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DIGITAL CONTROL OF THE HYDRAULIC ACTUATORS OF AN ADAPTIVE SUSPENSION VEHICLE

A Dissertation

Presented in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy in Mechanical Engineering

by

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VITA

Joseph Alan Dworak was born in North Platte, Nebraska on April 16, 1954. He graduated as an Honors Scholar with a B.S. in Mechanical Engineering from the University of Missouri-Columbia in 1976. As an undergraduate he was a member of Pi Tau Sigma and Tau Beta Pi. He enrolled in graduate studies in the Department of Nuclear Engineering at the University of Missouri-Columbia and was awarded an EPRI Fellowship. His graduate work concentrated on experimental methods in the area of thermal fluids. Upon completion of his masters degree in 1978, he accepted a position with Battelle Memorial Institute in Columbus, Ohio. In 1980 he began his Ph. D. studies in the Department of Mechanical Engineering at Ohio State and was awarded a Battelle Fellowship. His studies concentrated in the areas of system modeling and analog and digital control. His dissertation involved the application of these techniques to robotics. He is currently employed with the Advanced Technology Section of General Electric in Cleveland, Ohio.
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NOMENCLATURE

\( a \) = lead filter time constant
\( \alpha \) = ratio between numerator and denominator lead filter breakpoint frequencies
\( A_i \) = orifice area of valve \( i \) (in\(^2\))
\( A_p \) = piston area (in\(^2\))
\( A_s \) = servovalve constant, ratio of orifice area to current into servovalve (in\(^2\)/amps)
\( b \) = lead filter time constant
\( b_{ii} \) = element of matrix \( B \)
\( B_1 \) = coefficient of damping on actuator 1 (lbf-sec/in)
\( B_2 \) = coefficient of damping on actuator 2 (lbf-sec/in)
\( \beta \) = bulk modulus of fluid (psi)
\( C \) = flow coefficient (dimensionless)
\( d_{ij} \) = element of matrix \( D \)
\( D \) = differential operator with respect to time \( (d(variable)/dt) \)
\( f_i \) = element of matrix \( F \)
\( F_1 \) = external force on actuator 1 (lbf)
\( F_2 \) = external force on actuator 2 (lbf)
\( g_{ij} \) = element of matrix G

\( G_{pp}(s) \) = normalized pump transfer function (dimensionless)

\( G_{sv}(s) \) = normalized servovalve transfer function (dimensionless)

\( G_{lc}(s) \) = compensation plant

\( g \) = compensation plant

\( G_{9c}(s) \) = compensation plant

\( h_{ij} \) = element of matrix H

LHC = left hand chamber

LHS = left hand side

\( k_{ij} \) = element of matrix K

\( k_{1i} \) = element of matrix \( K_1 \)

\( K_{A1} \) = acceleration feedback gain for actuator 1 (amps/(in/sec^2))

\( K_{A2} \) = acceleration gain for actuator 2 (amps/(in/sec^2))

\( K_{C2} \) = flow pressure coefficient for valve 2 ((in^3/sec)/psi)

\( K_{C4} \) = flow pressure coefficient for valve 4 ((in^3/sec)/psi)

\( K_{E1} \) = proportional velocity error gain for actuator 1 (amps/(in/sec))
$K_{E2}$ = proportional velocity error gain for actuator 2 (amps/(in/sec))

$K_{Q2}$ = flow gain coefficient for valve 2 (in/sec)

$K_{Q4}$ = flow gain coefficient for valve 4 (in/sec)

$m_{ii}$ = element of matrix $M$

$M_1$ = reflected inertia seen by actuator 1 (lbf-sec$^2$/in)

$M_2$ = reflected inertia seen by actuator 2 (lbf-sec$^2$/in)

$P_i$ = valve inlet pressure (psi)

$P_o$ = valve outlet pressure (psi)

$P_1$ = pump outlet pressure (pressure in volume $V_1$) (psi)

$P_{11}$ = fluid pressure in volume $V_{11}$ (psi)

$P_{12}$ = fluid pressure in volume $V_{12}$ (psi)

$P_{21}$ = pressure in volume $V_{21}$ (psi)

$P_{22}$ = pressure in volume $V_{22}$ (psi)

$P_L$ = fluid pressure in volume $V_2$ (and $V_L$) (psi)

$P_R$ = return pressure (psi)

$q_{ij}$ = element of matrix $Q$

$Q$ = pump volumetric flow rate ($in^3$/sec)
\( Q_{11} \) = volumetric flow rate through LHS of valve 1 (in³/sec)

\( Q_{12} \) = volumetric flow rate through RHS of valve 1 (in³/sec)

\( Q_2 \) = volumetric flow rate through valve 2 (in³/sec)

\( Q_{31} \) = volumetric flow rate through LHS of valve 3 (in³/sec)

\( Q_{32} \) = volumetric flow rate through RHS of valve 3 (in³/sec)

\( Q_4 \) = volumetric flow rate through valve 4 (in³/sec)

\( \rho \) = fluid density (lbf·sec²/in⁴)

RHC = right hand chamber

RHS = right hand side

\( s \) = Laplace operator (sec\(^{-1}\))

\( \tau \) = lead filter time constant

\( T \) = sampling interval (sec)

\( V_1 \) = fluid volume between pump outlet and inlet of valves 1 and 2

\( V_{11} \) = fluid volume between valve 1 and actuator 1, including LHC of actuator 1 (in³)

\( V_{12} \) = fluid volume between valve 1 and actuator 1, including RHC of actuator 1 (in³)
\( V_2 \) = fluid volume between outlets of valves 1 and 2 and inlets of valves 3 and 4 (in\(^3\))

\( V_{21} \) = fluid volume between valve 3 and actuator 2, including LHC of actuator 2 (in\(^3\))

\( V_{22} \) = fluid volume between valve 3 and actuator 2, including RHC of actuator 2 (in\(^3\))

\( V_P \) = fluid volume between pump outlet and valve 2 and valve 1 and including LHC of actuator 1 (in\(^3\)) = \( V_1 + V_{11} \)

\( V_L \) = fluid volume \( V_2 \) plus RHC of actuator 1 and LHC of actuator 2 (in\(^3\)) = \( V_2 + V_{12} + V_{21} \)

\( X_i \) = position of actuator i (in)

\( \dot{X}_i \) = velocity of actuator i (in/sec)

\( \ddot{X}_i \) = acceleration of actuator i (in/sec\(^2\))

\( \dot{X}_M \) = maximum desired velocity (in/sec)

Subscripts

\( C \) = commanded value

\( C,B \) = commanded value using unequal desired velocity model matching algorithm

\( C,LF \) = commanded value after lead compensation
C,P = commanded value using proportional control
C,I = commanded value using integral control
C,PI = commanded value using proportional and integral control
D = desired value
e = equilibrium value
p = perturbation variable
Cp = perturbation in commanded value
Dp = perturbation in desired value
Mp = perturbation in maximum desired velocity
X% = X percent of fully open bypass valve
Chapter 1

INTRODUCTION

1.1 Description of the ASV Project

Throughout history man has used animals or machines as means of transportation. For land locomotion the two devices are vastly dissimilar; animals use legged locomotion while mechanized vehicles use wheels or tracks to provide motion. Since one half of the Earth's surface is inaccessible \cite{3} using wheeled or tracked vehicles but is easily accessible to animals using legged locomotion it would be desirable to construct a machine which has the capability for legged locomotion. Until recently two problems have prevented the practical construction of such a vehicle. The first major problem concerns the machine's capability to maneuver over uneven terrain. For static stability this capability requires a minimum of four legs with three independent actuators or degrees of freedom per leg. This places great demands on the coordination controller of the machine, and for practical purposes, excludes manual
control. The recent introduction of small, powerful microprocessors makes the automatic coordination control of the large number of actuators feasible.

The second major problem concerning mechanized legged locomotion is the autonomous, efficient and well controlled delivery of power to the legs. The power supply of a mobile vehicle should be self-contained and contend with the large number of independent actuators as opposed to the one or two on most wheeled or tracked vehicles. Experimental legged vehicles that have been built to date possess very poor mechanical efficiency. This dissertation will investigate a microprocessor controlled hydraulic power system which appears to be an efficient, reliable and reasonably controllable method of supplying power to the actuators of a legged walking machine.

In 1981, the Departments of Mechanical and Electrical Engineering at The Ohio State University contracted to design and build an Adaptive Suspension Vehicle (ASV) for The Defense Advanced Research Projects Administration. The proposed ASV will be a six-legged walking machine (See Figure 1-1) including the following capabilities:

1. The ASV will have an on board power supply,
Figure 1-1: Adaptive Suspension Vehicle
propulsion unit and all necessary equipment to control the vehicle motion during walking. In short, the ASV will be an independent self-contained unit.

2. In addition to the above equipment, the ASV will be able to support a driver and a 500 pound payload.

3. The ASV will have a maximum speed of 8 mph over smooth even terrain and be able to traverse up slopes of 60% grade at 2.5 mph.

As shown in Figure 1-2, the proposed leg design for the ASV has three degrees of freedom. They are referred to as the main drive movement, the shank movement and the abduction-adduction movement. Linear hydraulic cylinders are used as the actuators. The shank actuator controls vertical movement and will lift the foot during the return phase of the leg and also when it is necessary to avoid obstacles. The abduction-adduction actuator controls the side-to-side movement of the leg assembly and will be used in crab walking and turning maneuvers. The main drive actuator controls the horizontal forward-backward motion of the vehicle and is
Figure 1-2: Leg Design and Actuator Location for Adaptive Suspension Vehicle
primarily responsible for providing the "muscle" for the vehicle motion.

The energy and speed requirements of the vehicle dictate that hydraulics be used to supply power to the actuators. It is the purpose of this dissertation to investigate the dynamic characteristics of a hydraulic circuit which has possible applications in providing power for and controlling the main drive movement.

1.2 Description of Vehicle Modes

The proposed operational modes for the ASV are described by McGhee [38]. These modes are distinguished by varying degrees of human and computer control of vehicle motion. The ASV will utilize different gaits for different modes. Among the modes mentioned by McGhee, the following are considered important to the topic of this dissertation and are described below.

1.2.1 Precision Footing

In this mode, the operator will be able to directly control individual leg positions by means of a joystick, keyboard, or other interface. The operator will have the primary responsibility for vehicle stability and motion but will be assisted by information provided by the
on-board computer. As implied by the mode name, accurate leg positioning may be required. Fuel economy is of secondary importance in this mode.

1.2.2 Cruise

This mode will be the most efficient mode of operation with respect to fuel economy. It will be used for locomotion over reasonably smooth and level terrain. Since this mode will be used a majority of the time, limb motion coordination should be inherent in the hydraulic system, requiring a minimum of computer and driver supervision. The operator will input the desired forward speed. A motion planning computer will monitor leg positions and generate the appropriate desired horizontal leg velocities based on vehicle stability requirements and desired vehicle forward speed. If all three actuator velocities are controlled, a Jacobian matrix can be used to control leg motion [48]. If the Jacobian matrix technique is used, the position of each actuator will be monitored and used to constantly update the Jacobian elements. Based on the value of the Jacobian elements, the actuator positions will directly influence the desired velocity reference inputs to the individual leg controllers. A description of the ASV computer hierarchy is presented by McGhee [39].
This dissertation will be concerned with achieving the desired actuator velocities generated by the higher level motion planning computer using the Jacobian technique. The feedback control loops will be closed by an individual computer on each leg. The suitability of closing the feedback loops of each leg with a single board microcomputer will be evaluated in this dissertation. As indicated above, the performance of each loop will be monitored by the motion planning computer [39] and will influence the commands sent to the individual leg controllers.

In Cruise Mode, the vehicle will use a tripod gait in walking. A tripod gait is a periodic stride in which three legs are in contact with the ground at all times. For example, the middle leg on one side of the vehicle (e.g. leg 4 in Figure 1-1) and the front and back legs on the other side of the vehicle (e.g. legs 5 and 1 in Figure 1-1) may be in contact with the ground. The remaining three legs (legs 2, 3, and 6 in Figure 1-1) will be in the process of returning to the starting point of the contact phase of the stride. The average returning leg velocity is the same as the average contact leg velocity, resulting in a duty factor of one-half.
1.3 Alternative Main Drive Hydraulic Circuits

In the literature concerning hydraulic actuator control, two methods are usually advocated: valve control or pump control [7, 13, 36, 37, 42, 61]. In the following sections, both of these methods are evaluated for the main drive circuit. It will be seen that both control schemes have attractive features but also some have serious problems. The obvious design step would be to combine the two circuits to incorporate the advantages of both and avoid the disadvantages. This step has been taken and two hybrid schemes are presented and analyzed in this chapter. The first hybrid circuit is a combination which appears to be ideal for tripod gaits. Unfortunately it is shown to have problems in meeting the requirements of the other modes and gaits. The second hybrid circuit has been designed to correct the deficiencies of the first circuit. The analysis and subsequent experimental evaluation of this hybrid circuit forms the subject of this dissertation.
1.3.1 Valve Controlled Circuit

A possible valve controlled hydraulic circuit for the ASV is illustrated in Figure 1-3. In this circuit, each side of the vehicle is supplied by a constant displacement pump. Each pump provides fluid to three linear double ended pistons. One piston provides the main drive movement for a single leg. A servovalve governs the amount of flow to each cylinder and controls leg motion. The volumetric flow rate of each pump will be constant and proportional to the sum of the three largest desired piston velocities on each side of the ASV. In analyzing this circuit, it can be seen that the pressure at the pump outlet will increase to a value high enough to support the highest load across an individual cylinder in the circuit. If another cylinder in the circuit is subjected to a lower load, the pressure difference between the low and high loads must be dropped across the servovalve, thus wasting energy. This situation would exist in a tripod gait when the ASV is walking uphill. The front and back legs would be loaded when they are in contact with the ground, while the middle leg is unloaded and returning to the starting point of the stride. Energy efficiency is an important consideration for a vehicle which must carry its own
Figure 1-3: Servovalve Controlled Hydraulic Circuit
power supply. This potential for high energy losses for unequal leg loads is partially offset by the precise controllability and good dynamic performance offered by the primary controller, the servovalve. In a closed loop position feedback system, low position errors and closed loop velocity bandwidths of 60 Hz are possible for legs weighing up to 100 pounds and using medium size actuators [42].

As mentioned above, the legs are coupled by the fact that the pump outlet pressure will be determined by the largest load across a piston. Individual leg control is accomplished by servovalve control. However, a servovalve movement in one leg will affect the motion of the other legs. The affected legs will require some servovalve action to compensate for this disturbance.

1.3.2 Hydrostatic Circuit

A hydrostatic or variable displacement pump controlled hydraulic circuit is shown in Figure 1-4. Each leg would have a double ended piston directly controlled by a variable displacement pump. This type of pump varies the pump displacement and hence the flow rate in proportion to an electrical input signal. This electrical input would be proportional to the desired actuator velocity. Note that when a variable
Figure 1-4: Hydrostatic Circuit
displacement pump is used to control the motion of a hydraulic piston the flow out of the piston returns to the pump instead of a reservoir. This plumbing configuration is commonly referred to as a closed hydraulic circuit.

In this hydraulic power configuration, six pumps would be required instead of the two in the valve controlled circuit. However, these pumps would be smaller and weigh less because they only have to provide flow to one cylinder. Each pump would deliver the flow rate necessary to obtain the desired piston velocity. The pump outlet pressure increases to whatever value is necessary to overcome the load. Thus, essentially all the fluid power developed by the pump is used to move the load, resulting in a highly efficient system. However, the bandwidth of the hydrostatic configuration is relatively low when compared to a valve controlled system. This low bandwidth makes the hydrostatic system questionable for use for the ASV main drive circuit. The low bandwidth is primarily caused by the relatively large inertias and compressible volumes found in the pump and actuator. In contrast, when using valves for control, the only appreciable inertias found in the system are those of the piston and load. Also, due to size and weight considerations, the variable
displacement pump must often be located at a comparatively longer distance from the actuators. The hydraulic lines between the pump and actuator constitute compressible volumes which act as dynamic energy storage elements and decrease the system response speed. A servovalve's small size and weight allows it to be located much closer to the actuator, thus reducing the connecting line lengths.

Since each leg has its own pump and cylinder arrangement, the leg control circuits are effectively isolated from one another, allowing individual control. The circuits are however coupled via the mechanical drive that powers all the pumps. In a tripod gait, in which it is desirable for all legs to be moving at the same speed, the motion planning computer would have to continually monitor all six leg velocities and modulate individual velocities to ensure a coordinated control. In this respect the hydrostatic system requires more coordination at the computer level since limb motion coordination is not inherent in the hydraulic circuit design.
1.3.3 Series Circuit

From the discussions above it should be evident that both valve and pump control circuits have desirable features which could be incorporated in the ASV main drive hydraulic circuit. A hybrid system, shown in Figure 1-5, was proposed [58, 63] in order to combine the high dynamic response of the valve controlled system with the relatively high efficiency of the hydrostatic system. Again, each side of the vehicle has a variable displacement pump. The fluid leaves the pump outlet and flows to the servovalve-controlled piston of leg 5 to produce some leg motion. The fluid leaving the piston flows back through the servovalve and travels to the valve-cylinder arrangements of legs 3 and 1. The fluid then returns to the pump intake, completing the circuit.

Leg control will be implemented using a combination of pump and servovalve control. Specifically, the pump will be operated as an electronically pressure compensated pump. The pressure setting is slightly higher than the maximum anticipated load requirement and is continually adjusted by the computer. The servovalves are the primary controllers in the closed velocity loops and will be used to "trim" small velocity fluctuations due to unequal dynamic loading or slightly
Figure 1-5: Series Circuit
different cylinder velocity requirements. The high dynamic response of the servovalve will result in a high controller bandwidth.

In evaluating the efficiency of the series circuit it is important to note that the pump outlet pressure must be high enough to support the total load across all three cylinders on each side of the vehicle. A pressure drop occurs across each cylinder which is proportional to the load on the cylinder. Since the available supply pressure is only slightly higher than the required load pressure the corresponding energy losses are minimal, unlike the valve controlled circuit.

Recall that in tripod gait it is desirable for all legs to move at approximately the same speed. In this series circuit, since the fluid flows from the outlet of one cylinder to the input of the other cylinder, the velocities of the cylinders will be nearly the same. Also, the series arrangement handles the problem of distributing the drive load on one side of the vehicle, eliminating the need for load balancing control.

Another advantage of the series circuit in this situation is that the volume displacement of the pump is approximately one-third that of a pump and three cylinders arranged in the traditional parallel valve control configuration discussed previously. In the
parallel configuration, the pump must supply enough fluid to move three separate cylinders. Using the series circuit, the pump must only supply the requirements of one cylinder. However, the series circuit pump pressure must be approximately three times higher than the pump used in the parallel configuration. Since the physical characteristics of the pump are nominally proportional to the displacement, using the series circuit reduces the physical size and weight of the required pump.

A theoretical frequency response study of this circuit [58] revealed that the steady state accuracy and compliance of closed loop velocity feedback circuits coupled by the series circuit were identical to those of a single closed velocity loop. However, for equal proportional gains, the closed loop bandwidths of the series circuit were lower than for individual servovalve controlled actuators.

To summarize, the series circuit is an attempt to combine the desirable aspects of pump and valve control in order to meet the requirements of the ASV in tripod gait. The circuit incorporates some leg coordination in its design, thereby requiring a reduced degree of computer supervision. To provide the high efficiency necessary for the Cruise Mode the variable displacement pump will be used to supply the slowly varying flow
requirements of the cylinders for changing vehicle speed. Ideally, the servovalves will be nearly wide open during the stride minimizing the pressure drop and hence the wasted energy. They will be modulated to provide the high frequency control for small dynamic velocity adjustments in individual legs.

1.3.4 Series-Bypass Circuit

A major disadvantage of the series circuit is its inability to control individual leg movements independently. Notice that if the servovalve in leg 5 (Figure 1-5) is adjusted to decrease the piston velocity the velocities of legs 3 and 1 will also be decreased. Correspondingly, if leg 1 requires a large velocity, there is no means to accomplish this without also increasing the velocities of legs 5 and 3.

To remedy the above difficulties the series-bypass circuit was proposed for the ASV main drive [63]. This circuit is illustrated in Figure 1-6. Referring to Figure 1-6, it can be seen that the term series refers to the fact that the fluid flows from cylinder-to-cylinder. Bypass is used to describe the parallel flow path across each cylinder. The series-bypass circuit has all the characteristics of the series circuit. In addition, it has the capability to operate each leg
Figure 1-6: Series-Bypass Circuit
individually by bypassing fluid which the cylinder does not require.

It is proposed to control the series-bypass circuit with a combination of the variable displacement pump flow and bypass valve opening. The variable displacement pump will supply the amount of flow rate needed by the cylinder with the highest velocity requirement. If this large flow rate is not required by a leg the bypass valve will open, allowing a portion of the flow to bypass the leg cylinder. At present, the bypass path is envisioned as the primary control element for the cruise mode. For this reason, a servovalve has been specified as the bypass valve. It is felt that the larger power losses which occur with using a servovalve in the bypass path can be justified because the servovalve offers more precise control and faster response. Also, the power loss across the bypass valve is directly proportional to the amount of flow through the bypass path. If the method of control minimizes this flow, then the power loss caused by the larger pressure drops through the servovalve are minimized.

The four way directional valves will be used to reverse leg motion at the ends of the stride but will not be necessary for control in the tripod gait and will be at a constant (maximum) opening at all times during
the stride. To decrease the pressure drop through the four way directional valves, proportional four-way valves will be used. A proportional valve has larger orifice openings than a servovalve, resulting in a decreased speed of control but with a lower pressure drop across the valve. Since these valves will not be used for continuous control of leg motion, controllability can be sacrificed to lower the pressure drop across the valve. These valves were selected instead of directional solenoid valves because proportional valves offer the possibility of controlling leg motion in other modes.

The main objective of this dissertation is to combine the slower but more efficient control furnished by the variable displacement pump with the faster, finer control provided by a servo valve. It is thought that by using this hybrid circuit an energy efficient control scheme with good dynamic performance can be realized.

1.4 Research Issues to be Addressed by This Dissertation

There are three possible methods by which the series-bypass configuration for the ASV can be investigated. The most obvious, time consuming and expensive method would be full scale testing on an ASV built specifically with the series-bypass system. This
method can be rejected on the basis that it makes modification difficult if the hydraulic circuit has serious deficiencies. The second method involves construction of a prototype system which is similar to the proposed system but simpler in design. When the prototype system is subjected to similar loads and desired velocity inputs, it is expected to display characteristics similar to the actual system. The behavior of the actual system in the ASV can be inferred from the prototype behavior. The third method is mathematical analysis and computer simulation of the dynamics of the series-bypass configuration for the ASV. Since the full scale ASV hydraulic system will not be built and tested, more would be learned if the mathematical model described the prototype. Comparison of actual test data to simulation results would suggest whether the modeling techniques used were valid and could be applied in creating a mathematical model of the full scale ASV.

This dissertation is concerned with drive circuit control in the Cruise Mode, utilizing the series-bypass circuit. The techniques developed in Chapter 3 to decouple the actuators in the Cruise Mode may also be used in the Precision Footing Mode. Based on the investigation of the Cruise Mode, recommendations
concerning control of the Precision Footing Mode will be made. The extent of the decoupling in the Cruise Mode will determine whether individual position control is possible in the Precision Footing Mode. The research issues presented by the above problems can be summarized below and are described to a greater degree in the following sections.

1.4.1 Control of Cruise Mode

The nature of the drive circuit control algorithms is first defined. Based on the Cruise Mode requirements, velocity control of the drive actuators is necessary. This control will involve a combination of control of the variable displacement pump and bypass valve opening. In Chapter 2 control algorithms will be formulated which will include both the pump and bypass valves in the velocity control loops. These algorithms will attempt to control piston velocities in the presence of the load and dynamic coupling between drive actuators caused by the series-bypass configuration. The primary control algorithms will be augmented by precompensation controllers which will attempt to decrease this coupling to some degree. The precompensation controller design involves explicit consideration of the multivariable nature of the
controller design problem.

1.5 Scope and Organization of Thesis

This dissertation involves the construction of a prototype hardware system and the associated digital controllers and the use of linearized analysis and computer simulation of the nonlinear prototype system to investigate the dynamic behavior and control of the series-bypass circuit. The mathematical expressions which describe the prototype system are introduced in Chapter 2. Simple actuator velocity control algorithms are also introduced in the same chapter. These expressions can be combined to obtain a set of equations which describe the closed velocity feedback loop dynamics for the prototype system. The nonlinear expressions which describe the prototype system and control are linearized and are used to numerically obtain the small signal frequency responses of the open and closed velocity loops. The linearized equations which describe the open loop prototype system independent of the control algorithms are used to obtain symbolic transfer functions. These transfer functions are analyzed to gain some understanding of the generic behavior of the series-bypass system. A parameter study indicates how the system parameters contribute to the
system behavior.

In Chapter 3 the set of nonlinear equations is used in a time domain ACSL [1] simulation of the prototype system. The computer simulation of the prototype system presented in Appendix B will serve to build a basis for further studies of the whole ASV hydraulic system and if the simulation results agree with empirical data, modifications of the series-bypass configuration can be studied using the simulation exclusively, eliminating the need for modifying the physical hardware. Based on the results from Chapter 2, and ignoring the coupling between the two closed loop feedback systems, the series-bypass circuit is treated as two single-input single-output uncoupled systems in Chapter 3. Classical frequency response methods are used to design closed loop cascade compensators. The effectiveness of these compensators is evaluated by exercising the time domain simulation.

Methods to design controllers and/or compensators for multivariable systems are introduced briefly in Chapter 4. Rosenbrock's Inverse Nyquist Array (INA) technique [2, 21, 22, 23, 24, 40, 41, 47, 50, 51] is chosen as the best method to apply to the series-bypass system. A short review of the theory and actual applications of this technique is provided. The INA
technique is applied to the series-bypass circuit in an attempt to decouple the two circuits and derive controllers based on a decoupled or pseudo-decoupled system.

Based on the simulation and frequency response studies, controller designs are selected and implemented on the prototype system. The empirical data collected from the prototype tests is evaluated in Chapter 5 and compared to the simulation studies and linearized analysis. The prototype system introduced in Chapter 2 and in greater detail in Appendix A will be sufficiently similar to the full scale ASV circuit shown in Figure 1-6 so that recommendations concerning its suitability for use on the ASV could be made. These recommendations and conclusions are presented in Chapter 6.

1.6 General Significance of Dissertation

1. In this dissertation, the potential of the series-bypass circuit configuration to achieve an improved tradeoff between dynamic response and energy efficiency will be fully investigated and control strategy developed. The results will have immediate implications for the ASV operation in the Cruise Mode.
But, more generally, the results will be significant for hydraulic applications requiring control of single or multiple actuators with similar dynamic response and energy efficiency requirements.

2. The usefulness of linearized dynamic system analysis for complex classical controller design tasks will be evaluated. Unique features of this research will be the use of multivariable controller design techniques in an attempt to decouple the controllers in this hydraulic circuit.

3. Included in this research is the evaluation of the suitability of state of the art microprocessors for implementing complex control algorithms for hydraulic circuits.

