pressures.

The horsepower needed to lift one foot is 2.2 hp; to place three feet, 5.2 hp. A total of 11.4 hp is needed. The horsepower calculations are in Appendix G.

3.3.4 Dual-Cylinder Circuit

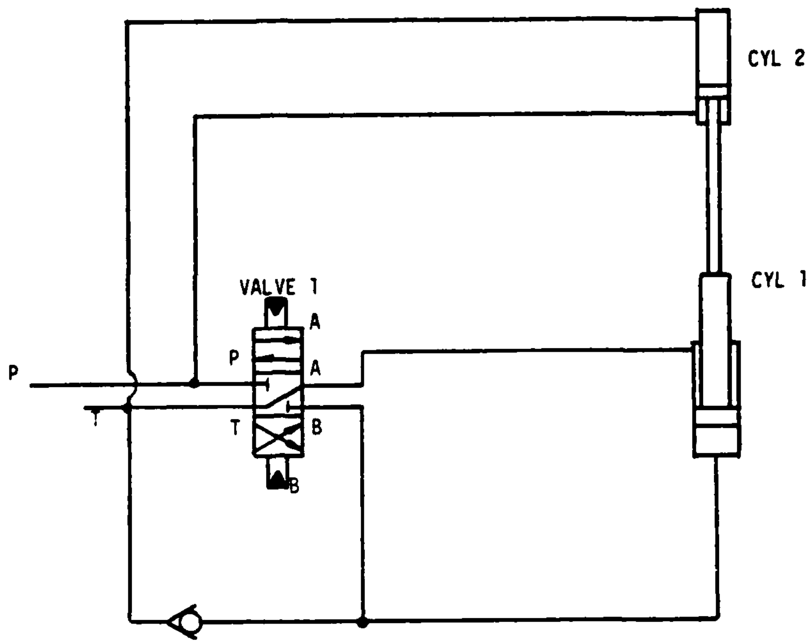
Dual-cylinder circuits use high-pressure, low-volume fluid on a small-bore cylinder to produce quick movement. The load is held by a large-bore cylinder; low flow at high pressure is used to make the adjustments under load. The three basic configurations of the dual-cylinder circuit shown in Figures 18 through 20 were selected as the most promising at the DARPA weekly review meeting on February 10, 1983. The horsepower requirements of these circuits are calculated in Appendix G and are summarized along with other data in Table 10.

The storage cylinder concept was selected for further study. During this study, several practical problems were dealt with, including the need to compensate for the fact that when the foot sinks into the



Holding Position:	While foot is down, the brake is applied and the load is transferred to cylinder 1 so cylinder sees no load. Valve 1 is shifted to the B side so that cylinder can be moved under load.
Foot Lift:	Valve 1 is shut and the brake unit is released by energizing valve 3. Valve 2 is shifted to the A side, driving the oil to the rod end of cylinder 2 and lifting the foot.
Foot Lower:	Valve 2 is shifted to the B position and cylinder 2 is extended, driving the foot down.
Hold and Adjust:	When the foot is fully down, deenergize valve 3 to apply the brake. Shift valve 1 to the B position to adjust under load.

FIGURE 17. HYDRAULIC BRAKE CIRCUIT



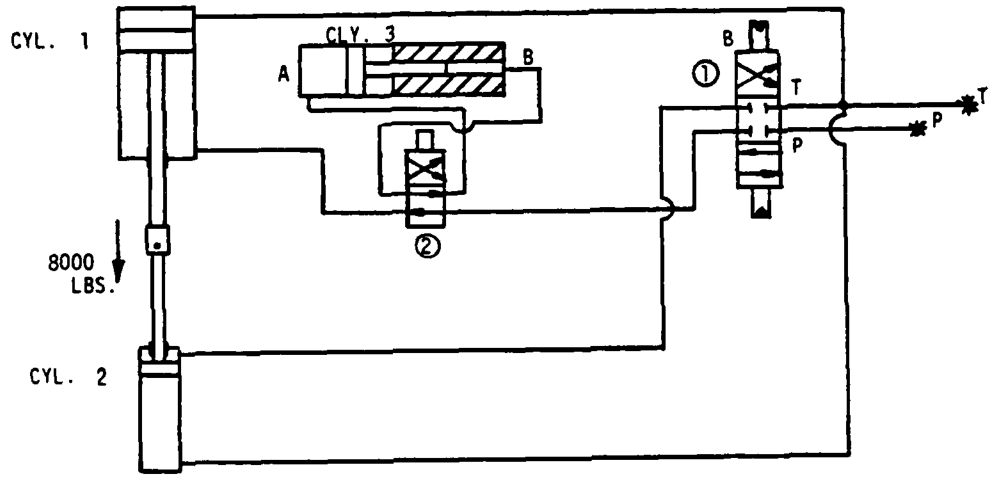
Holding Position: Valve 1 is in position B; pressure can be applied to the cap end of cylinder 1.

Lift Leg: Shift valve 1 to side A. Larger area (A_R) of cylinder 1 pulls against cylinder 2. Oil from cylinder 2 enters valve 1, making system go faster. Oil from end of cylinder 1 is forced out of system at valve 1.

Lower Leg: Center valve 1 so that pressure goes to A_B of cylinder 2 only, and A_R of cylinder 1 is exposed to tank. Cylinder 2 pulls leg down, and A_C of cylinder 1 is filled through a large check valve with low cracking pressure.

Hold and Adjust: Shift valve 1 to B position.

FIGURE 18. DUAL-CYLINDER CIRCUIT (PUSH-PULL COMPRESSION)



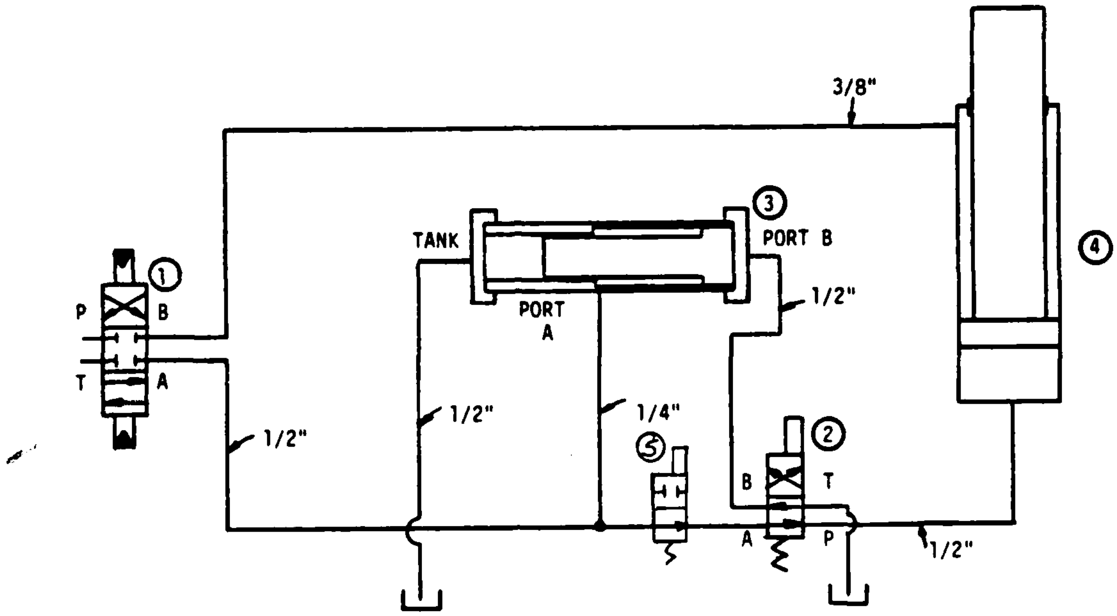
Holding Position: Valve 2 deenergized and valve 1 in position A.

Lift Leg: Shift valve 1 to position B and valve 2 to energized position. Pressure to rod end of cylinder 2 forces oil into A port of cylinder 3. Cap end of cylinder 1 fills from supercharge system of pump as cap end of cylinder 2 empties into supercharge.

Lower Leg: Shift valve 1 to A side and high pressure will be put to B part of cylinder 3, forcing oil of high volume and low pressure out of A port of cylinder 3 into rod end of cylinder 1. Cylinder 2 is exposed to tank (supercharge) pressure on both sides and is free to float.

Hold and Adjust: Shift valve 2 back to deenergized state to adjust under load.

FIGURE 19. DUAL-CYLINDER CIRCUIT (PUSH-PULL TENSION)



Holding Position:	Valve 1 is shifted to the B side and valve 2 is deenergized, allowing cylinder 4 to move the foot under load. Cylinder 3 is fully retracted.
Foot Lift:	Shift valve 1 to the A side and energize valves 2 and 5, porting oil out of the cap end of cylinder 4 into the cap end of cylinder 3. Oil from the rod end of cylinder 3 exits through valve 1.
Foot Lower:	Valve 1 is shifted to the B side, sending oil to the rod end of cylinder 3, which forces oil out of the cap end of cylinder 3 into the cap end of cylinder 4.
Hold and Adjust:	Valves 2 and 5 are deenergized; flow is available to adjust cylinder 4 under load.

FIGURE 20. DUAL-CYLINDER CIRCUIT (STORAGE CYLINDER)

TABLE 10. COMPARISON OF FOOT-LIFT CIRCUIT CHARACTERISTICS

Circuit Type	Pump Flow/Leg (gpm)	Pump Pressure (psi) Lift	Pump Pressure (psi) Hold	Engine* Horsepower (hp)	Complexity	Component Physical Compatability
Hydrostatic Transmission	8.4 (Loop) 2.0 (Charge)	155	2960	6.7	Low	Low
Dual-Pump	8.6 (Low Pressure) 1.0 (High Pressure)	400	2550	12.0	Medium	High
Hydraulic Brake	1.10	3000	3000	11.4	High	Low
Dual-Cylinder						
Storage Cylinder	2.5	2550	2550	12.6	Medium	High
Push-Pull Compression	0.93	2800	2800	10.5	Low	Low
Push-Pull Tension	1.83	1160	2960	9.3	High	Low

* Placement horsepower is approximately 5.1 for all concepts (except hydrostatic = 3.4), and is made on the assumption of 1-in foot movement in 200 msec at 8000 lb.

soil, it has to be raised more than it was lowered by the cylinder stroke. The circuit shown in Figure 20 was designed to eliminate this problem. In addition, components were selected for this particular design. Data on these components are listed in Table 11.

TABLE 11. COMPONENTS FOR THE STORAGE CYLINDER CONCEPT

Item	Description	Weight (lb)	Price ($)
Valve 1	Moog 760-104	2.3	
	Subplate 43586-1	2.4	
Valve 2	Denison A3D01-35-11-01-01-00A5-01	3.7	121.00
	Subplate	2.8	37.00
Cylinder 3	Special	10.00	
Cylinder 4	Special	22.5	
Hose	3/8-in valve 1 to cyl 4 rod (10 ft)	3.0	
	1/2-in valve 2 to cyl 4 cap (10 ft)	4.0	
	1/2-in valve 1 to valve 1 (1 ft)	0.4	
	1/4-in valve 2 to port A cyl 3 (1 in)	0.25	
	1/2-in cyl 3 tank to tank (1 ft)	0.4	
	1/2-in valve 2 to tank (1 ft)	0.4	
Fittings	Miscellaneous	2.0	
	Total	54.15	

3.3.5 Heat Dissipation

During operation of the hydraulic systems, a large amount of energy is wasted by conversion into heat. Although some heat is dissipated naturally throughout the vehicle by means of the hydraulic tank, lines, valves, and cylinders, a heat exchanger is needed to prevent the hydraulic oil temperature from increasing to a point where the system may be damaged. Because the vehicle is mobile, an oil-to-water heat exchanger is not practical; rather, an air-to-oil heat exchanger must be used.

Heat dissipation is a function of the difference between the temperatures of the fluid and the ambient air: the lower the ambient temperature and the higher the maximum fluid temperature, the better is the heat dissipation.

Heat dissipation is also a function of the air velocity through the oil cooler. Two general types of air movement are used for air-to-oil heat transfer. The first is the natural draft provided by vehicle and wind movement; the second is a forced draft provided by a driven blower. The first type produces a large variation in dissipation capability, depending on relative wind movements. The second produces a constant dissipation but requires some horsepower to drive the blowers.

For the vehicle, the heat to be dissipated is:

- Cruise Mode: 30 hp
- Dash Mode: 50 hp

The heat exchangers are to be placed in the hydraulic fluid loops of the drive circuit. Each drive loop was considered to have a flow of 10 gpm and a minimum pressure of 100 psi. The heat dissipated naturally by hydraulic components and by four air-to-oil heat exchangers was calculated by the methods illustrated in Appendix H. Selected heat dissipation values are listed in Table 12.

TABLE 12. HEAT DISSIPATION IN HYDRAULIC SYSTEM

Dissipation Means	Heat Dissipation per unit (hp) Cruise	Dash	Unit Weight (lb)	Units Required	Engine Horsepower Needed
Tank Dissipation	1.9	1.9		1	0
Natural Draft (5 mph)					
M-10	2.5	4.3	6	12	0
M-15	3.4	5.8	8	9	0
M-20	4.2	7.1	11	7	0
Forced Draft M-200	5.0	8.0	10	6	3

For natural drafts less than 5 mph, dissipation is lower; for drafts greater than 5 mph, dissipation is greater.

3.4 Results and Conclusions

The results of the actuation portion of the program are summarized below.

- o A simple hydraulic design for the foot-lift circuit was generated in response to an original set of design requirements.
- o A model was developed for analysis of this circuit design that can be easily modified to incorporate different components.
- o A series of energy efficient concepts for the foot-lift circuit were generated and analyzed in response to the revised design criteria.
- o Technical assistance was provided to O.S.U. in the selection and analysis of components for the foot-lift circuit.

Although the preliminary system design does not fulfill the revised design requirements, it does represent a simple, reliable design that provides the required operational capability. If the operational scenario for the ASV-84 vehicle is designed to minimize or eliminate the leg extension under load, this design represents an alternative that would be relatively simple to implement.

The system model developed as part of this effort can be easily modified for the analysis of alternative configurations. Application to the energy-efficient concepts would be relatively straightforward. The model can also be easily upgraded as performance data for the foot-lift circuit becomes available.

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APPENDIX A

FLOW CHART CONVENTIONS

The purpose of this appendix is to describe the conventions used in the flow charts (Figures 1, 2, and 3) in this report, where those conventions differ from those commonly used in flow charts.

The charts consist basically of vertical sequences of blocks, with control passing sequentially down through the sequences. However, interspersed in those sequences are decision (hexagonal) blocks, which require the execution of subsequences of blocks located to the right of the decision blocks. When the bottom of a vertical sequence of blocks is reached, control returns to the decision block from which that sequence began.

As to the decision blocks themselves, when the condition in the block is described by a FOR, WHILE, or UNTIL phrase, the subsequence to the right is executed, respectively, FOR the conditions stated, WHILE the condition stated is true, or UNTIL the condition stated is true. If the decision block contains an IF phrase, the subsequence indicated by the arrow exiting from the upper right of the block is executed IF the condition is true. If there is a subsequence indicated by an arrow exiting from the lower right of the block, that subsequence is executed IF the condition is false.

Finally, rectangular blocks bounded by thick lines indicate sequences described by other flow charts.

APPENDIX B

CALCULATION OF SYSTEM C_V

Servovalve

For the 15 gpm Moog servovalve (A076-104), vendor literature indicates a pressure drop of 1000 psi across the valve at a no-load flow of 15 gpm. Therefore,

$$15 \text{ gpm} \times \frac{236 \frac{in^3}{gas}}{60 \frac{sec}{min}} = C_V \sqrt{\frac{P}{S.G.}}$$

S.G. for hydraulic fluid = 0.865

$$15 \times \frac{236}{60} \sqrt{\frac{0.865}{1000}} = C_V = 1.74$$

For 1/2 inch Hose

According to Figure A-2 of Fluid Power Designers Lightning Reference Handbook, ΔP = 3.15 per foot for flows of 15.3 gpm for 1/2 inch hose.

Therefore,

$$C_V^2 = 15.3 \text{ gpm} \times \frac{236 \text{ in}^3/\text{gal}}{60 \text{ sec/min}} \qquad \sqrt{\frac{S.G.}{\Delta P}}$$

S.G. = 0.865 P = 3.15/ft

C_V = 31.54/1 ft

Since P = 31.5 psi for 10 feet of hose,

C_V = 9.97 for 10 feet of hose.

Filter (e.g., Schroeder NF30 with N10 Elements)

From Figure A-3, P = 4.5 @ 15 gpm

Therefore,

$$C_v = 15 \text{ gpm} \times \frac{236}{60} \sqrt{\frac{.87}{4.5} \text{ psi}} = 25.94$$

For 1/2 inch Pipe

For 1/2-in pipe (schedule 80 roughly the same I.D. as 1/2-in tubing) and using Figure A-2 again,

$\Delta P = 2.47$ psi @ 14.6 gpm

Therefore,

$$C_v = 14.6 \text{ gpm} \times \frac{236}{60} \sqrt{\frac{S.G.}{\Delta P}}$$

$= 33.98$ for 1 ft.

For 40 ft, $C_v = 5.37$.

Elbows

For 4 1/2-in elbows (schedule 80), each elbow is equivalent to 2.9 feet of 1/2-in pipe (schedule 80). From Figure A-2, $\Delta P = 2.47$ psi/ft at 14.6 gpm.

$$C_v = 14.6 \times \frac{236}{60} \sqrt{\frac{.87}{4 \times 2.9 \times 2.47}}$$

$= 9.98$.

To calculate the system C_v:

$$\frac{1}{C_v{}^2 total} = \sum \frac{1}{C_{v_i}{}^2}$$

Component	C_{v_i}
Valve	1.74
Hose (10 ft)	31.54
Filter	25.94
Pipe (40 ft)	5.37
Elbows (4)	9.98

$$\frac{1}{C_v^2 \text{ total}} = \frac{1}{(1.74)^2} + \frac{1}{(31.54)^2} + \frac{1}{(25.94)^2} + \frac{1}{(5.37)^2} + \frac{1}{(9.98)^2}$$

C_v total = 1.6.

For 1/2 in Pipe

For 1/2 in pipe (schedule 80) (roughly the same I.D. as 1/2 in tubing), using Figure A-2 again,

$$P = 2.47 \text{ psi @ } 14.6 \text{ gpm}$$

Therefore,

$$C_v = 14.6 \text{ gpm} \times \frac{236}{60} \sqrt{\frac{S.G.}{\Delta P}}$$

$$= 33.98 \text{ for 1 ft.}$$

For 40 ft, $C_v = 5.37$.

Elbows

For 4 1/2 in elbows (schedule 80), each elbow is equivalent to 2.9 feet of 1/2 in pipe (schedule 80). From Figure A-2, P = 2.47 psi/ft at 14.6 gpm.

$$C_v = 14.6 \times \frac{236}{60} \sqrt{\frac{.87}{4 \times 2.9 \times 2.47}}$$

$$= 9.98.$$

To calculate the system C_v:

$$\frac{1}{C_v{}^2 \text{total}} = \sum \frac{1}{C_{v_i}{}^2}$$

Component	C_{v_i}
Valve	1.74
Hose (10 ft)	31.54
Filter	25.94
Pipe (40 ft)	5.37
Elbows (4)	9.98

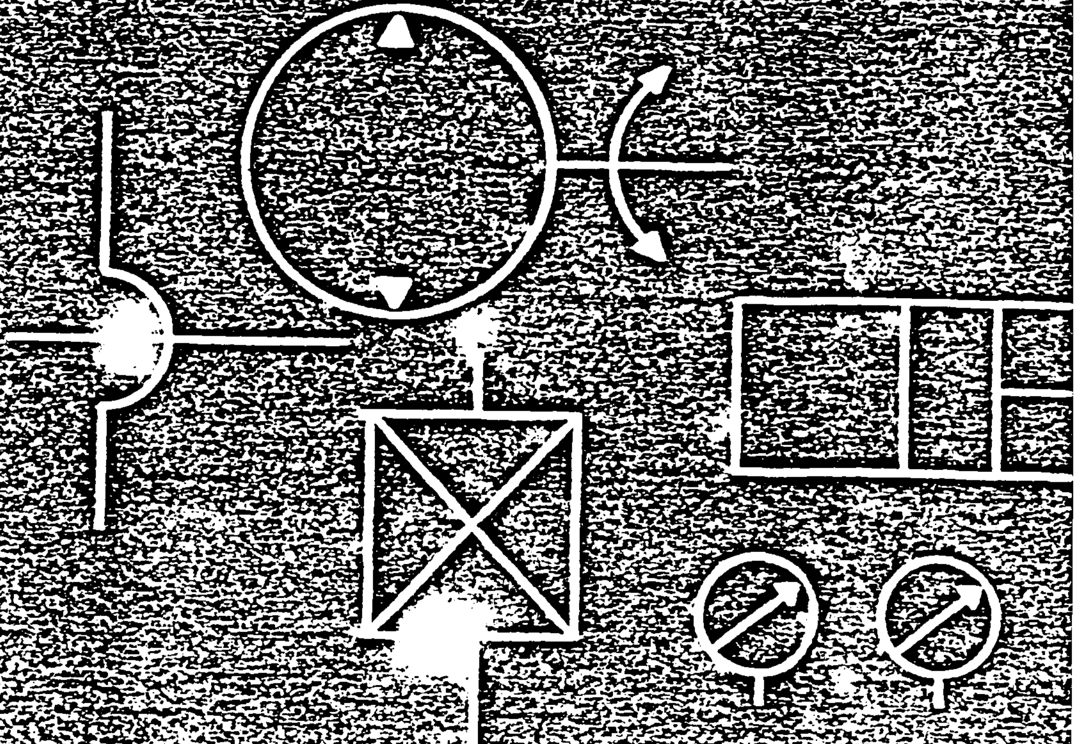
$$\frac{1}{C_v^2 \text{ total}} = \frac{1}{(1.74)^2} + \frac{1}{(31.54)^2} + \frac{1}{(25.94)^2} + \frac{1}{(5.37)^2} + \frac{1}{(9.98)^2}$$

$$C_v \text{ total} = 1.6.$$

Fluid Power Designers

Lightning Reference Handbook

STANDARD ENGINEERING DATA



THIRD EDITION

PAUL-MUNROE HYDRAULICS INC

[illegible] (213) 723-0692 · San Francisco (415) [illegible] · [illegible] (714) [illegible] · [illegible] (805) 832-3630

FLUID POWER DESIGNER'S LIGHTING REFERENCE HANDBOOK

HYDRAULIC CHARACTERISTICS

Unless specified otherwise, all performance parameters are given for valve operation on Mobil DTE-24 fluid at 100°F (38°C).

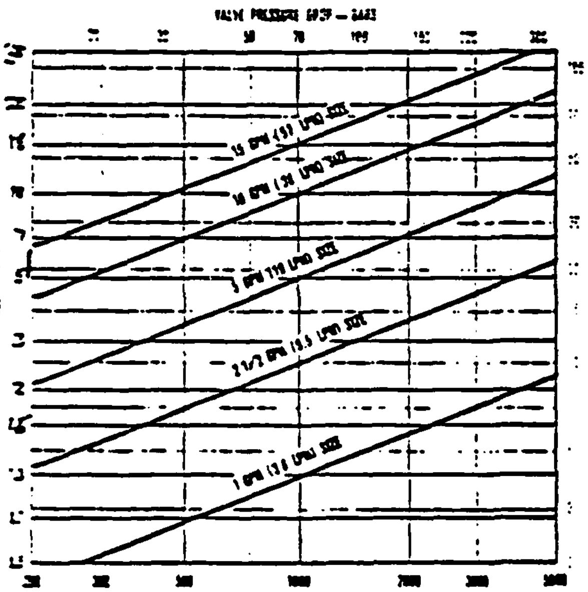


FIGURE 1 CHANGE IN RATED FLOW WITH PRESSURE

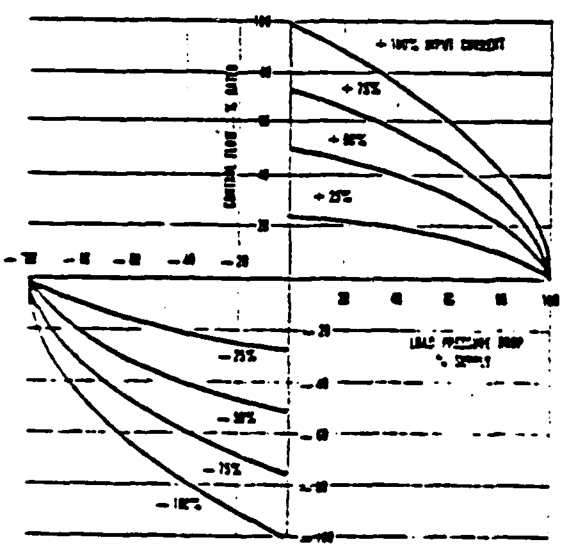


FIGURE 2 — CHANGE IN CONTROL FLOW WITH CURRENT AND LOAD PRESSURE

FLUID SUPPLY AO76 Servovalves are intended to operate with constant supply pressure.

Supply Pressure	
minimum	200 psi (14 bars)
maximum standard	3000 psi (210 bars)
maximum special order	5000 psi (350 bars)
Proof Pressure	
at pressure port	150% supply
at return port	100% supply
NFPA static pressure rating* (test pressure 10,700 psi)	6900 psi
NFPA cyclic pressure rating (pressure port)* (cyclic test pressure 4350 psi for > 10^6 cycles)	3000 psi
Fluid†	petroleum base hydraulic fluids 60-450 SUS @ 100° F (10-97 cST @ 38° C)
Supply filtration required	10μm nominal (25μm absolute) or finer recommended
Operating temperature	
minimum	− 40° F (− 40° C)
maximum (unless limited by fluid)	+ 275° F (+ 135° C)

*Method of verifying static and fatigue pressure ratings per NFPA/T2.6.1-1974, category 3/90.

†Buna N seals are standard; Viton A and EPR available on special order.

RATED FLOW Five standard sizes are available having rated flows of 1, 2½, 5, 10, and 15 gpm at 1000 psi valve drop (3.8, 9.5, 19, 38, and 57 lit/min at 70 bars). See plot at left for corresponding rated flows at other supply pressures.

Flow with various combinations of supply pressure and load pressure drop can be determined by calculating the valve pressure drop.

$$P_V = (P_S - P_R) - P_L$$

P_V = valve pressure drop
P_S = supply pressure
P_R = return pressure
P_L = load pressure drop

FLOW-LOAD CHARACTERISTICS Control flow to the load will change with load pressure drop and electrical input as shown in Figure 2. These characteristics follow closely the theoretical square-root relationship for sharp-edged orifices, which is

$$Q_L = K\, i \sqrt{P_V}$$

Q_L = control flow
K = valve sizing constant
i = input current
P_V = valve pressure drop

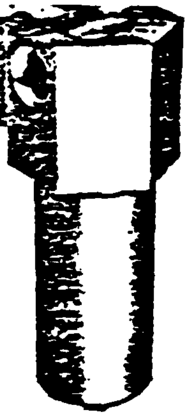
INTERNAL LEAKAGE Maximum internal leakage for each size servovalve is:

Flow with 1000 psi (70 bars) Supply

Rated Flow	Internal Leakage
1 gpm (3.8 lit/min)	< 0.17 gpm (0.66 lit/min)
2½ gpm (9.5 lit/min)	< 0.22 gpm (0.83 lit/min)
5 gpm (19 lit/min)	< 0.35 gpm (1.32 lit/min)
10 gpm (38 lit/min)	< 0.35 gpm (1.32 lit/min)
15 gpm (57 lit/min)	< 0.35 gpm (1.32 lit/min)

FIGURE A-1. Servo Valve Data Sheet

NF30 15 GPM/3000 PSI



3 to 60 MICRONS

Pressure Rating: 3000 PSI (204 Bar) operating
10,000 PSI (680 Bar) minimum yield

Maximum Operating Temperatures:
250° F (121° c) with N elements
300° F (149° c) with NM elements

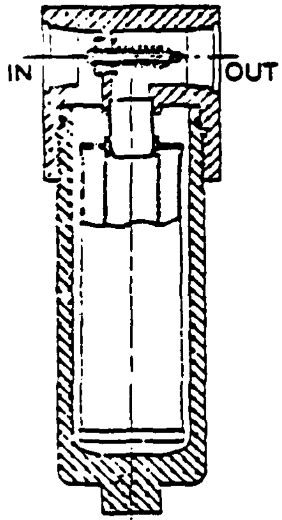
Bypass Valve Setting: 30 PSI (cracking pressure)

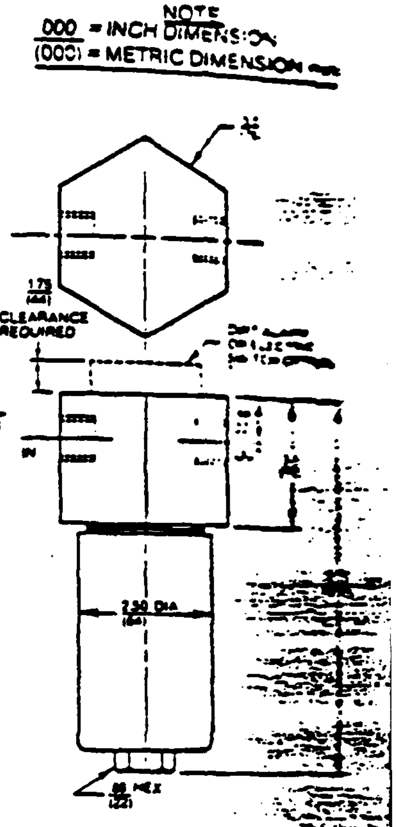
Material: Porting Head: Aluminum
Element Case: Aluminum

Compatibility: All commonly used mineral base and phosphate esters fluids. DF30 recommended for high water base and water glycol fluids. H seal adder recommended for phosphate esters. Consult factory for indicator availability for phosphate esters.

Servicing: Element case threaded into porting head. Minimum clearance required 2½" (64 mm.)

Weight: 3 lbs. (1 kg.)



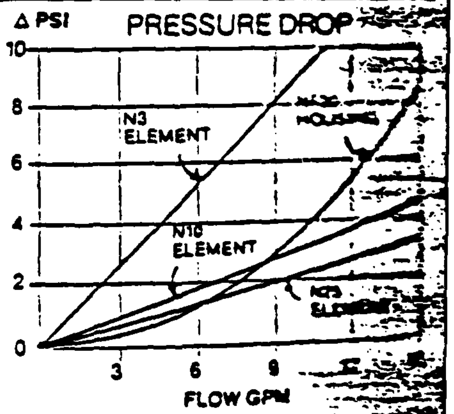


ORDERING INFORMATION

MODEL NUMBER SELECTION*

FILTER SERIES	NUMBER OF ELEMENTS	ELEMENT TYPE	SEAL MATERIAL	PORTING OPTIONS	DIRT ALARM TYPE
NF30	-1 = one element	N3 N10 N25 NM10 NM60	(Omit) = Buna N V = Viton	-P = ¾" NPTF -S = 1 1/16" - 12 SAE straight	(Omit) = none -D = color coded visual dirt alarm -MS2 = electrical switch

*Example: NF30-1N10V-P-D
NF30 filter with single 10 micron throw away element, viton seals, ¾" pipe ports, and visual indicator.