4. The usefulness of symbol manipulation programs such as FORMAC to assist in obtaining the symbolic transfer functions and formulating the control algorithms for complex controller design tasks will be evaluated.
5. Hydraulic control algorithms will be developed to control actuator velocity and position using a combination of control of a variable displacement pump, a servo bypass valve and a proportional four-way valve.
Chapter 2

SYSTEM EQUATIONS

2.1 Series-Bypass Prototype Description

A simple schematic of the prototype system is exhibited in Figure 2-1. It is described in greater detail in Chapter 5 and Appendix A. Essentially, it consists of two series connected cylinders, each with its own bypass path. As described earlier, the bypass valves are servovalves and the four way valves are proportional valves. The flow for the system is provided by an electronically actuated variable displacement pump. Two separate hydraulic circuits, not shown in the figure, will simulate the axial load seen by the ASV main drive actuators during a stride. Each load circuit consists of an unbalanced area actuator, a constant displacement pump providing flow to the circuit and an electronically adjustable pressure relief valve. This relief valve controls the pressure in the circuit in response to an electronic signal. The load circuit will be calibrated in terms of force (both tension and
compression) versus voltage input to the relief valve. The load circuit is described in detail in Appendix A.

Figure 2-1 also illustrates the definition of some of the symbols which are defined in the nomenclature. These symbols are primarily used in this chapter. Figure 2-2 and Table 2-1 show the line lengths and the line volumes associated with the prototype system. Whenever possible these line lengths were constructed to be as close as possible to the actual lengths that would be needed on the ASV.

To facilitate the study of the system response during a simulated stride and to allow comparisons with computer simulation results, the prototype system and load circuit have been instrumented with various transducers. Strain gage differential pressure transducers are located across both main drive cylinders to monitor differential piston chamber pressure. The velocity and position of each actuator are measured with potentiometers and tachometers respectively. Finally, the force on each piston is measured with a strain gage load cell.

As mentioned previously, the purpose of this dissertation is to design a controller which will effectively control actuator motion. Classical analog controllers for valve controlled cylinders and
Figure 2-1: Prototype Circuit
Figure 2-2: Line Lengths in Prototype System
Table 2-1: Fluid Volumes of Prototype System

<table>
<thead>
<tr>
<th>Segment</th>
<th>Length (in)</th>
<th>Line Diameter (LD) (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A-B</td>
<td>118</td>
<td>.75</td>
</tr>
<tr>
<td>B-C</td>
<td>8</td>
<td>.75</td>
</tr>
<tr>
<td>B-D</td>
<td>8</td>
<td>.75</td>
</tr>
<tr>
<td>E-F</td>
<td>60</td>
<td>.75</td>
</tr>
<tr>
<td>G-H</td>
<td>60</td>
<td>.75</td>
</tr>
<tr>
<td>I-J</td>
<td>20</td>
<td>.75</td>
</tr>
<tr>
<td>J-M</td>
<td>90</td>
<td>.75</td>
</tr>
<tr>
<td>K-L</td>
<td>19</td>
<td>.75</td>
</tr>
<tr>
<td>N-O</td>
<td>60</td>
<td>.75</td>
</tr>
</tbody>
</table>

Cylinder Volume = 32.8 in³

\[
V_P = \{\text{Volumes of (A-B + B-C + B-D + E-F)}\} \left[\frac{\pi \cdot \text{LD}^2}{4}\right] + \frac{1}{2} \text{Cylinder Volume} = V_1 + V_{11} \text{ (in}^3\text{)}
\]

\[
V_L = \{\text{Volumes of (G-H + I-J + J-M + K-L + N-O)}\} \left[\frac{\pi \cdot \text{LD}^2}{4}\right] + \text{Cylinder Volume}
\]

\[
= V_2 + V_{12} + V_{21} \text{ (in}^3\text{)}
\]
hydrostatic systems include proportional and phase lag controllers [42, 61]. Recently, 16-bit microprocessors have been introduced into the commercial market with the ability to add, subtract, multiply and divide sixteen bit integer numbers very quickly. In addition, some of these processors have floating point co-processors which enable the processor to handle real arithmetic, trigonometric, logarithmic and other transcendental functions in real time very quickly. In short, microprocessors have the sophistication necessary to implement complex control algorithms which are similar in affect to analog controllers and which minimize the detrimental effects of sampling.

It has been decided to use this powerful 16-bit computing tool to implement the control for the hydraulic actuators in the prototype circuit. The main advantage of a digital controller when compared to an analog controller in this application is its ability to implement complex control algorithms, change control algorithms or control set points in response to changing inputs or system parameters. After an extensive search of the existing microprocessors and floating point packages commercially available the Intel 8086 CPU [25] was selected. The main features offered by this processor are its high speed real arithmetic
co-processor, the 8087, and higher level language support, in this case, PASCAL.

The 8086-8087 combination and necessary peripherals are packaged as a single board computer (SBC). This SBC will be used with analog to digital (A-D) converter boards and digital to analog (D-A) converter boards to control the series-bypass circuit and load circuits. The feasibility of the Intel SBC to control the similar hydraulic system on the ASV will also be evaluated by these prototype tests.

2.2 Basic System Equations

The equations for conservation of mass describing the series-bypass configuration illustrated in Figure 2-1 can be written as [423]:

\[
\frac{\Delta P}{\beta} \frac{V_1}{Q_{11}} = Q_{11} - Q_2 \tag{2.1}
\]

\[
\frac{\Delta P}{\beta} \frac{V_{11}}{Q_{11}} + \dot{X}_1 A_P = Q_{11} \tag{2.2}
\]

\[
\frac{\Delta P}{\beta} \frac{V_{12}}{Q_{12}} - \dot{X}_1 A_P = -Q_{12} \tag{2.3}
\]

\[
\frac{\Delta P}{\beta} \frac{V_2}{Q_{12}} = Q_{12} + Q_2 - Q_{31} - Q_4 \tag{2.4}
\]
These equations were formulated by assuming an incompressible fluid, no external or internal leakages and modeling fluid line volumes simply as single capacitances with no friction or inertia effects. In words, the equations simply state that the mass flow rate into the volume minus the mass flow rate out of the volume is equal to the change in the mass of the volume per unit time.

All volumetric flow rates through the valves can be modeled by the equation for turbulent flow through sharp edged orifices [42].

\[ Q_i = CA_i \sqrt{2 \left( \frac{P_i - P_o}{\rho} \right)} \]  

Good correlations between equation (2.7) and empirical tests have been achieved when \( C \) equals .61. The servovalve orifice area is relatively small when compared to the piston area of a hydraulic cylinder. To accommodate the flow rate required by the piston the fluid velocity through the servovalve is very high, and the flow can be considered turbulent.
Therefore:

\[ Q_{11} = C_A \sqrt{2(P_1 - P_{11})/\rho} \]  \hspace{1cm} (2.8)

\[ Q_2 = C_A \sqrt{2(P_1 - P_L)/\rho} \]  \hspace{1cm} (2.9)

\[ Q_{12} = C_A \sqrt{2(P_{12} - P_L)/\rho} \]  \hspace{1cm} (2.10)

\[ Q_{31} = C_A \sqrt{2(P_L - P_{21})/\rho} \]  \hspace{1cm} (2.11)

\[ Q_4 = C_A \sqrt{2(P_L - P_R)/\rho} \]  \hspace{1cm} (2.12)

\[ Q_{32} = C_A \sqrt{2(P_{22} - P_R)/\rho} \]  \hspace{1cm} (2.13)

The conservation of mass equations can now be written:

\[ \frac{\partial V_1}{\partial t} + \frac{\partial (C_A \sqrt{2(P_1 - P_{11})/\rho})}{\partial x} = Q_{11} - C_A \sqrt{2(P_1 - P_L)/\rho} \]  \hspace{1cm} (2.14)

\[ \frac{\partial V_{11}}{\partial t} + \frac{\partial (C_A \sqrt{2(P_1 - P_{11})/\rho})}{\partial x} = C_A \sqrt{2(P_1 - P_{11})/\rho} \]  \hspace{1cm} (2.15)

\[ \frac{\partial V_{12}}{\partial t} + \frac{\partial (C_A \sqrt{2(P_1 - P_{12})/\rho})}{\partial x} = -C_A \sqrt{2(P_1 - P_{12})/\rho} \]  \hspace{1cm} (2.16)

\[ \frac{\partial V_{L}}{\partial t} = C_A \sqrt{2(P_{12} - P_L)/\rho} + (C_A \sqrt{2(P_1 - P_L)/\rho}) \]
\[-(C_A \sqrt{2(P_L - P_{21})/\rho}) \]
\[-(C_A \sqrt{2(P_L - P_R)/\rho}) \]  \hspace{1cm} (2.17)

\[ \frac{\partial V_{21}}{\partial t} + \frac{\partial (C_A \sqrt{2(P_L - P_{21})/\rho})}{\partial x} = C_A \sqrt{2(P_L - P_{21})/\rho} \]  \hspace{1cm} (2.18)

\[ \frac{\partial V_{22}}{\partial t} + \frac{\partial (C_A \sqrt{2(P_L - P_R)/\rho})}{\partial x} = -C_A \sqrt{2(P_L - P_R)/\rho} \]  \hspace{1cm} (2.19)

For the average duty cycle, the required flow rates through valves 1 and 3 will be relatively small when compared to the rated valve flow. As a result the pressure drops across these valves will also be small.
For example, the Rexroth literature (Appendix A) lists a 150 psi pressure drop for a 22 GPM flow rate through the proportional valve. A typical value of actuator velocity is 8 inches per second. Assuming steady state behavior, an 8 inch per second actuator velocity is equivalent to an 8.5 GPM flow rate which corresponds to a 22.4 psi pressure drop across the valve. This pressure can be assumed negligible in comparison to the loads across the piston as the vehicle is walking. Therefore the assumptions made are $P_{11} = P_1$, $P_{12} = P_L$ and $P_{21} = P_L$.

If the pressure drops across the valves 1 and 3 can be neglected, the system can now be described by the following equations.

\[
\frac{(V_p)}{\beta}DP_1 + \dot{X}_1 A_P = 0 - CA_2 \sqrt{2(P_1 - P_L)}/\rho \tag{2.20}
\]

\[
\frac{(V_L)}{\beta}DP_L - \dot{X}_1 A_P + \dot{X}_2 A_P = CA_2 \sqrt{2(P_1 - P_L)}/\rho - CA_2 \sqrt{2(P_L - P_R)}/\rho \tag{2.21}
\]

In addition, two equations can be written to describe actuator motion. On actuator 1:

\[
F_1 - B_1 \dot{X}_1 + (P_1 - P_L)A_P = M_1 \ddot{X}_1 \tag{2.22}
\]

On actuator 2:

\[
F_2 - B_2 \dot{X}_2 + (P_L - P_R)A_P = M_2 \ddot{X}_2 \tag{2.23}
\]

In this dissertation, $F_1$ and $F_2$ are considered to
be independent. For the ASV, the forces would be low and independent during the return phase of the stride. In the contact phase, the forces would be kinematically coupled since the legs are attached to the body of the vehicle. The loads during the contact phase would be relatively low when the vehicle is at a constant velocity and traversing over even terrain. The forces would increase during rapid accelerations and when the vehicle is climbing gradients.

2.3 Linearization of Open Loop System Equations

In order to facilitate theoretical analysis it is necessary to linearize equations (2.20) and (2.21). The linear equations can be used to define transfer functions and identify system parameters useful in applying closed loop control techniques. This can be accomplished by performing a Taylor Series expansion of these equations about an assumed equilibrium point. The equilibrium point can be defined as a stable, steady state operating condition of the system. The general equation (2.7) for flow through an orifice can be linearized as:

\[ Q = Q_e + \left[ \frac{\partial Q}{\partial A} \right] A + \left[ \frac{\partial Q}{\partial P} \right] P + \left[ \frac{\partial Q}{\partial P_{op}} \right] P_{op} \]

\[ + \left[ \frac{\partial Q}{\partial P_{i}} \right] P_{i} \]  

(2.24)
The subscript \( p \) designates a perturbation variable while \( I_e \) indicates that the derivative is evaluated at the equilibrium point. A perturbation variable can be defined as the difference between the value of the variable at a specified time and the value of the variable at the equilibrium condition. The subscript \( e \) refers to the equilibrium value about which the perturbation takes place. A variable \( x \) may be written as:

\[
x = x_e + x_p
\]  

(2.25)

For a constant equilibrium value \( x_e \) the derivative of \( x \) is:

\[
\frac{dx}{dt} = \frac{dx_e}{dt} + \frac{dx_p}{dt} = \frac{dx_p}{dt}
\]  

(2.26)

Performing the required differentiation on equation (2.7) and defining \( Q_e \) as:

\[
\frac{\partial Q}{\partial A_e} = CA \sqrt{\frac{2(P_i - P_o)}{\rho}} = K_Q
\]  

(2.27)

\[
\frac{\partial Q}{\partial P_i e} = \frac{C A(2/\rho)^{5/2}}{2(P_i - P_o)^{5/2}} = K_C
\]  

(2.28)

\[
\frac{\partial Q}{\partial P_o e} = -\frac{CA(2/\rho)^{5/2}}{2(P_i - P_o)^{5/2}} = -K_C
\]  

(2.29)

\[
Q_e = CA \sqrt{\frac{2(P_i - P_o e)}{\rho}}
\]  

(2.30)

In this text \( K_Q \) will be referred to as the flow
gain coefficient and $K_C$ as the flow pressure coefficient.

Based on the above definition of the Taylor Series expansion and the definitions of the flow gain and flow pressure coefficient, Equation (2.20) can be linearized in terms of equilibrium and perturbation variables as:

$$\left(\frac{V_p}{\beta}\right)D_{lp} + \left(\dot{X}_{lp} + \dot{X}_{lp}^2\right)A_p = Q_p + Q_p$$

$$-CA_2e\sqrt{2(P_{le}-P_{Le})/\rho}$$

$$-CA_2p\sqrt{2(P_{le}-P_{Le})/\rho}$$

$$-[CA_2e\sqrt{2/[\rho(P_{le}-P_{Le})]}]P_{lp}/2$$

$$+[CA_2e\sqrt{2/[\rho(P_{le}-P_{Le})]}]P_{lp}/2$$

Equation (2.31)

The following equation is valid for all time.

$$\dot{X}_{le}A_p = Q_p - CA_2e\sqrt{2(P_{le}-P_{Le})/\rho}$$

Equation (2.32)

If equation (2.32) is subtracted from (2.31) the result is an equation which describes the perturbation about the equilibrium condition.

$$\left(\frac{V_p}{\beta}\right)D_{lp} + \dot{X}_{lp}A_p = Q_p - CA_2e\sqrt{2(P_{le}-P_{Le})/\rho}$$

$$- [CA_2e\sqrt{2/[\rho(P_{le}-P_{Le})]}]P_{lp}/2$$

$$+[CA_2e\sqrt{2/[\rho(P_{le}-P_{Le})]}]P_{lp}/2$$

Equation (2.33)

Writing equation (2.33) in terms of the flow pressure and flow gain coefficients:

$$\left(\frac{V_p}{\beta}\right)D_{lp} + \dot{X}_{lp}A_p = Q_p - K q^2 A_2p - K C_2 P_{lp} + K C_2 P_{lp}$$

Equation (2.34)
Following a similar procedure, equations (2.21) through (2.23) can be written in terms of perturbation variables as below. In expressions (2.21) and (2.23), $P_R$, the return pressure is assumed constant. Correspondingly its perturbation is zero.

\[
\begin{align*}
\frac{V_L}{\beta}D P_{Lp} + (\dot{X}_{2p} - \dot{X}_{1p}) A_p &= K_{C2} A_2 + K_{C4} P_{Lp} \\
&= -K_{C2} P_{Lp} - K_{C4} A_4 P_{Lp} \\
&= K_{C2} A_{2p} - K_{C4} A_4 P_{Lp}
\end{align*}
\]

(2.35)

\[
F_{lp} - B_1 \dot{X}_{1p} + (P_{lp} - P_{Lp}) A_p = M_1 \dot{X}_{1p} \\
F_{2p} - B_2 \dot{X}_{2p} + (P_{lp} - P_{Lp}) A_p = M_2 \dot{X}_{2p}
\]

(2.36) \quad (2.37)

Figure 2-3 is a block diagram representation of equations (2.34) through (2.37). This figure illustrates the complex nature of the coupling that exists between the two systems.

It should be mentioned that fluid volumes $V_P$ and $V_L$ (see Figure 2-1) consist of fixed line and actuator volumes. Therefore equilibrium actuator positions $X_{1e}$ and $X_{2e}$ are included in the calculations of $V_P$ and $V_L$.

Due to the complexity of the equation and the uncertainties of the model parameters, it has been decided to use frequency domain based techniques to select controller gains and to design compensators. Frequency response curves are conveniently obtained numerically by using the software package, SPEAKEASY [57]. Basically, this consists of programming
Figure 2-3: Block Diagram of Linearized Series-Bypass System
the following matrix equations, taken from equations (2.34) through (2.37), into the form of expression (2.38).

\[
\begin{bmatrix}
(V_p/\beta)D+K_{c2} & -K_{c2} & A_p & 0 \\
-K_{c2} & (V_L/\beta)D+K_{c2}+K_{c4} & -A_p & A_p \\
A_p & -A_p & -M_1D-B_1 & 0 \\
0 & A_p & 0 & -M_2D-B_2
\end{bmatrix} * 
\]

\[
\begin{bmatrix}
\dot{P}_{1s} \\
\dot{P}_{s} \\
\dot{x}_{s} \\
\dot{x}_{s}^2
\end{bmatrix} = \begin{bmatrix}
1 & -K_{q_2} & 0 & 0 & 0 \\
0 & K_{q_2} & -K_{q_4} & 0 & 0 \\
0 & 0 & 0 & -I & 0 \\
0 & 0 & 0 & 0 & -1
\end{bmatrix} * \begin{bmatrix}
Q_p \\
A_{2s} \\
A_{4s} \\
F_{1s} \\
F_{2s}
\end{bmatrix} \quad (2.38)
\]

To obtain the transfer functions between any independent input and dependent output variables the matrix equations can be arranged as...
where [A] equals

\[
\begin{bmatrix}
(V_p/β)D + K_c^2 & -K_c^2 & A_p & 0 \\
-K_c^2 & (V_L/β)D + K_c^2 + K_c^4 & -A_p & A_p \\
A_p & -A_p & -M_1D - B_1 & 0 \\
0 & A_p & 0 & -M_2D - B_2 \\
\end{bmatrix}
\]

and [B] equals

\[
\begin{bmatrix}
1 & -K_q^2 & 0 & 0 & 0 \\
0 & K_q^2 & -K_q^4 & 0 & 0 \\
0 & 0 & 0 & -1 & 0 \\
0 & 0 & 0 & 0 & -1 \\
\end{bmatrix}
\]

In obtaining the transfer functions, all independent variables other than the desired input variable in the transfer function are set to zero.

It is interesting to note that equations (2.34) through (2.37) can be easily manipulated into a
state space description of the series-bypass circuit which is both controllable and observable. In one description the states would be the pressures $P_{1p}$ and $P_{2p}$ and the actuator velocities $\dot{X}_{1p}$ and $\dot{X}_{2p}$. All of these states can be measured with physical instruments. The author investigated the possibilities of using state space methods to control the series-bypass circuit. However, when the closed loop control algorithms and actuator dynamics are included in the model (see following sections) the physical analogies between the states of the state space description and system variables deteriorates rapidly (see equation (2.66)).

Also, state space control methods are not known to be as robust as other methods of controller design.

2.4 Closed Loop Control Equations

As described in Chapter 1, we wish to investigate control problems in the Cruise Mode. It was also stated that the actuator velocities would be controlled by the bypass valves in conjunction with the variable displacement pump. The bypass valve velocity control algorithms will primarily affect the magnitude of the velocity error and the closed loop speed of response when there is a small change in desired velocity. In order to retain a margin of control, it is desirable
that $A_2$ and $A_4$ remain open a nominal amount throughout the leg stride. The phrase 'margin of control' is used to signify that a closed bypass valve does not exert any control over actuator velocity. The proposed velocity control algorithms can be written as:

$$A_{2c} = A_{2X^%} - K_{E1} A_s (\dot{X}_1 - \ddot{X}_1)$$

(2.40) $$A_{4c} = A_{4X^%} - K_{E2} A_s (\dot{X}_2 - \ddot{X}_2)$$

(2.41)

The subscript $X^%$ refers to the nominal valve opening which will be the commanded bypass valve areas for zero velocity errors. Except for the nominal bypass valve opening term, equations (2.40) and (2.41) are proportional control algorithms. Examining equations (2.40) and (2.41), it can be seen that an increase in the desired velocity results in a smaller bypass valve opening, which routes more fluid to the actuator, increasing its speed. Correspondingly, a decrease in the desired velocity results in a larger bypass valve opening which allows more fluid to travel through the bypass path, decreasing actuator speed.

For the preceding linearized analysis, all perturbations will occur about a zero velocity error condition which implies that the bypass valves will be open $X^%$ of their full open area at the beginning of the perturbation. Therefore $A_{2X^%}$ and $A_{4X^%}$ and $A_2e$ and $A_4e$
are synonymous when referring to the equilibrium condition before the perturbation. It will be shown in the following text that zero steady state errors occur for step inputs when the proposed control algorithms are used.

Equations (2.40) and (2.41) can be written in terms of perturbation variables to describe the perturbation in the commanded bypass valve area.

\[ \dot{A}_{2c} = -K_{2c} A_s (\dot{X}_1 D - \dot{X}_1 p) \] (2.42)

\[ \dot{A}_{4c} = -K_{4c} A_s (\dot{X}_2 D - \dot{X}_2 p) \] (2.43)

It has been stated by several investigators [6, 19, 54, 61] that linear hydraulic actuators have relatively small amounts of damping and it is often necessary to augment this with acceleration feedback. If necessary, acceleration feedback can be incorporated into the velocity control equations (2.40) and (2.41) by the addition of a single term. These equations become:

\[ \dot{A}_{2c} = A_{2c} + A_s K_{2c} (\dot{X}_1 D - \dot{X}_1) + A_s K_{1c} \dot{X}_1 \] (2.44)

\[ \dot{A}_{4c} = A_{4c} + A_s K_{4c} (\dot{X}_2 D - \dot{X}_2) + A_s K_{2c} \dot{X}_2 \] (2.45)

The following perturbation equations can be written for perturbations about zero velocity errors at equilibrium.

\[ \dot{A}_{2c} = A'_{2c} + K_{2c} A_s \dot{X}_1 p \] (2.46)
The terms $K_{A_1}A_{1p} \ddot{X}_{1p}$ and $K_{A_2}A_{2p} \ddot{X}_{2p}$ represent the acceleration feedback. Qualitatively, it is seen that, as the actuators accelerate in the positive sense the bypass valve area increases, which will tend to allow more fluid through the bypass valve and less to the actuator, decreasing its acceleration.

In addition to the velocity control loops around the piston it is necessary to formulate some algorithm to control the actual flow rate, $Q$ from the pump. If the flow through the bypass path is ignored and the pump is only required to supply enough flow for maximum actuator velocity a steady state error will occur whenever the bypass valve is not closed. As an example of a possible pump control algorithm, the pump flow rate command could be set at a constant value, which would be large enough to provide a specific amount of excess flow through the bypass valve. This excess "flow margin" would be sufficiently high such that the bypass valve is never completely closed. The pump command would be fixed during the Cruise Mode, eliminating the slower pump dynamics from the leg control loop. If an increase in actuator velocity is desired, the bypass valve would close, allowing more flow to the actuator. The
uncertainty in this method is the amount of flow margin. If the flow margin is too large, positive velocity errors (actual velocity greater than desired) may result. If the flow margin is decreased, negative velocity errors are possible. In addition, it is important to keep in mind that any flow through the bypass valve results in energy waste and should be minimized in this mode.

In order to provide enough flow to meet the maximum desired velocity between the two actuators, supply the flow through the bypass path and provide zero steady state velocity errors for step inputs, the pump control law is formulated as:

$$Q_C = \dot{X}_M A_P + \left( (2A_{2X} - A_{2C}) \sqrt[2]{(P_1 - P_L)/\rho} \right) + \left( (2A_{4X} - A_{4C}) \sqrt[2]{(P_L - P_R)/\rho} \right) / 2 \quad (2.48)$$

Referring to equation (2.48), it can be seen that the pump flow depends on the maximum desired actuator velocity (denoted by subscript M) and the bypass flows. It is also important to note that the expression involves nonlinear terms in the variables $A_2$, $A_4$, $P_1$ and $P_L$ and requires measurement of the pressures $P_1$ and $P_L$. It also combines the closed loop velocity feedback variables $A_{2C}$ and $A_{4C}$ with a nonlinear feedforward term, $\dot{X}_M$. The disadvantage in using this sophisticated
algorithm is that the closed loop control of an actuator velocity is essentially achieved by using both the bypass valve and variable displacement pump and hence is influenced by the adjacent bypass valve. Also, both the bypass valve and pump control algorithms use an identical variable (bypass valve area) to control piston velocity. The problems associated with coordinating two control actuators to control one output (basically, every device has different operating characteristics) will present themselves in the following chapters.

The pump control algorithm of equation (2.48) was proposed because it has the potential to provide zero velocity errors in a steady state condition when the desired velocities of both actuators and the external loads on the actuators are equal. Notice that, under steady state conditions, if the desired velocity equals the actual velocity and actuator accelerations are zero, \( A_{2C} \) and \( A_{4C} \) in equations (2.44) and (2.45) will be at \( A_{2X^\%} \) and \( A_{4X^\%} \), and the pump flow rate command will equal:

\[
Q_{Ce} = \dot{X}_P M_c P_f + \left( A_{2X^\%} C \sqrt{Z} \frac{(P_{Le} - P_{Le})}{\rho} \right) + \left( A_{4X^\%} C \sqrt{Z} \frac{(P_{Le} - P_R)}{\rho} \right)/2 \tag{2.49}
\]

\( Q_{Ce} \) is the actual amount of fluid required by the
actuators and bypass valves in steady state, if the desired actuator velocities and the actuator loads are identical. In a dynamic situation, if $\dot{x}_1 < \dot{x}_{1D}$, then the bypass valve opening, $A_{2C}$ will decrease because of the control law stated in equation (2.44). Referring to the pump control equation (2.48), if $A_{2C}$ decreases, the pump flow will increase because the quantity, $(2A_{2X} - A_{2C})$, will increase. This is exactly the desired corrective action if the desired actuator velocity exceeds the actual actuator velocity. Correspondingly, if $\dot{x}_1 > \dot{x}_{1D}$, the bypass valve area will increase, thereby decreasing the quantity $(2A_{2X} - A_{2C})$ and decreasing the pump flow. Similar control actions will occur with the second actuator for positive and negative velocity errors in $\dot{x}_2$.

To verify the statement claiming zero velocity errors when the proposed control algorithms were used, a program was written which uses the Newton-Raphson technique to solve nonlinear simultaneous equations. This routine was used to solve equations (2.48), (2.44), (2.45) and (2.20) through (2.23) for the steady state actual velocity. The program, HYDSIM, is listed in Appendix B.2. For matched actuators (i.e. $M_1 = M_2$, $B_1 = B_2$) and when the external loads, equilibrium bypass valve areas and desired velocities of both actuators are
identical, the actual velocities equaled the desired velocities. This routine is used to solve for the initial values of all state variables in the simulation studies described in Chapter 3.