Element ΔP = (ΔP from curve) X (SSU Viscosity/150)
Filter ΔP = Housing ΔP + element ΔP
Element ΔP based on 150 SSU

SELECTION CHART

PRESSURE	ELEMENT Type	ELEMENT Micron	ENCLOSURES AND ELEMENTS TO MEET FLOW REQUIREMENTS (Based on use of 150 SSU mineral base hydraulic fluid)	
TO 3000 PSI	N Throw-Away	3	NF30-1N3-P	SEE MODEL DF30
		10	NF30-1N10-P	
		25	NF30-1N25-P	
	NM Clean-able	10	NF30-1NM10-P	
		60	NF30-1NM60-P	
Flow GPM			10	

MATERIAL						PRESSURE LOSS (PSI/FOOT LENGTH) IN PIPES AT AVERAGE FLOW VELOCITY (FT/SEC) OF														EQUIVALENT PIPE LENGTHS (FT.) FOR CIRCUIT COMPONENTS					
SIZE (INCHES)	PIPE HOSE	O.D. INCHES	I.D. INCHES	WALL INCHES	I.D. AREA SQ. IN.	5		7		10		15		20		25		30		TEE			ELBOW		
						LOSS	GPM	LOSS	GPM	LOSS	GPM	LOSS	GPM	LOSS	GPM	LOSS	GPM	LOSS	GPM						
1/8	PIPE-SCH 40	.405	.269	.068	.057	1.25	.89	1.79	1.24	2.60	1.75	3.16	2.67	5.47	3.56	6.20	4.45	7.07	5.34						
	PIPE-SCH 80	.405	.215	.095	.036	1.89	.56	3.05	.78	4.26	1.12	5.20	1.68	8.38	2.24	11.1	2.80	12.7	3.36						
	HOSE	[illegible]	.125	[illegible]	.012	5.96	.186	8.37	.260	11.9	.372	16.0	.558	24.0	.744	30.0	.930	35.7	1.11	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1/4	PIPE-SCH 40	.540	.364	.088	.104	.67	1.62	1.05	2.27	1.64	3.24	1.92	4.96	2.97	6.48	3.23	8.10	3.73	9.72						
	PIPE-SCH 80	.540	.302	.119	.072	1.11	1.12	1.49	1.57	2.11	2.24	2.84	3.36	4.15	4.48	5.08	5.60	6.30	6.72						
	HOSE	[illegible]	.250	[illegible]	.049	1.57	.756	2.17	1.08	3.00	1.49	4.49	2.23	6.04	2.98	7.49	3.72	8.95	4.44	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
3/8	PIPE-SCH 40	.675	.493	.091	.191	.39	2.98	.57	4.18	.86	5.96	1.05	8.94	1.69	11.92	4.27	14.9	5.78	16.9	2.7	.8	2.7	1.2	2.7	.6
	PIPE-SCH 80	.675	.423	.126	.140	.54	2.18	.74	3.06	1.10	4.36	1.34	6.54	1.97	8.72	5.19	10.9	7.20	13.1						
	HOSE	[illegible]	.375	[illegible]	.110	.885	1.71	1.07	2.43	1.54	3.35	2.02	5.03	2.88	6.71	3.33	8.38	3.99	10.0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1/2	PIPE-SCH 40	.840	.622	.109	.304	.24	4.74	.36	6.65	.49	9.48	.68	14.22	2.09	18.98	3.38	23.7	4.28	28.4	3.5	1.05	3.5	1.5	3.5	.75
	PIPE-SCH 80	.840	.546	.147	.234	.30	3.65	.45	5.12	.71	7.30	.78	10.9	2.47	14.6	3.61	18.2	5.00	21.9	2.9	.9	2.9	1.4	2.9	.68
	PIPE-SCH XX	.840	.252	.294	.050	1.54	.78	2.19	1.09	3.08	1.56	3.65	2.34	6.13	3.12	7.48	3.90	9.55	4.68						
	HOSE	[illegible]	.500	[illegible]	.196	.387	3.03	.547	4.30	.755	5.94	1.13	8.90	2.47	11.9	3.15	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
3/4	PIPE-SCH 40	1.050	.824	.113	.533	.14	8.32	.22	11.7	.27	16.6	.78	25.0	1.47	33.3	2.19	41.6	3.00	49.9	4.5	1.4	4.5	2.1	4.5	1.0
	PIPE-SCH 80	1.050	.742	.154	.432	.16	4.74	.26	9.45	.37	13.5	.87	20.2	1.71	27.0	2.48	33.7	3.52	40.4	4.0	1.2	4.0	1.6	4.0	.8
	PIPE-SCH XX	1.050	.434	.308	.148	.53	2.31	.67	3.24	1.05	4.62	1.31	6.93	1.94	9.24	5.06	11.6	7.02	13.9						
	HOSE	[illegible]	.750	[illegible]	.442	.171	6.82	.240	9.92	.336	13.4	.502	20.1	.733	26.8	2.07	33.3	2.60	41.3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1	PIPE-SCH 40	1.315	1.049	.133	.863	.10	13.5	.13	18.9	.34	26.9	.57	40.4	1.42	53.8	1.64	67.3	2.24	80.7	5.7	1.7	5.7	2.6	5.7	1.2
	PIPE-SCH 80	1.315	.957	.179	.719	.11	11.2	.15	15.7	.24	22.4	.62	33.6	1.23	44.8	1.84	56.1	2.93	67.3	5.2	1.6	5.2	2.5	5.2	1.1
	PIPE-SCH XX	1.315	.599	.358	.863	.26	4.39	.37	6.16	.53	8.78	.67	13.2	2.25	17.6	3.29	22.0	4.20	26.3	3.0	1.0	3.0	1.5	3.0	.75
	HOSE	[illegible]	1.00	[illegible]	.785	.097	12.2	.136	17.1	.194	24.4	.410	36.6	.987	48.8	1.61	61.2	1.03	73.4	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1-1/4	PIPE-SCH 40	1.660	1.380	.140	1.496	.05	23.4	.08	31.7	.25	46.7	.39	70.1	.78	93.4	1.18	117	1.47	140	7.5	2.4	7.5	3.7	7.5	1.6
	PIPE-SCH 80	1.660	1.278	.191	1.280	.07	20.0	.09	28.1	.26	39.9	.44	58.9	.85	79.8	1.27	99.8	1.80	120	7.0	2.1	7.0	3.5	7.0	1.5
	PIPE-SCH XX	1.660	.896	.382	.630	.13	9.83	.16	13.8	.24	19.7	.71	29.5	1.35	39.3	2.01	49.2	2.76	59.0	4.9	1.5	4.9	2.3	4.9	1.1
	HOSE	[illegible]	1.25	[illegible]	1.23	.062	19.1	.087	26.8	.125	38.2	.436	57.3	.738	76.4	1.08	95.5	1.52	115	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1-1/2	PIPE-SCH 40	1.900	1.610	.145	2.046	.04	31.8	.06	44.5	.19	63.5	.33	95.3	.64	127	.96	159	1.26	191	9.0	2.8	9.0	4.3	9.0	2.0
	PIPE-SCH 80	1.900	1.500	.200	1.767	.04	27.6	.06	38.6	.21	55.1	.42	82.7	.71	110	1.06	138	1.36	166	8.2	2.6	8.2	4.2	8.2	1.8
	PIPE-SCH XX	1.900	1.100	.400	.950	.09	14.8	.09	20.8	.32	29.6	.51	44.4	1.05	59.2	1.51	74.1	2.14	88.9	6.5	2.0	6.5	3.0	6.5	1.4
	HOSE	[illegible]	1.50	[illegible]	1.77	.044	27.7	.061	38.6	.180	55.1	.353	82.7	.594	110	.86	138	1.21	166	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
2	PIPE-SCH 40	2.375	2.067	.154	3.355	.03	52.3	.08	73.4	.14	105	.24	159	.48	209	.69	262	.85	324	11.0	3.5	11.0	5.5	11.0	2.5
	PIPE-SCH 80	2.375	1.939	.218	2.953	.03	46.0	.09	64.6	.15	92.0	.26	138	.52	184	.73	230	.98	275	10.8	3.4	10.8	5.0	10.8	2.4
	PIPE-SCH XX	2.375	1.503	.436	1.773	.04	27.7	.12	38.8	.21	55.3	.36	82.9	.72	111	1.34	138	1.36	166	8.2	2.6	8.2	4.0	8.2	1.8
	HOSE	[illegible]	2.00	[illegible]	3.14	.024	48.9	.034	68.5	.123	97.8	.256	147	.415	196	.60	245	.89	293	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
2-1/2	PIPE-SCH 40	2.875	2.469	.203	4.788	.03	74.8	.07	105	.11	149	.20	224	.37	299	.53	374	.72	449	14.0	4.2	14.0	6.5	14.0	3.0
	PIPE-SCH 80	2.875	2.323	.276	4.238	.03	66.1	.07	92.6	.12	132	.21	198	.39	264	.57	331	.87	397	13.0	4.0	13.0	6.1	13.0	2.9
	PIPE-SCH XX	2.875	1.771	.552	2.464	.03	38.5	.10	53.4	.17	76.9	.30	115	.59	154	.79	193	1.15	231	10.3	3.1	10.3	4.8	10.3	2.2
	HOSE	[illegible]	2.50	[illegible]	4.91	.014	76.5	.045	107	.095	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

APPENDIX C

DARPA 150-POUND LEG SIMULATION

TIME	ACCELERATION	VELOCITY	DISPLACEMENT
0.00010	9337.3105	0.9557	0.0001
0.00020	9532.4668	1.9090	0.0003
0.00030	9408.1963	2.8542	0.0006
0.00040	9335.6543	3.7884	0.0010
0.00050	9166.9785	4.7051	0.0014
0.00060	8955.2217	5.6006	0.0020
0.00070	8704.2168	6.4710	0.0026
0.00080	8418.4414	7.3123	0.0034
0.00090	8102.8438	8.1231	0.0042
0.00100	7762.6699	8.8994	0.0051
0.00110	7403.2798	9.6397	0.0060
0.00120	7029.9927	10.3427	0.0071
0.00130	6647.9302	11.0075	0.0082
0.00140	6261.9023	11.6337	0.0093
0.00150	5876.3027	12.2213	0.0105
0.00160	5495.0479	12.7708	0.0118
0.00170	5121.5342	13.2830	0.0132
0.00180	4758.6221	13.7599	0.0145
0.00190	4408.6377	14.1997	0.0159
0.00200	4073.4023	14.6071	0.0174
0.00210	3754.2615	14.9825	0.0189
0.00220	3452.1321	15.3277	0.0204
0.00230	3167.5513	15.6445	0.0220
0.00240	2900.7268	15.9345	0.0236
0.00250	2651.5913	16.1997	0.0252
0.00260	2419.8489	16.4417	0.0269
0.00270	2205.0227	16.6622	0.0285
0.00280	2006.4938	16.8628	0.0302
0.00290	1823.5422	17.0452	0.0319
0.00300	1655.3723	17.2107	0.0336
0.00310	1501.1456	17.3608	0.0354
0.00320	1359.9987	17.4968	0.0371
0.00330	1231.0653	17.6199	0.0389
0.00340	1113.4873	17.7313	0.0407

0.00350	1006.4296	17.8319	0.0424
0.00360	909.0832	17.9228	0.0442
0.00370	820.6812	18.0049	0.0460
0.00380	740.4878	18.0790	0.0478
0.00390	667.8171	18.1457	0.0497
0.00400	602.0219	18.2059	0.0515
0.00410	542.5013	18.2602	0.0533
0.00420	488.6968	18.3091	0.0551
0.00430	440.0905	18.3531	0.0570
0.00440	396.2091	18.3927	0.0588
0.00450	356.6118	18.4284	0.0607
0.00460	320.8979	18.4604	0.0625
0.00470	288.7042	18.4893	0.0643
0.00480	259.6916	18.5133	0.0662
0.00490	233.5560	18.5386	0.0681
0.00500	210.0189	18.5596	0.0699
0.00510	188.8277	18.5785	0.0718
0.00520	169.7558	18.5955	0.0736
0.00530	152.5926	18.6108	0.0755
0.00540	137.1522	18.6245	0.0774
0.00550	123.2619	18.6368	0.0792
0.00560	110.7708	18.6479	0.0811
0.00570	99.5374	18.6578	0.0829
0.00580	89.4372	18.6668	0.0848
0.00590	80.3590	18.6748	0.0867
0.00600	72.1978	18.6820	0.0885
0.00610	64.8618	18.6885	0.0904
0.00620	58.2694	18.6943	0.0923
0.00630	52.3451	18.6996	0.0942
0.00640	47.0228	18.7043	0.0960
0.00650	42.2395	18.7085	0.0979
0.00660	37.9399	18.7123	0.0998
0.00670	34.0798	18.7157	0.1016
0.00680	30.6101	18.7188	0.1035
0.00690	27.4928	18.7215	0.1054
0.00700	24.6936	18.7240	0.1073
0.00710	22.1783	18.7262	0.1091
0.00720	19.9188	18.7282	0.1110

0.00730	17.8904	18.7300	0.1129
0.00740	16.0681	18.7316	0.1147
0.00750	14.4324	18.7330	0.1166
0.00760	12.9609	18.7343	0.1185
0.00770	11.6386	18.7355	0.1204
0.00780	10.4533	18.7365	0.1222
0.00790	9.3878	18.7375	0.1241
0.00800	8.4311	18.7383	0.1260
0.00810	7.5724	18.7391	0.1279
0.00820	6.8001	18.7398	0.1297
0.00830	6.1063	18.7404	0.1316
0.00840	5.4849	18.7409	0.1335
0.00850	4.9241	18.7414	0.1354
0.00860	4.4224	18.7418	0.1372
0.00870	3.9713	18.7422	0.1391
0.00880	3.5669	18.7426	0.1410
0.00890	3.2037	18.7429	0.1429
0.00900	2.8762	18.7432	0.1447
0.00910	2.5830	18.7435	0.1466
0.00920	2.3201	18.7437	0.1485
0.00930	2.0829	18.7439	0.1504
0.00940	1.8713	18.7441	0.1522
0.00950	1.6808	18.7443	0.1541
0.00960	1.5097	18.7444	0.1560
0.00970	1.3557	18.7446	0.1579
0.00980	1.2180	18.7447	0.1597
0.00990	1.0936	18.7448	0.1616
0.01000	0.9823	18.7449	0.1635

APPENDIX D

TECHNICAL SPECIFICATIONS ON VICKERS PVB 10 HYDRAULIC PUMP

Pump Specifications

Vickers pumps PVB 10 with pressure compensator and maximum stroke adjustor (all mechanical settings with adjustment screws and nuts).

Pumps capable of 3200 rpm with 12 psig supercharge. Maximum pressure = 3000 psi; compensation range = 250 - 3000 psi/hput HP @ 1800 rpm and 3000 psi; extrapolated to 3200 rpm and 3000 psi, HP = 35.6, volume = 18 gpm, weight = 31 lb each.

VARIABLE DISPLACEMENT PISTON PUMPS

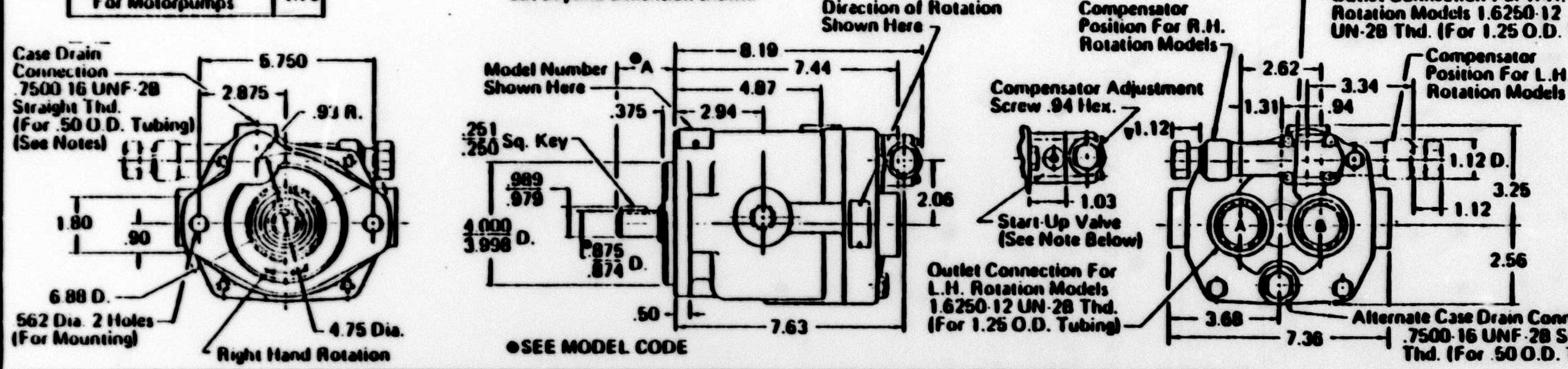
SPERRY VICKERS T.M.

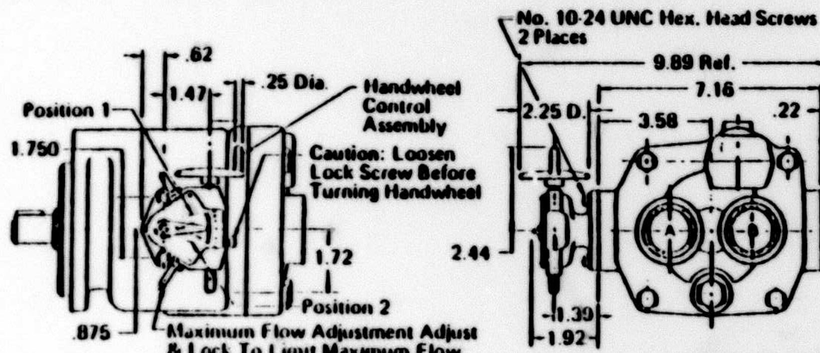
PVB10 AND PVB15 - 30 DESIGN (IN-LINE TYPE)
WITH PRESSURE COMPENSATOR, HANDWHEEL AND LEVER CONTROLS

Shaft Dimension "A" ●See Code For Models	
Standard	2.31
Optional -- For Motorpumps	1.75

"CG" REMOTE COMPENSATOR SEE PAGE 508400-1.

▼Caution – While pump is operating do not back compensator adjustment screw out beyond dimension shown.

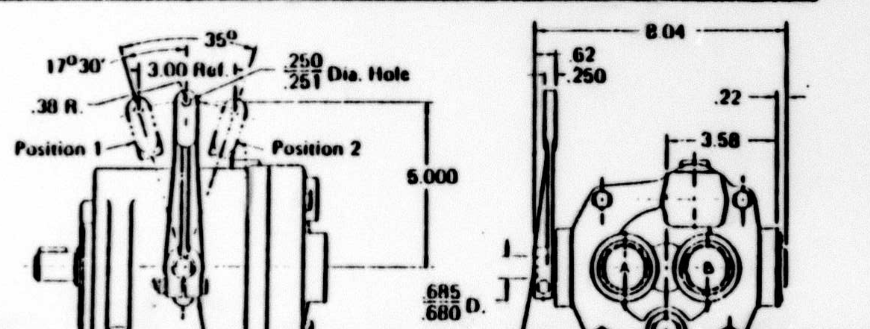




HANDWHEEL CONTROL

Provides manual selection of pump displacement. Handwheel controlled units may be operated on both sides of center permitting bi-directional fluid flow characteristics.

Shaft Rotation	Pointer Position	Handwheel Rotation From Zero	Outlet Port
R.H.	1	Clockwise	A
R.H.	2	Counter-Clockwise	B
L.H.	1	Clockwise	B
L.H.	2	Counter-Clockwise	A



LEVER CONTROL

The lever control provides mechanical or manual selection of pump displacement. Lever controlled units may be operated on both sides of center permitting bi-directional fluid flow characteristics. Hand lever controls must be secured by suitable linkage arrangement to maintain desired setting. The control pintle may be rotated 17-1/2° on each side of center position to permit full reversal of flow. Pintle travel is limited to 35° by internal stops. Torque required to rotate control pintle is approximately 60 lbf·in at 1800 RPM and 1000 PSI. (Note: Torque varies with speed and pressure.)

Shaft Rotation	Lever Position	Outlet Port
R.H.	1	A

PRESSURE COMPENSATOR CONTR

The pressure compensator control a matically adjusts pump delivery to mai volume requirements of the system preselected operating pressure. Maxi pump delivery is maintained to app mately 75 PSI (PVB10), 100 PSI (PV below the pressure control setting b being reduced. The pressure compen control operates on one side of center has an adjustment range of 250 to 150 on "CM" models and 250 to 3000 PS "C" models as designated in the m coding. Customer must limit PVB1 2000 PSI when using "C" compensator

WITH START-UP VALVE

The pressure compensator control ca ordered with a start-up valve. This unique device that permits starting a sure compensated pump with a smaller electric motor than is normally requir rated conditions. It is designed for use high flow-low pressure and low flow pressure applications in which torqu quirements are considerably lower tha rated conditions. This device is compl automatic with no external signal need

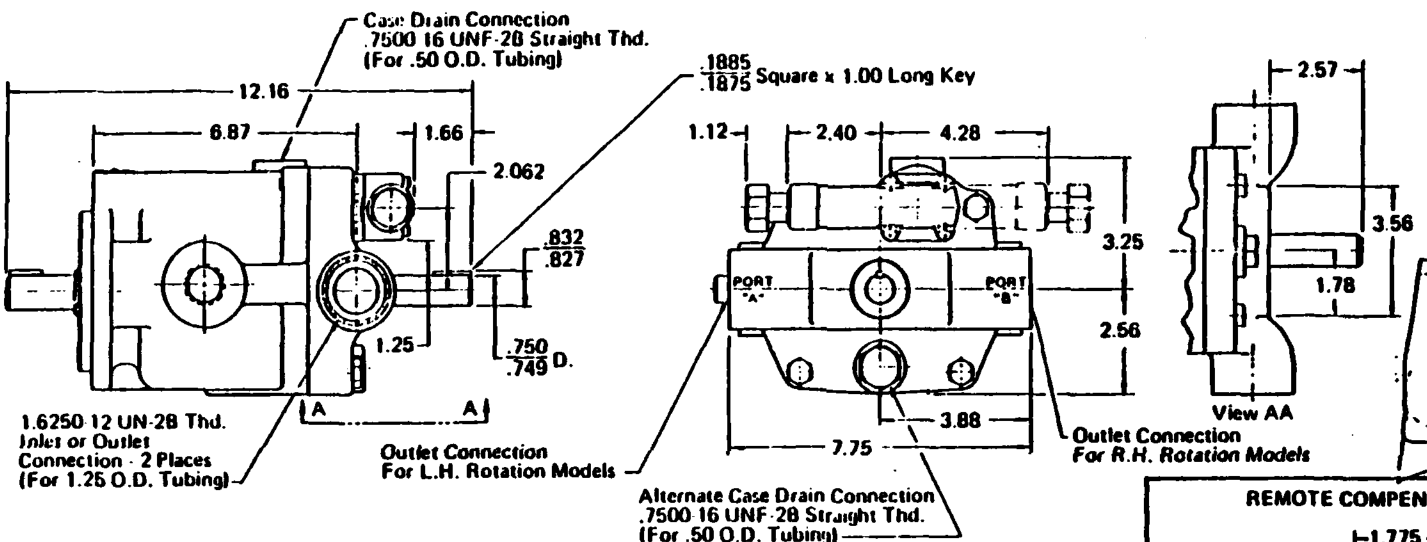
Operational data for this device is as foll

1. Start-up pressures can be doubled u a start-up valve and the same ele motor.
2. Electric motor size can be halved usi start-up valve with start-up pressure maining the same.

SPERRY VICKERS VARIABLE DISPLACEMENT PISTON PUMPS

T.M.

THRU SHAFT AND/OR SIDE PORTS
WITH PRESSURE COMPENSATOR, HANDWHEEL AND LEVER CONTROLS



PRESSURE COMPENSATOR CONTROL - THRU SHAFT AND SIDE PORTS

General Data

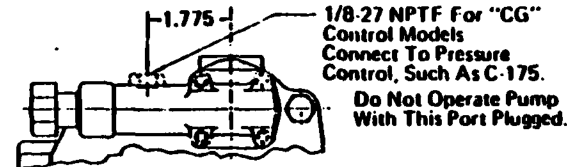
Side Ports

Rated and maximum speeds. Pressure and general performance of the side port pumps are identical to the standard units.

Thru Shaft

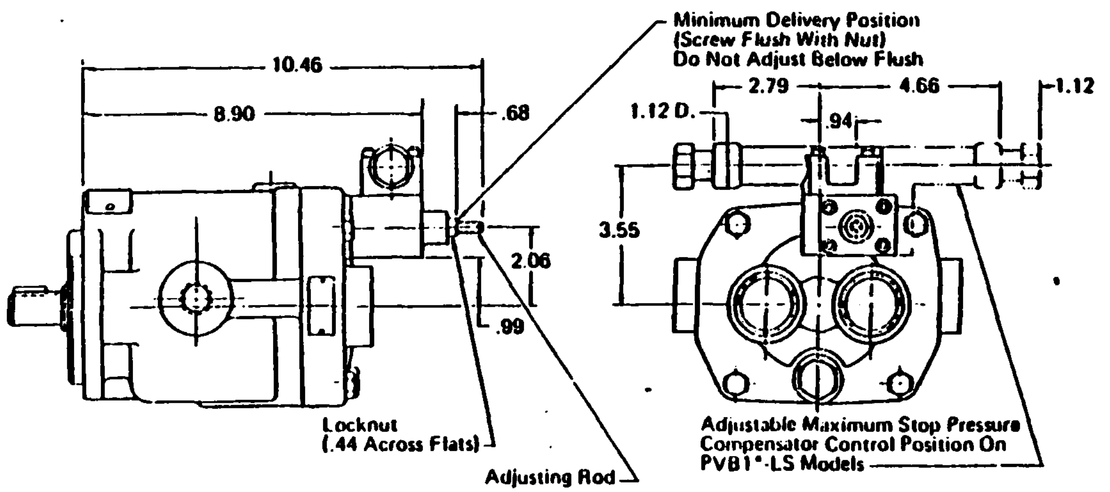
Rated and maximum speeds, pressure and general performance of the thru shaft pumps are identical to the standard units, except the input torque is limited to 735 lbf-in.

REMOTE COMPENSATOR "CG" (SEE MODEL CODE)



Adjustment procedure for "CG" models
1. Turn pressure control (such as C-175) CCW to minimum setting.
2. Turn compensator adjustment plug to desired minimum pressure (250 PSI or higher).
3. Full pressure range can now be obtained with pressure control.

Caution: Effective compensator pressure will be compensator control plus remote relief valve setting.



PRESSURE COMPENSATOR CONTROL WITH ADJUSTABLE MAXIMUM DISPLACEMENT STOP

Control Description

The adjustable maximum stop pressure control enables the maximum pump delivery to be externally adjusted from 25% to 100% while maintaining all of the standard features of a pressure compensated pump. To assist initial priming, manual adjustment control setting must be at least 40 percent of maximum flow position.

Control Adjustment

Loosen locknut on adjusting rod. Turn adjusting rod clockwise (CW) to decrease maximum pump delivery or counterclockwise (CCW) to increase maximum pump delivery until desired setting is obtained. Secure this setting by tightening locknut.

Note:
Not available with (X) thru shaft port models.

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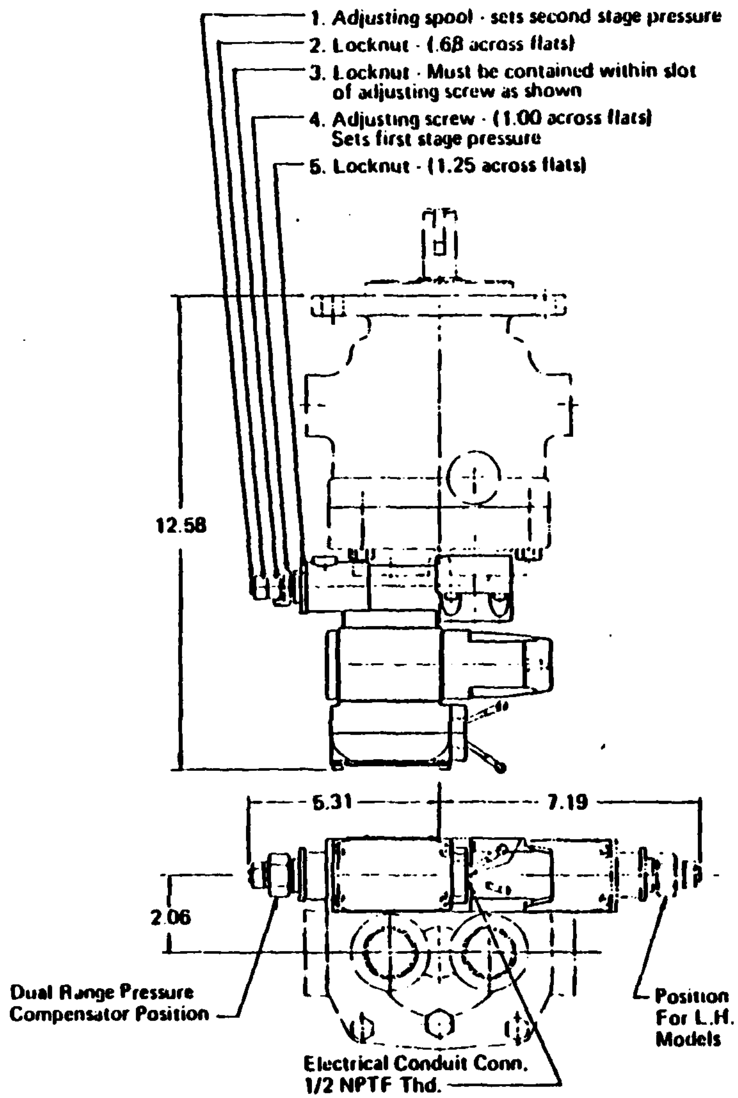

ELECTRIC DUAL-RANGE
PRESSURE COMPENSATOR CONTROL

PVB10 AND PVB15 - 30 DESIGN
VARIABLE DELIVERY INLINE PISTON PUMP
WITH DUAL RANGE PRESSURE COMPENSATOR CONTROL

Control Description

The dual range pressure compensator control automatically adjusts pump delivery to maintain volume requirements of the system at either of two preselected operating pressures.

Maximum pump delivery is maintained to approximately 75 PSI (PVB10), 100 PSI (PVB15) below either of the pressure control settings before being reduced.

Electric control of the dual range pressure compensator is available in two pressure ranges of 250 to 1600 PSI or 250 to 3000 PSI. Control type and pressure range is designated in the model code.

Control Features

The dual range pressure compensator offers the following advantages:

1. Low - pressure setting for –
   A–Low horsepower start-up
   B–Tool or equipment tryout
   C–Low power consumption and low heat generation when the machine or circuit is at rest

2. High - pressure setting for –
   A–Machining or circuit applications

Control Adjustment

1. With the directional valve de-energized, loosen locknut "5" and turr the adjusting screw "4" to the desired first stage pressure setting anc tighten locknut "5".

2. With solenoid de-energized, turn adjusting spool "1" counterclockwis (CCW) until nut "3" is bottomed in adjusting screw slot. (Second stag setting is now equal to first stage pressure setting). Turn adjusting spoo clockwise (CW) to desired second stage pressure requirements. (On complete turn of adjusting spool equals approximately 600 PSI o "CD" models, and 300 PSI on "CMD" models.) Energize solenoid an check pressure setting. De-energize solenoid and re-adjust if necessary Secure this setting by tightening locknut "2".

Note:
Not available with (X) thru shaft models.

▲ **Note:**
Any sliding spool valve, if held shifted under pressure for long perioc of time, may stick and not spring return due to fluid residue formatio and, therefore, should be cycled periodically to prevent this fro happening.

▲Solenoid data (110 V ac 50 Hz and 115/120 V ac 60 Hz)

Solenoid Current	Inrush amps (R.M.S.) ✱	Holding amps
115/120 V ac 60 Hz – 110 V ac 50 Hz	2.0	.4

✱Maximum peak inrush amps approximately 1.4 x R.M.S. value shown.

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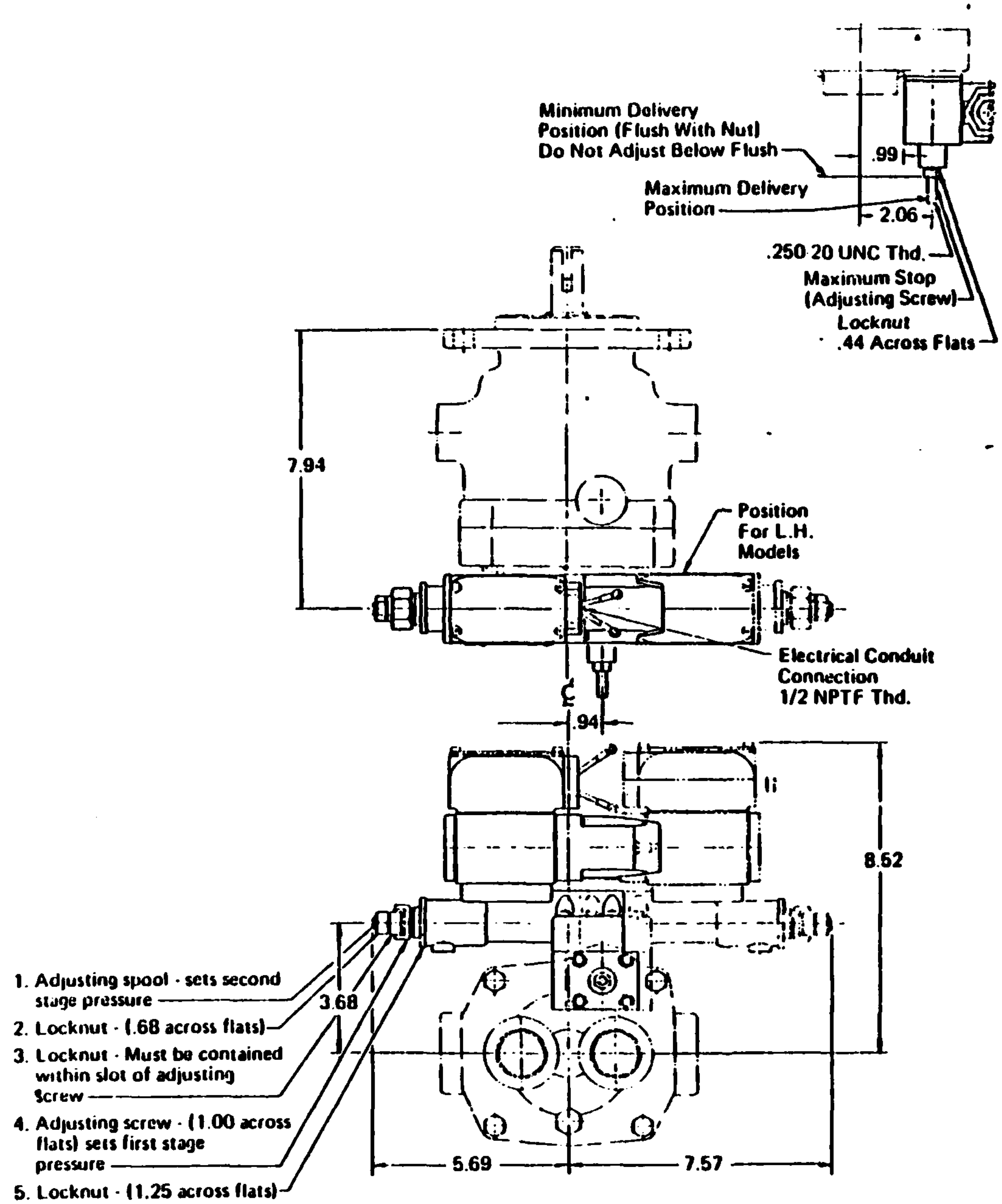


ELECTRIC DUAL-RANGE PRESSURE COMPENSATOR AND MAXIMUM STOP CONTROL

PVB10 AND PVB15 - 30 DESIGN VARIABLE DELIVERY INLINE PISTON PUMP WITH DUAL RANG PRESSURE COMPENSATOR CONTROL & MAXIMUM FLOW ADJUSTABLE STOP

Control Description

The dual range pressure compensator control with max. flow adjust: stop automatically adjusts pump delivery to maintain volume requireme of the system at either of two preselected operating pressures.