Again, it is necessary to linearize the pump control equation so that it can be used in conjunction with equations (2.34) through (2.37). Applying a Taylor Series expansion to equation (2.48) and using X% of the fully open bypass valve area as the equilibrium value for $A_{2e}^c$ and $A_{4e}^c$, the linearized pump flow rate control equation can be written as:

$$Q_C = \dot{X}_M A_P^c$$

$$-\left(2A_{2X%}-A_{2X%}^c\right)C\sqrt{2(\rho(P_{le}-P_{le}^c))/\rho J/2}$$

$$+\left(2A_{2X%}^c-A_{2X%}\right)C\sqrt{2/[\rho(P_{le}-P_{le}^c)]}P_{lp}/4$$

$$-\left(2A_{2X%}^c-A_{2X%}\right)C\sqrt{2/[\rho(P_{le}-P_{le}^c)]}P_{lp}/4$$

$$-\left(2A_{4X%}^c-A_{4X%}\right)C\sqrt{2(P_{le}-P_{le}^c)/\rho J/2}$$

$$+\left(2A_{4X%}^c-A_{4X%}\right)C\sqrt{2/[\rho(P_{le}-P_{le}^c)]}P_{lp}/4 \quad (2.50)$$

Subtracting the equilibrium condition (equation (2.49)), equation (2.50) can be linearized and written in terms of perturbation variables as:
In terms of the flow and pressure coefficients defined in equations (2.27) and (2.28), equation (2.51) can be written as:

$$Q_{Cp} = \dot{X}_{Mp}^2 + L_A_{2Xp} \sqrt{\frac{2}{\rho (P_{Le} - P_{Le})}} J_{Plp} / 4$$

$$- L_A_{2Xp} \sqrt{\frac{2}{\rho (P_{Le} - P_{Le})}} J_{Plp} / 4$$

$$- L_A_{2Cp} \sqrt{2 (P_{Le} - P_{Le}) / \rho J / 2}$$

$$+ L_A_{4Xp} \sqrt{2 / \rho (P_{Le} - P_{R})} J_{Plp} / 4$$

$$- L_A_{4Cp} \sqrt{2 (P_{Le} - P_{R}) / \rho J / 2}$$

(2.51)

Notice that the equations (2.38) or (2.39) represent only equations (2.34) through (2.37) and do not describe the feedback control system. Specifically, $A_2$ and $A_4$ in equations (2.34) and (2.35) represent actual servovalve areas. In the closed loop system, $A_{2C}$ and $A_{4C}$ will be determined by equations (2.53) and (2.54) respectively. The actual servovalve opening will be the commanded servovalve opening modified by the dynamic response of the servovalve. Specifically,

$$A_{2p} = A_{2Cp} G_{sv} (s)$$

(2.53)

$$A_{4p} = A_{4Cp} G_{sv} (s)$$

(2.54)
where $G_{sv}(s)$ is a nondimensionalized servovalve transfer function. Hereafter we will drop the explicit dependence of $G_{sv}(s)$ on the Laplace variable $s$, for convenience of notation.

Recalling equations (2.42) and (2.43) and Laplace transforming these equations with zero initial conditions, the following equations are obtained.

$$A_{2Cp}'(s) = -K_{E1} A_{s} (\dot{X}_{1Dp}(s) - \dot{X}_{lp}(s))$$

(2.55)

$$A_{4Cp}'(s) = -K_{E2} A_{s} (\dot{X}_{2Dp}(s) - \dot{X}_{2p}(s))$$

(2.56)

Laplace transforming equations (2.46) and (2.47) similarly yields:

$$A_{2Cp}(s) = A_{2Cp}'(s) + K_{A1} A_{s} \dot{X}_{lp}(s)$$

(2.57)

$$A_{4Cp}(s) = A_{4Cp}'(s) + K_{A2} A_{s} \dot{X}_{2p}(s)$$

(2.58)

$A_{2p}$ and $A_{4p}$ can be written, therefore, as:

$$A_{2p}(s) = [ -K_{E1} A_{s} (\dot{X}_{1Dp}(s) - \dot{X}_{lp}(s)) + K_{A1} A_{s} \dot{X}_{lp}(s) ] G_{sv}$$

(2.59)

$$A_{4p}(s) = [ -K_{E2} A_{s} (\dot{X}_{2Dp}(s) - \dot{X}_{2p}(s)) + K_{A2} A_{s} \dot{X}_{2p}(s) ] G_{sv}$$

(2.60)

Referring to equations (2.59) and (2.60) it can be seen that $A_{2p}(s)$ and $A_{4p}(s)$ are derived from contributions from a closed loop proportional control term and an acceleration feedback term. The acceleration feedback term can be viewed as a minor loop
within the closed loop velocity feedback system.

Also, from equation (2.52) it can be seen that the pump command \( Q_{sp} \) is derived from the commanded servovalve areas \( A_{2sp} \) and \( A_{4sp} \). Laplace transforming equation (2.52) and inserting equations (2.57) and (2.58), the following equation is obtained:

\[
Q_{sp}(s) = \hat{x}_{sp}(s)A_{sp} - A_{sp}'(s)K_{Q2}/2 - K_{A1}s\hat{x}_{1p}(s)
- K_{Q2}/2 + K_{C2}P_{sp}(s)/2 - K_{C2}P_{Lp}(s)/2
- A_{sp}'(s)K_{Q4}/2
- K_{A2}s\hat{x}_{2p}(s)K_{Q4}/2 - K_{A2}s\hat{x}_{2p}(s)K_{Q4}/2
- K_{C4}P_{Lp}(s)/2
\]  

Again, \( Q_{sp}(s) \) is the command sent to the pump. The actual pump flow \( Q_{p}(s) \) is obtained by multiplying \( Q_{sp}(s) \) by the pump transfer function, \( G_{pp}(s) \). Thus:

\[
Q_{p}(s) = Q_{sp}(s)G_{pp}(s)
\]  

Hereafter the transfer function \( G_{pp}(s) \) will be represented simply as \( G_{pp} \).

Laplace transforming equations (2.34) and (2.35) and combining with equations (2.57), (2.58) and (2.63) the following expressions are obtained.
\[
\{V_p/\beta \} \mathbf{p} \mathbf{1}_p(s) + \mathbf{X}_1p(s) \mathbf{A}_p =
\begin{align*}
&\frac{\ddot{X}_{mp}(s)}{s^2} \mathbf{A}_p - \mathbf{A}_{2cP}(s)\frac{Q_2}{2} \\
&-K_{a1}A_s\dot{X}_{1p}(s)\frac{Q_2}{2} + K_{c2}P_l(s)\frac{s}{2} \\
&-K_{c2}P_{lp}(s)\frac{s}{2} \\
&-A^4_{4cP}(s)\frac{Q_4}{2} - K_{a2}A_s\dot{X}_{2p}(s)\frac{Q_4}{2} \\
&+K_{c4}P_{lp}(s)\frac{s}{2}\mathbf{G}_{pp} \\
&-K_{Q2}A_{2cP}(s) + K_{a1}A_s\dot{X}_{1p}(s)\mathbf{G}_{sv} \\
&-K_{c2}P_{lp}(s) + K_{c2}P_{lp}(s) 
\end{align*}
\] (2.64)

\[
\{V_L/\beta \} \mathbf{p} \mathbf{L}_p(s) - \mathbf{X}_1p(s) \mathbf{A}_p + \mathbf{X}_2p(s) \mathbf{A}_p =
\begin{align*}
&K_{Q2}A_{2cP}(s) + K_{a1}A_s\dot{X}_{1p}(s)\mathbf{G}_{sv} \\
&+K_{c2}P_{lp}(s) - K_{c2}P_{lp}(s) \\
&-K_{Q4}A_{4cP}(s) + K_{a2}A_s\dot{X}_{2p}(s)\mathbf{G}_{sv} \\
&-K_{c4}P_{lp}(s) 
\end{align*}
\] (2.65)

After Laplace transforming equations (2.36) and (2.37) they can be combined with equations (2.64) and (2.65) and written in matrix form as expression (2.66). As indicated in section 2.3 the complexity of the equations represented in equation (2.66) prohibits obtaining a state space description of the system with states which are easily measurable (i.e., states which can be measured without constructing observers) and controllable.

These equations can be numerically solved by SPEAKEASY to obtain the open loop frequency response for
\[
\begin{bmatrix}
(V_p/\beta)s - G_{pp} K_{c2}/2 + K_{c2} & G_{pp}/2(K_{c2} - K_{c4}) - K_{c2} & A_p + K_{a1} A_3 K_{q2}(G_{pp}/2 + G_{sv})s & (G_{pp} K_{a2} A_5 K_{q4}/2)s \\
-K_{c2} & (V_p/\beta)s + K_{c2} + K_{c4} & -A_p - K_{q2} K_{a1} A_3 G_{sv}s & -A_p - K_{q2} K_{a1} A_3 G_{sv}s \\
A_p & -A_p & -M_1 s - B_1 & 0 \\
0 & A_p & 0 & -M_2 s - B_2 \\
\end{bmatrix} \ast \begin{bmatrix}
P_p(s) \\
0 \\
0 \\
0 \\
\end{bmatrix} = \begin{bmatrix}
\dot{X}_{mp}(s) \\
A_{2c}(s) \\
A_{4c}(s) \\
F_{lp}(s) \\
F_{2p}(s) \\
\end{bmatrix}
\]

(2.66)
any input-output pair. A representative frequency response program, MFRSIM, is listed in Appendix B.3. For example the plots of $\dot{X}_{1p}/A_{2Cp}(i\omega)$ and $\dot{X}_{2p}/A_{4Cp}(i\omega)$, will be primarily used to determine the gains for the proportional closed loop velocity feedback control. Note that $K_{Al}$ and $K_{A2}$ in these expressions represent the acceleration feedback gains in the minor loops. These gains can be selected by obtaining plots of $\dot{X}_{1p}/A_{2Cp}(i\omega)$ and $\dot{X}_{2p}/A_{4Cp}(i\omega)$ for various values for $K_{Al}$ and $K_{A2}$ and selecting the most desirable response (see section 2.7). The response of $\dot{X}_{1p}$ and $\dot{X}_{2p}$ can be written in terms of the transfer functions obtained from equation (2.66) as:

$$\dot{X}_{1p} = \frac{\dot{X}_{1p}(s)}{A_{2Cp}} + \frac{\dot{X}_{1p}(s)}{A_{4Cp}} + \frac{\dot{X}_{1p}(s)}{F_{1p}} + \frac{\dot{X}_{1p}(s)}{F_{2p}}$$

$$+ \frac{\dot{X}_{1p}(s)}{\dot{X}_{Mp}}$$

(2.67)
\[ \dot{X}_{2p} = \frac{\dot{X}_{2p}(s)}{A_2C_p} A_2' C_p + \frac{\dot{X}_{2p}(s)}{A_4C_p} A_4' C_p \]

\[ + \frac{\dot{X}_{2p}(s)}{F_{1p}} F_{1p} + \frac{\dot{X}_{2p}(s)}{F_{2p}} F_{2p} \]

\[ + \frac{\dot{X}_{2p}(s)}{X_{mp}} X_{mp} \]

The *to obtain the closed loop system frequency response for various values of $K_{e1}, K_{e2}, K_{a1}$ and $K_{a2}$ merely substitute expressions (2.55) and (2.56) into equations (2.64) and (2.65).

\[ (2.68) \]

\[ \{V_p/\beta\} s P_{1p}(s) + \dot{X}_{1p}(s) A_P = G_p C_X(s) M_p(s) A_P \]

\[ -K_{Q2} \{ -K_{e1} A_S(\dot{X}_{1Dp}(s) - \dot{X}_{1p}(s)) \}/2 \]

\[ -K_{A1} A_S \dot{X}_{1p}(s) K_{Q2}/2 + K_{C21p}(s) /2 - K_{C2p} P(s) /2 \]

\[ -K_{Q4} \{ -K_{e2} A_S(\dot{X}_{2Dp}(s) - \dot{X}_{2p}(s)) \}/2 \]

\[ -K_{A2} A_S \dot{X}_{2p}(s) K_{Q2}/2 + K_{C4p} P(s) /2 \]

\[ -K_{Q2} \{ -K_{e1} A_S(\dot{X}_{1Dp}(s) - \dot{X}_{1p}(s)) \}

\[ + K_{A1} A_S \dot{X}_{1p}(s) G_s - K_{C2p} P(s) + K_{C2p} L_p(s) \]

\[ (2.69) \]
The set of closed loop system equations, written in matrix SPEAKEASY form, appears as Equation (2.71).

The closed loop response of $\dot{X}_{1p}$ and $\dot{X}_{2p}$ can be written in terms of the transfer functions obtained from equation (2.71) as:

\[
\dot{X}_{1p} = \frac{\dot{X}_{1p}(s)}{X_{1Dp}} \dot{X}_{1Dp} + \frac{\dot{X}_{1p}(s)}{\dot{X}_{2Dp}} \dot{X}_{2Dp} + \frac{\dot{X}_{1p}(s)}{F_{1p}} F_{1p} + \frac{\dot{X}_{1p}(s)}{F_{2p}} F_{2p} + \frac{\dot{X}_{1p}(s)}{\dot{X}_{Mp}} \dot{X}_{Mp}
\]  

(2.72)
\[
\begin{bmatrix}
(V_o/2)(I_s-K_{C2}G_{op}/2)K_{C2} & (V_o/2)(I_s+K_{C2}+K_{C4})-K_{C2} & A_pK_{C4}A_pG_{0o}(G_{op}/2+G_{op})I_s+K_{C2}G_{0o}G_{op}+K_{C4}(G_{op}/2+G_{op}) + K_{C2}

-K_{C2} & (V_o/2)I_s+K_{C2}+K_{C4} & -A_p

A_p & -A_p & 0

0 & 0 & A_p
\end{bmatrix}
\]

\[
\begin{bmatrix}
-\Delta_{t-S}/B_s

-\Delta_{t-S}/B_s

0

-M_{t-S}/B_2
\end{bmatrix}
\]

\[
\begin{bmatrix}
P_{1}(s)

P_{2}(s)

\dot{x}_{1}(s)

\dot{x}_{2}(s)
\end{bmatrix}
\]

\[
\begin{bmatrix}
P_{1}(s)

P_{2}(s)

\dot{x}_{1}(s)

\dot{x}_{2}(s)
\end{bmatrix}
\]
\[ \dot{x}_{2p} = \frac{\dot{x}_{2p}(s)}{\dot{x}_{1Dp}} \dot{x}_{1Dp} + \frac{\dot{x}_{2p}(s)}{\dot{x}_{2Dp}} \dot{x}_{2Dp} \]

\[ + \frac{\dot{x}_{2p}(s)}{F_{1p}} F_{1p} + \frac{\dot{x}_{2p}(s)}{F_{2p}} F_{2p} \]

\[ + \frac{\dot{x}_{2p}(s)}{\dot{x}_{Mp}} \dot{x}_{Mp} \]

(2.73)

2.5 Series-Bypass Symbolic Transfer Functions

It was originally intended to analyze the series-bypass configuration and design the compensators in the closed velocity loops using the open loop symbolic transfer functions. Many techniques to design closed loop compensators utilize the symbolic open loop transfer functions. Specifically, in similar hydraulic applications, open loop symbolic transfer functions are primarily used in designing closed loop compensators for classical servovalve controlled pistons [37, 42, 53, 54, 61]. Closed loop compensators that have been used for servovalve controlled actuators have included lag compensators to increase system stiffness to low frequency load disturbances and lead compensators to increase the system bandwidth. The
requirements on the servovalve controlled pistons are also compatible with the requirements of the series-bypass circuit. Consequently, it seemed feasible to use the open loop symbolic transfer functions to design compensators for the series-bypass circuit.

Other advantages in obtaining the open loop symbolic transfer functions are:

1. It is possible to determine the effect of different features and components of the series-bypass circuit on system performance. The symbolic expressions also lend themselves to a parameter sensitivity analysis.

2. The coupling mechanisms between the two circuits can be identified in terms of the system parameters. The symbolic transfer functions may suggest methods to decouple the system or equilibrium conditions which minimize or maximize the system coupling.

3. The analysis of the symbolic transfer functions should complement the linearized frequency response methods and time domain simulations of the prototype system.
Specifically, frequency response curves could be obtained from the symbolic transfer functions which could be used to confirm the frequency response curves obtained using the matrix expressions derived in section 2.4.

Equations (2.34) through (2.37) describe the open loop series-bypass circuit but neglect servovalve and pump dynamics, acceleration feedback (last terms in equations (2.44) and (2.45)) and the pump volumetric flow rate control algorithm, equation (2.48). Equations (2.64), (2.65), (2.36) and (2.37) include these four factors. If the two sets of equations are compared it becomes apparent that it would involve considerably more effort to solve for symbolic transfer functions using equations (2.64), (2.65), (2.36) and (2.37). Also, if the symbolic transfer functions were obtained from this set of equations, they could conceivably be too unwieldy and contain too many terms to permit a simple parameter sensitivity analysis.

Despite the apparent simplicity of equations (2.34) through (2.37) resulting from neglecting actuator dynamics, acceleration feedback and the pump volumetric flow rate control algorithm it was felt that the symbolic transfer functions obtained from the system
equations would provide some insight into the effects of cylinder pressures, bypass valve areas, piston locations and reflected inertias on the series-bypass system response.

To obtain the symbolic open loop transfer functions, two approaches were considered. The simultaneous equations can be solved by hand or the symbolic manipulation computer program FORMAC [49] could be used. Initially, it was decided to use FORMAC for two reasons.

1. Gain experience in the limitations and capabilities of symbol manipulation programs.

2. It was believed that using FORMAC would eliminate algebraic errors which could occur given the large number of manipulations involved in solving these equations by hand.

In order to use FORMAC, it was necessary to reduce the equations to a form which involved only one of the dependent variables $\dot{X}_{lp}$, $\dot{X}_{2p}$, $P_{lp}$ or $P_{lp}$. An example of a FORMAC program is displayed in Appendix B.4. The transfer functions involving $\dot{X}_{lp}$ were derived first. These transfer functions are $\dot{X}_{lp}/Q_p(s)$, $\dot{X}_{lp}/A_{2p}(s)$,
To obtain the FORMAC transfer functions in Table 2-2, the original set of four equations had to be solved for each dependent variable. The FORMAC output also had to be reduced from seventh order to fourth order after identification of the common term in the numerator and denominator. Extensive algebraic manipulations were required to factor this term from the numerator and denominator and the effort showed that FORMAC did not offer an easy alternative to solving the system of equations by hand. For this reason, equations (2.34) through (2.37) were solved manually for the transfer functions $\dot{X}_{1p}/A_4p(s)$, $\dot{X}_{1p}/F_{1p}(s)$ and $\dot{X}_{1p}/F_{2p}(s)$ and are also listed in Table 2-2.

To verify that the transfer functions are correct, frequency response curves obtained by using the transfer functions in Table 2-2 were compared to frequency response curves obtained by solving equations (2.34) through (2.37) using SPEAKEASY. For all the transfer functions examined, the frequency response curves were virtually identical.
Table 2-2: Symbolic Transfer Functions

\[ F_{AC1} = (V_p s + K_{C2}^p) \] (2.74)

\[ F_{AC2} = [V_L s + \beta (K_{C2} + K_{C4})] (V_p s + K_{C2}^p) - K_{C2}^p \beta^2 \] (2.75)

\[ \text{DENOMINATOR} = \beta A_p^4 + \beta A_p^2 [F_{AC1} (M_1 s + B_1)] + \beta A_p^2 [V_L + V_p] s + K_{C4} \beta (M_2 s + B_2) + [F_{AC2} (M_1 s + B_1) (M_2 s + B_2)] \] (2.76)

\[ \dot{X}_{LP}(s) = \frac{\beta V A_K P Q_{2}^2 (M_2 s + B_2) s}{A_4 p} \text{ DENOMINATOR} \] (2.77)

\[ \dot{X}_{LP}(s) = \frac{\beta A_K P Q_{2}^2 (M_2 s + B_2) s}{A_4 p} \text{ DENOMINATOR} \] (2.78)

\[ \dot{X}_{LP}(s) = \frac{(M_2 s + B_2) [F_{AC2} + A_p^2 \beta [F_{AC1}]}{F_{LP} \text{ DENOMINATOR}} \] (2.79)

\[ \dot{X}_{LP}(s) = \frac{A_p^2 \beta V_p s}{F_{2p} \text{ DENOMINATOR}} \] (2.80)

\[ \dot{X}_{LP}(s) = \frac{A_p^2 \beta (V_L s + K_{C4} \beta) (M_2 s + B_2)}{Q_p \text{ DENOMINATOR}} \] (2.81)

\[ \dot{X}_{2P}(s) = \frac{\beta V A_K P Q_{2}^2 (M_1 s + B_1) s}{A_2 p} \text{ DENOMINATOR} \] (2.82)
Table 2-2: Symbolic Transfer Functions

(continued)

\[ \dot{X}_{2p}(s) = -\beta K A_P \frac{\left\{ (M_1 s + B_1)(V_P s + K C_2 \beta) + \beta A_P^2 \right\}}{A_{4p} \text{DENOMINATOR}} \] (2.83)

\[ \dot{X}_{2p}(s) = A_P^2 \beta V_P s \] (2.84)

\[ \dot{X}_{2p}(s) = (M_1 s + B_1)(FAC2) + A_P^2 \beta ((V_L + V_P) s + K C_4 \beta) \] (2.85)

\[ \dot{X}_{2p}(s) = A_P \beta \frac{A_P^2 \beta + K C_2 \beta (M_1 s + B_1)}{Q_p \text{DENOMINATOR}} \] (2.86)
2.6 Analysis of Symbolic Transfer Functions

The system constants used to obtain the open and closed loop frequency response data in the preceding sections are listed in Table 2-3. The transfer function \( G_{pp}(s) \) and values for piston damping \( B_1 \) and \( B_2 \) and averaged reflected leg inertias \( M_1 \) and \( M_2 \) at the actuator were obtained from Gardner [18]. The transfer function for the servovalve \( G_{sv}(s) \) was obtained by curve fitting to a frequency response curve supplied by the servovalve manufacturer.

The constant term in the symbolic denominator in Table 2-2 can be simplified by examining the relative magnitudes of the various terms in it. Nominal values of \( K_{C2}^2 \) and \( K_{C4}^2 \) are approximately 0.01 in\(^3\)/sec/psi and a nominal value of piston damping for \( B_1 \) and \( B_2 \) is 10 lbf-sec/in. Substituting these values into the symbolic expression and comparing the relative magnitudes of the terms, it is apparent that the constant term can be simplified to \( \beta^2 A_P^4 \). If all denominator coefficients are divided by \( \beta^2 A_P^4 \), the constant term becomes unity and the denominator appears as the product of two standard second order terms [14].

\[
(s^2/\omega_{n1}^2 + 2\zeta_{n1}/\omega_{n1}s + 1)(s^2/\omega_{n2}^2 + 2\zeta_{n2}/\omega_{n2}s + 1)
\]  

(2.87)
Table 2-3: System Constants for Series-Bypass Prototype System

\[ M_1 = M_2 = 1.0 \text{ lbf-sec}^2/\text{inch} \]
\[ B_1 = B_2 = 10.0 \text{ lbf-sec/inch} \]
\[ A_p = 4.1 \text{ inch}^2 \]
\[ \beta = 1.0 \times 10^5 \text{ psi} \]
\[ A_s = 0.53 \text{ inch}^2/\text{rpm} \]

\[
G_{pp}(s) = \frac{1}{s^2 + 2\zeta_p s + \frac{1}{w_p}}
\]
\[ w_p = 100 \text{ rad/sec} \]
\[ \zeta_p = 1.0 \]

\[
G_{sv}(s) = \frac{1}{s^2 + 2\zeta_{sv} s + \frac{1}{w_{sv}}}
\]
\[ w_{sv} = 345 \text{ rad/sec} \]
\[ \zeta_{sv} = 1.0 \]
\[ \rho = 0.78 \times 10^{-4} \text{ lbf-sec}^2/\text{inch}^4 \]
\[ C = 0.61 \]
\( \zeta \) is defined as the damping ratio and \( \omega \) is referred to as the natural frequency of the system.

When two second order terms are multiplied, the fourth order term involves both system natural frequencies, \( \omega_{n1} \) and \( \omega_{n2} \). Merritt [42] defines the natural frequency of a servovalve controlled piston as

\[
\omega_n = \left( \frac{\beta A_p}{(\sqrt{VM})} \right)^{1/2}
\]  

(2.88)

where

- \( \omega_n \) = natural frequency of servovalve controlled piston
- \( \beta \) = fluid bulk modulus
- \( V \) = fluid volume
- \( M \) = external piston inertia

The inverse of the coefficient of the fourth order term in the denominator can be written as:

\[
\left[ \left( \frac{\beta^2 A_p^4}{(M_2 V_p V_L M_1)} \right)^{1/2} \right]^2
\]  

(2.89)

It appears reasonable to equate this coefficient to the product of the squares of the two natural frequencies of the series-bypass circuit. Notice that the coefficient consists of parameters describing both actuators. The terms of this coefficient suggest an increase in the system natural frequencies as \( M_2 \) and \( M_1 \) decrease and as the compressible volumes \( V_p \) and \( V_L \) decrease. Equally important, this coefficient suggests that the natural frequencies of the series-bypass system
are independent of system pressure, bypass valve opening and actuator velocity.

For the nominal values of $K_{C2}$, $K_{C4}$, $B_1$ and $B_2$ specified above, the frequency response curves of $\dot{X}_{1p}/A_{2p}(i\omega)$ and $\dot{X}_{2p}/A_{4p}(i\omega)$ display sharp resonances. These resonances are shown in Figures 2-4 and 2-5. By examining the coefficients in the denominator it can be seen that as $K_{C2}$, $K_{C4}$ and $B_1$ and $B_2$ decrease the magnitude of the denominator decreases. It can be suggested that an increase or decrease in the magnitude of the denominator may lead to a subsequent increase or decrease in the resonance peaks displayed in the open loop frequency response curves. From equation (2.28) and (2.29) it can be seen that $K_{C2}$ and $K_{C4}$ are calculated from the system pressures and bypass valve openings. When $B_1$ and $B_2$ are set equal to 0 and $K_{C2}$ and $K_{C4}$ are lowered to .005 in$^3$/sec/psi the magnitude of the resonance peaks is greater, as shown by the plots of the magnitudes of $\dot{X}_{1p}/A_{2p}(i\omega)$ and $\dot{X}_{2p}/A_{4p}(i\omega)$ in Figures 2-6 and 2-7. Merritt and other researchers [6, 19, 42, 53, 54, 61] report that this attribute is also displayed by the underdamped response of the servovalve controlled piston. Note also that the locations of the resonance peaks were not affected by the change in the damping or flow pressure values. The
Figure 2-4: Magnitude of Open Loop Frequency Response, $|\hat{X}_{1p}/A_{2p}(i\omega)|$

Figure 2-5: Magnitude of Open Loop Frequency Response, $|\hat{X}_{2p}/A_{4p}(i\omega)|$
Figure 2-6: Magnitude of Open Loop Frequency Response, $|\hat{X}_1p/A_{2p}(i\omega)|$

Figure 2-7: Magnitude of Open Loop Frequency Response, $|\hat{X}_2p/A_{4p}(i\omega)|$
locations of the resonance peaks are directly related to the natural frequencies of the system. It was suggested in equation (2.89) that the natural frequencies of the series-bypass system did not depend on the external damping or flow pressure values.