Maximum pump delivery is maintained to approximately 75 PSI (PVB 100 PSI (PVB15) below *either of the pressure control settings before b* reduced. The maximum delivery is adjustable and can be set at any set from 25% to 100% flow.

Electric control of the dual range pressure compensator is available in pressure ranges of 250 to 1500 PSI or 250 to 3000 PSI. Control type pressure range is designated in the model code.

Compensator Control

1. With the directional valve de-energized, loosen locknut "5" and turn adjusting screw "4" to *the desired first stage pressure setting and tigh* locknut "5".
2. With directional valve de-energized, turn adjusting spool "1" coun clockwise until nut "3" is bottomed in adjusting screw slot. (Sec *stage setting is now equal to first stage pressure setting).* Turn adjus spool clockwise to desired second stage pressure requirements (complete turn of adjusting spool equals approximately 600 PSI "CCD" models, and 300 PSI on "CMCD" models). Energize soler and check pressure setting. De-energize solenoid and re-adjust pres: if necessary. Secure this setting by tightening locknut "2".

Maximum Flow Adjustment

With the system pressure below both compensator settings, loosen m mum stop adjusting screw locknut and adjust screw to desired I *position (turning screw clockwise decreases flow and turning sc* counterclockwise increases flow). To lock screw in position tigh locknut. To assist initial priming, adjust control setting to at least of maximum flow position.

Note:

Not available with (X) thru shaft models.

▲ **Note:**

Any sliding spool valve, if held shifted under pressure for long period time, may stick and not spring return due to fluid residue formation a therefore, should be cycled periodically to prevent this from happening

▲ Solenoid data (see note on page 508400B-2).

General Data

These units are of the axial piston, variable displacement, in-line design, displacement is varied by means of either a pressure compensator, handwheel, or lever control.

Installation Information

Horizontal mounting is recommended to maintain necessary case fluid level. The case drain line must be full size unrestricted and connected from the uppermost drain port directly to the reservoir in such a manner that the case remains filled with fluid. Piping of drain line must prevent siphoning. Pipe drain line so that it terminates below reservoir fluid level. No other lines are to be connected to this drain line. Caution must be exercised to never exceed 5 PSI unit case pressure.

Starting

Before starting, fill case with system fluid thru uppermost drain port. Case must be kept full at all times to provide internal lubrication.

When first starting, it may be necessary to bleed air from pump outlet line to permit priming and reduce noise. Bleed by loosening an outlet connection until solid stream of fluid appears. An air-bleed valve is available for this purpose. See drawing 521601.

To assist initial priming, control setting must be at least 40% of maximum flow position.

Operating Specifications

Model Number	Theoretical Displacement In.^{3}/Rev.	Delivery GPM At 1800 RPM	Delivery GPM At Max. RPM	Operating Speed RPM (Maximum)	Pressure PSI (Maximum)	■ Sound Data dB(A)	Input Horsepower At Max. PSI & 1800 RPM
PVB10	1.29	10	18	3200	3000	62	20
PVB15	2.01	15.7	26.2	3000	2000	65	21

Refer to curve for inlet pressure requirements.

■ Straight port model at cutoff, 1200 RPM, 2000 PSI pressure, SAE 10W oil at 120°F., 5" Hg inlet vacuum per NFPA standard T3.9.70.12.

Drive Rotation

Shaft rotation is not reversible and must be specified when ordering. (See drawing 517010 for indirect drive data).

Case Pressure

Not to exceed. . . . . 5 PSI

Handwheel Data

Approximate displacement for one turn of handwheel is .32 in.^{3}/rev. for 10 size and .5 in.^{3}/rev. for 15 size.

Filtration

Pressure or Return Line . . . . 10 Micron Absolute or Less
Inlet. . . . . 149 Microns

Fluids

Clean petroleum antiwear industrial hydraulic oil, water-in-oil emulsions, HWBF, water glycol, phosphate esters, or automotive crankcase oil designated SC, SD or SE. Running viscosity range 70-250 SUS. Operating temeprature 120°F. recommended, 150°F. usual maximum. Refer to data sheet I-286-S for hydraulic fluid and temperature recommendations. Also see fluid life chart 507000

This unit is designed to meet specifications as outlined. To insure maximum unit performance, in conjunction with your specific application, consult your application engineer if your:

Speed is above maximum RPM.

Fluid does not meet the specifications of data sheet I-286-S.

Mounting attitude is other than horizontal.

Model Code

F3 P V B 15 - (F) R S W Y - 30 -*C* (*) (L) - 10

Special Seals (Omit if not needed)

Pump

Variable Displacement

In-line Design

GPM - Rating at 1800 RPM
10 - 10 GPM 15 - 15 GPM

Mounting
F - Foot Bracket (Omit for Flange)
For Separate "Foot Bracket Kit" Order Model FB-B-10

Rotation (Viewing Shaft End)
R - Right Hand
L - Left Hand

Displacement
S - One Side of Center (Pressure Compensator Models)
D - Both Sides of Center (Handwheel & Lever Models)

Optional Ports & Shaft
W - Side Ports (Standard Drive Shaft)
X - Side Ports and Thru Shaft (Standard Drive Shaft)

●Drive Shaft
Y Standard (Omit For Short Shaft)

Pump Design Number
Design Numbers Subject To Change. Installation Dimensions Remain As Shown For Design Numbers 30 Thru 39.

*Control Type
C - Pressure Compensator (250-3000 PSI See Compensator Control Informatio On Front Page.
CM - Pressure Compensator (250-1500 PSI
H - Handwheel
M - Lever
V - No Control

Control Option
C - Adjustable Maximum Displacemen Stop (With Compensator) Not Avai able With Thru Shaft

Compensator Variations
D - Dual-Range (Electric Control (Not Available With Thru Shaft
E - Start-Up Valve
G - Remote Compensator (Use CCC For L.H. Side Port Models)

Other Controls or Options
L - Left Hand Location Viewing Shaft End (H & M Only)

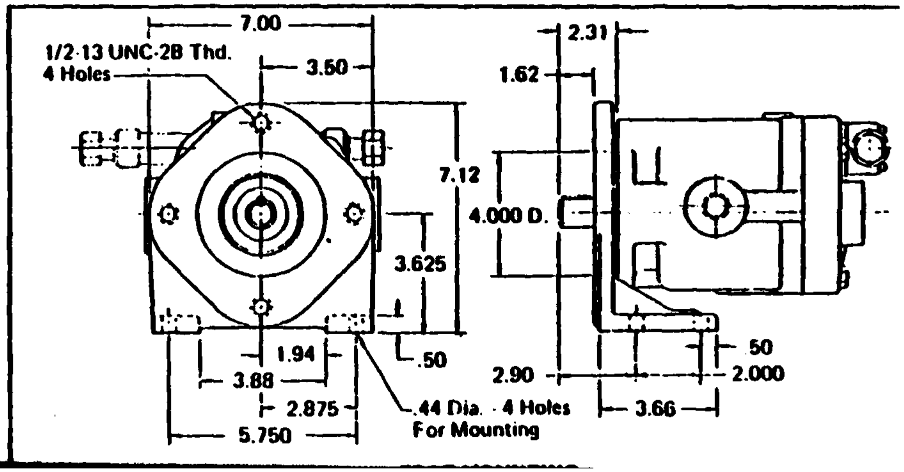
Control Design Number
Design Numbers Subject To Change. Installation Dimensions Remain As Shown For Design Numbers 10 Thru 19 20 Thru 29 For "D" Contro

*Note: Pressure shown defines the minimum adjustable pressure range. It might be possible to mak settings outside of this range.

Note: For thru-drive spline shafts, add S124 suffix to model code.

Weight Lb. (Approx.)

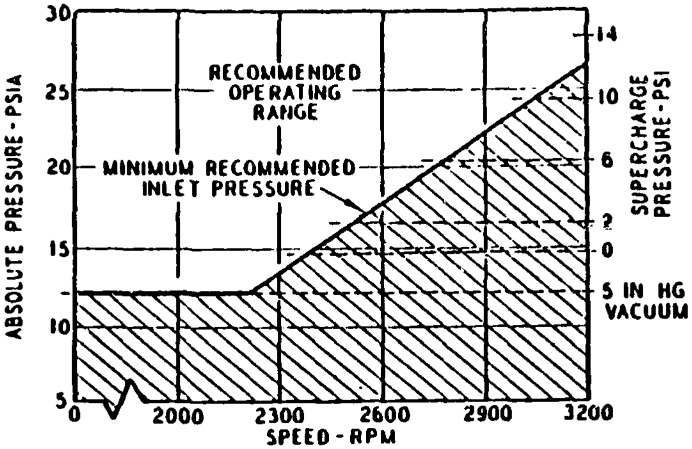
Flange Mounting. . . . .
Foot Mounting. . . . .



TYPICAL PERFORMANCE CURVES

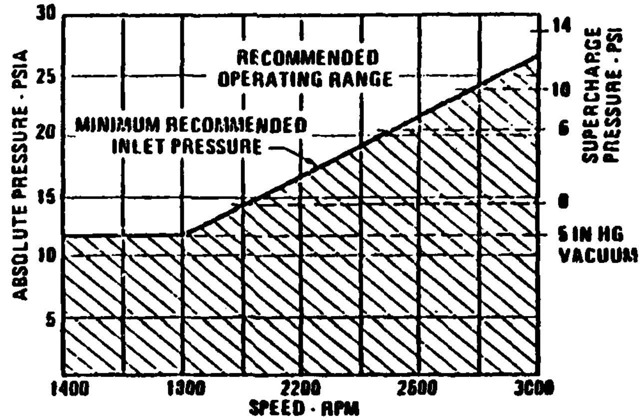
PVB10 SERIES

INLET PRESSURE CURVE
BASED ON OIL TEMPERATURE OF 120° F (100 SUS)
VISCOSITY OF 150 SSU @ 100° F
INLET PRESSURE REQUIREMENT VERSUS PUMP SPEED

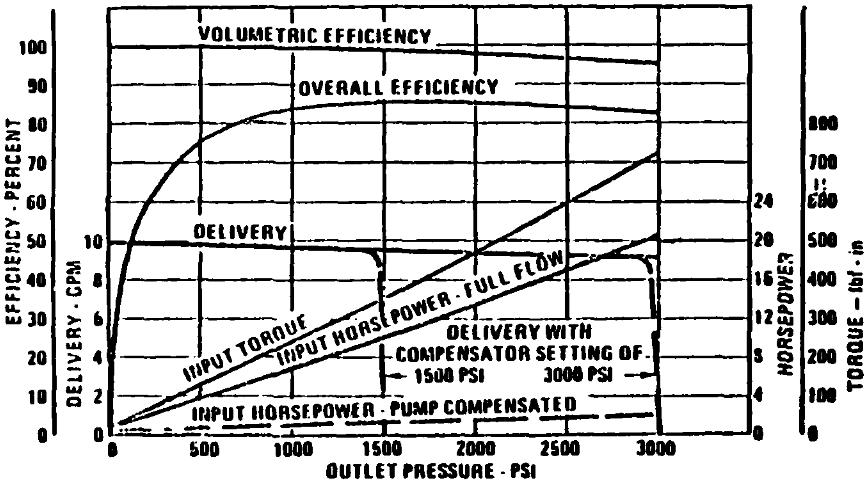


PVB15 SERIES

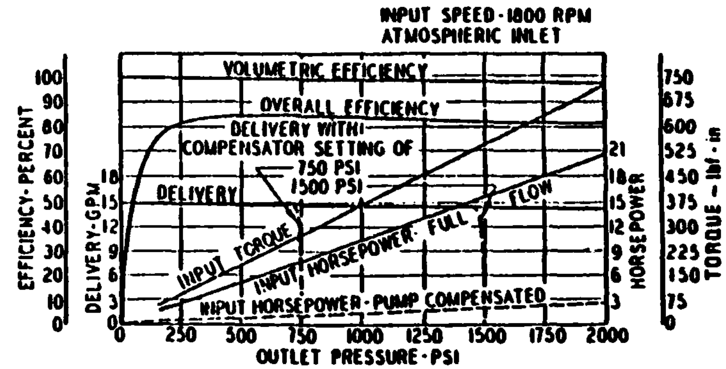
INLET PRESSURE CURVE
BASED ON OIL TEMPERATURE OF 120° F (100 SUS)
VISCOSITY OF 150 SSU @ 100° F
INLET PRESSURE REQUIREMENT VERSUS PUMP SPEED



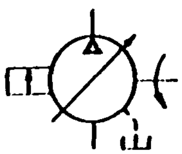
PERFORMANCE CHARACTERISTICS
BASED ON OIL TEMPERATURE OF 120° F (100 SUS)
1800 RPM INPUT — ATMOSPHERIC INLET



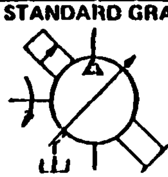
PERFORMANCE CHARACTERISTICS
BASED ON OIL TEMPERATURE OF 120° F (100 SUS)
1800 RPM INPUT — ATMOSPHERIC INLET



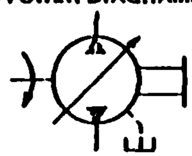
STANDARD GRAPHICAL SYMBOLS FOR FLUID POWER DIAGRAMS



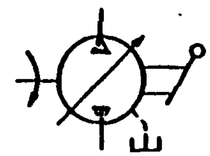
PRESSURE COMPENSATOR



PRESSURE COMPENSATOR
WITH MAXIMUM ADJUSTABLE STOP



HANDWHEEL CONTROL



LEVER CONTROL

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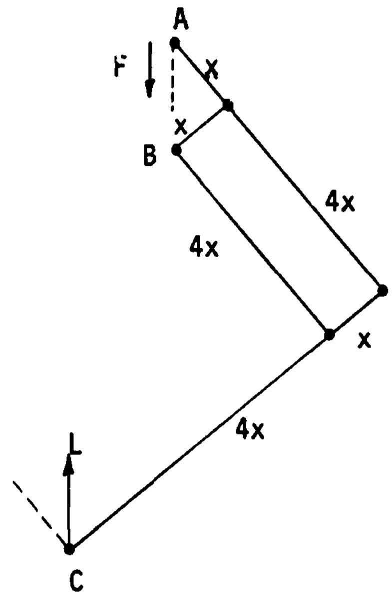
APPENDIX E

DESIGN CONCEPTS FOR THE FOOT-LIFT HYDRAULIC CIRCUIT

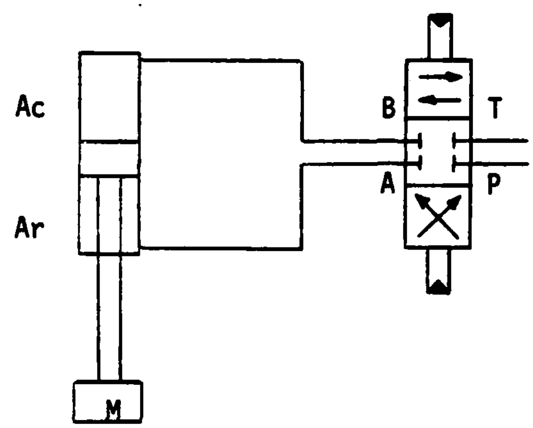
Foot-Lift Circuit Design for Pantograph Leg (Revised)

This figure shows the proportions of the members in the pantograph leg. These proportions result in the displacement at C 4 times that at A. The loads are also multiplied by a factor of four: dC = 4dA, or F = 4L.