Several important features of the series-bypass behavior can be inferred from the numerator terms of Table 2-2. The transfer functions for actuator 1 are analogous to those for actuator 2. $\frac{\dot{X}_{1p}}{A_{2p}}(s)$ and $\frac{\dot{X}_{2p}}{A_{4p}}(s)$, $\frac{\dot{X}_{1p}}{F_{1p}}(s)$ and $\frac{\dot{X}_{2p}}{F_{2p}}(s)$, $\frac{\dot{X}_{1p}}{F_{2p}}(s)$ and $\frac{\dot{X}_{2p}}{F_{1p}}(s)$, $\frac{\dot{X}_{1p}}{A_{4p}}(s)$ and $\frac{\dot{X}_{2p}}{A_{2p}}(s)$ and $\frac{\dot{X}_{1p}}{Q_{p}}(s)$ and $\frac{\dot{X}_{2p}}{Q_{p}}(s)$ are analogous pairs. It can be seen that the forms of the numerators of all analogous pairs are similar. For example, the numerator of $\frac{\dot{X}_{1p}}{A_{2p}}(s)$ is:

$$-\beta K_{Q_{2}A_{p}}(M_{2}s+B_{2})(V_{L}+V_{p})s+\beta K_{C_{4}}+\beta A_{p}^{2}$$ (2.90)

The numerator of $\frac{\dot{X}_{2p}}{A_{4p}}(s)$ is:

$$-\beta K_{Q_{4}A_{p}}[(M_{1}s+B_{1})(V_{p}s+K_{C_{2}})+\beta A_{p}^{2}]$$ (2.91)

These two pairs as well as the other analogous pairs differ in that the numerator of the transfer functions for $\dot{X}_{1p}$ contain terms associated with the second actuator while the numerator of the transfer functions for $\dot{X}_{2p}$ contain terms associated with the first actuator. Note that the numerator of $\frac{\dot{X}_{1p}}{A_{2p}}(s)$
contains the terms \( M_2, B_2 \) and \( K_{C4} \) while the numerator of \( \dot{X}_{2p}/A_{4p}(s) \) contains the terms \( M_1, B_1 \) and \( K_{C2} \). This illustrates one aspect of the coupling between the two actuators. The transfer functions which consist of both independent and dependent variables from the same actuator, \( \dot{X}_{1p}/A_{2p}(s), \dot{X}_{1p}/F_{1p}(s), \dot{X}_{2p}/A_{4p}(s) \) and \( \dot{X}_{2p}/F_{2p}(s) \), contain system constants and linearized parameters associated with the adjacent actuator. Another aspect of the coupling is illustrated by the direct effect of one actuator on the other actuator. The transfer functions \( \dot{X}_{1p}/A_{4p}(s), \dot{X}_{1p}/F_{2p}(s), \dot{X}_{2p}/A_{2p}(s) \) and \( \dot{X}_{2p}/F_{1p}(s) \) display this aspect of the coupling. Because of these two effects, the series-bypass configuration is coupled not only by the action of adjacent independent variables, but also, the adjacent system parameters determines the response of an actuator to a direct input on the actuator. Note that because of the similarity in the transfer functions of both actuators, the steady state values of analogous pairs will be equal for equal actuator loads and bypass valve areas.

In the following chapters it will become important to minimize the coupling which exists between the bypass valves and adjacent actuators. This would allow the velocity of both actuators of the series-bypass system
to be primarily controlled by their immediate bypass valves and operated as two independent closed velocity loop systems. Using the symbolic transfer functions in Table 2-2 the degree of coupling which exists between a bypass valve and the actuators can be expressed as:

\[ \frac{\dot{X}_{1p}/A_{4p}(s)}{\dot{X}_{2p}/A_{4p}(s)} = \frac{-V_p(M_2s+B_2)s}{(M_1s+B_1)(V_p+K_{C2}B)+A_p^2B} \quad (2.92) \]

\[ \frac{\dot{X}_{2p}/A_{2p}(s)}{\dot{X}_{1p}/A_{2p}(s)} = \frac{-V_p(M_1s+B_1)s}{[(M_2s+B_2)((V_p+V_L)s+K_{C4}B)+A_p^2B]} \quad (2.93) \]

Assuming constant and equal system inertias and damping, the only variables in equations (2.92) and (2.93) are the linearized flow pressure coefficients, \( K_{C2} \) and \( K_{C4} \). Therefore, the equilibrium values of bypass valve areas and pressure drops determine the degree of coupling between a bypass valve and the two actuators. Since these terms are in the denominator, any increase in the flow pressure coefficients will decrease the effect of a bypass valve on the velocity of the adjacent actuator. From the definition of the flow pressure coefficients given in equation (2.28), an increase in the equilibrium bypass valve area or a decrease in the equilibrium pressure drop across the valve will decrease the coupling. The equilibrium bypass valve area is commanded
by the control program while the pressure drop across the valve is determined by the loads on the actuators. Therefore, the open loop velocity coupling should be minimized for large bypass valve areas and low actuator loads. Intuitively, it can be argued that an alternative flow path is created as the bypass valve opening increases. This flow path decreases an actuator's dependence on the flow from the adjacent actuator.

Note also that the entries in Table 2-2 indicate only dynamic coupling between a bypass valve or external load and the adjacent actuator velocity. No steady state coupling exists using these transfer functions. When the pump control algorithm (equation (2.48)) is included in the model, the bypass valve area and pressure drop terms in the algorithm create steady state coupling between bypass valves, external loads and the adjacent actuator velocities.

It was stated at the beginning of this section that the transfer functions obtained from the simplified series-bypass model of equations (2.34) through (2.37) would be similar to the response of the model which included pump and servovalve dynamics and the pump flow rate control law. Figures 2-8 and 2-9 are frequency response curves of $\dot{X}_{1p}/A_{2p}(i\omega)$ and $\dot{X}_{2p}/A_{4p}(i\omega)$ for the simplified system and Figures 2-10 and 2-11 show
\[ \dot{X}_{lp}/A_{2CP}(i\omega) \text{ and } \dot{X}_{2p}/A_{4Cp}(i\omega) \]

for the more complete model (equations (2.64), (2.65), (2.36) and (2.37)), with acceleration feedback gains equal to zero. Comparing the figures for the two models, it can be seen that Figures 2-8 and 2-10 are similar as are the curves of Figures 2-9 and 2-11, implying that the system characteristics are similar. The differences in the curves are in the steady state values and the amplitudes of the resonance peaks. The difference in the steady state values can be attributed to the additional bypass flow terms included in the pump control equation. The difference in the resonance peaks is due to the attenuation of the pump dynamics which occurs beyond 100 rad/sec (see Table 2-3).

In summary, it appears that the symbolic transfer functions obtained by solving the system equations (2.34) through (2.37), can be used to analyze the series-bypass open loop behavior. Specifically, the transfer functions can be used to predict the effect of system pressures, bypass valve openings, fluid compressible volumes and reflected inertias on system response. The transfer functions also give an indication of the coupling between the systems. If the ASV requirements did not dictate that the series-bypass circuit be constructed from identical components
Figure 2-8: Magnitude of Open Loop Frequency Response, $|\tilde{X}_{1p}/A_{2p}(i\omega)|$, Simplified Model

Figure 2-9: Magnitude of Open Loop Frequency Response, $|\tilde{X}_{2p}/A_{4p}(i\omega)|$, Simplified Model
Figure 2-10: Magnitude of Open Loop Frequency Response, $|X_p/A_{2Cp}(i\omega)|$, Complex Model

Figure 2-11: Magnitude of Open Loop Frequency Response, $|X_{2p}/A_{4Cp}(i\omega)|$, Complex Model
methods to decouple the circuit could be suggested by
the numerator of the symbolic transfer functions.

2.7 Effect of Equilibrium Conditions on System Behavior

To predict the behavior of the series-bypass circuit under different operating conditions, it is necessary to specify the equilibrium values of actuator pressures $P_{1e}$ and $P_{Le}$, bypass valve areas $A_{2e}$ and $A_{4e}$, and piston positions $X_{1e}$ and $X_{2e}$. The actuator pressures will be determined by the external loads while the equilibrium bypass valve areas will be specified by the controller, based on the considerations presented in this section. The equilibrium values for actuator pressures and bypass valve areas appear in the $K_Q$ parameters in equation (2.27). The equilibrium values for piston pressures also appear in the $K_C$ parameters in equations (2.28) and (2.29). Piston position appears in the calculations for volumes $V_P$ and $V_L$. After all other parameters are specified, acceleration feedback gains $K_{A1}$ and $K_{A2}$ will be selected by examining the frequency response curves for $\dot{X}_{1p}/A_{2Cp}(i\omega)$ and $\dot{X}_{2p}/A_{4Cp}(i\omega)$ obtained by using the system constants in Table 2-3 and for specific equilibrium conditions. In this section, the relationships between the behavior of the
series-bypass system and the operating conditions will be investigated. The symbolic transfer functions in Table 2-2 and frequency response curves obtained from equation (2.66) will be used to analyze the dependencies.

Linear hydraulic actuators possess very small amounts of system damping. Therefore it is common to observe large resonance peaks in the frequency response of closed loop systems which employ linear actuators (see Figures 2-12 and 2-13). Large resonance peaks are recognized as undesirable in control applications because the large magnitude and phase gradients cause an undesirable time response which must be modified by the controller. It was shown in section 2.6 that a decrease in the magnitude of the coefficients in the denominator caused an increase in the resonant peaks. Examination of the approximate open loop transfer functions $\dot{X}_{1p}/A_{2c}^p(s)$ and $\dot{X}_{2p}/A_{2c}^p(s)$ in Table 2-2 reveals that the magnitudes of the denominator coefficients decrease as the values of $K_{C2}$ and $K_{C4}$ decrease. From the definition of $K_C$ given in equations (2.28) and (2.29), it is apparent that $K_{C2}$ and $K_{C4}$ will decrease if the valve pressure drops increase or valve areas decrease. Therefore, for control applications, the worst case would be at high valve pressure drops and zero bypass valve areas. Valve
pressure drops are maximum when the vehicle is traveling uphill and for legs in the contact phase of the stride. ASV specifications list a maximum speed of 2.5 mph when on a slope of 50% gradient. For a 6000 pound vehicle in tripod gait and using the current 5 to 1 ratio of horizontal foot force to main drive actuator force, the pressure drops across each bypass valve of the two loaded actuators on one side of the vehicle can be as high as 1400 psi.

The equilibrium piston position determines values of $V_P$ and $V_L$. It has been stated by Merritt [42] that the natural frequencies of a servovalve controlled piston are relatively constant with respect to piston location and are inversely proportional to the product of the square root of the external inertia ($M$) and compressible fluid volume ($V$). The coefficient of the quadratic term in the denominator of the FORMAC transfer functions listed in Table 2-2 is $M_1 M_2 V_P V_L$. Therefore it appears that this coefficient for the series-bypass can be associated with the inverse of the system natural frequencies. The natural frequencies determine the location of the resonance peaks. If this quadratic term remains constant throughout the stride then the natural frequencies of the system should also remain constant. The values of external inertia were assumed to be
relatively constant throughout the stride for the purpose of simplifying the analysis. The fluid volumes $V_p$ and $V_L$ depend on $X_1$ and $X_2$ (See Figure (2.1)) specifically, $V_p$ increases and $V_L$ decreases with an increase in $X_1$ while $V_L$ increases with an increase in $X_2$. Figure 2-2 shows the approximate line lengths for the series-bypass circuit of Figure 2-1. The line volumes of $V_p$ and $V_L$ equal 86 and 110 cubic inches respectively. The total fluid volumes in the actuators are 32.8 cubic inches. If pistons 1 and 2 move in similar trajectories as expected in the Cruise Mode, $V_L$ will remain constant (142.8 cubic inches) throughout the stride. As $X_1$ increases from the LHS of the piston to the RHS, $V_p$ increases by 38%. $V_p$ will be a maximum at the end of the stride, which corresponds to lower system natural frequencies and a more difficult system to control. However, it is doubtful that any meaningful control will be applied at the end of the stride. Consequently for this system it appears reasonable to select the piston midpoint as an equilibrium value on the basis that $V_p$ does not change considerably throughout the stride. Also, for simulation studies starting from equilibrium conditions, this position would leave one half the piston length to compare the effects of different control algorithms on piston
velocity. As suggested by Merritt, SPEAKEASY frequency response curves for various values of piston position (Figures 2-21 and 2-22) show little difference.

The equilibrium bypass valve area is an important parameter in the bypass valve and pump control algorithms. Its value will determine many features of the ASV performance. It has been shown that the actuator velocity displays high resonance peaks when the bypass valve areas are decreased. The relationship between the bypass valve area and the amplitude of the resonance peaks has a simple physical interpretation. Note that the bypass flow is proportional to the bypass valve area. The apparent damping due to the bypass valve occurs because the bypassed flow does no work on the load and its fluid energy is totally dissipated by the bypass valve. As the bypass valve area increases, the damping increases and the amplitude of the resonant peak decreases. Note however that if the bypass valve is opened a large amount to satisfy system damping requirements, the overall energy efficiency of the vehicle would suffer since a larger amount of fluid energy would be dissipated across the bypass valve. The actuator’s speed of response would also decline for larger values of bypass valve opening. Another consideration for the nominal valve area is its margin
of control over the actuator velocity. As the valve area decreases it has less ability to increase the actuator velocity. If the bypass valve is completely closed it effectively loses control of the actuator velocity. Finally, it was shown in section 2.6 that the system coupling decreases as the bypass valve area increases.

Therefore the considerations for selecting the equilibrium bypass valve area are system damping and speed of response, margin of control, energy efficiency and the degree of coupling between actuators. Recall that in section 2.4 acceleration feedback could be added to augment system damping, eliminating the need to control system damping by using the bypass flow. In section 3.5 compensation will be included in the series-bypass control algorithms to increase the system's speed of response. The margin of control depends on both the equilibrium bypass valve area and the proportional controller gains $K_{E1}$ and $K_{E2}$. Regardless of the equilibrium bypass valve area, the bypass valve could close completely for small positive velocity errors if these constants are relatively large. These gains will be determined primarily from stability considerations (see Chapters 3, 4 and 5). Consequently it appears reasonable to select equilibrium bypass valve areas based on other criteria and judge whether the
margin of control determined from $K_{E1}$ and $K_{E2}$ is sufficient. As stated previously, it would be desirable if both actuators of the series-bypass system would be operated as independent closed loop velocity feedback systems. Another method to decouple the actuators will be presented in Chapter 3. Attempts to decouple the actuator velocities will primarily depend on the use of this procedure. It was decided therefore to select the nominal bypass valve area primarily on the basis of energy efficiency. The steady state energy efficiency of piston 1 in the series-bypass configuration can be stated as (neglecting leakage flows and pressure drops across proportional valves):

\[
\frac{\text{flow into piston 1}}{\text{total flow to bypass path and piston 1}}
\]

Or in the notation of Figure (2.1),

\[
\text{EFF} = \frac{Q_{11}}{Q_{11} + Q_2} (100)
\] (2.94)

\[
Q_{11} = \dot{X}_{1} A_p
\] (2.95)

\[
Q_2 = C_A \sqrt{\frac{2(P_1 - P_L)}{\rho}}
\] (2.96)

\[
Q_S = Q_{11} + Q_2
\] (2.97)

\[
Q_{11} = Q_S - C_A \sqrt{\frac{2(P_1 - P_L)}{\rho}}
\] (2.98)

\[
\text{EFF} = \left[1 - \frac{C_A \sqrt{2(P_1 - P_L)/\rho}}{Q_S}\right] (100)
\] (2.99)

Equation (2.98) shows that for maximum efficiency the bypass valve should be closed and that efficiency
decreases with increased bypass valve pressure drop.

A nominal value for the preferred operating efficiency of a servovalve controlled piston can be stated as 66% [42]. To ensure that each actuator of the series-bypass circuit would be more energy efficient than a servovalve controlled piston, 66% was chosen as the lowest operating efficiency for the series-bypass circuit. Since the energy efficiency of the series-bypass circuit decreases in proportion to valve pressure drops, the 1400 psi pressure drop recognized as the worst case for controller designs is also the worst case for efficiency considerations. As an example of computing a nominal bypass valve area, if 66% is substituted for EFF and 1400 for \( P_1 - P_L \), equation (2.99) can be solved for the nominal bypass valve area, which is .0049 in\(^2\).

As an example of selecting the acceleration feedback gains \( K_{A1} \) and \( K_{A2} \), open loop frequency response curves for \( \dot{X}_{1p}/A_{2Cp}(i\omega) \) and \( \dot{X}_{2p}/A_{4Cp}(i\omega) \) were obtained from SPEAKEASY by solving equation (2.66) and selecting equilibrium bypass valve pressure drops of 1400 psi, valve areas of .004 in\(^2\), pistons in the middle of the cylinders and \( K_{A1}K_{A2} = 0 \). These plots are shown in Figures 2-12 and 2-13. Note that these figures are identical to Figures 2-10 and 2-11 which displayed the
magnitude of the open loop frequency response of the complex model with no acceleration feedback. The low damping of the series-bypass circuit caused by the large pressures and small bypass valve openings can be clearly seen from the large resonance peaks in these plots. For most systems an optimum tradeoff between speed of response (or rise time) and settling time for a step input occurs when a slight overshoot occurs in the time response. A slight peak in the frequency response can be loosely associated with the transient overshoot in the response to a step input and therefore should not be considered detrimental to the open loop performance. Also, it would be desirable to maintain a flat amplitude frequency response curve over the largest frequency range while minimizing the phase lag. Figures 2-14, 2-15, 2-16 and 2-17 show the responses of $\hat{X}_{1p}/A_{2p}^2(i\omega)$ and $\hat{X}_{2p}/A_{4p}^2(i\omega)$ using the same pressures, bypass valve areas and piston positions as in Figures 2-12 and 2-13 but for increasing values of $K_{A1}$ and $K_{A2}$. As the acceleration gains are increased, the resonant peaks as well as the amplitude ratios between the peaks are decreased. It would be expected that the system speed of response would suffer for the values of acceleration gains in Figures 2-16 and 2-17. For these equilibrium conditions and based on the above criterion, the values
Figure 2-12: Magnitude of Open Loop Frequency Response, $|\dot{X}_{1p}/A_2^{'}C_p(i\omega)|$

Figure 2-13: Magnitude of Open Loop Frequency Response, $|\dot{X}_{2p}/A_4^{'}C_p(i\omega)|$
Figure 2-14: Magnitude of Open Loop Frequency Response, $|\dot{X}_1p/A_2Cp(i\omega)|$

Figure 2-15: Magnitude of Open Loop Frequency Response, $|\dot{X}_2p/A_4Cp(i\omega)|$
Figure 2-16: Magnitude of Open Loop Frequency Response, $|X_1p/A_2^p(i\omega)|$

Figure 2-17: Magnitude of Open Loop Frequency Response, $|X_2^p/A_4^p(i\omega)|$
or $K_{A1}$ and $K_{A2}$ in Figures 2-14 and 2-15 were judged to be optimum.

The series-bypass system of Figures 2-12 through 2-17 included the pump control algorithm and acceleration feedback. Similar to the symbolic transfer function analysis in the previous section, these frequency response curves predict equal steady state actuator velocities for equal loads and bypass valve areas even though these two features were added to the control structure.

The following figures illustrate some of the statements above and introduce further observations regarding the frequency response of the series-bypass circuit. These curves were obtained by using SPEAKEASY and equation (2.66). Unless stated otherwise, these figures use the acceleration feedback gains, equilibrium pressures, bypass valve areas and actuator positions listed in Table 2-4. When the characteristics of both actuators are similar, only the response of one will be shown.

Figures 2-18 through 2-20 were generated to display the effects of actuator pressure and bypass valve area on the frequency response of an actuator. Comparing Figure 2-18 to 2-14 it can be seen that the amplitude ratio of the frequency response decreases over the
Table 2-4: Equilibrium Parameters

Linearized Analyses

\[ \begin{align*}
K_{A1} &= 4.0 \times 10^{-6} \text{ amps/in/sec}^2 \\
K_{A2} &= 11.0 \times 10^{-6} \text{ amps/in/sec}^2 \\
P_{Le} - P_{Le} &= 1400 \text{ psi} \\
P_{Le} - P_{Re} &= 1400 \text{ psi} \\
A_{2e} &= 0.004 \text{ in}^2 \\
A_{4e} &= 0.004 \text{ in}^2 \\
K_{C2} &= 0.0052 \text{ in}^3/\text{sec/psi} \\
K_{C4} &= 0.0052 \text{ in}^3/\text{sec/psi} \\
K_{Q2} &= 3667 \text{ in}^3/\text{sec/in}^2 \\
K_{Q4} &= 3667 \text{ in}^3/\text{sec/in}^2 \\
V_P &= 102.1 \text{ in}^3 \\
V_L &= 142.8 \text{ in}^3
\end{align*} \]

Simulation Studies (if different from above)

\[ \begin{align*}
P_1 &= 2964.4 \text{ psi} \\
P_{11} &= 2953.2 \text{ psi} \\
P_{12} &= 1553.2 \text{ psi} \\
P_L &= 1542.2 \text{ psi} \\
P_{21} &= 1531.2 \text{ psi} \\
P_{22} &= 131.2 \text{ psi}
\end{align*} \]
entire frequency range as the system pressures decrease. It appears as if a constant value separates the magnitudes of the two curves. This would seem to indicate a linear correspondence between amplitude ratio and system pressure. Comparing Figure 2-19 to 2-14 it can be seen that as a consequence of the control algorithms, the steady state value of the frequency response is independent of the equilibrium bypass valve position. However larger bypass valve areas attenuate the amplitude ratio at higher frequencies (higher values of system damping). Figure 2-20 shows that an actuator's frequency response is decreased over the entire frequency range for lower pressures and larger bypass valve openings.

To illustrate the relative independence of system natural frequencies on piston position, Figure 2-21 shows the system frequency response for \( \frac{\dot{x}_p}{A_4C_p}(i\omega) \) at the same equilibrium values for bypass area and valve pressure drops as above but with \( X_1 = 0 \) inches and \( X_2 = 8 \) inches. In these locations, \( V_p \) becomes a minimum and \( V_L \) is a maximum. Figure 2-22 shows \( \frac{\dot{x}_p}{A_4C_p}(i\omega) \) for \( X_1 = 8 \) inches and \( X_2 = 0 \) inches. For these values, \( V_p \) is a maximum and \( V_L \) a minimum. Both these figures are similar to Figure 2-15 in which both piston positions were chosen to be at the midpoint.
Figure 2-18: Magnitude of Open Loop Frequency Response, $|\dot{X}_p/A_{2Cp}(i\omega)|$

Figure 2-19: Magnitude of Open Loop Frequency Response, $|\dot{X}_p/A_{2Cp}(i\omega)|$, $A_{2e}, A_{4e} = .02 \text{ in}^2$
Figure 2-20: Magnitude of Open Loop Frequency Response, $|\dot{X}_{1p}/A'_{2Cp}(i\omega)|$

Some additional characteristics of the series-bypass circuit are listed below. They were observed after examination of frequency response curves obtained at the conditions listed in Table 2-4 and identify the frequency response characteristics of the series bypass circuit which exist at these equilibrium conditions. The analysis illustrates the procedures which may be used to identify the behavior of the circuit at other conditions.

- The phase angle plots of $\dot{X}_{1p}/A'_{2Cp}(i\omega)$ and $\dot{X}_{2p}/A'_{4Cp}(i\omega)$ displayed in Figures 2-23 and
Figure 2-21: Magnitude of Open Loop Frequency Response, \( |\frac{X_2}{X_4(i\omega)}| \), \( X_{1e} = 0'' \), \( X_{2e} = 8'' \)

Figure 2-22: Magnitude of Open Loop Frequency Response, \( |\frac{X_2}{X_4(i\omega)}| \), \( X_{1e} = 8'' \), \( X_{2e} = 0'' \)
2-24 begin at 180 degrees, indicating that an increase in bypass valve area decreases actuator velocity.

Amplitude and phase angle plots of the \( \dot{X}_{1p}/A_{1Cp}^i(i\omega) \) and \( \dot{X}_{2p}/A_{2Cp}^i(i\omega) \) in Figures 2-25 through 2-28 illustrate the magnitude and nature of the coupling in the series-bypass circuit. It can be seen that the magnitude of these figures is less than the magnitude of Figures 2-14 and 2-15 indicating that, at these equilibrium conditions, the velocity of an actuator is primarily controlled by its immediate bypass valve. Since equations (2.92) and (2.93) show that these conditions result in extreme coupling, it can be expected that the coupling would decrease at lower pressures and larger bypass valve areas. From the phase angle figures it can be seen that an increase in the bypass valve area decreases the velocity of the adjacent actuator. This is due to the decrease in pump flow rate caused by the increased valve opening. As shown in section 2.6, without the pump control algorithm, a bypass valve has no effect on the
Figure 2-23: Phase Angle of Open Loop
Frequency Response, $X_{1p}/A_{2Cp}$($\omega$)

Figure 2-24: Phase Angle of Open Loop
Frequency Response, $X_{2p}/A_{4Cp}$($\omega$)
steady state velocity of the adjacent actuator.

- A representative frequency response curve illustrating \( \dot{X}_{1P}/F_{1p}(i\omega) \) is shown in Figure 2-29. The frequency response plots of \( \dot{X}_{1P}/F_{2p}(i\omega), \dot{X}_{2P}/F_{1p}(i\omega) \) and \( \dot{X}_{2P}/F_{2p}(i\omega) \) are similar. Note the very small effect the load disturbance has on the steady state velocity of the actuator. These small values of amplitude ratio substantiate the statements regarding zero steady state velocity errors. As stated previously, the values at steady state are the result of the pump control algorithm. Over the entire frequency range the magnitude of the load disturbance on the velocity is very small when compared to the other open loop frequency response magnitudes. Although not included in this document, the phase angle plots show that a positive/negative perturbation in loads (see the positive direction assigned to the variables in Figure 2-1) increases/decreases the velocity of the actuators.

- The frequency response curves of the transfer
Figure 2-25: Magnitude of Open Loop Frequency Response, $|\dot{X}_1p/A_{4Cp}(i\omega)|$

Figure 2-26: Phase Angle of Open Loop Frequency Response, $\dot{X}_1p/A_{4Cp}(i\omega)$
Figure 2-27: Magnitude of Open Loop Frequency Response, $|\frac{X_2}{A_2'}(i\omega)|$

Figure 2-28: Phase Angle of Open Loop Frequency Response, $\frac{X_2}{A_2'}(i\omega)$
functions $\frac{X_{lp}}{X_{Mp}(i\omega)}$ and $\frac{X_{2p}}{X_{Mp}(i\omega)}$ are displayed in Figures 2-30 through 2-33. Note that the steady state values of the amplitude ratios in these figures are unity.

The phase angle plots of $\frac{X_{lp}}{A_{2Cp}(i\omega)}$, $\frac{X_{2p}}{A_{4Cp}(i\omega)}$, $\frac{\dot{X}_{lp}}{\dot{X}_{Mp}(i\omega)}$ and $\frac{\dot{X}_{2p}}{\dot{X}_{Mp}(i\omega)}$ in Figures 2-23, 2-24, 2-31 and 2-33 show that the phase lag of $\frac{X_{lp}}{A_{2Cp}(i\omega)}$ or $\frac{X_{2p}}{A_{4Cp}(i\omega)}$ is much less than the phase lag of $\frac{\dot{X}_{lp}}{\dot{X}_{Mp}(i\omega)}$ or $\frac{\dot{X}_{2p}}{\dot{X}_{Mp}(i\omega)}$ when compared over an identical frequency range. This indicates that the time
Figure 2-30: Magnitude of Open Loop Feedforward Frequency Response, $|\dot{X}_{lp}/\dot{X}_{mp}(i\omega)|$

Figure 2-31: Phase Angle of Open Loop Feedforward Frequency Response, $\dot{X}_{lp}/\dot{X}_{mp}(i\omega)$
Figure 2-32: Magnitude of Open Loop Feedforward Frequency Response, $|X_{2p}/X_{mp}(i\omega)|$

Figure 2-33: Phase Angle of Open Loop Feedforward Frequency Response, $\angle X_{2p}/X_{mp}(i\omega)$
response of the valve controlled actuator is faster than the variable displacement pump controlled actuator. It must be kept in mind that the dynamics of the variable displacement pump are included in $\dot{X}_{1p}/A'_{2C_p}(i\omega)$ and $\dot{X}_{2p}/A'_{4C_p}(i\omega)$ since the pump control algorithm, equation (2.48), uses $A'_{2C_p}$ and $A'_{4C_p}$.

- All the open loop frequency response curves display two resonance peaks, suggesting two dominant natural frequencies. In section 2.6, it was shown that the location of these two natural frequencies appears primarily dependent on the physical constants associated with the components of series-bypass circuit such as the external inertias, piston damping and area, and fluid volumes and pressures, rather than the pump and servovalve dynamics.

- The frequency response curves for the dependent variable $\dot{X}_{1p}$ have greater amplitude ratios past the frequency of the first resonance peak than the curves which have $\dot{X}_{2p}$ as the dependent variable. This implies that the time response of the first system is faster than the second for these equilibrium
conditions.

2.8 Summary

In this chapter, the mathematical expressions modeling the series-bypass circuit were formulated. These nonlinear expressions were linearized and used to obtain open loop symbolic transfer functions and open loop frequency response transfer functions. These open loop descriptions identified relationships between the system constants and equilibrium conditions and the natural frequencies, damping and frequency response characteristics of the system. The symbolic transfer functions were also used to describe the degree of open loop coupling between the bypass valves and actuator velocities. The analysis indicated that the coupling between the adjacent actuators decreases as the system pressures decrease and bypass valve areas increase. The linearized frequency response curves predicted that the magnitude of the transfer functions between the bypass valve areas and piston velocities would decrease as system pressures decrease and bypass valve areas increase.

The control structure of the series-bypass circuit was also defined in this chapter. The primary objective was to control the ASV in an energy efficient manner.
during tripod gait over smooth and even terrain. Under these conditions, the vehicle will be subjected to nominally equal desired leg velocities and load conditions. It is desired that the bypass valves allow independent operation of adjacent closed velocity loop systems and increase the system's speed of response when compared to a pump controlled system. It was also desired (for reasons made evident in Chapters 3 and 4) that the formulation of the control algorithms would allow the series-bypass circuit to be described mathematically as a two input, two output system even though three control actuators (two bypass valves and variable displacement pump) are being used to control the velocities of two hydraulic pistons. The control algorithm for the variable displacement pump was formulated to supply the maximum hydraulic power required by the actuators in the series-bypass circuit. The pump control expression included a nonlinear feedforward term and nonlinear terms implicitly involving the closed loop variables (commanded bypass valve areas derived from actual and desired actuator velocities) and system pressures. The bypass valves were primarily controlled by a proportional control algorithm that included a provision for adding acceleration feedback to increase system damping. It was shown that
the pump controller contributed to the steady state coupling of the system but without acceleration feedback, the open loop transfer functions between the bypass valves and actuator velocities could be approximated by the symbolic transfer functions.

To further characterize the open loop system, the linearized pump and bypass valve control algorithms were included in the previously linearized open loop expressions. A frequency response analysis of this augmented open loop system predicted the following:

1. The actuators would respond more quickly to a change in the bypass valve area rather than a change in the pump displacement. The curves also suggested that the first actuator would respond more quickly to a change in its desired velocity than the second actuator would respond to a change in its desired velocity.

2. The natural frequencies of the series-bypass circuit are independent of piston positions.

3. The series-bypass circuit is relatively stiff to equal static and dynamic loads on the
actuators.

4. The series-bypass circuit possesses two resonant peaks.
3.1 Closed Loop Block Diagram

The closed loop block diagram of the series-bypass circuit is shown in Figure 3-1. The elements of the matrices in Figure 3-1 were obtained by solving equation (2.66) for the various transfer functions. \( G(s) \) is the 2x2 matrix with the general elements:

\[
\begin{bmatrix}
g_{11}(s) & g_{12}(s) \\
g_{21}(s) & g_{22}(s)
\end{bmatrix}
\]

For the series-bypass configuration \( G(s) \) can be represented symbolically as

\[
\begin{bmatrix}
\dot{X}_{1P}(s) & \dot{X}_{1P}(s) \\
A'_{2Cp} & A'_{4Cp} \\
\dot{X}_{2P}(s) & \dot{X}_{2P}(s) \\
A'_{2Cp} & A'_{4Cp}
\end{bmatrix}
\]
Figure 3-1: Matrix Block Diagram of Series-Bypass Circuit
where \( \dot{X}_{1p} \) and \( \dot{X}_{2p} \) are the actual velocities of actuators 1 and 2 respectively and \( A'_{2Cp} \) and \( A'_{4Cp} \) are the commanded servovalve inputs to valves 2 and 4 respectively.

\[ D \text{ is the } 2 \times 2 \text{ matrix of} \]
\[
\begin{bmatrix}
\dot{X}_{1p}(s) & \dot{X}_{2p}(s) \\
F_{1p} & F_{2p} \\
\dot{X}_{2p}(s) & \dot{X}_{2p}(s) \\
F_{1p} & F_{2p}
\end{bmatrix}
\]

where \( F_{1p} \) and \( F_{2p} \) are the external loads on actuators 1 and 2.

From equation (2.63), the pump flow rate will equal:
\[
Q_{p} = G_{pp} \left[ \dot{X}_{1p}A_{p} - (A'_{2Cp} + K_{A1}A_{s}\dot{X}_{1p})K_{Q2}/2 \right. \\
\left. + P_{1p}K_{C2}/2 - P_{2p}K_{C2}/2 \right. \\
\left. - (A'_{4Cp} + K_{A2}A_{s}\dot{X}_{2p})K_{Q4}/2 \right. \\
\left. + P_{2p}K_{C4}/2 \right] \quad (2.63)
\]

The term \( G_{pp} \dot{X}_{1p}A_{p} \) represents the product of the variable displacement pump dynamics, piston area and magnitude of the largest desired velocity, \( \dot{X}_{1Dp} \) or \( \dot{X}_{2Dp} \). None of these terms are associated with any closed loop system variables and consequently the term represents the feedforward branch in Figure 3-1. This feedforward
branch is indicated by the diagonal matrix M. M can be expressed as:

\[
\begin{bmatrix}
\dot{X}_{1p}^p(s) & 0 \\
0 & \dot{X}_{2p}^p(s) \\
0 & \dot{X}_{1p}^p(s) \\
0 & \dot{X}_{2p}^p(s)
\end{bmatrix}
\]

NL refers to the switching function.

\[
\dot{X}_{1p}^p = \dot{X}_{1p}^p \text{ if } \dot{X}_{1p}^p > \dot{X}_{2p}^p \\
\dot{X}_{2p}^p = \dot{X}_{2p}^p \text{ if } \dot{X}_{2p}^p > \dot{X}_{1p}^p
\]

(3.1)

(3.2)

Note that because of the function of NL, the feedforward branch can be designated as nonlinear.

The matrix K(s) is a precompensation matrix consisting of:

\[
\begin{bmatrix}
k_{11}(s) & k_{12}(s) \\
k_{21}(s) & k_{22}(s)
\end{bmatrix}
\]

The matrix Kl(s) is a diagonal matrix composed of:

\[
\begin{bmatrix}
k_{11}(s) & 0 \\
0 & k_{22}(s)
\end{bmatrix}
\]
For the series-bypass circuit, if $K=I$, the identity matrix, the elements of $Kl(s)$ can be written symbolically as:

$$
\begin{bmatrix}
A_2'(s)

\dot{x}_{1Dp} - \dot{x}_{1p}

0

0

\end{bmatrix}
$$

The matrix $B$ is a diagonal matrix with either loops closed or 0 (loops open) as the diagonal elements and is defined as:

$$
\begin{bmatrix}
b_{11} & 0 \\
0 & b_{22}
\end{bmatrix}
$$

(3.3)

Using the matrices shown in Figure 3-1, the open loop expression can be written as:

$$
\dot{x}_p = G(s)K(s)Kl(s)\dot{x}_{Ep} + M(s)\dot{x}_{Mp} + D(s)L_p
$$

(3.4)

where

$$
\dot{x}_p = \begin{bmatrix} \dot{x}_{1p} \\ \dot{x}_{2p} \end{bmatrix} \quad \text{and} \quad \dot{x}_{Ep} = \dot{x}_{Dp} - B\dot{x}_p
$$
The closed loop expression can be written as:

\[
\frac{L_p}{F_1p} \quad \text{and} \quad \frac{X_{DP}}{F_2p} = \left[ \begin{array}{c} X_{1DP} \\ X_{2DP} \end{array} \right]
\]

\[
\frac{X_{MP}}{F_1p} = \left[ \begin{array}{c} X_{1MP} \\ X_{2MP} \end{array} \right]
\]

The closed loop expression can be written as:

\[
\begin{align*}
\dot{X}_p &= [I + G(s)K(s)K(s)]B^{-1}G(s)K(s)K(s)J\dot{X}_{DP} \\
&\quad + [I + G(s)K(s)K(s)]B^{-1}M(s)\dot{X}_{MP} \\
&\quad + [I + G(s)K(s)K(s)]B^{-1}D(s)L_p
\end{align*}
\]  

(3.5)

The matrix \( K(s)K(s) \) will contain controller designs to improve system operation. The contents of this matrix will be the subject of this chapter.

This chapter will be concerned with optimizing the system performance when \( k_{11} = k_{22} = 1 \) and \( k_{12} = k_{21} = 0 \). An identity matrix \( K \) will have no effect on the system response. In the next chapter these gains will be altered in an attempt to decouple the closed loop system outputs and obtain more desirable performance characteristics.

3.2 Time Response of Series-Bypass Circuit Using Proportional Control

The matrices in Figure 3-1 can be expanded into the block diagram of Figure 3-2. This figure shows the
internal structure of the controller and series relationships between the controller, actuators and plant. The blocks $G_{1c}(s)$ through $G_{9c}(s)$ are possible locations for compensation filters. As shown in Figure 3-2 it is possible to filter the input command signals $A_{2cP}$ and $A_{4cP}$ at many different locations before using them as inputs to the actuators. In addition, the acceleration signals $\ddot{X}_1$ and $\ddot{X}_2$ can also be filtered before they are fed back into the system.

The primary objective of the series-bypass configuration is to achieve the nearly identical leg velocities required in the Cruise Mode. The series-bypass circuit was designed to accomplish a significant amount of coordination hydraulically, eliminating the need for complex control algorithms. Therefore $K_{E1}(s)$ and $K_{E2}(s)$ will be selected as constant gains for the initial closed loop design. Figures 3-3 and 3-4 show the closed loop velocity response of the prototype time domain simulation for $K_{E1}$ and $K_{E2} = .0001 \text{ amps/in/sec}$. The ACSL program is listed in Appendix B.1 and basically consists of programming the nonlinear expressions (2.14) through (2.19) and the control algorithms (2.44), (2.45) and (2.48). Unless otherwise specified, for these figures and all others in the chapter, the initial values of the system variables at the beginning of the
Figure 3-2: Block Diagram of Series-Bypass Circuit
Figure 3-3: Velocity of Actuator 1, \( K_{E1}, K_{E2} = 0.001 \)

Figure 3-4: Velocity of Actuator 2, \( K_{E1}, K_{E2} = 0.001 \)
simulation or equilibrium values for the frequency response curves are listed in Table 2-4. These values were selected primarily because the frequency response analysis presented in section 2.7 predicted that these conditions are least desirable in regard to system control and efficiency, in other words, worst case conditions.

In these figures, the desired velocities $\dot{x}_{1D}$ and $\dot{x}_{2D}$ are identical, as anticipated for the ASV actuators during Cruise Mode. From the velocity response, several static and dynamic characteristics of the series-bypass circuit can be observed. Qualitatively, the velocity response of the two adjacent systems appear very similar, specifically in terms of rise and settling time. The second system appears to have a greater time delay in reacting to the step input and has a greater initial overshoot. As stated in section 2.4 and shown in the frequency response analysis of section 2.7, both actuators achieve the desired velocity with essentially zero velocity error, despite the implicit lack of a free integrator in the denominator and the presence of a constant opposing load of 5740 lbf on each actuator. For both actuators the response to an increasing or decreasing velocity step appears to be identical. This was expected from previous examination of the FORMAC
transfer functions in section 2.6. The system response to either an increasing or decreasing step input depends on the equilibrium values of system pressures and bypass valve areas and the fluid volumes. For either an increasing or decreasing step, the system pressures are the same, the fluid volume does not change severely as the pistons translate through the cylinders. Also the control algorithms presented in section 2.4 return the bypass valves to their equilibrium values for equal desired velocity inputs. Since it has been shown that the velocity loop response of this stable system is independent of the direction of the desired input, some subsequent figures will only show the system response to a positive step in the desired velocity. Also, when the operating characteristics of both actuators are similar, the response of only one will be shown to illustrate some particular behavior.

It was shown in the frequency response curves of Figures 2-12 through 2-17 that acceleration feedback increased the system damping. Figure 3-5 displays the response of \( \dot{x}_1 \) with no acceleration feedback. The effects of acceleration feedback on the velocity oscillations can be clearly seen by comparing this figure to Figure 3-3. As stated by other researchers [19, 54], the speed of response of the
system has not decreased with the addition of these moderate amounts of acceleration feedback. In selecting these gains, the designer must choose between system stability and speed of response.

Figure 3-5: Velocity of Actuator 2, No Acceleration Feedback, $K_{E1}, K_{E2} = .0001$

The response of both actuators in Figures 3-3 and 3-4 could be described as sluggish. To improve the performance of the closed loop system, the gains $K_{E1}$ and $K_{E2}$ were increased to .0007 amps/in/sec. The velocity output of the system is shown in Figures 3-6 and 3-7.

As a result of the gain increase, the system speed
Figure 3-6: Velocity of Actuator 1, $K_{E1}, K_{E2} = 0.0007$

Figure 3-7: Velocity of Actuator 2, $K_{E1}, K_{E2} = 0.0007$
of response has increased slightly, but the response has also become more oscillatory for both actuators. Again, there appears to be a greater delay to the step input in the response of the second actuator. This delay is further illustrated by the plot of the actuator accelerations shown in Figure 3-8. The acceleration of the second actuator is displaced with respect to time when compared to the acceleration of the first actuator. Since the inertial properties of the fluid were neglected in the analytical and simulation model the delay must be due to the participation of the variable displacement pump in the control loop and the fact that it affects the first actuator before the second actuator. As observed previously, the magnitude of the initial overshoot is greater in the second actuator. In addition, the frequency of the velocity oscillations of the second actuator appear to be less than the first actuator although both actuators settle down to the desired value in the same amount of time. These comparisons could have been predicted by recalling from the open loop frequency response curves in Figures 2-14 and 2-15 that the response of the second actuator had less amplitude at higher frequencies and a larger resonant peak at the lower natural frequency.

Figures 3-10, 3-11 and 3-9 show the time response
Figure 3-8: Actuator Accelerations, $X_1$, $X_2$

Figure 3-9: Pump Flow Rate, $Q$, $K_{E1}, K_{E2} = 0.0007$
Figure 3-10: Bypass Valve Area, $A_2$, $K_{E1}\cdot K_{E2} = 0.0007$

Figure 3-11: Bypass Valve Area, $A_4$, $K_{E1}\cdot K_{E2} = 0.0007$
of $A_2^2$, $A_4^4$ and $Q$. Figures 3-10 and 3-11 show that the servovalve orifice area varies by only a small percentage of its total available area, due to the relatively small value of the proportional gain. If these values were to increase, the variation in the valve area would increase and the servovalve would be capable of exerting greater control over the series-bypass system. Comparing the time response of $A_2^2$, $A_4^4$ and $Q$ it can be seen that the time response of $Q$ is slower than the servovalves', supporting the argument for a servovalve controlled bypass path in the series circuit for the purpose of controlling actuator velocity. Figures 3-12 and 3-13 illustrate the differences in the open loop velocity response of the actuators to changes in either the bypass valve area or pump flow rate. Figure 3-12 displays the velocity response of actuator 1 to a step change of 0.0007 in$^2$ in the bypass valve area, $A_2^2$. Figure 3-13 shows the velocity response of the first actuator to a step change of 1 in$^3$/sec in the pump volumetric flow rate. By carefully comparing these two figures it can be seen that the actuator responds more quickly to a step in servovalve area. This agrees with the frequency response analysis presented in section 2.7. For the low values of proportional gains used in this simulation, it appears
Figure 3-12: Velocity of Actuator 1, $\dot{x}_1$

Figure 3-13: Velocity of Actuator 1, $\dot{x}_1$
the pump dynamics are the primary influence on the velocity response of the actuators. As the proportional gains are increased, the servovalves could control greater amounts of flow to the actuators and it would be expected that the system speed of response would increase due to the greater servovalve bandwidth.

3.3 Stability of Series-Bypass Circuit

Additional simulations show that further proportional gain increases do not improve the system speed of response but do increase oscillations in the response, leading to an unstable system for $K_{E1} = K_{E2} = 0.011$ amps/in/sec. It would be desirable to predict the stability limit on the gain values analytically rather than increasing the gains in the simulation on a trial and error basis. To predict the gain limits on stability for an interacting multivariable closed loop system three procedures were found which could be applied to the series-bypass configuration.

The Nyquist stability criterion [51] can be used to determine the stability of a multivariable system. In the notation of Figure 3-1, the characteristic equation for this multivariable system can be written as:
\[ \det[I + G(s)K(s)K_l(s)B] = 0 \quad (3.6) \]

Equation (3.6) simply equates the determinant of the closed loop equation (3.5) to zero.

With the characteristic equation in this form, the Nyquist stability criterion states that a system with stable open loop poles is asymptotically stable if the Nyquist plot of the determinant, expression (3.7), does not encircle the origin.

\[ \det[I + G(i\omega)K(i\omega)K_l(i\omega)B] \quad (3.7) \]

This method addresses the multivariable system as a whole, not as separate closed loop subsystems which compose the complete system. Therefore the Nyquist plot of expression (3.7) is useful to determine the gross stability of the overall system, not the relative stability of the various components.

Another procedure considers the stability of the two loops simultaneously by expanding the determinant in expression (3.7) into expression (3.8) [35, 51].

\[ \prod_{i=1}^{k} (1 + b_{ii}c_i) \quad (3.8) \]

where \( c_i \) is the closed loop transfer function between input \( i \) and output \( i \) with the \( i \)th loop open \( (b_{ii}=0) \) and all other loops closed with specified gains. The loops are opened one at a time and left open during evaluation.
of subsequent $c_i$. For the $2\times2$ matrix of the series-bypass system expression (3.8) can be written in the notation of Figure 3-1 as:

$$\Gamma(l+g_{11}(i\omega)K_{E1})$$

$$\begin{align*}
\frac{[(l+g_{11}(i\omega)K_{E1})(l+g_{22}(i\omega)K_{E2})
-\bar{g}_{12}(i\omega)g_{21}(i\omega)K_{E1}K_{E2}]}{(l+g_{11}(i\omega)K_{E1})}
\end{align*}$$

(3.9)

Note that $(1 + g_{11}(i\omega)K_{E1})$ can be either loop 1 or 2. The common design procedure is to plot

$$(l+g_{11}(i\omega)K_{E1})$$

(3.10)

and select a $K_{E1}$ based on the usual Nyquist stability criterion for the $i$th single loop system (plot must not encircle the origin). Using this gain $K_{E1}$, the other gain must be selected such that the plot of

$$\begin{align*}
\frac{[(l+g_{11}(i\omega)K_{E1})(l+g_{22}(i\omega)K_{E2})
-\bar{g}_{12}(i\omega)g_{21}(i\omega)K_{E1}K_{E2}]}{1+g_{11}(i\omega)K_{E1}}
\end{align*}$$

(3.11)

also excludes the origin. The reasoning behind this procedure can be explained by considering that if both plots approach but exclude the origin their phase angles in the vicinity of the origin will be approximately 0 degrees and their magnitudes will be relatively small (see Figure 3-16). When the two plots are combined (expressions (3.10) and (3.11) are multiplied) the
resulting plot of expression (3.12) will also have a small magnitude and a phase angle less than -180 degrees, ensuring that the origin will not be encircled.

\[(l+g_{11}(i\omega)K_{E1})(l+g_{22}(i\omega)K_{E2})
-\bar{g}_{12}(i\omega)g_{21}(i\omega)K_{E1}K_{E2}\]  \hspace{1cm} (3.12)

This process allows the designer to select one gain independent of the other, permitting a different margin of stability for each loop. In the strict sense an independent stability margin for each individual loop cannot be defined. Qualitatively however, as the plot of expression (3.10) approaches the origin, the stability of the ith loop should decrease. In contrast, as the plot of expression (3.11) approaches the origin, a qualitative description of the relative stability of both loops cannot be made.

The method of 2-Hodographs [46] involves rewriting expression (3.7) as

\[
\frac{\frac{1}{(l+g_{11}(i\omega)K_{E1})(l+g_{22}(i\omega)K_{E2})}}{\frac{1-C(i\omega)g_{11}(i\omega)g_{22}(i\omega)K_{E1}K_{E2}}{(l+g_{11}(i\omega)K_{E1})(l+g_{22}(i\omega)K_{E2})}}
\]

where \(C(i\omega)\) equals

\[
\frac{(g_{12}(i\omega)g_{21}(i\omega))}{(g_{11}(i\omega)g_{22}(i\omega))}
\]

For stability, the half closed systems \((b_{11}=1, b_{22}=0)\)
and \( b_{11} = 0, b_{22} = 1 \) should be stable. These requirements are satisfied since simulation studies and Nyquist diagrams have shown (not presented here) that the half closed systems are inherently stable for gains which exceed the stable gains for the fully closed system. If the half closed systems are stable, their effect on expression (3.13) can be neglected and the stability of the fully closed multivariable system can be determined from the Nyquist plot of:

\[
1 + \frac{-C(i\omega)}{\left( \frac{1}{q_{11}(i\omega)K_{E1}} + 1 \right) \left( \frac{1}{q_{22}(i\omega)K_{E2}} + 1 \right)}
\]

Expression (3.15) is in a form which is similar to the Nyquist stability criterion for single loop systems. The method of 2-Hodographs considers the expression:

\[
\left( \frac{1}{q_{11}(i\omega)K_{E1}} + 1 \right) \left( \frac{1}{q_{22}(i\omega)K_{E2}} + 1 \right) - C(i\omega)
\]

In this form, the \( C(i\omega) \) polar plot now plays the same role for the inverse controller (IC\((i\omega)\)) plot of:

\[
\left( \frac{1}{q_{11}(i\omega)K_{E1}} + 1 \right) \left( \frac{1}{q_{22}(i\omega)K_{E2}} + 1 \right)
\]

as the point \(-1 + j0\) for the Nyquist plot of:

\[
\left( \frac{1}{q_{11}(i\omega)K_{E1}} + 1 \right) \left( \frac{1}{q_{22}(i\omega)K_{E2}} + 1 \right) - C(i\omega)
\]

The definitions of phase and gain margins for expression
(3.16) are illustrated in Figure 3-14. Comparing these definitions to those applying to single loop systems \cite{12}, it can be seen that as the controller gains increase, the graph of $\text{IC}(i\omega)$ shrinks toward the graph of $\text{C}(i\omega)$, decreasing the distance ratio $OB/OA$ and therefore the stability. Similarly it can be shown that as the phase lag of $\text{IC}(i\omega)$ increases, $\alpha$ decreases.

Several advantages exist in using the method of 2-Hodographs to determine the stability limits for the proportional gains of the series-bypass circuit.

1. The plot of $\text{C}(i\omega)$ is an indication of the amount of interaction which exists in the open loop system. The magnitude of $\text{C}(i\omega)$ is directly proportional to the coupling in the system.

2. Changing the controller gains only affects the plot of expression (3.17). In some systems, it allows a degree of graphical insight into the synthesis of controllers.

3. This procedure permits a rapid graphical parameter sensitivity analysis.
Gain Margin at $\omega_1$ rad/sec = OB/OA
Phase Margin at $\omega_2$ rad/sec = $\alpha$

Figure 3-14: Method of 2-Hodographs
The three procedures discussed above were evaluated with respect to their ability to predict the stability limits of the proportional gains in the series bypass system and their usefulness in providing some insight into the characteristics of the series-bypass configuration. It should be realized that the transfer functions used in these procedures were derived from a linearized analysis for small deviations from an equilibrium point. The time domain simulation includes the nonlinear effects and uses the actual values of the system variables. Figures 3-15, 3-16 and 3-17 are plots of two of the procedures described above over the frequency range of 0 to 240 rad/sec with $K_{E_1} = K_{E_2} = 0.0007$ amps/in/sec. Inspection of the frequency response figures in Chapter 2 show that this range includes the two resonance peaks of the open loop response. Because of the amplitude attenuation after 240 rad/sec, it was felt that any indication of instability would occur in this region. The Nyquist plot of the system determinant, expression (3.7), displayed in Figure 3-15, predicts a stable system since it does not encircle the origin. Figure 3-16 is a plot of the expanded determinant, expressions (3.10) and (3.11), with $1 + g_{11}(i\omega)K_{E_1}$ as the first term. Figure 3-17 is another plot of expressions (3.10) and (3.11) with $1 + g_{22}(i\omega)K_{E_2}$ as the
Figure 3-15: Nyquist Plot of Determinant, Expression (3.7), $K_{E1}, K_{E2} = 0.0007$

first term. Both figures predict a stable system, since in either figure, no plot encircles the origin. However comparing the plots of the first terms, it can be seen that the plot of $1 + g_{11}(i\omega)K_{E1}$ approaches closer to the origin than the plot of $1 + g_{22}(i\omega)K_{E2}$. This suggests that the stability of the first loop is less than the stability of the second at these equilibrium conditions. Comparing the time history plots of Figures 3-6 and 3-7 it can be seen that the velocity response of the first actuator displays some high frequency, low
Figure 3-16: Nyquist Plot of Expressions (3.10),(3.11)

Figure 3-17: Nyquist Plot of Expressions (3.10),(3.11)
amplitude oscillations, whereas the response of the second actuator has lower frequency oscillations and appears to damp out completely.

The magnitudes and phase angles of $C(\omega)$ and the inverse controller expression ($IC(\omega)$), expression (3.17) are listed in Table 3.1. The variation in the magnitude of $IC(\omega)$ creates some difficulty in plotting these two functions over the entire frequency range. From Table 3.1 two regions can be identified in which a gain margin can be specified, 97.5 to 105 rad/sec and 225 to 232 rad/sec. The system gain margin will be the lower value of gain margin found in these two regions. Each range was divided into smaller increments and a minimum value of gain margin was found to be 4.42 at 101 rad/sec. Figure 3.18 is a plot of the 2-Hodographs from 0 to 110 rad/sec illustrating the gain margin.

The magnitude of $C(\omega)$ in Table 3.1 begins at .11 for 0 rad/sec and increases with frequency to a maximum of 9.39 at 150 rad/sec. Over the range from 97.5 to 225 rad/sec, $C(\omega)$ is greater than unity. This can be interpreted as an indication that in this region the velocity of the actuators will be significantly influenced by the adjacent bypass valve, not by the immediate bypass valve. In terms of the elements of the matrix $G(\omega)$, at these higher frequencies, one or both
Table 3-1: Magnitude and Phase Angle for C(iω) and IC(iω)

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<th>PHASE</th>
<th>MAG</th>
<th>PHASE</th>
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<td>27.63</td>
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<td>197.50</td>
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<td>24.56</td>
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<td>1.41</td>
<td>22.21</td>
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<td>1.26</td>
<td>29.32</td>
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<td>1.13</td>
<td>18.76</td>
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<td>1.64</td>
<td>17.46</td>
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<tr>
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<td>0.96</td>
<td>16.23</td>
<td>4.16</td>
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<tr>
<td>232.50</td>
<td>0.90</td>
<td>15.28</td>
<td>5.70</td>
</tr>
<tr>
<td>240.00</td>
<td>0.85</td>
<td>14.29</td>
<td>9.40</td>
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of the coupling terms exerts a significant influence over actuator control.