Point A is to be moved by a hydraulic cylinder powered by the foot-lift circuit. In this Appendix, several different means of driving point A are considered. It is important to note that the inertia of the system was not considered because those values were not available. In addition, it was assumed that point A is constrained to move vertically only and that this constraint imposes no frictional forces. All calculations are based on raising point C 12 inches in 0.25 seconds and lowering it 12 inches in 0.25 seconds in one cycle.

Method 1



Cylinder: 2-in bore, 1-in rod, 13-in stroke. Cushions on rod end.

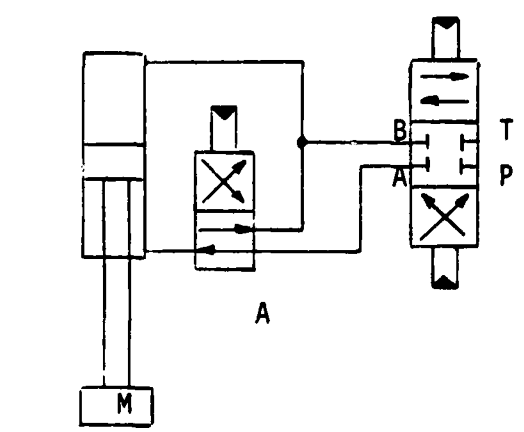
	Load Pressure	Movement Flow	Average Flow
Ac = 3.14 in^2	Low	9.8 gpm	8.6 gpm
Ar = 2.36 in^2	2200 psi	7.4 gpm	

This configuration has the disadvantage of having a large area on the cap end, which requires a large volume of oil to move a distance under no load (just lifting foot).

Using the same configuration with the following cylinder, 1-1/2-in bore, 5/8-in rod, 13-in stroke, cushions on rod end:

	Load Pressure	Movement Flow	Average Flow
Ac = 1.76 in^2	Low	5.3 gpm	4.9 gpm
Ar = 1.46 in^2	3560 psi	4.4 gpm	

Method 2



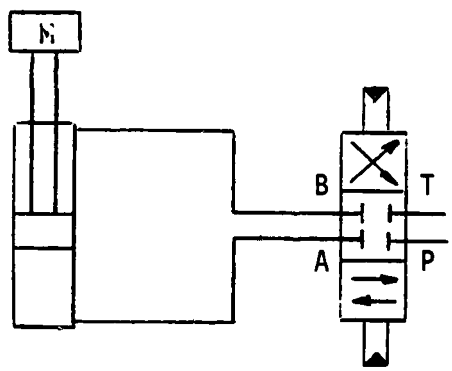
Cylinder: 2-in bore, 1-in rod, 13-in stroke. Cushions on rod end.

This circuit uses a regenerating valve A to help extend the cylinder down (leg up) quickly. This size rod produces a four-fold increase in extension speed.

	Load Pressure	Movement Flow	Average Flow
$Ar = 2.36\ in^2$	2200 psi	7.4 gpm	5.0 gpm
$Ac = 3.14\ in^2$	Low	2.5 gpm	

Two disadvantages result from the addition of valve A: (1) Valve A would have to be a servovalve comparable in size to the main servo so that pressure drops are not too large, and it should be located as close to the cylinder as possible. (2) If Valve A sticks in the shifted position, a load cannot be held on the leg.

Method 3

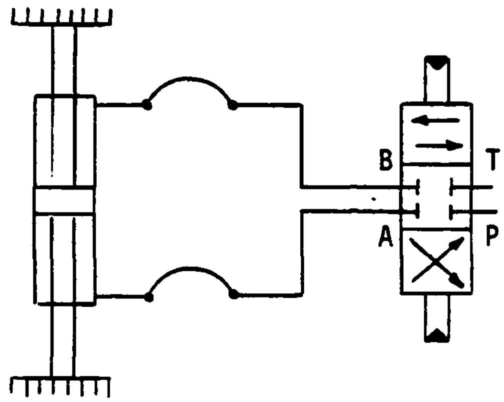


Cylinder: 1-1/2-in bore, 1-in rod, 13-in stroke. Cushions on cap end.

	Load Pressure	Movement Flow	Average Flow
$Ac = 1.76\ in^2$	2950 psi	5.5 gpm	4.3 gpm
$Ar = 0.96\ in^2$	Low	3.1 gpm	

This circuit takes advantage of the smaller rod area, causing the upward foot movement under little load. One disadvantage is that if the circuit sits idle for a long time, leakage will occur across the piston seals, giving pressure intensification. This can be dealt with easily by draining the rod end of the cylinder back to the tank through a small orifice.

Method 4



Cylinder: 2-in bore, 1-in double rod, 13-in stroke. Cushions on one end.

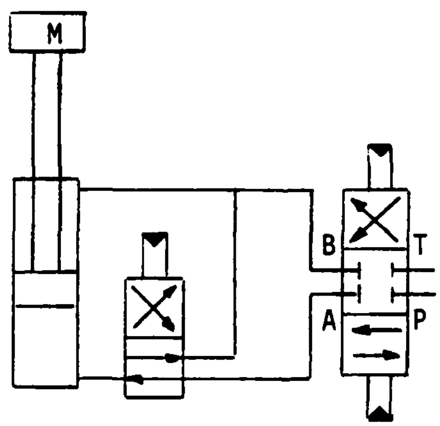
Cylinder moves with point A on pantograph sketch, and rod is stationary.

	Load Pressure	Movement Flow	Average Flow
$Ac = Ar = 2.36\ in^2$	2200 psi	7.4 gpm	7.4 gpm

The only advantage of this circuit is that it would probably make a small, lightweight package.

Method 5

Cylinder: 1-1/2-in bore, 1-in rod, 13-in stroke. Cushions on cap end.



For Regeneration	Load Pressure	Movement Flow	Average Flow
$Ac = 1.76 - 0.78 = 0.$[illegible] in^2	2950 psi	3.1 gpm	3.1 gpm
$Ar = 0.98\ in^2$	Low	3.1 gpm	

This circuit has the same advantages and disadvantages as the other regeneration circuit of Method 2.

SUMMARY OF THE FEATURES OF THE FOOT-LIFT CIRCUIT CONCEPTS

Method	Load Pressure (psi)	Movement Pressure (psi)	Movement Pressure Moog Servo Size (gpm)	Average Movement Flow (gpm)	Estimated Horsepower (hp)
1a	2200	800	10	8.6	4.6
1b	3560	1100	5	5.0	3.66
2	2200	1100	5	5.0	3.66
3	2950	600	5	4.3	1.72
4	2200	675	10	7.4	3.33
5	2950	500	5	3.1	1.0

APPENDIX F

FOOT-LIFT CIRCUIT MODEL

Pressure loss in a pipe (Laminar)

$$\Delta P = \frac{Q \;\; \mu 128 \; L}{\pi \cdot D^4}$$

μ = absolute viscosity, $\frac{\text{lb sec}}{\text{in}^2}$

L = tube length (inches)

D = tube diameter (inches)

Pressure loss in valve

$$\Delta P = \left(\frac{Q}{C_v}\right)^2 SG$$

Cv ~ loss constant

SG ~ specific gravity of fluid

Acceleration of cylinder rod/mass

Acc = A PV/M

A ~ area of piston (in^2)

PV ~ cylinder pressure (psi)

M ~ mass (lbm)

Velocity of cylinder rod/mass

V = Q A

Some assumptions:

Incompressible fluid

Laminar flow

Zero fluid inertia

$P_{valve} = P_{cylinder}$

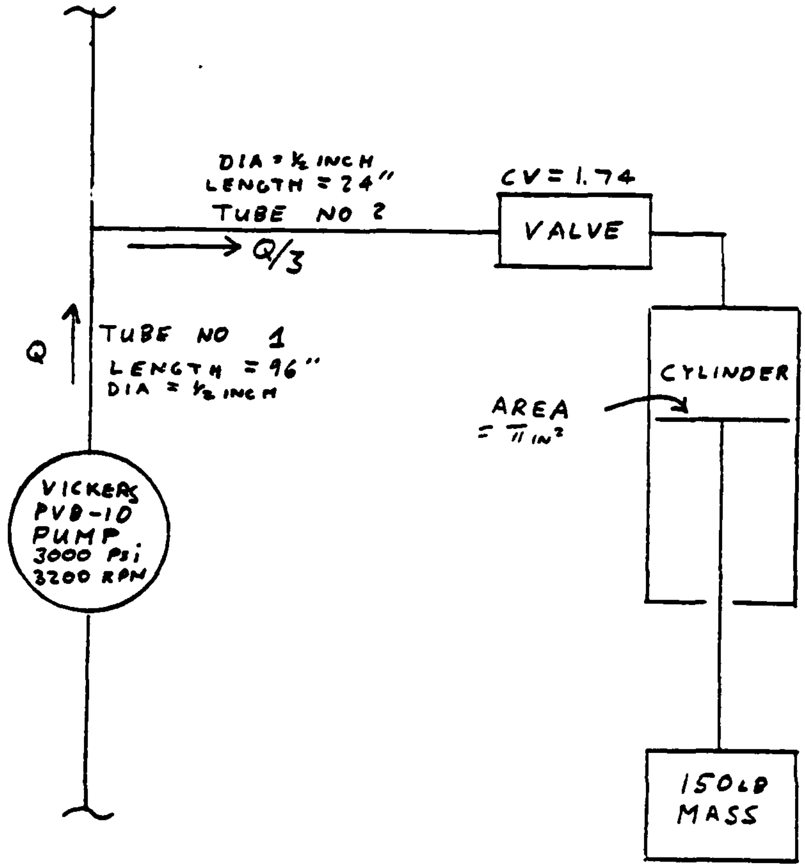


FIGURE F-1. MODELED CIRCUIT

Oil viscosity = 150 S.U.S.

Flow rate

$$Q = (W\ DP\ \emptyset) - K_{pt} PP$$

W ~ rotational velocity (rps)

DP ~ pump displacement per revolution (in^3/rev)

Ø ~ output factor (1.0 = full flow)

K_{pt} ~ total pump leakage $\frac{in^3}{sec\ psi}$

PP ~ pump pressure (psi)

APPENDIX G

HORSEPOWER CALCULATIONS FOR ENERGY - EFFICIENT FOOT-LIFT CIRCUIT DESIGN

Hydrostatic Drive

(See Figure G-1)

Foot-Lift Calculation

Flow is to A port of cylinder 1 (2-in bore with 3/4-in diameter rod)

$A = 2.7\ in^2$ System pressure = 2960 psi

The flow to lift and lower 3 inches in 0.5 sec:

$$Q = \frac{2.7\ in^2 \times 3\ in \times 2\ in}{231\ in^3/gal} \left(\frac{60\ sec/min}{0.5\ sec}\right) = 8.4\ gpm$$

The pressure required for this flow:

$P = P_{drop} + P_{inertial}$

for 150 lb leg, $P_{inertial}$ = 55 psi

estimate P_{drop} = 100 psi

P = 155 psi.

Horsepower needed for the main pump:

hp = 155 psi x 8.4 gpm/1500 = 0.87 hp.

Horsepower needed for the charge pump:

hp = 2 gpm x 150 psi/1500 = 0.2 hp.

Total horsepower = 0.87 + 0.2 = 1.07 hp.

The hydrostatic pump is shifted by a servo-controlled valve to port pump flow to the cylinder.

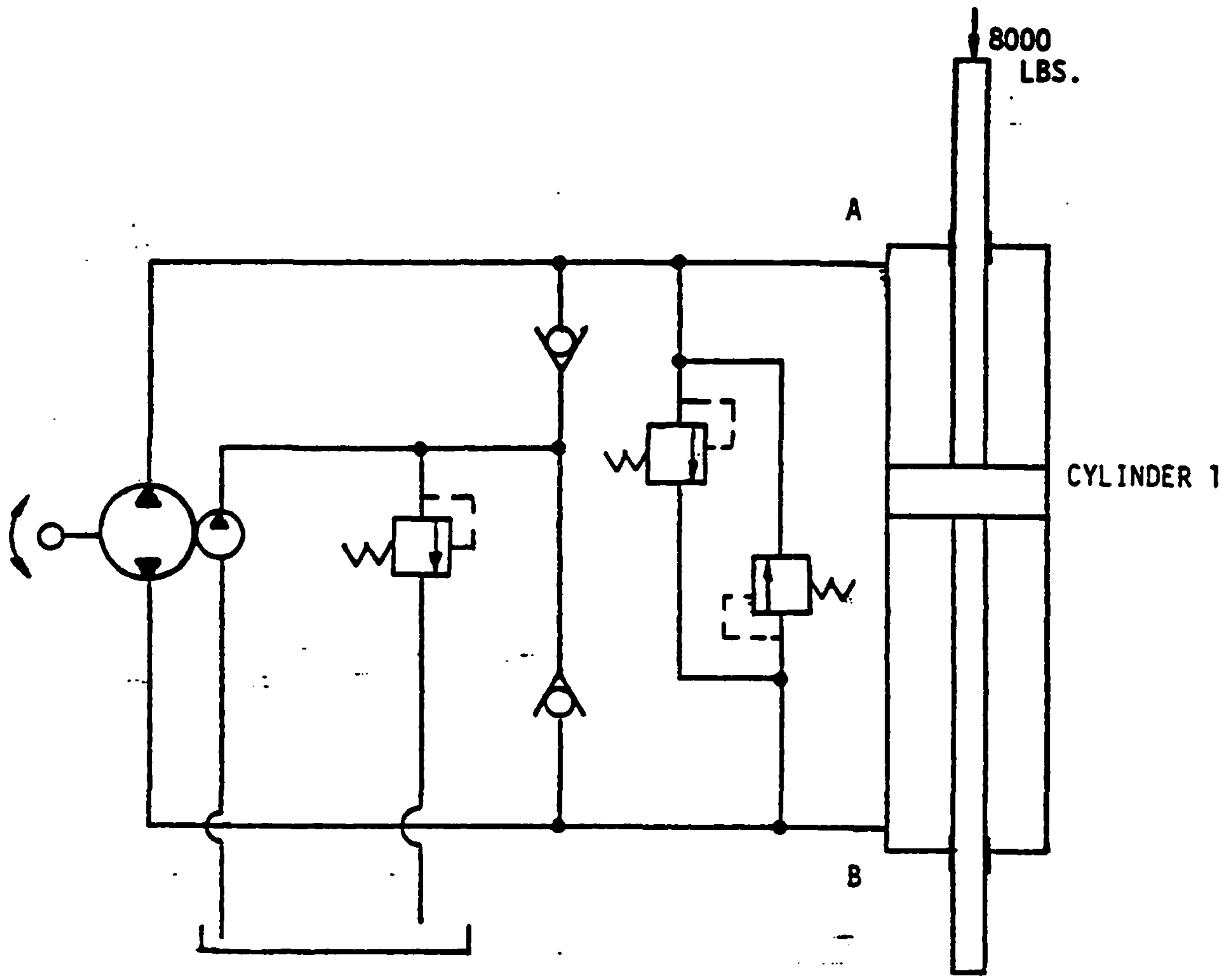


FIGURE G-1. HYDROSTATIC TRANSMISSION CIRCUIT

Placement Horsepower

It is assumed that the foot sinks 1 inch into the ground, so the cylinder must move 1/4 inch in 200 msec.

$$Q = \frac{2.7 \text{ in}^2 \times 0.25 \text{ in}}{231 \text{ in}^3/\text{gal}} \quad \frac{60 \text{ sec/min}}{0.2 \text{ sec}} = 0.88 \text{ gpm}$$

P = 2960 psi for 8 kips, 1480 psi for 4 kips

hp = 2960 psi x 0.88/1500 = 1.73 hp; 0.86 hp

The foot-lift circuit total power will be the sum of these values:

Total horsepower = (1.07 hp/leg)(3 legs) + 1.73 hp/leg + 0.86 hp/leg (2 legs) = 6.7 hp.

Dual-Pump Circuit

(See Figure G-2)

Foot-Lift Calculation

In order to lift and lower the foot in 0.50 sec,

$$Q = \frac{(3.14 \text{ in}^2)\ 3 \text{ in} + (2.36 \text{ in}^2)\ 3 \text{ in}}{231 \text{ in}^3/\text{gal}} \quad \frac{60 \text{ sec/min}}{0.5 \text{ sec}} = 8.57 \text{ gpm}$$

High flow pressure drop:

$$P = \Delta P_{valve} + P_{inertia}$$

$P_{inertia} \approx 50$ psi

$\Delta P_{valve} \approx 350$ psi (Moog 760-104).

P = 400 psi

Horsepower = 400 x 8.57/1500 = 2.3 hp

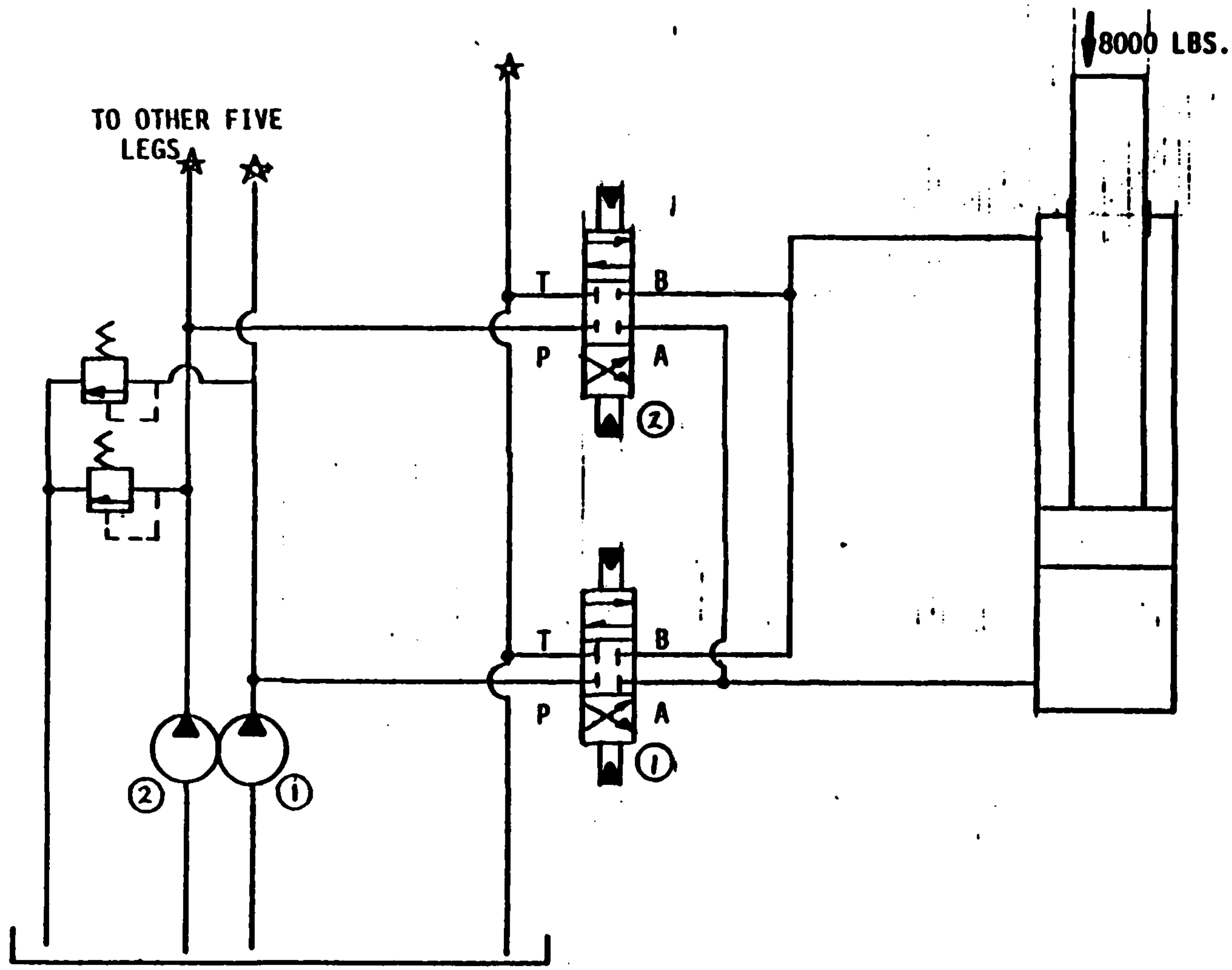


FIGURE G-2. DUAL-PUMP CIRCUIT

Placement Horsepower

$$Q = \frac{(3.14\ in^2)\ 0.25\ in}{231\ in^3/gal} \quad \frac{60\ sec/min}{200\ msec} = 1.0\ gpm$$

$P = 2550$ psi

Horsepower = 2550 x 1.0/1500 = 1.7 hp

Total horsepower = (2.3)3 +1.7(3) = 12.0 hp.

Hydraulic Brake Circuit

(See Figure G-3)

Foot-Lift Calculation

For cylinder 2 with a 3/4-in bore and 1/2-in diameter rod,

$A_C = 0.44\ in^2$

$A_R = 0.24\ in^2$

To move up and down in 0.480 seconds,

$$Q_{lift} = \frac{(0.44\ in^2 + 0.24\ in^2)\ 3\ in}{231\ in^3/gal} \quad \frac{60\ sec/min}{0.480\ sec} = 1.10\ gpm$$

The pump must put out 3000 psi because this is the pressure required to hold the brake units.

Brake Unit

With an 8,000-lb load, and assuming a brake with a coefficient of friction of 0.5, the force of squeeze must be 16,000 lb.

At 3000 psi, the area of the brake must be

$$A_{brake} = \frac{16,000\ lb}{3000\ psi} = 5.33\ in^2$$

Assuming a clearance of 0.010-in and a maximum application time of 20 msec,

$$Q_{brake} = \frac{(5.33\ in^2)\ 0.01\ in}{231\ in^3/gal} \quad \frac{60\ sec/min}{0.020\ sec} = 0.7\ gpm$$

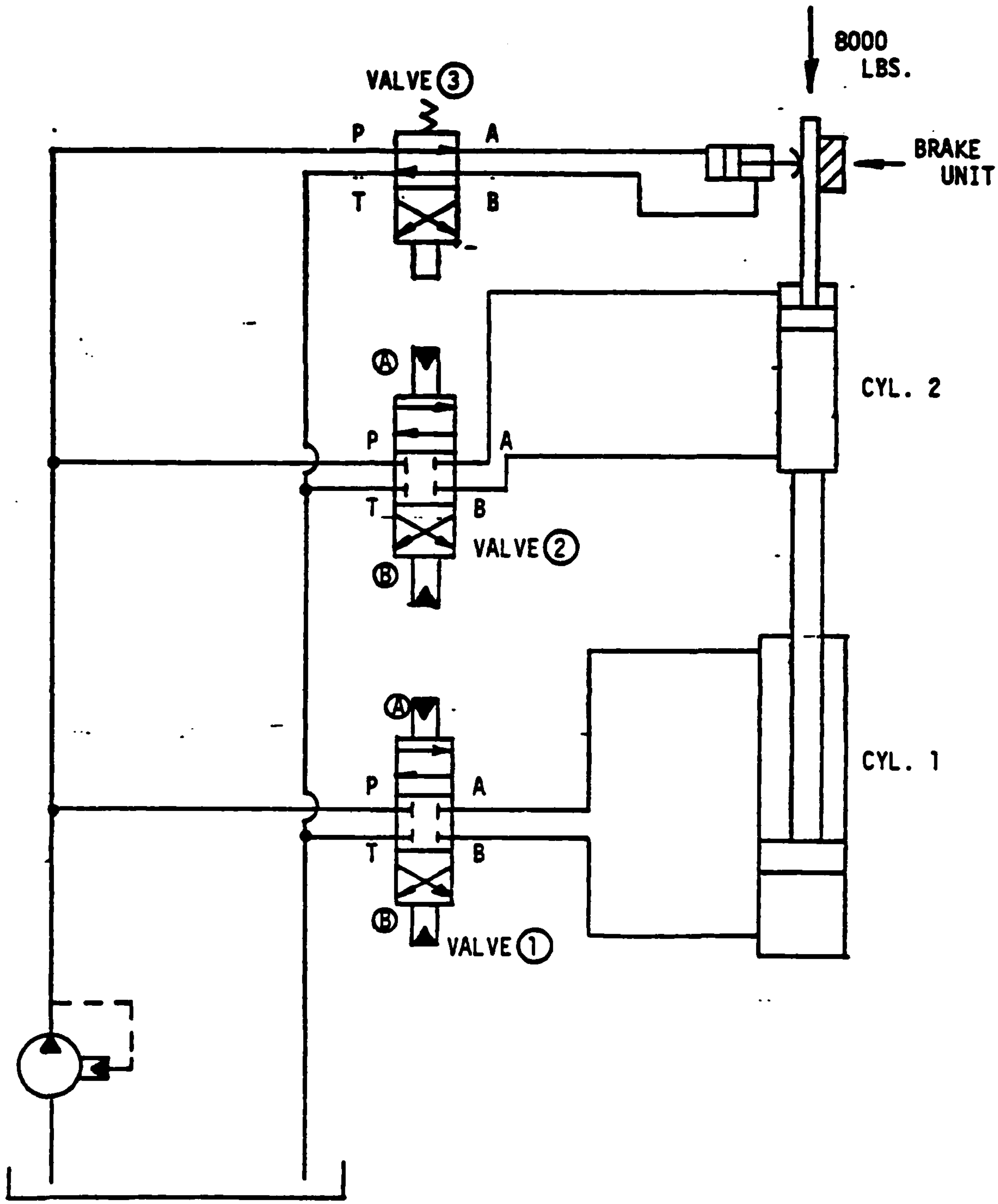


FIGURE G-3. HYDRAULIC BRAKE CIRCUIT

Since Q_{brake} is less than Q_{lift} and they are never being used at the same time,

$$Q_{total} = Q_{lift} = 1.10 \text{ gpm.}$$

Therefore, using the placement horsepower of 1.7 hp, the total power requirement for all systems is

engine horsepower = lift horsepower + placement horsepower

$$= \left(1.10 \text{ gpm} \times \frac{3000}{1500}\right) 3 + (1.7)\ 3$$

engine horsepower = 11.4 hp

Push-Pull Compression Circuit

(See Figure G-4)

Lift-Leg Stroke

$$P = P_{R1} \quad + \quad P_{R1},\ P_{R2} = P$$

Balance Forces

$$A_{R1}\ (P_{R1}) = A_{R2}P + A_{C1}P_{C1}$$

$$P_{C1} = \quad P_{C1} \text{ for flow to tank.}$$

If we pull down 3 inches in 0.25 seconds,

$$Q_{R1} = 3 \text{ in } (0.50 \text{ in}^2/231 \text{ in}^3/\text{gal})\ (0.25 \text{ sec})\ 60 \text{ sec/min} = 1.6 \text{ gpm}$$

$$Q_{C1} = 1.6 \text{ gpm } (A_{C1}/A_{R1}) = 10.05 \text{ gpm}$$

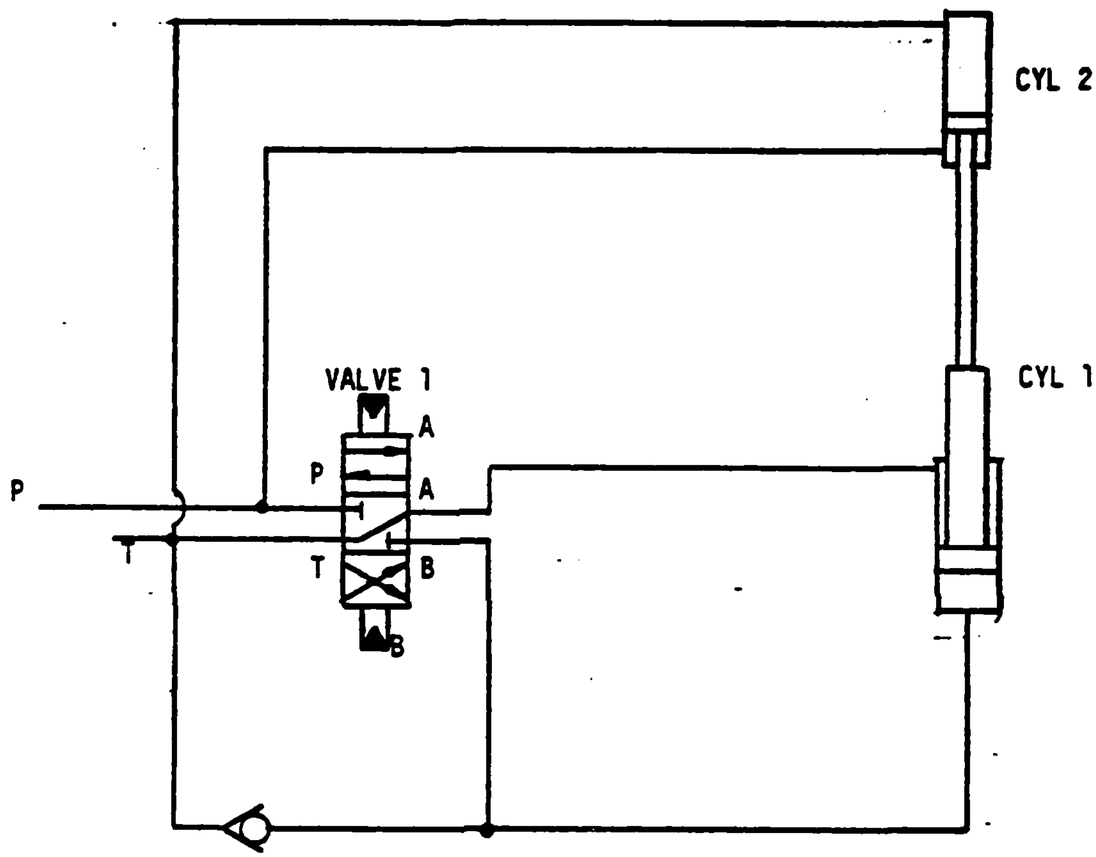


FIGURE G-4. DUAL-CYLINDER CIRCUIT (PUSH-PULL COMPRESSION)

From the Moog valve chart, find ΔP for Model 760-104

ΔP_{R1} = 50 psi (for 1/2 loop flow)

*ΔP_{C1} = 250 psi (for 1/2 loop flow)

From the balance-forces equation,

$$0.5 \text{ in}^2 (P - \Delta P_{R1}) = 0.21 \text{ in}^2 (P) + 3.24 \text{ in}^2 \Delta P_{C1}$$

$$0.29 \text{ in}^2 P = 25 \text{ lbf} + 785 \text{ lbf} = 810 \text{ lbf}$$

$$P = 2800 \text{ psi}$$

Actual pump flow to A_{R1} in this case is less than 1.6 gpm since part of the flow is coming from A_{R2}:

$$Q_{pump} = 1.6 \text{ gpm} \left(\frac{A_{R1} - A_{R2}}{A_{R2}} \right) = 0.93 \text{ gpm}$$

Theoretical horsepower = 0.93 gpm (2800 psi)/1714 = 1.52 hp

Practical horsepower = 0.93 gpm (2800 psi)/1500 = 1.74 hp.

<u>Lower-Leg Stroke</u>

Balance Forces

$$A_{R2} P = A_{R1} P_{R1} + A_{C1} P_{C1},$$

where $P_{R1} = \Delta P_{R1}$ over valve 1 and P_{C1} is a vacuum of $\approx$15 psi

Q_{R1} = 1.6 gpm for 3 inch movement in 0.25 sec.

ΔP_{R1} = 50 psi

$$0.21 \text{ P} = 0.5 \text{ in}^2 (50 \text{ psi}) + 3.14 \text{ in}^2 (15 \text{ psi})$$

$$P = 1/0.21 \text{ in}^2 (25 \text{ lbf} + 47 \text{ lbf})$$

$$P = 343 \text{ psi}$$

*If the check valve is pilot-operated from A port of the valve, ΔP can be lowered below 100 psi, so horsepower can be saved. Mininum P is 2550 psi, so savings are not as great as would be calculated.