The gain was increased in the simulation until instability occurred for $K_{E1} = K_{E2} = .0011$ amps/in/sec. The velocity responses for these values of gain are illustrated in Figures 3-19 and 3-20. It is interesting to note that the velocity oscillations in the step response of the first actuator appear to limit cycle
HODOGRAPH - 1 -- C(ω), 2 -- INVERSE CONTROLLER

Figure 3-18: Method of 2-Hodographs, $K_{E1}, K_{E2} = 0.0007$
Figure 3-19: Velocity of Actuator 1, $\dot{X}_1, K_{E1}, K_{E2} = 0.011$

Figure 3-20: Velocity of Actuator 2, $\dot{X}_2, K_{E1}, K_{E2} = 0.011$
while the oscillations in the step response of the second actuator damp out, again suggesting that the closed loop feedback system of the first actuator in the series-bypass circuit is more oscillatory than the second actuator, for the chosen set of conditions and equal proportional gains. However, it is unlikely that the limit cycle is limited to the first closed velocity loop since a limit cycle in any part of the series-bypass circuit indicates poles in the right half plane of the system characteristic equation. It is probable that the second loop is limit cycling at a reduced amplitude.

Figure 3-21 shows that the Nyquist plot of expression (3.7) for these higher gains encircles the origin between 225 and 232 rad/sec, predicting an unstable system. It can be shown from the Nyquist stability criterion that the frequency of oscillations of an unstable system corresponds to the frequency at which the Nyquist plot encircles the origin. From Figure 3-19 it can be seen that the frequency of oscillations is approximately 210 rad/sec.

The Nyquist plot of the expanded determinant shown in Figures 3-22 and 3-23 also predict instability since the plot of expression (3.11) encircles the origin. As seen previously, the plot of $1 + q_{11}(i\omega)K_{E1}$ lies closer
to the origin than $1 + q_{22}(i\omega)K_{E_2}$, supporting the argument that the first velocity loop is more oscillatory than the second velocity loop in the series at these equilibrium conditions.

Table 3-2 lists the values for $C(i\omega)$ and $IC(i\omega)$ for gains of .0011 amps/in/sec. Again, a gain margin can be calculated in two frequency intervals, 97.5 to 105 rad/sec and 225 to 232.5 rad/sec. When these ranges were closely examined, it was found that for $\omega = 103$ rad/sec, the magnitudes of $C(i\omega)$ and $IC(i\omega)$ were 1.45
Figure 3-22: Nyquist Plot of Expressions (3.10),(3.11)

Figure 3-23: Nyquist Plot of Expressions (3.10),(3.11)
and 2.04 respectively while for \( w = 228 \) rad/sec the magnitudes were .93 and .61 yielding a gain margin of .66, obviously indicating instability.

<table>
<thead>
<tr>
<th>( \omega ) (rad/sec)</th>
<th>( w = \omega ) (rad/sec)</th>
<th>0.93</th>
<th>0.61</th>
</tr>
</thead>
<tbody>
<tr>
<td>228</td>
<td>228</td>
<td>0.93</td>
<td>0.61</td>
</tr>
</tbody>
</table>

Table 3-2: Magnitude and Phase Angle for \( C(i\omega) \) and \( IC(i\omega) \)

<table>
<thead>
<tr>
<th>( \omega ) (rad/sec)</th>
<th>( w = \omega ) (rad/sec)</th>
<th>0.93</th>
<th>0.61</th>
</tr>
</thead>
<tbody>
<tr>
<td>228</td>
<td>228</td>
<td>0.93</td>
<td>0.61</td>
</tr>
</tbody>
</table>

To summarize, the three frequency response methods were found to correctly predict the stability of a nonlinear multivariable closed loop system. In the examples above, all three methods agreed in predicting the stability of the simulated series-bypass system at
specific operating conditions and also were close to the time domain simulation limit cycle frequency. The Nyquist plot of the determinant of equation (3.5) requires a single plot to determine whether the system is stable or not. The expanded determinant can be used to determine system stability as well as to yield a qualitative description of the relative stability of the two loops. The method of 2-Hodographs is another graphical procedure which can be used for a quantitative measure of the loop interactions and permits a designer to compare different controller designs on the basis of a gain margin defined for the whole system. It is also used to predict instability. For large variations in the magnitude of the Hodographs and small gain margins it was found to be difficult to plot satisfactorily.

3.3.1 Relative Stability of the Two Closed Velocity loops

To investigate the relative stability of the two loops of the series-bypass circuit, one controller gain was held constant while the other was varied. For the first tests, $K_{E2}$ was held at $0.007 \text{ amps/in/sec}$ and $K_{E1}$ was increased to $0.0013 \text{ amps/in/sec}$. The Nyquist plot of the determinant in Figure 3-24 encircles the origin.
between 225 and 232 rad/sec. As shown in Figures 3-25 and 3-26 the simulation results agree with this prediction.

Figure 3-24: Nyquist Plot of Determinant, Expression (3.7), $K_{E1} = 0.0013, K_{E2} = 0.0007$

It is interesting to note in this figure that both the first and second loops are highly oscillatory, in contrast to Figures 3-19 and 3-20 where only the first loop was highly oscillatory. Also, the frequency of oscillations is in agreement with the Nyquist criterion prediction.

From the Nyquist plot of the determinant in Figure
Figure 3-25: Velocity of Actuator 1, $\dot{X}_1$

Figure 3-26: Velocity of Actuator 2, $\dot{X}_2$
3-27, it can be seen that the series-bypass system would be unstable for $K_{E1} = .0007$ and $K_{E2} = .0018$ amps/in/sec. In contrast, the nonlinear time domain response plotted in Figures 3-28 and 3-29 show that the system is stable for these values of gains. The method of 2-Hodographs and Nyquist plots of the expanded determinant also predicted instability for the above gains.

Figure 3-27: Nyquist Plot of Determinant, Expression (3.7), $K_{E1} = .0007, K_{E2} = .0018$

$K_{E2}$ was increased to .003 amps/in/sec where the Nyquist plot of the determinant shown in Figure 3-30 encircles the origin twice. Simulation results showed that both
Figure 3-28: Velocity of Actuator 1, $\dot{x}_1$

Figure 3-29: Velocity of Actuator 2, $\dot{x}_2$
Figure 3-30: Nyquist Plot of Determinant, Expression (3.7), $K_{E1} = .0007, K_{E2} = .003$

loops were unstable for these values of gain. These results were independent of the size of the step. This discrepancy between the simulation and the stability methods will be explained in the next section.

3.4 Effects of Step Magnitude, Equilibrium Velocity and System Pressures on Transient Response

To gain additional knowledge concerning the sensitivity of the velocity response of the series-
bypass system to changes in equilibrium values and step sizes, simulation studies were conducted with equilibrium pressures, initial velocities and velocity step inputs which were different from the previous values. From the development presented in sections 2.1 through 2.6, it can be seen that the linearized frequency response curves are independent of the velocity step size and equilibrium actuator velocity. Consequently, any difference in the simulated transient response at different mean velocities could be due either to the effect of the open loop term $\dot{X}_M A_F G_{pp}$ in the pump control algorithm or the nonlinearities in the conservation of mass equations discussed in section 2.2 and used in the simulation. Moreover, the linearized analysis ignored the pressure drops in the proportional four-way valves whereas the simulation included the corresponding valve flow equations.

Figures 3-31 and 3-32 show the response of the first and second actuator to a 4 in/sec increase in the desired velocity from an equilibrium actuator speed of 1 in/sec. The responses of both actuators to the 4 in/sec step input are very similar to the responses shown in Figures 3-6 and 3-7. The percent overshoots are slightly larger and the velocity oscillations require a longer period of time to damp out. In contrast, the unit step
Figure 3-31: Velocity of Actuator 1, $K_{E1}, K_{E2} = 0.0007$

Figure 3-32: Velocity of Actuator 2, $K_{E1}, K_{E2} = 0.0007$
about an equilibrium actuator speed of 1 in/sec, shown in Figures 3-33 and 3-34 display a much less damped response. Both the percent overshoot and decay envelopes are greater than those in Figures 3-6 and 3-7. This behavior was not expected, since it was not predicted by any previous analysis. Since the velocity errors were identical it was expected and confirmed by examining plots of bypass valve area, that the bypass valve movements were identical for a step input about an equilibrium value of 1 or 8 in/sec. If the bypass valve movements and equilibrium values (pressures, valve areas and piston locations) are identical, any dissimilarities in the transient response must be due to the difference in the pump flow rate, Q. The pump flow rate term \( \dot{M} A_p G_p \) assumes much less importance (in terms of magnitude) in comparison to the terms associated with the bypass flows when the desired actuator velocities are 1 in/sec rather than 8 in/sec and for a 1400 psi bypass valve pressure drop. Therefore, it appears that as the magnitude of the desired velocities decreases and correspondingly the contribution from the feedforward term, the system becomes less damped. At this time, the physical reason for the lower damping at decreased velocities is believed to be caused by the decreased pressure drops across the proportional valves.
Figure 3-33: Velocity of Actuator 1, $K_{E1}, K_{E2} = .0007$

Figure 3-34: Velocity of Actuator 2, $K_{E1}, K_{E2} = .0007$
At this point, the conflict between the stability prediction and simulation results for $K_{E1} = 0.0007$ and $K_{E2} = 0.0018$ amps/in/sec$^2$ was recalled. A relationship may exist between system stability and actuator equilibrium velocity which is not comprehended by the linearized frequency response analysis. This relationship may cause instability to occur in the series-bypass system for $K_{E1} = 0.0007$ and $K_{E2} = 0.0018$ amps/in/sec when the desired actuator speed is decreased below 8 in/sec. Figures 3-35 and 3-36 show the velocity response for a step input about 1 in/sec. The system is definitely unstable, agreeing with the Nyquist plot of the determinant displayed in Figure 3-27. Also, it is important to observe that the Nyquist plot in Figure 3-27 encircles the origin between 97 and 105 rad/sec. The frequency of oscillations in Figures 3-35 and 3-36 are in this range.

The next question which needs to be addressed concerns whether a stable system having gains selected by using the criteria from one of the frequency response methods above will exhibit instability when the desired velocity is decreased. Simulation studies were performed at low actuator velocities with gains which were slightly below $K_{E1} = 0.0011$ and $K_{E2} = 0.0007$ amps/in/sec and $K_{E1} = 0.0007$ and $K_{E2} = 0.0018$ amps/in/sec. Figure 3-37 is a Nyquist plot of the determinant for $K_{E1} = 0.0007$ and
Figure 3-35: Velocity of Actuator 1, $\dot{x}_1$

Figure 3-36: Velocity of Actuator 2, $\dot{x}_2$
Figure 3-37: Nyquist Plot of Determinant

Figure 3-38: Nyquist Plot of Determinant
$K_{E2} = 0.0013 \text{ amps/in/sec}$. The plot predicts a stable system, which is verified by simulation results for a step input from 1 in/sec to 2 in/sec. To investigate the system when the first gain is greater than the second, Figure 3-38 shows the Nyquist plot of the determinant for $K_{E1} = 0.001$ and $K_{E2} = 0.0007 \text{ amps/in/sec}$. This plot also indicates a stable system which is also in agreement with the simulated velocity responses to a step input about an equilibrium velocity of 1 in/sec shown in Figures 3-39 and 3-40.

Stability investigations were conducted at other equilibrium bypass valve areas and pressures. No differences between Nyquist stability predictions and the simulation results were noticed in stability studies concerning the first actuator. The stability discrepancies concerning the second actuator did not appear at decreased equilibrium pressures and increased bypass valve areas. In addition, the prototype system was simulated using the linearized equations of (2.71). No discrepancies were noted between the Nyquist stability predictions and the system response. Therefore it is hypothesized that the discrepancy may be the result of neglecting the proportional valve pressure drops. The simulated system approaches the condition of zero pressure drops across the proportional valves as
Figure 3-39: Velocity of Actuator 1, $\dot{X}_1$

Figure 3-40: Velocity of Actuator 2, $\dot{X}_2$
the actuator velocities decrease. This corresponds to
the observation that the linearized analysis predictions
agree with the simulation behavior at low mean actuator
velocities. At higher mean actuator velocities, the
pressure drops across the proportional valves are
higher, resulting in additional damping of the simulated
response and greater stability than is predicted by the
linear analysis.

To illustrate the effect of pressure on the system
time response, the external loads $F_1$ and $F_2$ were reduced
to 1640 lbf, which corresponds to a bypass valve
pressure drop of 400 psi. The velocity step responses of
actuator 1 for $K_{E1} = K_{E2} = .0007 \text{ amps/in/sec}$ is shown in
Figure 3-41. As predicted by the analysis of the open
loop symbolic transfer functions in section 2.6, the
response is less oscillatory due to the lower system
pressures. The speed of response has also decreased.
The gain margin calculated from the method of two
hodographs was larger than the gain margin calculated
when the cylinder pressure was 1400 psi, which indicates
a system with a greater degree of stability.

In summary, the transient response of the
series-bypass appears to be significantly dependent on
the desired operating speed and external load. As the
system pressures decrease, the system response time
decreases and the response becomes increasingly damped. The relationship between system pressure and dynamic performance was explained in section 2.6. Comparing the time response plots, the rise time does not appear to be a function of the desired speed. However as the desired velocity decreases or system pressure increases, the velocity response to a step input becomes increasingly oscillatory, less damped and as shown above, less stable.

The evidence suggests that the linearized model
used and the frequency response methods which determine stability yield conservative values for the stability limits on the controller gains, which are independent of actuator speed. The gains can apparently be increased above these limits, depending on the magnitude of the desired velocity. The absolute upper limit appears to be the value of gain which causes the Nyquist plot of the determinant to encircle the origin. It is felt that this observation is not general and would not apply to a series-bypass circuit with more than two closed loop systems.

At the conditions listed in Table 2-4, both the simulation and frequency response criteria suggest that the stability of the series-bypass configuration is primarily determined by the first closed loop system in the series. Simulations show that the first system can be highly oscillatory when the second exhibits less oscillations but no simulations showed the reverse. When one proportional gain was held constant and the other proportional gain was increased until instability occurred, the increase in the gain for the first control loop was less than for the second.
3.5 Lead Compensation of Series-Bypass System

Quantitative performance requirements for the series bypass configuration have not been specified. A nontechnical description could be stated "as fast as the system will respond, as accurate as possible, using the existing hardware". In regard to accuracy, the series-bypass circuit has displayed essentially zero steady state errors in the presence of equal and constant loads and velocity commands. The accuracy of the system in situations of unequal velocity commands and load sharing will be addressed in section 3.6. This section will be concerned with increasing the system's speed of response, in an attempt to minimize the rise and settling times.

Classical control theory advocates lead compensation to increase a system's speed of response. Filters \( G_1(s) \) through \( G_9(s) \) in Figure 3-2 are possible locations for these compensators. The velocity feedback, equilibrium valve area, acceleration feedback and commands to the servovalve and pump can all be filtered to offer the greatest possible generality in terms of compensation. The compensators in this figure are not the typical cascade compensators because of the introduction of the variable displacement pump and
acceleration feedback into the control scheme. For these reasons, it is difficult to apply the classical cascade compensation rules to design lead compensators. It can be argued that the algorithm to control the pump flow rate (equation (2.48)) does not introduce any closed loop feedback variables which are different from those in the bypass valve control equations (2.44) and (2.45). Equation (2.48) includes the feedforward term $\dot{X}_M A_P G_{pp}$ in addition. Therefore, the pump is not an independent actuator because its output $Q$ is a function of the same independent input variables $\dot{X}_{1D}$ and $\dot{X}_{2D}$ which are used in the closed loop feedback system involving the bypass valves. Certainly in the system matrix representation of equation (2.66) in section 2.4, the pump as an actuator does not explicitly appear in the final form. The system becomes a two-input, two-output multivariable system with only two closed loop actuators and feedforward action included.

Since the immediate objective is to increase the speed of the total system and because the symbolic transfer functions, open loop frequency response and closed loop time response indicated that each loop behaves similarly, the lead compensators were designed ignoring the interaction between the systems. For single-input, single-output closed loop feedback systems
lead compensators can be designed using the root locus or frequency response of the open loop system. The root locus approach involves solving for the poles and zeros of the open loop transfer function, which requires the open loop symbolic transfer functions. As discussed in section 2.5 the symbolic transfer functions are not easily obtained, when servovalve and pump dynamics and acceleration feedback are included in the system model. However, the open loop frequency response curves for the system were obtained in section 2.6 and 2.7 from SPEAKEASY. It was decided to use these magnitude and phase angle diagrams as an aid in designing the lead compensators. Although these compensators will be designed using the frequency response plots, they will be evaluated by their affect on the closed loop step response of the system. A lead compensator increases the system bandwidth and it is commonly recognized that a high system bandwidth is analogous to a high speed of response. The time domain evaluation was used because a step response is easily included in the time domain ACSL model and is representative of an actual operating input.

The transfer function for a lead compensator can be written as
\[ \frac{Ts + 1}{\alpha Ts + 1} \]  \hspace{1cm} (3.19)

where \( T \) is referred to as the time constant of the compensator [12] and \( 1/T \) is referred to as the breakpoint frequency [14] of the numerator and \( 1/\alpha T \) is the breakpoint frequency of the denominator. \( \alpha \) is a constant less than 1 which is equal to the ratio of the numerator and denominator breakpoint frequencies. For most applications, \( \alpha \) is set equal to .1. As \( \alpha \) is decreased, the lead compensator approaches a pure proportional plus derivative controller which can worsen the noise response of the system.

For a type zero system such as the series-bypass circuit, D'Azio and Houpis [12] recommend that in the frequency domain, the breakpoint frequency of the numerator of the lead compensator should be placed around the second largest natural frequency of the system. Examination of the open loop frequency response plots of \( \hat{X}_{1p}/A''_{2Cp}(i\omega) \) and \( \hat{X}_{2p}/A''_{4Cp}(i\omega) \) shown in Figures 2-14 and 2-15 reveal two easily distinguishable natural frequencies, at approximately 85 and 230 rad/sec. The second largest of these would be at 80 rad/sec. The appropriate value of the time constant in equation (3.19) would be .0125 seconds.

The filters \( G_c(s) \) through \( G_9(s) \) can be classified
as belonging to fundamental classes based on their locations.

1. Before the commanded bypass valve openings $A_{2C}$ and $A_{4C}$ are used to compute the pump flow rate ($G_3^C(s)$, $G_4^C(s)$, $G_5^C(s)$ and $G_6^C(s)$).

2. Before the commanded bypass valve openings $A_{2C}$ and $A_{4C}$ are output to the appropriate bypass valve ($G_1^C(s)$, $G_2^C(s)$, $G_7^C(s)$ and $G_8^C(s)$).

3. Before the commanded pump flow rate $Q_C$ is output to the pump ($G_9^C(s)$).

Since the variable displacement pump has been shown to be the slowest actuator, a lead compensator was initially placed at $G_9^C(s)$. The time constant of the compensator was varied between .02 and .005 seconds and the effects were studied in the simulation. Figures 3-42 and 3-43 show the step response of actuator 1 with lead compensators having time constants of .01 and .02 seconds respectively. As expected, both closed loop systems of the series-bypass circuit behaved similarly. Therefore only the response of actuator 1 is shown.
Figure 3-42: Velocity of Actuator 1, Lead Filtered

Figure 3-43: Velocity of Actuator 1, Lead Filtered
Comparing these two figures with Figure 3-6 it can be seen that as the time constant increases (numerator breakpoint frequency decreases) from .01 to .02 seconds the velocity response becomes faster (rise time decreases), but also more oscillatory.

To compensate for this oscillatory behavior, lead compensation was applied to the bypass valve opening. System damping has been shown to be proportional to the bypass valve opening. In a dynamic situation, such as a step change in velocity input, a lead compensator after $A_2C$ and $A_4C$ would increase the magnitude of the changes in valve position, increasing the system damping during transients, and thereby decreasing the oscillations. To implement this filter, $G_3^c(s)$, $G_4^c(s)$, $G_5^c(s)$ and $G_6^c(s)$ were set equal to 1 and lead compensators of the form of equation (3.19) with $\alpha=0.1$ seconds were installed at $G_1^c(s)$, $G_2^c(s)$, $G_7^c(s)$ and $G_8^c(s)$ using various time constants. Figure 3-44 shows the system response with $\tau = .01$ seconds and the pump filter at $G_9^c(s)$. From this figure, it can be seen that increased damping has been achieved with approximately the same speed of response as in Figure 3-43. Simulation studies showed that the system response was unaffected for $\tau < .005$ seconds. The system speed of response decreased for $\tau > .01$ seconds. In an attempt to decrease the large velocity
overshoots seen in Figures 3-43 and 3-44, lead compensators of the form of equation (3.19) were inserted at $G_3(s)$, $G_4(s)$, $G_5(s)$ and $G_6(s)$ to filter the bypass valve command input to the pump. Since this bypass valve command was computed using acceleration feedback, this should increase the damping in the system, hopefully decreasing the overshoot and also ensuring that the commanded bypass areas to the pump and servo valve would have the same approximate magnitude at any instant of time. Again, the time constant for this
compensator was varied and the effects on the velocity response was studied from the simulation results. On the basis of this study, a time constant of .02 seconds was selected for this lead filter. Figure 3-45 shows the velocity response using filters $G_1(s)$ through $G_9(s)$ as specified above.

figure 3-45: Velocity of Actuator 1, Lead Filtered

As the time constant of $G_3(s)$, $G_4(s)$, $G_5(s)$ and $G_6(s)$ was increased to .02 seconds, the velocity response displayed the discontinuity circled in Figure 3-45. This effect became more pronounced as this time
constant was increased. As this time constant is increased, the numerator breakpoint frequency of the lead compensator decreases, increasing the magnitude of the filter output over the frequency range of interest (0 to approximately 250 rad/sec). Time history plots of actuator velocity and acceleration, commanded and actual bypass valve area and pump flow rate were examined to determine the cause of this discontinuity. From these plots it was observed that as the actual velocity approached the desired velocity the bypass valve area increased, since the error term in equations (2.44) and (2.45) is decreasing and the actuator acceleration is positive. This increase in area is accentuated by the lead filter and used as input to the commanded bypass valve area. This increased bypass valve area has a double effect, permitting more flow to bypass and at the same time decreasing the pump output. This significant decrease in flow rate to the actuator causes the velocity discontinuity circled in Figure 3-45. After several corrective methods were tried, the best solution found was to omit the acceleration feedback term in the calculation of the bypass valve area sent to the pump and also reduce the proportional gain from .0007 to .0005 amps/in/sec. Both these changes increase the amount of flow to the actuator as the velocity error
approaches zero. The result is shown in the velocity responses of actuators 1 and 2 plotted in Figures 3-46 and 3-47 respectively. Comparing these figures to Figures 3-6 and 3-7 it can be seen that the system responds faster and with greater stability when lead compensation is used. The final lead compensation design is tabulated in Table 3-3.

Table 3-3: Lead Filter Time Constants, Equation (3.19), α=.1

| $G_1(s)$, $G_2(s)$, $G_7(s)$, $G_8(s)$ | $\tau = .01$ seconds |
| $G_3(s)$, $G_4(s)$, $G_5(s)$, $G_6(s)$ | $\tau = .01$ seconds |
| $G_9(s)$ | $\tau = .02$ seconds |

3.6 Reduction of Velocity Errors Using Integral Control

3.6.1 Velocity Errors Due to Unequal Velocity and Load Inputs

Figures 3-48, 3-49, 3-50 and 3-51 show the response of the series-bypass system to unequal velocity inputs. No lead compensation was used to obtain these simulation results. Figures 3-48 and 3-49 are plots of the desired
Figure 3-46: Velocity of Actuator 1, Lead Filtered

Figure 3-47: Velocity of Actuator 2, Lead Filtered
and actual velocities of the two actuators when only the first actuator is subjected to a step increase in the desired velocity. Figures 3-50 and 3-51 show the actuator velocities when only the second actuator is subjected to a step increase in velocity. Comparing all four figures it can be seen that, except for the differences in dynamic characteristics, the velocity responses of the actuators which are subjected to the step input are independent of their location in the series circuit. The same statement can be made regarding the velocity responses of the actuators which are not subjected to a step input in desired velocity. Sections 2.6 and 2.7 predicted equal open loop steady state velocities (not including $\dot{X}_M$) when the actuators are subjected to equal loads, and equal proportional gains and bypass valve areas. Note that the steady state error of the actuator subjected to the step input is much less than the velocity error of the undisturbed actuator. This is due to the increase in pump flow rate caused by the step increase in the maximum desired velocity in the circuit. This additional flow is necessary to increase the velocity of the actuator subjected to the step input but must be bypassed by the actuator which did not receive the step input.

Previous figures have shown that low proportional
Figure 3-48: Velocity of Actuator 1, $K_{E1} = K_{E2} = 0.005$

Figure 3-49: Velocity of Actuator 2, $K_{E1} = K_{E2} = 0.005$
Figure 3-50: Velocity of Actuator 1, $K_{E1},K_{E2}=.0005$

Figure 3-51: Velocity of Actuator 2, $K_{E1},K_{E2}=.0005$
gains result in inadequate bypass valve areas. For proportional control systems, if the proportional gains are increased, the steady state errors should decrease. Figures 3-52 and 3-53 verify this by showing that the nominal values of actuator velocity are closer to their desired values when $K_{E1}$ and $K_{E2}$ are increased to .001 amps/in/sec. However, as expected, the increase in proportional gains has decreased the system stability. Again, the responses shown in these figures were not lead filtered.

As described previously, the actuators of the series-bypass circuit will be subjected to symmetrical but opposite load variations about an equilibrium value as the leg travels along its contact trajectory. One of the perceived advantages of the series-bypass configuration is the sharing of these load variations throughout the stride such that the pressure at the pump outlet remains constant. In order to determine the response of the series-bypass circuit to this type of disturbance input, the load variation as a function of leg position was represented by the following equations in the simulation.
Figure 3-52: Velocity of Actuator 1, $K_{E1}, K_{E2} = .001$

Figure 3-53: Velocity of Actuator 2, $K_{E1}, K_{E2} = .001$
\[ F_1 = F_{le} + \left( \frac{\text{Fraction}}{F_{le}} \right) \left( \sin \left( \frac{X_1}{\text{Stridelength}} \right) \right) \]  
(3.20)
\[ F_2 = F_{2e} + \left( \frac{\text{Fraction}}{F_{2e}} \right) \left( \sin \left( \frac{X_2}{\text{Stridelength}} \right) \right) \]  
(3.21)

Figures 3-54 and 3-55 show the velocity response of the actuators when the first actuator encounters a sinusoidally varying load having a peak amplitude 50% greater than the equilibrium value of -5740 lbf and the second actuator is subjected to a load which varies in a similar manner but decreases by 50%. No lead compensation was used in these figures.

It can be seen from these figures that these load variations cause velocity errors of up to 15%. These velocity errors are not due to the opposing resistance of the disturbance loads on the actuator as in a conventional servo controlled actuator but primarily from the effect of the unequal disturbance loads on the bypass flows and to a lesser extent, the change in compressibility flows due to the dynamic variation of the loads. Increasing loads on the actuators cause negative velocity errors while decreasing actuator loads result in positive errors.

The major part of the error is due to the difference in the pressure drops across the bypass valves. This difference is a function of the unequal
Figure 3-54: Velocity of Actuator 1, Increasing Load

Figure 3-55: Velocity of Actuator 2, Decreasing Load
Figure 3-56: Volumetric Flow Rates, $F_1$ Increasing

Figure 3-57: Volumetric Flow Rates, $F_2$ Decreasing
loads seen by each actuator. Figures 3-56 and 3-57 (see Figure 2-1 and symbol definitions in the Nomenclature) show the resulting flow rates in and out of the actuators and the bypass flows across the actuators. The flows $Q_{11}$ and $Q_{31}$ are the flow rates into the actuators, $Q_{12}$ and $Q_{32}$ are the flow rates out of the actuators and $Q_2$ and $Q_4$ are the bypass valve flow rates. An increasing pressure drop allows more flow through the valve, subtracting from the flow to the actuator and decreasing the velocity. A decreasing pressure drop will lower the flow rate through the bypass valve, providing additional flow to the actuator and increasing its velocity. As the actuator velocity deviates from the desired value, the bypass valve area will change proportionally in an attempt to correct the velocity error. Figures 3-58 and 3-59 show the bypass valve areas in response to the increasing load on actuator 1 and decreasing load on actuator 2. Due to the low values of proportional gains the valve areas are not adequate to compensate for the change in pressure across the piston. Figures 3-60 and 3-61 show the velocity response to identical loading conditions but with $K_{E1}$ and $K_{E2}$ equal to .001 amps/in/sec and without lead compensation. As shown by the figures, the mean velocity errors have decreased but so has the system stability.
Figure 3-58: Bypass Valve Area, $A_2$, Increasing Load

Figure 3-59: Bypass Valve Area, $A_4$, Decreasing Load
Figure 3-60: Velocity of Actuator 1, $K_{E1}K_{E2} = 0.01$

Figure 3-61: Velocity of Actuator 2, $K_{E1}K_{E2} = 0.01$
Figure 3-62 shows the nonsymmetric loading which has occurred during the stride. The reason for this nonsymmetric loading can be seen from the plot of the actuator positions in Figure 3-63. Notice that due to the positive velocity errors, the position of the second actuator leads the first. Since the load specified by equations (3.20) and (3.21) is a function of their piston position, the load cycle defined by equation (3.21) for the second actuator leads the first. The net result is that the pressure at the pump outlet, $P_L$, shown in Figure 3-64 increases after .45 seconds. The pressure variations in the second actuator and in the connecting line between the first and second actuators, $P_L$, are plotted in Figure 3-65.

Figure 3-55 shows that the velocity of the second actuator is higher than the desired value during the time period 0 to .5 seconds and then decreases below the desired value even though the magnitude of the opposing force shown in Figure 3-62 is less than the equilibrium value. This phenomenon illustrates the second type of velocity errors; those caused by the unequal dynamic load characteristics of the loading. As illustrated above, the system pressures increase for time greater than .5 seconds. The fluid compressed in the actuator and line volumes is directly proportional to magnitude.
Figure 3-62: Actuator Loads, $F_1, F_2$

Figure 3-63: Actuator Positions, $X_1, X_2$
Figure 3-64: Fluid Pressures, $P_1, P_{11}, P_{12}$

Figure 3-65: Fluid Pressures, $P_L, P_{21}, P_{22}$
of the pressure gradient. Specifically, as the pressures $P_{2L}$ in the LHS of actuator 2 and in the connecting line $P_L$ increase, a greater amount of fluid is being stored. Consequently, this flow is not available to the actuator and the velocity of second actuator drops below the desired value.

This decrease in velocity is not evident in the plot of the velocity of actuator 1 because its velocity is already below the desired value. Correspondingly the bypass valve area is less than its equilibrium value and according to equation (2.44), this will increase the flow rate from the pump as well as diverting a comparatively greater flow to the actuator. Another reason for the relative independence of the first actuator with regard to the dynamic pressure gradients can be attributed to the fact that the pressure gradients shown in Figure 3-64 have the same sign on both sides of the piston. Therefore the compressibility effects tend to cancel out, in contrast to the second actuator which in this example has an essentially constant pressure (return) in the RHC.

Figures 3-66 and 3-67 are plots of the velocity response of the series-bypass configuration to a sinusoidally increasing force on the second actuator and a decreasing force on the first. As in Figures 3-54 and
the increasing load on the second actuator creates negative velocity errors while the decreasing load on the first actuator causes positive velocity errors. It can be seen from Figure 3-67 that the speed of actuator 2 increases above its desired value as the load on the actuator approaches the equilibrium value. This overshoot can again be attributed to the dynamic characteristics of the fluid in the cylinders and connecting lines. Again, no lead filters were used to obtain the velocity responses shown in Figures 3-66 and 3-67.

Figures 3-68 and 3-69 show that as before, the pump outlet pressure, $P_1$, increases because of the net load increase caused by the non constant relative leg displacements. The decreasing pressures in the connecting line $V_2$ and the LHC of the second actuator after .40 seconds will increase the amount of fluid available to the actuator. This will increase the actuator velocity and cause the velocity overshoot seen in Figure 3-67.

From Figures 3-68 and 3-69 it can be seen that after .45 seconds, the pressure in the fluid volume between the pump and first actuator $V_1$, is increasing while the pressure in the series line $V_2$ is decreasing. This opposing pressure gradient on the opposite ends of
Figure 3-66: Velocity of Actuator 1, $F_1$ Decreasing

Figure 3-67: Velocity of Actuator 2, $F_2$ Increasing
Figure 3-68: Fluid Pressures, $P_1, P_{11}, P_{12}$

Figure 3-69: Fluid Pressures, $P_L, P_{21}, P_{22}$
the first piston will assist the control effort of the bypass valve in slowing the actuator. Consequently it would be reasonable to expect a velocity undershoot for actuator 1 if the time limit of Figure 3-66 were increased past .7 seconds. Recall that for the previous loading example, the pressure gradients on both sides of the piston were in the same direction and therefore the dynamic effects of the fluid on the actuator velocity would tend to cancel.

3.6.2 Design of an Integral Controller

To summarize the results of the previous section, unequal desired velocity inputs and loading distributions introduced velocity errors when the existing control algorithms are used to control the system. An increase in the proportional gains decreased these errors but the system became increasingly unstable. In single input, single output systems, integral control is commonly used to reduce errors due to disturbances or steady state errors similar to those caused by unequal desired velocity and load inputs. This integral controller can be incorporated into the existing control structure by cascading an integrating algorithm in the forward loop or incorporating an integrating algorithm parallel to the existing
proportional controller. Because the previous acceleration gains and lead filters had been selected on the basis of a proportional controller it was decided to incorporate an integral controller in a parallel path, rather than in series. The resulting controller is a PI controller. This method permits more flexibility in designing the control algorithms around the integral controller. The basic integral controller was implemented in the simulation as:

\[ A_{2,\text{INT}} = -K_{\text{INTEGRAL}} s f(\dot{x}_{1D} - \dot{x}_1) \, dt \quad (3.22) \]
\[ A_{4,\text{INT}} = -K_{\text{INTEGRAL}} s f(\dot{x}_{2D} - \dot{x}_2) \, dt \quad (3.23) \]

Equations (3.22) and (3.23) were added to equations (2.44) and (2.45) respectively. Various options exist as to whether this term would be lead filtered and the specification of the magnitude of \( K_{\text{INTEGRAL}} \). All the velocity responses shown in this section were lead compensated. The simulation was exercised using various combinations of filters and values of \( K_{\text{INTEGRAL}} \). To summarize the system velocity responses for various integral configurations tested, the actuator velocity response became increasingly oscillatory when the integral gain was increased or when the integral term was lead filtered. A compromise was obtained by passing the integral terms through the same lead filters as...
equations (2.44) and (2.45) (parameters listed in Table 3-3) and decreasing $K_{\text{INTEGRAL}}$ below its unfiltered value. Figures 3-70 and 3-71 show the velocity response to a step input in the desired velocity of actuator 1 when $K_{\text{INTEGRAL}} = 0.04$ amps/in. As shown by the figures, the steady state velocity errors gradually go to zero, an improvement over Figures 3-48 and 3-49. The system speed of response has not been affected by the addition of the integral controller but the percent overshoot of the system subjected to the step input has increased. Note that the velocity of the second actuator increases sharply due to the step increase in pump flow rate but the integral control term reduces the error to zero.

As shown above, integral control successfully decreased the velocity errors for unequal velocity inputs to the series-bypass circuit. The large velocity errors which occurred for the load sharing examples above should also be decreased by the integral controller. Figures 3-72 and 3-73 show the velocity response to the load variations shown in Figure 3-74. Comparing Figure 3-72 to 3-54 it can be seen that the velocity errors are reduced but the integral controller has produced positive errors in the response. The negative velocity errors are caused by the dynamic load variation on actuator 1 and were discussed above. After
Figure 3-70: Velocity of Actuator 1, Integral Control

Figure 3-71: Velocity of Actuator 2, Integral Control
Figure 3-72: Velocity of Actuator 1, Integral Control

Figure 3-73: Velocity of Actuator 2, Integral Control
the force peaks and begins to decrease (after .4 seconds) the velocity of actuator 1 increases above the desired value because the output from the integral controller has decreased the bypass valve to compensate for the negative velocity errors. Due to the integral nature of the controller, some positive velocity errors must occur to subtract from the effect of the negative velocity errors. The bypass valve area of actuator 1, $A_2$ is shown in Figure 3-75. Actuator 2 exhibits the same type of positive and negative velocity error behavior,
but as seen previously, has a much greater overshoot when Figure 3-73 is compared to 3-72. However, because of the integral controller, the velocity errors in Figure 3-73 are much less than those of Figure 3-55.

Figures 3-76 and 3-77 are plots of the velocity response of the actuators to the external loading shown in Figure 3-78. The characteristics of the velocity response to the external loads are very similar to the velocity responses to unequal velocity inputs shown above. Again, positive and negative velocity errors
Figure 3-76: Velocity of Actuator 1, Integral Control

Figure 3-77: Velocity of Actuator 2, Integral Control
occur, but their magnitudes are much less than the system response without the integral controller, shown in Figures 3-66 and 3-67. Also, as seen above, the velocity responses display a larger overshoot because of the addition of the integral controller.
3.7 Consideration of Total System Response

From equation (3.5), the expression for \( \dot{X}_p \) can be written as:

\[
\dot{X}_p = [I+G(s)K(s)K_1(s)B^{-1}G(s)K(s)K_1(s)]\dot{X}_{dp} \\
+ [I+G(s)K(s)K_1(s)B^{-1}M(s)]\dot{X}_{mp} \\
+ [I+G(s)K(s)K_1(s)B^{-1}D(s)]L_p
\]  

Equation (3.5) and Figure 3-1 show that \( \dot{X}_p \) is derived by contributions from the closed loop velocity term, \([I+G(s)K(s)K_1(s)B^{-1}G(s)K(s)K_1(s)]\), the feedforward term, \([I+G(s)K(s)K_1(s)B^{-1}M(s)]\) and the contribution from the external loads \([I+G(s)K(s)K_1(s)B^{-1}D(s)]\). The following chapter will focus on decoupling the closed loop term to facilitate individual leg motion control. In this section, the effects of the feedforward and closed loop terms on the system response are presented and analyzed and methods to improve the system response to these two terms are presented and implemented.

3.7.1 Interaction of Feedforward and Closed Loop Controllers

It is enlightening to investigate how the feedforward and closed loop components contribute to the actual actuator velocities. Using the equilibrium
conditions in Table 2-4, a unity precompensation matrix, proportional gains of .0005 amps/in/sec and no integral control or lead filters, the frequency response curves of $|\dot{X}_{1p}/\dot{X}_{1Dp}(i\omega)|$, $|\dot{X}_{1p}/\dot{X}_{2Dp}(i\omega)|$, $|\dot{X}_{2p}/\dot{X}_{1Dp}(i\omega)|$, $|\dot{X}_{2p}/\dot{X}_{2Dp}(i\omega)|$, $|\dot{X}_{1p}/\dot{X}_{Mp}(i\omega)|$, and $|\dot{X}_{2p}/\dot{X}_{Mp}(i\omega)|$ are displayed in Figures 3-79 thru 3-84. These curves show that the closed loop response of actuator 1 is significant to 250 rad/sec, much greater than the closed loop frequency response of actuator 2. The contributions of the off diagonal terms in Figures 3-80 and 3-81 are also significant to 250 rad/sec. As noticed in section 2.7, the frequency contributions of the feedforward terms (approximately 100 rad/sec) are less than those of the closed loop terms due to the differences in the dynamic response of the variable displacement pump and bypass valve. For this value of proportional gain however, the magnitude of the feedforward contribution in Figures 3-83 and 3-84 is greater than the closed loop.

To display the relative contribution of these two control elements and also illustrate agreement between the time response simulation and linearized frequency response curves, the steady state values of the closed loop and feedforward matrices in equation (3.5) are shown in equation (3.24).
Figure 3-79: Magnitude of Closed Loop Frequency Response, \(|\tilde{X}_p/\tilde{X}_1D_p(i\omega)|\)

Figure 3-80: Magnitude of Frequency Response, \(|\tilde{X}_p/\tilde{X}_2D_p(i\omega)|\)
Figure 3-81: Magnitude of Frequency Response, $|\dot{X}_2p/\dot{X}_1Dp(i\omega)|$

Figure 3-82: Magnitude of Closed Loop Frequency Response, $|\dot{X}_2p/\dot{X}_2Dp(i\omega)|$
Figure 3-83: Magnitude of Frequency Response, $|\hat{X}_p / \hat{X}_m(i\omega)|$

Figure 3-84: Magnitude of Frequency Response, $|\hat{X}_2 / \hat{X}_m(i\omega)|$
For a unit step increase in desired velocity for both actuators and constant loads \((F_{1p}, F_{2p}=0)\), the steady state velocities of the first and second actuators is 1 in/sec, yielding a zero steady state error. This agrees with the time response plots of Figures 3-3, 3-4, 3-6 or 3-7. For a step increase in the desired velocity of the second actuator alone, the predicted steady state perturbation velocities for the first and second actuators are .744 and .935 in/sec respectively. These values agree with the time domain plots in Figures 3-50 and 3-51.

If the proportional gains are increased to .005 amps/in/sec, the steady state values in equation (3.25) are obtained.

\[
\begin{align*}
\begin{bmatrix}
\dot{X}_{1p} \\
\dot{X}_{2p}
\end{bmatrix}
&= 
\begin{bmatrix}
.764 & .061 \\
.061 & .764
\end{bmatrix}
\begin{bmatrix}
\dot{X}_{1Dp} \\
\dot{X}_{2Dp}
\end{bmatrix}
+ 
\begin{bmatrix}
.236 & -.061 \\
-.061 & .236
\end{bmatrix}
\begin{bmatrix}
\dot{X}_{Mp} \\
\dot{X}_{Mp}
\end{bmatrix}
\end{align*}
\]

(3.25)

Note that the relative magnitudes of the closed loop terms in equation (3.25) have increased when compared to the feedforward terms. Also note that the magnitudes of the off diagonal closed loop terms have not increased
because of the increase in gain. The higher steady state values of the closed loop velocity feedback system reflect the increase in the commanded bypass valve area made possible by using large proportional gains for a given error (see equations (2.40) and (2.41)). The increased bypass valve area permits the closed loop feedback system to exert greater control of the output. Consequently, the feedforward contribution to the actual velocity for a unit step in desired velocity will not have to be as large. Because both the pump and bypass valve are used as control actuators, it would be difficult to assign an increased amount of control action to one actuator as the gains are increased.

It is also important to note that the steady state values of \( \dot{x}_{1p} \) and \( \dot{x}_{2p} \) are unity for unit step perturbations of the desired velocities, which are identical with the steady state responses when proportional gains of .0005 amps/in/sec were used. The open loop investigations in sections 2.6 and 2.7 indicated equal steady state actuator velocities for equal actuator loads and bypass valve areas. For practical purposes (see section 3.7.3), the zero steady state closed velocity loop error is independent of the proportional gain for equal actuator loads and bypass valve areas. The pump and bypass valve control
algorithms were specifically formulated to achieve zero steady state error at these conditions. The advantages of higher proportional gains are the increased decoupling (shown in Chapter 4), increased closed loop frequency response (not shown) and decreased velocity errors for unequal actuator loads. However, simulation and frequency response studies have shown the system to be unstable at gains slightly above .0005 amp/in/sec for this set of conditions. Therefore the advantages of a high gain system will never be realized and basically the actuators are primarily controlled by the feedforward loop (pump controlled) in the system.

Sections 2.6 and 2.7 showed that the magnitude of the transfer functions between the bypass valves and actuator velocities decreased at low system pressures and high bypass valve areas. The steady state velocity loop values for 100 lbf actuator loads and .03 in$^2$ bypass valve areas are shown in equation (3.26).

\[
\begin{bmatrix}
\dot{X}_{1P} \\
\dot{X}_{2P}
\end{bmatrix} =
\begin{bmatrix}
.042 & .016 \\
.016 & .042
\end{bmatrix}
\begin{bmatrix}
\dot{X}_{1Dp} \\
\dot{X}_{2Dp}
\end{bmatrix} +
\begin{bmatrix}
.958 & -.016 \\
-.016 & .958
\end{bmatrix}
\begin{bmatrix}
\dot{X}_{Mp} \\
\dot{X}_{Mp}
\end{bmatrix}
\]

(3.26)

Note that for $K_{E1} = K_{E2} = .0005$ amps/in/sec, the contribution of the closed loop terms to the velocity response is much less than the open loop contribution.
In section 4.6 it will be shown that the magnitude of the closed loop terms can be increased significantly by increasing the proportional gains. The permissible increases in the gains are greater at low pressures and large bypass valve areas because the system is more stable at these conditions.

3.7.2 Decoupling the System Response to $\dot{X}_{mp}$

The previous sections have documented the system response to the closed loop feedback term in equation (3.5). In the next chapter, an attempt will be made to decouple the effect of the bypass valve commands on adjacent actuators. The second term in equation (3.5) represents the feedforward command of the maximum desired velocity input to the variable displacement pump. As the power source of the series-bypass circuit, the pump must provide the flow rate required by the maximum desired velocity. As shown in section 3.6, unequal velocity commands cause steady state velocity errors. Section 3.6.2 showed that these velocity errors could be decreased by increasing the proportional gains and adding an integral control algorithm in the commanded bypass valve area. In this section another algorithm will be introduced which will decouple the
actuators specifically for unequal velocity commands.

Figures 3-83 and 3-84 show the frequency response curves for $|\hat{x}_{1p}/\hat{x}_{np}(i\omega)|$ and $|\hat{x}_{2p}/\hat{x}_{np}(i\omega)|$ respectively. For example, a step increase in the desired velocity of the second actuator will affect the velocity of the first actuator according to $|\hat{x}_{1p}/\hat{x}_{np}(i\omega)|$. It is desirable to prevent this disturbance in the velocity of the first actuator by eliminating the dependence of $\hat{x}_{1p}$ on $\hat{x}_{np}$ when $\hat{x}_{1p} \neq \hat{x}_{np}$. Two methods have been commonly proposed in the past (model matching techniques [47, 52]), for multivariable systems to filter an output from undesirable influences of this nature. One method involves the insertion of a cascade filter which would have a frequency response transfer function $|\hat{x}_{np}/\hat{x}_{1p}(i\omega)|$. The second method would use a parallel filter with the frequency response transfer function $-|\hat{x}_{1p}/\hat{x}_{np}(i\omega)|$. For this system a cascade filter could not be physically introduced into the hardware control scheme to discriminate only against the actuator whose $\hat{x}_{np} \neq \hat{x}_{np}$. Since the additional flow due to $\hat{x}_{np}$ is a serial input to the system, such a filter would also affect the actuator for which $\hat{x}_{np} = \hat{x}_{np}$.

A parallel filter can be implemented in an indirect manner. It has been shown in section 3.3 that the series-bypass system is unstable for proportional gains.
greater than .0011 amps/in/sec. These gains control the commanded bypass valve area change for a velocity error. As an alternative to increasing these gains the bypass valve area will be adjusted to bypass the flow rate which is equivalent to the difference between its actuators $\dot{X}_D$ and $\dot{X}_M$. This correction will only occur for an actuator when $\dot{X}_D \neq \dot{X}_M$. Because Figures 3-83 and 3-84 show that the response of the actuators to $\dot{X}_M$ is sharply attenuated above 90 rad/sec, a steady state correction will be employed.

The proposed algorithm can be written as

$$A_U = \frac{(\dot{X}_M - \dot{X}_D)A_P}{C\sqrt{2\Delta P/\rho}} \quad (3.27)$$

where $A_U$ is the adjustment in the commanded bypass valve area because of the difference between the maximum desired velocity and the desired velocity and $\Delta P$ is the pressure drop across the bypass valve.

In the control structure this adjustment will be added to the commanded bypass valve area input to the bypass valve only. It will not be added in the commanded bypass valve area input to the pump control algorithm. This enables each individual bypass valve to correct for any velocity difference independently of the other actuators.
Figures 3-85 and 3-86 show the response of the system to a unit step velocity input in $\dot{X}_{1D}$ when equation (3.27) is included in the control structure. In these figures, $K_{E1} = K_{E2} = .0005$ amps/in/sec and $A_U$ was input to the lead filters $G_{1c}(s)$ and $G_{7c}(s)$. The parameters of these filters are listed in Table 3-3. The system response in these figures is superior to the response to an identical input with and without integral compensation (Figures 3-48, 3-49, 3-70 and 3-71). The sharp velocity discontinuity circled in Figure 3-85 is the result of the coupling between $A_4$ and $\dot{X}_1$. This discontinuity illustrates the failure to decouple the actuators from adjacent bypass valves.

Figure 3-86 displays sharp velocity spikes at the initial instants when $\dot{X}_{2D} \neq \dot{X}_M$ and when the desired velocities return to equality. Investigation of the time plots of other system variables revealed that the spikes were caused by the difference in the time responses between the bypass valves and the variable displacement pump. As an example, for an increase in $\dot{X}_M$ and at the initial moments when $\dot{X}_D \neq \dot{X}_M$ the command to the bypass valve will suddenly increase the bypass valve area faster than the pump can provide the additional flow requested by $\dot{X}_M$. Consequently the velocity of the actuator will decrease due to the larger bypass valve
Figure 3-85: $\dot{x}_1$, Decoupled From $\dot{x}_M$.

Figure 3-86: $\dot{x}_2$, Decoupled From $\dot{x}_M$. 
area before the increase in pump flow rate is available to increase the actuator's velocity. This is a common problem when two actuators with different time constants are used to simultaneously control an output. As a remedy, a low pass filter of the form of equation (3.28) was inserted after equation (3.27) to delay the command to the bypass valve. The filters were empirically tuned using the simulation. The time constants, $\tau_{LP}$, for the first and second actuators were chosen to be .0075 and .01 seconds respectively.

$$G_{LP}(s) = \frac{1}{\tau_{LP}^s + 1}$$

(3.28)

Figures 3-87 and 3-88 show the system response to conditions identical to those of Figures 3-85 and 3-86 when these filters were installed. These figures show that the sharp peaks have been attenuated.

3.7.3 System Response to $F_{1p}$ and $F_{2p}$

Tables 3-4 and 3-5 list data for the closed loop transfer functions, $\dot{X}_{1p}/F_{1p}(i\omega)$, $\dot{X}_{1p}/F_{2p}(i\omega)$, $\dot{X}_{2p}/F_{1p}(i\omega)$ and $\dot{X}_{2p}/F_{2p}(i\omega)$. These values are the elements of the third term in equation (3.5). The values were obtained without lead filter or integral compensation. Comparing these magnitudes to those of
Figure 3-87: $\dot{X}_1$, Decoupled From $\dot{X}_M$, Filtered

Figure 3-88: $\dot{X}_2$, Decoupled From $\dot{X}_M$, Filtered
the other terms in equation (3.5) (see Figures 3-79 through 3-84) and recalling the time response studies of the previous sections it can be concluded that for these conditions constant loads have little effect on the velocity response of the series-bypass system. This stiffness is a result of the system construction, closed and feedforward control algorithms and the relatively high tolerance of the velocity and position control requirements. Therefore, it is not considered necessary to introduce additional compensation to correct or decouple the system from the maximum anticipated static loads. The integral compensation designed in section 3.6.2 can be retained to correct for nonstatic and unequal loading.

3.8 Summary

In this chapter the bypass valves and actuators of the series-bypass circuit were considered to be individual closed velocity loops and the coupling in the circuit was not explicitly incorporated in the controller design process. Several multivariable frequency response methods were selected to identify the stability boundaries of the series-bypass circuit. Using these methods, it was shown that the linearized frequency response curves could be used to identify the
Table 3-4: $\dot{X}_l / F_l(iw), (1,1)$ and $\dot{X}_l / F_2(iw), (1,2)$, $K_{E1}, K_{E2} = 0.0005$

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stability region of the simulated prototype system.

Classical closed loop compensators for SI-SO systems were used to improve the performance of the series-bypass circuit. Lead filters for the commanded bypass valve areas and commanded variable displacement pump flow rate were designed by using the linearized SI-SO open loop frequency response curves as a guide in selecting filter time constants. The time domain simulation showed that using these lead filters increased the series-bypass system's speed of response.
Without lead compensation, simulation results showed the velocity response to a change in the bypass valve area to be slightly faster than the response to a change in the pump flow rate.

The simulation studies agreed with the frequency response analysis which predicted that the series-bypass circuit was sufficiently stiff for equal and constant actuator loads. However, dynamic and unequal actuator loads created large velocity errors. These errors were caused by the inability of the bypass valve (using only
proportional control) to compensate for the changing flow rates across the valve (due to load variations) and the fluid compressibility effects in the serially connected volumes of the series-bypass circuit. Large velocity errors were also observed when only the proportional control algorithms were used to correct for unequal desired velocity inputs. An integral controller was included in the algorithms which calculate the commanded bypass valve areas. Simulation results showed that the integral compensation reduced the magnitude of the velocity errors caused by unequal desired velocity inputs and actuator loads.

A nonlinear algorithm obtained from the mathematical model of the bypass flow rate was proposed to correct for static differences in the desired velocities. The algorithm calculated the bypass flow rate corresponding to the difference in desired velocities and adjusted the appropriate bypass valve proportionally. It was shown in the simulation that this term successfully eliminated the static velocity errors which were caused by unequal desired velocities. The final controller designs based on these techniques and for the conditions listed in Tables 2-3 and 2-4 are summarized in Tables 3-6, 3-7 and 3-8. Although the controller parameters in Tables 3-6 through 3-8 were
calculated for one set of equilibrium conditions, the procedures in this chapter illustrate the methodology which would be used to specify controllers at other operating conditions. Using a digital computer it would not be difficult to switch the controllers depending on the operating conditions of the series-bypass circuit. Note that, for Tables 3-6 and 3-7, the integral controller and the algorithm to compensate for dissimilar desired velocities were not used simultaneously.

This completes the section on compensation design for the series-bypass circuit with \( K \) equal to a unity matrix. In one sense, the integral controller and model matching algorithm attempted to decouple the two actuators of the series-bypass circuit by decreasing the velocity errors created by having two closed velocity loop systems in the circuit. However, these methods are viewed to be deficient for the following reasons.

- It is important to note that the integral method of compensation could only compensate for velocity errors after they occurred.