The flow necessary for this pressure to P_{R2} is

$$Q_{R2} = 0.21 \text{ in}^2 \times 3 \text{ in}/231 \text{ in}^3/\text{gal} \quad (1/0.25 \text{ sec}) \left(\frac{60 \text{ sec}}{\text{min}}\right) = 6.7 \text{ gpm}$$

Theoretical horsepower = 0.67 (343/1714) = 0.15 hp

Practical horsepower = 0.67 (343/1714) = 0.18 hp.

Fluid must be drawn from the tank line to fill the A_{C1} during this stroke:

$$Q_{AC1} = \frac{(A_{C1} - A_{R1} - A_{C2}) \ 3 \text{ in}}{231 \text{ in}^3/\text{gal}} \left(\frac{1}{0.25 \text{ sec}}\right)\left(\frac{60 \text{ sec}}{\text{min}}\right) = 6.9 \text{ gpm}$$

This flow will have to made up by the supercharge circuit: for three legs, a supercharge flow of 3(6.9 gpm) + 3(0.67 gpm) + 3(height adjust flow = 0) = 29 gpm. This flow may be a problem because it is so much larger than the pump flow of 5 gpm (3 gpm lift + 2 gpm adjust).

Push-Pull Tension Circuit

(See Figure G-5)

Lift Leg

$$P_{R1} = \Delta P_{valve\ 2} + \Delta P_{valve\ 1} \text{ and } P_{R1} = P_{pump} - \Delta P_{valve}$$

Because $P_{valve\ 2} >> \Delta P_{value\ 1}$, ignore $\Delta P_{valve\ 1}$. $Q_{v1} << Q_{v2}$.

Balance Forces

$$A_{R2}\, P_{R2} = P_{R1}\, A_{R1} + A_{C2}\, P_{C2} + A_{C1}\, P_{C1}$$

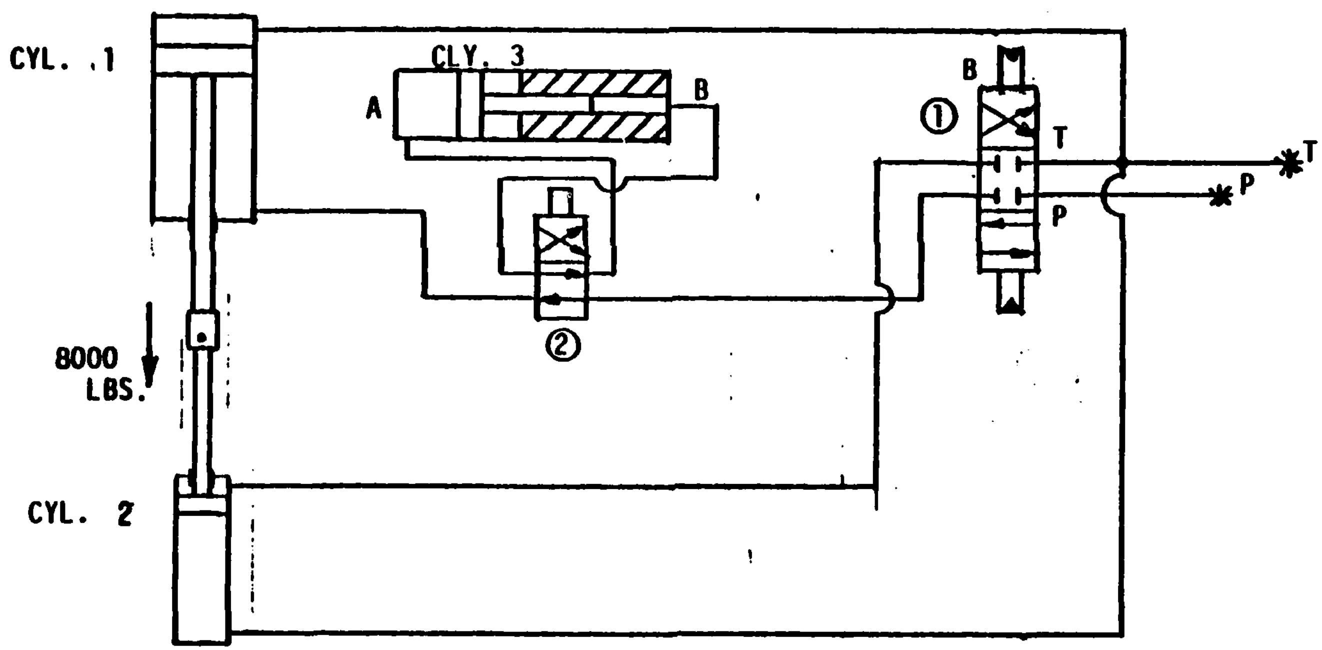


FIGURE G-5. DUAL CYLINDER CIRCUIT (PUSH-PULL TENSION)

P_{C2} <<10 psi by large lines; therefore ignore.
P_{C1} <<15 psi vacuum, therefore ignore.

$A_{R2}\ P_{R2} = P_{R1}\ A_{R1}.$

To move 3 inches in 0.25 sec,

$$Q_{R2} = 3 \text{ in } (0.59 \text{ in}^2) \frac{60}{(231)(.25)} = 1.83 \text{ gpm}$$

$$Q_{R1} = Q_{R2}\ (A_{R1}/A_{R2}) = 8.4 \text{ gpm}$$

$$\Delta P_{valve\ 2} = P_{R1} = 200 \text{ psi} \quad \text{(Moog 760-104)}$$

$$P_{pump} = P_{R2} + P_{inert} = 200 \text{ psi} \left(\frac{A_{R1}}{A_{R2}}\right) + \frac{150 \text{ lb}}{0.59 \text{ in}^2} = 1160 \text{ psi}$$

$$\text{Theoretical horsepower} = \frac{1.83 \times 1160}{1714} = 1.23 \text{ hp}$$

$$\text{Practical horsepower} = \frac{1.83 \times 1160}{1500} = 1.41 \text{ hp.}$$

Lower Leg

Because A_{R3} is chosen to be the same as A_{R2}, the values will be the same; so

$Q_{pump} = 1.83$ gpm
$P_{pump} = 1160$ psi

Horsepower will also be the same.

Storage Cylinder

(See Figure G-6)

Calculations

The system pressure must be above 2550 psi because this is the pressure required to hold an 8,000 lb load on cylinder 4. The areas of cylinder 3 and cylinder 4 are equal and have values of $A_R = 0.40$ in^2 and $A_C = 3.14$ in^2.

Leg-Lift Calculations

Valve 2 has a shift time of 40 msec from unenergized to energized. This delay time should be subtracted from the overall time allowed for the movement:

$$t_{available} = t_A = 250 \text{ msec} - 40 \text{ msec} - 210 \text{ msec}$$

To move a distance of 3 inches, the flow at rod end of cylinder 4 is

$$Q_R = \frac{3 \text{ in x } 0.40 \text{ in}^2}{231 \text{ in}^3/\text{gal}} \left(\frac{60 \text{ sec/min}}{0.21 \text{ sec}}\right) = 1.48 \text{ gpm.}$$

The pressure required will be the flow losses caused by flow Q_C through valve 2 and the pressure required to lift the leg.

$$P_{lift} = \text{Leg wt}/A_R = 100 \text{ lb}/0.40 \text{ in}^2 = 250 \text{ psi}$$

$$Q_C = Q_R\ (A_C/A_R) = 11.6 \text{ gpm.}$$

$$P_C = P_{V2} = 100 \text{ psi } \frac{11.6 \text{ gpm}}{7 \text{ gpm}} = 274 \text{ psi} \quad \text{(from Denison catalog).}$$

Therefore,

$$P_R = P_{lift} + P_C\ (A_C/A_R)$$
$$P_R = 250 \text{ psi} + 2150 \text{ psi} = 2400 \text{ psi.}$$

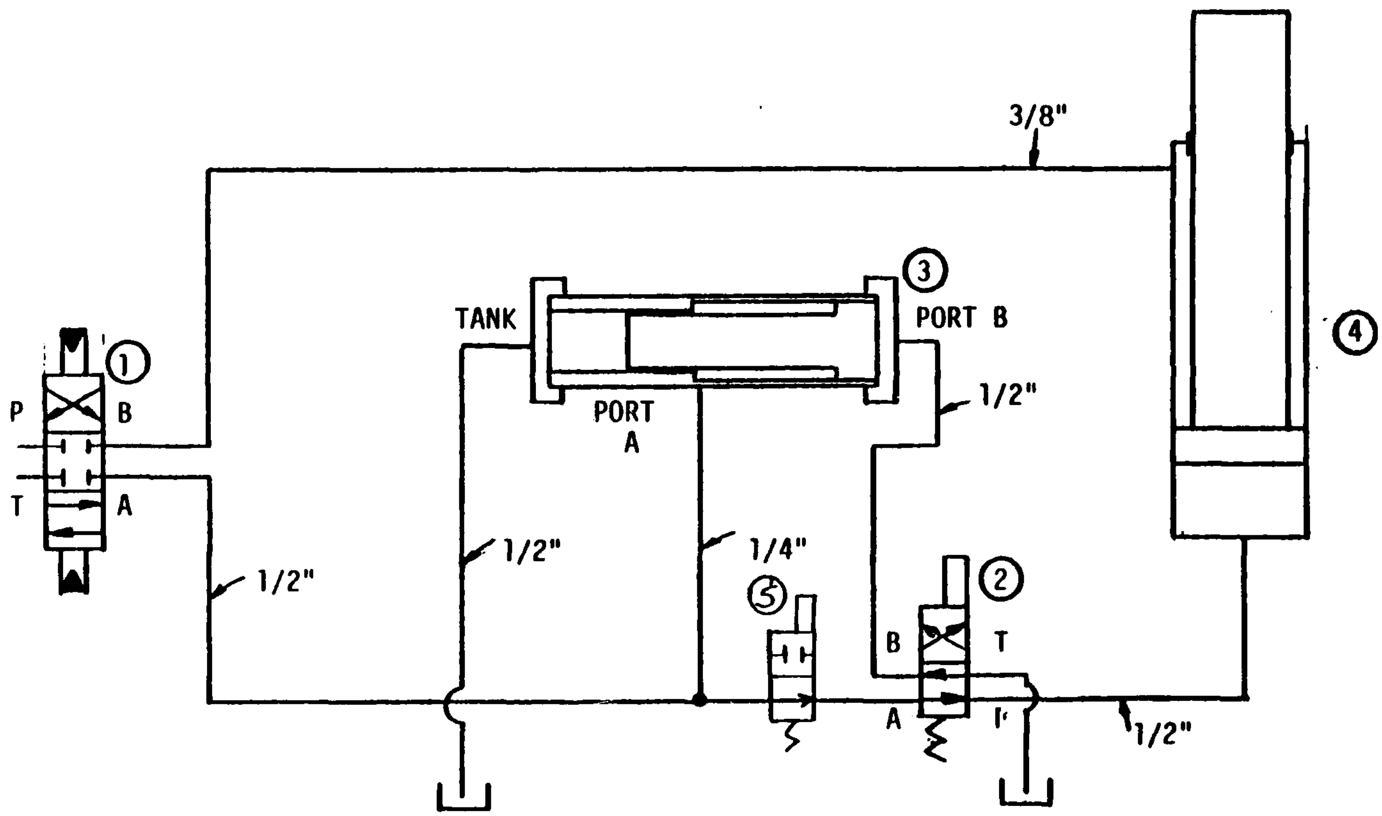


FIGURE G-6. DUAL-CYLINDER CIRCUIT (STORAGE CYLINDER)

The full pressure drop of 2550 psi is required to calculate horsepower per leg, theoretical and practical. So,

Theoretical horsepower = 1.48 gpm (2550 psi)/1714 = 2.2 hp

Practical horsepower = 1.48 gpm (2550 psi)/1500 = 2.5 hp.

Placement Horsepower

In addition to the flow required to lift the leg, flow is required to move the leg downwards under load if the foot sinks into the ground. The maximum distance of foot sinkage is 1 inch so cylinder 4 must move 0.25 inch, but the time of this adjustment is short in the first 200 msec of stroke.

$$Q_S = \frac{3.14 \text{ in}^2 \times 0.25 \text{ in}}{231 \text{ in}^3/\text{gal}} \left(\frac{60}{0.2 \text{ sec}} \right) = 1.02 \text{ gpm}$$

$P = 2550$ psi

Theoretical horsepower = 1.02 x 2550/1714 = 1.51 hp

Practical horsepower = 1.02 x 2550/1550 = 1.73 hp.

System Horsepower

The flow requirements at the peak are

Q_{max} = 1.48 gpm + 1.02 gpm = 2.5 gpm

For a set of opposite legs,

Theoretical horsepower = 3.71 hp

Practical horsepower = 4.23 hp.

Actual power could be less than this if a longer time were allowed for adjusting the load. If the full stroke were allowed for adjustment (500 msec), the following horsepowers are needed per pair of legs:

Theoretical horsepower = 2.7 hp

Practical horsepower = 3.1 hp.

Horsepower Reduction

Overall horsepower could be reduced by operating a two-pump system. The middle set of legs would operate as shown in the calculations because they will see a full load of 2550 psi. The outside sets of legs do not see this full load, so the pressure of the pump must be lost in the valves to compensate. The actual pressure that these legs would work at is 1275 psi. Therefore, the system horsepower would be (theoretical with 200 msec adjustment time):

$$\text{Total horsepower} = 3.71 + 2(3.71)/2 = 7.42 \text{ hp.}$$

A practical figure would be 8.46 hp.

Thermal Transfer Unit	Price ($)	Weight (lb)
M-10	69	6
M-15	78	8
M-20	114	11

Blower Type Air-Oil Coolers (Thermal Transfer Model 200)

These items have a maximum pressure of 150 psi, so the charge circuit should be limited to 100 psi.

ΔP @ 5 gpm = 17 psi

Heat Losses

Assume flow = 5 gpm and $\Delta t = 70^{o}F$ between the ambient air and oil entering, fan speed = 3200 rpm.

$\dot{Q}$ = horsepower dissipated = 4.9 hp

Approximately 1/2 hp is needed to drive the blower. Therefore, for two blowers,

$\dot{Q}$ = horsepower dissipated = 9.8 hp (or approximately 1 hp per gpm).

Horsepower required by engine = 1.0 hp.

For fast mode operation,

flow = 10 gpm and $\Delta T = 100^{o}F$.

$\dot{Q}$ = horsepower dissipated = 8 hp.

For two coolers, $\dot{Q}$ = 16 hp.

Engine horsepower = 1.0 hp.

APPENDIX H

HEAT DISSIPATION CALCULATIONS

Tank Dissipation

General rule for tank heat dissipation:

$$Q = 1.0 \times 10^{-3} \times \Delta T \times \text{Area (sq ft)}$$

For ambient air at $80^{o}F$ and a tank temperature of $140^{o}F$,

$$Q = 7.0 \times 10^{-2} \times \text{Area}$$

For a 20 gal square tank 4 inches high, A= 22 sq ft and
$Q = 1.5$ hp (circulation only).

If a breeze can be considered to be on one side of the tank, then dissipation can be increased by 25%, and
$Q = 1.9$ hp.

Heat Exchangers (Thermal Transfer (300 psi))

The heat exchanger is to be mounted so a breeze can pass over the face of the unit. Assume flow = 10 gpm and $\Delta T = 70^{o}F$

Breeze (mph)	Breeze (fpm)	$\dot{Q}$ (hp) M-10	$\dot{Q}$ (hp) M-15	$\dot{Q}$ (hp) M-20
1	88	1.7	1.8	2.0
2	176	2.0	2.1	2.5
3	264	2.20	2.8	3.2
5	440	2.5	3.4	4.2
7	616	3.0	3.7	4.5
10	880	3.4	4.0	4.8

Thermal Transfer Unit	Price ($)	Weight (lbs)
M-10	69	6
M-15	78	8
M-20	114	11

Blower Type Air-Oil Coolers (Thermal Transfer Model 200)

These items have a maximum pressure of 150 psi, so the charge circuit should be limited to 100 psi.

P @ 5 gpm = 17 psi

Heat Losses

Assume flow = 5 gpm and $\Delta t = 70^{o}F$ between the ambient air and oil entering, fan speed = 3200 rpm.

$\dot{Q}$ = horsepower dissipated = 4.9 hp

Approximately 1/2 hp is needed to drive the blower. Therefore for two blowers,

$\dot{Q}$ = horsepower dissipated = 9.8 hp (or approximately 1 hp per gpm).

Horsepower required by engine = 1.0 hp.

For fast mode operation,

flow = 10 gpm and T = $100^{o}F$.

$\dot{Q}$ = horsepower dissipated = 8 hp.

For two coolers, $\dot{Q}$ = 16 hp.

Engine horsepower = 1.0 hp.