- The model matching algorithm to correct for unequal desired velocity inputs attempted to
Table 3-6: Bypass Valve Control Algorithms, Valve 2

\[ K_{E1} = 0.005 \text{ amps/in/sec}, \quad K_{Al} = 4.0 \times 10^{-6} \text{ amps/in/sec}^2 \] and
\[ K_{\text{INTEGRAL}} = 0.04 \text{ amps/in} \]
(Other constants defined in Table 2-3)

**Commanded Bypass Valve Area**

\[
A_{2C} = A_{10\%} - K_{E1} A_S (\dot{x}_{1D} - \dot{x}_1)
+ K_{Al} A_S \ddot{x}_1
- K_{\text{INTEGRAL}} A_S \int (\dot{x}_{1D} - \dot{x}_1) \, dt \tag{3.29}
\]

**Lead Filter for Valve 2 Commanded Bypass Valve Area.**

Equation (3.29)

\[
\frac{(.01s+1)}{(.001s+1)}
\]

**Algorithm to Compensate for Unequal Desired Velocities**

\[
A_{2U} = \frac{(\dot{x}_M - \dot{x}_{1D}) A_P}{C\sqrt{2(P_1 - P_L)/\rho}} \tag{3.30}
\]

**Low Pass Filter for Equation (3.30)**

\[
\frac{1}{.0075s+1}
\]

Command to bypass valve 2 is sum of lead filtered equation (3.29) and low pass filtered equation (3.30).
Table 3-7: Bypass Valve Control Algorithms, Valve 4

\[ K_{E2} = 0.0005 \text{ amps/in/sec}, \quad K_{A2} = 11.0 \times 10^{-6} \text{ amps/in/sec}^2 \] and \[ K_{\text{INTEGRAL}} = 0.04 \text{ amps/in} \]

(other constants defined in Table 2-3)

**Commanded Bypass Valve Area**

\[
A_{4C} = A_{10\%} - K_{E2} A_S \left( \dot{X}_{2D} - \dot{X}_2 \right) \\
+ K_{A2} A_S X_2 \\
- K_{\text{INTEGRAL}} A_S \int \left( \dot{X}_{2D} - \dot{X}_2 \right) \, dt \tag{3.31}
\]

**Lead Filter for Valve 4 Commanded Bypass Valve Area, Equation (3.31)**

\[
(G_7^c, G_8^c, \text{ in Figure 3-2}) \\
\frac{(.01s+1)}{(.001s+1)}
\]

**Algorithm to Compensate for Unequal Desired Velocities**

\[
A_{4U} = \frac{\left( \dot{X}_M - \dot{X}_{2D} \right) A_P}{C\sqrt{2(p_L - p_R)/\rho}} \tag{3.32}
\]

**Low Pass Filter for Equation (3.32)**

\[
\frac{1}{.01s+1}
\]

Command to bypass valve 4 is sum of lead filtered equation (3.31) and low pass filtered equation (3.32).
Table 3-8: Pump Control Algorithm

\( K_{E1}, K_{E2} = 0.0005 \) amps/in/sec, \( K_{\text{INTEGRAL}} = 0.04 \) amps/in

(other constants defined in Table 2-3)

\[
A_{2C} = A_{10\%} \cdot K_{E1} \cdot A_s (\dot{X}_{1D} - \dot{X}_1) - K_{\text{INTEGRAL}} A_s \int (\dot{X}_{1D} - \dot{X}_1) \, dt \quad (3.33)
\]

Lead Filter for Equation (3.33)

\( G_{3c} \) in Figure 3-2

\[
\frac{(0.01s+1)}{(0.001s+1)}
\]

\[
A_{4C} = A_{10\%} \cdot K_{E2} \cdot A_s (\dot{X}_{2D} - \dot{X}_2) - K_{\text{INTEGRAL}} A_s \int (\dot{X}_{2D} - \dot{X}_2) \, dt \quad (3.34)
\]

Lead Filter for Equation (3.34)

\( G_{5c} \) in Figure 3-2

\[
\frac{(0.01s+1)}{(0.001s+1)}
\]

Pump Command

\[
Q_c = [\dot{X}_M A_p + \frac{(2A_{10\%} - A_{2C}) \sqrt{2(P_1 - P)}/\rho)}{2} + \frac{(2A_{10\%} - A_{4C}) \sqrt{2(P - P_R)/\rho}}{2}] \quad (3.35)
\]

Lead Filter for Equation (3.35)

\( G_{9c} \) in Figure 3-2

\[
\frac{(0.02s+1)}{(0.002s+1)}
\]

Command to pump is lead filtered equation (3.35).
decouple the system by applying a constant magnitude correction which depended on the desired velocity inputs and not the magnitude of the actual velocity errors which exist at any instant.

In Chapter 4, Rosenbrock's Inverse Nyquist Array (INA) technique will be discussed. This technique will be used in an attempt to eliminate the closed loop velocity errors caused by the coupling and as an alternative method of compensation design. This method will use coupling information provided by the linearized frequency response model to calculate values for all the elements of the precompensation matrix K. This matrix compensation is different from the two methods of decoupling discussed above in that it is designed to predict the effect that the adjacent actuator has on the immediate actuator's velocity and will adjust the commands sent to the bypass valve accordingly. In this manner, Rosenbrock's INA technique attempts to minimize the coupling disturbances before they appear as velocity errors in the system.
Chapter 4

MULTIVARIABLE SYSTEM METHODS

4.1 Inverse Nyquist Array Technique

Figure 4-1 is a block diagram illustrating the open loop transfer functions which can be obtained by solving equation (2.66) for the specified transfer functions. The transfer functions were defined as the elements of the matrices in Figure 3-1. The coupling between the two closed loop velocity feedback systems is evident in this figure.

The block diagram labels are defined as:
\[ q_{11}(s), q_{22}(s) \] Transfer functions between bypass valve opening and immediate actuator velocity.

\[ m_{11}(s), m_{22}(s) \] Transfer functions between maximum desired actuator velocity and actuator velocity.

\[ q_{12}(s), q_{21}(s) \] Transfer functions between bypass valve opening and adjacent actuator velocity.

\[ d_{11}(s), d_{22}(s) \] Transfer functions between actuator load and immediate actuator velocity.

\[ d_{12}(s), d_{21}(s) \] Transfer functions between actuator load and adjacent actuator velocity.

From Figure 4-1 it can be seen that the two
Figure 4-1: Block Diagram of Series-Bypass Circuit - $K, K_I = I$
hydraulic circuits are coupled in three explicit ways.

- A load applied to either actuator will "propagate through" and affect the performance of the other actuator ($d_{12}(s)$ and $d_{21}(s)$).

- If a bypass valve area changes in order to correct the immediate actuator velocity, this change will disturb the actuator velocity in the other circuit ($g_{12}(s)$ and $g_{21}(s)$).

- A pump flow rate variation will directly change both piston velocities ($m_{11}(s)$ and $m_{22}(s)$).

This coupling between the piston motions is intentional and is a direct consequence of the series connection. However, there may be a need to modify some aspects of the coupling by appropriate controller design. The nature of the desired decoupling will be investigated in this chapter. Using multivariable control techniques [47, 51], methods will be proposed which will attempt to achieve the desired decoupling.

As stated in section 1.2.2, when the ASV is in the cruise mode it is desirable for all legs to move at the
same velocity. The series-bypass configuration was designed to achieve this hydro-mechanically. To control actuator velocities, closed loop feedback systems were constructed between $\dot{X}_1$ and $A_2$ and between $\dot{X}_2$ and $A_4$. Specifically:

$$A_{2C} = A_{2X} - A_S K_{E1} (\dot{X}_{1D} - \dot{X}_1) + A_S K_{A1} \ddot{X}_1 \quad (2.44)$$

$$A_{4C} = A_{4X} - A_S K_{E2} (\dot{X}_{2D} - \dot{X}_2) + A_S K_{A2} \ddot{X}_2 \quad (2.45)$$

These control equations were proposed because it was felt that controlling each actuator with its own immediate bypass valve would result in the greatest amount of freedom when individual leg motion was required, as in the precision footing mode. For example, it is the intent of equation (2.44) to control $\dot{X}_1$ by using only the velocity error and acceleration of actuator 1 and utilizing only the bypass valve area, $A_2$. Notice from the elements of matrix $G$ in Figure 4-1 however, that $\dot{X}_1$ consists of contributions from both $A_2$ and $A_4$. Therefore any changes in $A_4$ due to variations in either $\dot{X}_{2D}$, $\dot{X}_2$ or $\ddot{X}_2$ will affect $\dot{X}_1$ as a disturbing input in the velocity control of actuator 1. The same situation applies in the control of $\dot{X}_2$ due to variations in $A_2$. It would be desirable to decrease this disturbance so that the velocity of actuator 1 would be determined primarily by $A_2$, and similarly the velocity
of actuator 2 would be determined primarily by $A_4'$. Representative frequency response curves for the elements of matrix $G$ are shown in Figure 4-2. These curves were obtained by solving equation (2.66) and using the system constants listed in Table 2-3 and the equilibrium conditions in Table 2-4. Notice that the magnitudes and phase angles of $\dot{X}_{1p}/A_{2Cp}'(i\omega)$, $\dot{X}_{1p}/A_{4Cp}'(i\omega)$, $\dot{X}_{2p}/A_{2Cp}'(i\omega)$ and $\dot{X}_{2p}/A_{4Cp}'(i\omega)$ are similar, indicating that $A_{4Cp}'$ will affect $\dot{X}_{1p}$ almost to the same extent as $A_{2Cp}'$. A similar statement can be made regarding $\dot{X}_{2p}$, $A_{2Cp}'$ and $A_{4Cp}'$.

To illustrate the closed velocity loop coupling in the time domain, Figures 4-3 and 4-4 show the response of the actuators to a step increase in $\dot{X}_{1D}$ and Figures 4-5 and 4-6 show the response to a step increase in $\dot{X}_{2D}$. The system was at the equilibrium conditions in Table 2-4, $K_{E1}$, $K_{E2} = .0005$ amps/in/sec and integral and lead compensation are not included. To represent the coupling in the transfer functions between $\dot{X}_{1p}$, $\dot{X}_{2p}$ and $A_{2Cp}'$, $A_{4Cp}'$ the commanded bypass valve areas were the only inputs to the bypass valves and variable displacement pump (i.e., $\dot{X}_M$ equals a constant). The coupling in these figures is evident in the steady state velocity errors of both actuators and oscillations in the velocity response of the actuator not subjected to
Figure 4-2: Nyquist Plots of Matrix G
Figure 4-3: Velocity of Actuator 1, $\dot{X}_M=\text{constant}$

Figure 4-4: Velocity of Actuator 2, $\dot{X}_M=\text{constant}$
Figure 4-5: Velocity of Actuator 1, $\dot{X}_M=$constant

Figure 4-6: Velocity of Actuator 2, $\dot{X}_M=$constant
the step increase in desired velocity. Figures 4-3 through 4-6 are different from Figures 3-48 through 3-51 because the feedforward loop has not been included in the system. The feedforward loop was not included because it would alter the transient and steady state response of the system and make it difficult to compare the affects of different decoupling methods. Note that the steady state velocity errors of Figures 4-3 and 4-6 and Figures 4-4 and 4-5 are equal for identical proportional gains. The linearized open loop frequency response analysis in sections 2.6 and 2.7 predicted equal steady state open loop velocities for equal actuator loads, proportional gains and bypass valve areas. These figures show that the steady state velocity errors from the closed loop contribution to the total actuator velocities are independent of the location of the actuator in the series-bypass circuit.

To maximize the independent closed loop operation of the actuators it is necessary to reduce the magnitude of the coupling between the two closed loop systems by reducing the magnitude of the off diagonal terms of the matrix $G$. Notice from Figure 4-1 and equations (2.72) and (2.73) that the actuator velocities are derived not only from $\dot{x}_{1D}$ and $\dot{x}_{2D}$ but also contributions from $\dot{x}_M$, the external disturbance loads $F_1$ and $F_2$. 
Their effects on the actuator velocities were discussed in section 3.7.3. $X_M$ is a feedforward input and an algorithm to decouple the actuators from this input was presented in section 3.7.2. This chapter will address the coupling in the series-bypass circuit due to the commanded bypass valve areas.

The Inverse Nyquist Array technique (INA) proposed by Rosenbrock [47, 50, 51] is a frequency response method which has been used in a variety of applications to analyze the stability of and design controllers for multivariable systems. This technique appeared attractive in the series-bypass application for the following reasons.

1. It is a frequency response technique in which numerical or empirical frequency response curves may be used. This eliminates the need for symbolic transfer functions.

2. A prerequisite for its application is the establishment of a diagonally dominant matrix. Roughly defined, a diagonally dominant matrix has diagonal elements which are greater in magnitude than the sum of the magnitude of the off diagonal terms. As
shown in section 4.3, many authors have contributed techniques to obtain diagonally dominant matrices. The majority use the elements of the precompensation matrix (see Figure 3-1) $K$ to pseudo-diagonalize $G$. Pseudo-diagonalization refers to a procedure which reduces the magnitude of the off diagonal terms or increases the magnitude of the diagonal terms to make a matrix as close to diagonal as possible. If the off diagonal terms of $GK$ have less magnitude than the diagonal terms, the interaction between the control loops is reduced. Some multivariable techniques attempt to reduce the coupling to zero or compensate the system such that the coupling is canceled [47] (matrix is diagonal). In practice, pseudo-diagonalization of a matrix is much easier to obtain than completely cancelling the off diagonal elements.

3. The pseudo-diagonalization method enables the use of classical single-input, single-output frequency response compensation techniques in multivariable system applications. These
classical techniques have proven to be robust, and have been successfully used in numerous servovalve controlled piston applications where decoupling was not attempted.

4. This procedure was selected over state variable methods because state variable formulation of expression (2.66) resulted in state variables which were not measurable, and required state observers. In addition, the INA technique leads to robust controllers.

To agree with Rosenbrock's notation [51] the closed loop system of Figure 4-7 is transformed to Figure 4-8. The matrix \( K_1B \) will be referred to as \( F \). From the definitions given in section 3.1, \( F \) is a diagonal matrix with diagonal elements since \( B \) and \( K_1 \) are both diagonal matrices. \( F \) is defined as:

\[
\begin{bmatrix}
f_1 & 0 \\
0 & f_2
\end{bmatrix}
\quad (4.1)
\]

\( F \) contains all the proportional gains and dynamic compensation for \( GK \). The matrix \( K_1 \) outside the loop does
not affect system stability. When the transformed system in Figure 4-8 is actually implemented it is probably easier to move the matrix $K_1$ back to the forward loop to obtain Figure 4-7.

In the preceding discussions the following terminology will be used. For a general matrix $X$:

- $x_{ij}^{-1}$ = inverse of the element $x_{ij}$ of the matrix $X$
- $\hat{X}$ = $X^{-1}$, inverse of the matrix $X$
- $\hat{x}_{ij}$ = element $ij$ of the matrix $\hat{X}$

Rosenbrock has also shown that stability can be determined by examining only the diagonal terms of the matrix $\hat{KG}$ if $\hat{KG}$ is made diagonally dominant. Rosenbrock defines diagonal dominance for a $k \times k$ matrix as

$$|\hat{q}_{ii}(i\omega)| - \sum_{j=1, j\neq i}^{k} |\hat{q}_{ij}(i\omega)| > 0$$

for $i=1, 2, \ldots, k$

and for $0 < \omega < \omega_{max}$ (row) \hspace{1cm} (4.2)

or
Figure 4-7: Block Diagram of Series-Bypass Circuit

Figure 4-8: Block Diagram, Rosenbrock's Representation
for $i=1,2,...,k$

and for $0<\omega<\omega_{\text{max}}$ (column)

where $K\hat{G}=\hat{Q}$ and $\hat{q}_{ij}$ are the elements of the matrix $\hat{Q}$. $\omega_{\text{max}}$ is the maximum frequency of interest. For the remainder of this text dominance will designate diagonal dominance.

Equation (4.2) refers to the situation when the sum of the magnitudes of the off diagonal terms in row $i$ is smaller than the diagonal term in row $i$ (row dominance) and equation (4.3) refers to the situation when the sum of the magnitudes of the terms in column $i$ is smaller in magnitude than the diagonal term in column $i$ (column dominance).

Rosenbrock advocates a graphical means to check for dominance. The procedure involves plotting $\hat{q}_{ii}(i\omega)$ over the range of interest and then plotting circles of radius,

$$\sum_{j=1, j\neq i}^{k} |\hat{q}_{ij}(i\omega)|$$

for row dominance

(4.4)

with their centers at the corresponding points of
\( q_{ii}(i\omega) \). See Figure 4-9 for an example. If the envelope created by the perimeters of these circles has the origin as an interior point or on its exterior, then the system is not dominant. If the envelope excludes the origin then the system is dominant. Rosenbrock refers to these circles as Gershgorin circles and the exterior boundaries of the envelope created by the circles as Gershgorin bands.

\[ \text{Figure 4-9: Gershgorin Circles and Bands} \]

If the system is dominant then the Gershgorin bands
can be used to establish the stability of the system.

Rosenbrock's stability theorem is stated as:

- Let each of the Gershgorin bands based on the diagonal elements \( q_{ii} \) of \( Q \) exclude the origin and the point \((-f_i, 0)\). Let these bands encircle the origin \( \hat{N}_{qi} \) times and encircle the point \((-f_i, 0)\), \( \hat{N}_{hi} \) times. Then the closed loop system is asymptotically stable if and only if

\[
\sum \hat{N}_{qi} - \sum \hat{N}_{hi} = p_o
\]

\( p_o \) is the number of open loop poles in the right half plane. For this application, the number of poles in the right half plane is equal to the number of roots with positive real parts of the denominator of the open loop transfer functions \( \dot{X}_{1p}/A_{2Cp}(s) \), \( \dot{X}_{2p}/A_{2Cp}(s) \), \( \dot{X}_{1p}/A_{4Cp}(s) \) or \( \dot{X}_{2p}/A_{4Cp}(s) \). These could be obtained from equation \((2.66)\).

From a graphical viewpoint this theorem can be interpreted as saying that asymptotic stability is assured if the gain \( f_i \) is between the origin the Gershgorin band nearest the origin. If the gain is increased to lie within the Gershgorin envelope, the theorem cannot predict the system stability. If \( f_i \) is located outside the outer Gershgorin band, the system is unstable.
In addition to Gershgorin bands, Rosenbrock also defines Ostrowski bands for dominant systems. H is defined as the closed loop transfer function, (between $\hat{x}_p$ and $K\hat{x}_D$, see equation (3.5))

$$H = (I + QF)^{-1}Q$$

(4.6)

and $\hat{H}$ is the inverse closed loop transfer function,

$$\hat{H} = F + Q$$

(4.7)

If $Q$ and $\hat{H}$ are dominant, for each frequency $\omega$, the diagonal elements $h^{-1}_{ii}$ satisfy

(row dominant)

$$|h^{-1}_{ii}(\omega) - (f_i + q_{ii}(\omega))| \leq \phi_i(\omega) \delta_i(\omega)$$

(4.8)

or

(column dominant)

$$|h^{-1}_{ii}(\omega) - (f_i + q_{ii}(\omega))| \leq \phi_i(\omega) \delta_i(\omega)$$

(4.9)

accordingly as $\hat{H}$ is row or column dominant at $\omega$.

$\phi_i$ is defined as

$$\phi_i(\omega_o) = \max_{j, j \neq i} \frac{\delta_j(\omega_o)}{|(f_j + q_{jj}(\omega_o))|}$$

(4.10)

or
and (for a diagonal $F$)

$$
\phi'(i\omega) = \max_{j \neq i} \frac{\delta'(i\omega)}{|(f_i + q_{ij}(i\omega))|} \tag{4.11}
$$

and (for a diagonal $F$)

$$
S_i(i\omega) = \sum_{j \neq i} |q_{ij}(i\omega)| \tag{4.12}
$$

$$
S_i'(i\omega) = \sum_{j \neq i} |\hat{q}_{ij}(i\omega)| \tag{4.13}
$$

$S_i$ define the Gershgorin bands for $Q$ when $F$ is diagonal. $\phi_i$ can be simply interpreted as the maximum ratio of the summation of the off diagonal terms over the diagonal term, determined by examining each row (for row dominance) of $\hat{H}$ except the $i$th. Negative feedback is specified since it is possible for some closed loop multivariable systems to be stable for positive feedback.

This theorem has the following graphical interpretation. If we define $h^{-1}_i(i\omega)$ as $h^{-1}_{ii}(i\omega)-f_i$ then $h^{-1}_i(i\omega)$ lies within a circle centered on $\hat{q}_{ii}(i\omega)$ and having radius $\phi_i(i\omega)\delta_i(i\omega) < S_i(i\omega)$. As $\omega$ is varied from 0 to higher frequencies these circles sweep out bands which lie inside the Gershgorin bands, since $\phi_i(i\omega) < 1$ for a dominant $\hat{H}$. These narrower bands are
called the Ostrowski bands and are illustrated in Figure 4-10.

\[ h_i \text{ is the closed loop transfer function between input } i \text{ and output } i \text{ when the } i \text{th loop is open (} f_i = 0) \text{ and all other loops are closed with specified gains. It is similar to the open loop transfer function between input } i \text{ and output } i \text{ for single-input, single-output (SI-SO) systems, but in this multivariable system other closed loop feedback systems contribute to the output} \]
i. $h_i^{-1}$ is the inverse of this closed loop transfer function and is analogous to the inverse open loop transfer function of a SI-SO system. Recall that for SI-SO systems, the Nyquist plot of the inverse open loop transfer function can be used to design compensators for the corresponding closed loop system. If we wish to design a closed loop compensator for the $i$th loop of a multivariable system, we can design it using $h_i^{-1}(i\omega)$ and the classical frequency response techniques developed for SI-SO systems. As long as $\hat{H}$ and $\hat{Q}$ are dominant, the Ostrowski envelope eliminates the need to calculate $h_i^{-1}(i\omega)$ exactly. Notice that as the gains $f_1, f_2, \ldots, f_i-1, f_{i+1}, \ldots, f_k$ are increased (other closed loops are "tightened"), $\phi_i(i\omega)$ will decrease and the Ostrowski envelope will narrow, decreasing the ambiguity in the location of $h_i^{-1}(i\omega)$.

Recall that $\phi_i$ (equation (4.10)) depends on the gains in the other loops so that the $i$th Ostrowski band depends on these other gains. As the loop gains, $f_1, f_2, \ldots, f_i-1, f_{i+1}, \ldots, f_k$ vary, $h_i^{-1}(i\omega)$ will change. As long as dominance is maintained however, $h_i^{-1}(i\omega)$ lies within the appropriate Ostrowski band, evaluated for the gains $f_1, f_2, \ldots, f_i-1, f_{i+1}, \ldots, f_k$. The Gershgorin band is independent of these other loop gains and gives an outside bound for the Ostrowski band. The stable values
of the gains $f_i$ can be conservatively determined from the Gershgorin bands.

The second function of the Ostrowski bands is to determine the stability margins of the loops. Notice that as only one of the $f_i$ varies, a single loop situation occurs. Consequently appropriate phase and gain margins can be determined if we know $h_i^{-1}(i\omega)$ and this is within the $i$th Ostrowski band. Descriptively speaking, the Nyquist stability criterion for the inverse open loop plot for SI-SO systems states that the Nyquist plot must encircle the point $(-f_i,0)$ and the origin an equal number of times. As mentioned above, when the Ostrowski envelope is narrow enough it is reasonable in practice to treat it as though it was a single-loop inverse Nyquist plot and use it to calculate phase and gain margins. The distance from the inner Ostrowski band intersection with the real axis to the gain $f_i$ is an indication of the gain margin. If this distance is large, the gain margin is large. As the distance decreases the gain margin decreases. If $f_i$ is outside the outer Ostrowski band, the closed loop system is unstable (see Figure 4-10).

To summarize, the usual design procedure is to initially diagonalize $\hat{K}\hat{G}$. Some methods for attempting this will be described in section 4.3. Then $\hat{q}_{ii}(i\omega)$ and
its appropriate Gershgorin bands are drawn and preliminary loop gains $f_i$ are selected. These gains are used to compute the closed loop transfer function, $H=(I+QF)^{-1}Q$ and using $\phi_i$ and $\delta_i$ the appropriate Ostrowski bands are drawn. If the Ostrowski bands are narrow enough, $h_i^{-1}(i\omega)$ can be located with a small amount of ambiguity. The gains can be specified based on gain or phase margin considerations and single loop compensation can proceed. Note that as $f_i$ changes, the Ostrowski bands for the other loops also change. Therefore as $f_i$ changes the procedure must be repeated in the other loops in order to determine the change in the Ostrowski bands due to these changes in the magnitude of $f_i$.

The question naturally comes to mind, why attempt to formulate the theory using $\hat{Q}$ instead of $Q$. Rosenbrock [51] gives three reasons to justify using $\hat{Q}$ instead of $Q$.

1. The relation $\hat{H}=\hat{F}+\hat{Q}$ gives an easy transition from open loop to closed loop properties.

2. There appears from practical experience to be a tendency for $\hat{Q}$ to be more dominant than $Q$. 
3. For some given \( \omega \), suppose that the critical points \((-f_i, 0)\) in all loops except the jth can be moved so that the distance from \((-f_i, 0)\) to \(q_{ii}(i\omega)\) becomes infinitely large. Then the width of the Ostrowski band for the jth loop shrinks to zero at this frequency.

From the definition of Ostrowski bands (equations (4.8) to (4.13)) it can be seen that as the width of the bands goes to zero \( (\phi_i \to 0) \), the diagonal terms in the other loops go to infinity, or, in other words, the diagonal terms become increasingly dominant. This means that, if in all other loops except the jth, the gain is so high that the corresponding outputs are held to zero when the jth input is sinusoidal with frequency \( \omega_o \), then the transfer function \( h_j \) between input j and output j at \( \omega_o \) is \( \hat{q}_{jj}^{-1}(i\omega_o) \).

Algebraically, from equation (4.8), it can be determined that as \( \phi_j \to 0 \), \( h_{jj}^{-1} = \hat{q}_{jj} \) and \( (h_{jj}^{-1} r_j)^{-1} = \hat{q}_{jj}^{-1} \) becomes the open loop transfer function between input j and output j when all other loops except the jth are closed. The elements \( q_{jj} \) of Q have the opposite property; they are the values
obtained by \( h_j \) as all loop gains go to zero. Naturally, we are much more interested in the closed loop properties and the case when all loop gains go to zero is not of practical interest.

In the remainder of this section, several points concerning the INA technique are clarified. These items are not explicitly stated in the literature and are considered important to the application of this technique.

The stability of a nondominant system can be assessed using the methods presented in section 3.3. For a dominant system, assessing the stability using the INA method has the advantage of determining the entire gain space (\( k \) dimensional region of stable gains) by examining Gershgorin or Ostrowski plots instead of using the primarily trial and error procedure of using simulation results or Nyquist plots of the system determinant, equation (3.7) (section 3.3) to select optimum filter designs. In addition, for dominant matrices, gain and phase margins can be defined for each loop as opposed to the method of 2-Hodographs in which a gain and phase margin is defined for the total system.

Single loop compensation is possible regardless of
whether \( \hat{Q} \) and/or \( \hat{H} \) are dominant. However, if these matrices are dominant, it is much easier to use the Ostrowski plots as an aid in determining the compensation required and the effects of a compensator on the closed loop system response. For nondominant systems, the location of \( h_{i}^{-1}(i\omega) \) is not guaranteed to lie within the Ostrowski and Gershgorin bands and \( h_{ii}^{-1}(i\omega) - f_{i} \) must be computed each time a loop gain is changed. This involves much more computation than using the Ostrowski plots.

It is important to understand the qualitative difference between the definitions of a dominant or decoupled system. Both these terms are relative to the designer and the requirements of the system. A dominance ratio for a \( k \times k \) matrix \( X \) can be defined as

\[
\lambda_i = \frac{\sum_{j=1}^{k} |x_{ij}|}{|x_{ii}|} \quad \text{(row dominance)}
\]

or

\[
\lambda_i' = \frac{\sum_{j=1}^{k} |x_{ji}|}{|x_{ii}|} \quad \text{(column dominance)}
\]

For ratios less than 1, the row or column of the \( X \)
matrix can be defined as dominant, according to equations (4.2) and (4.3). Note that if this ratio is .99, for example, for all rows or columns, this system is still defined as dominant. However, the off diagonal terms have almost as much magnitude as the diagonal term. In this sense the system could not realistically be classified as decoupled but it would be possible to use the INA technique in this application. Many multivariable systems do not require decoupled closed loop systems to achieve their objectives. Multivariable compensation may take many forms, not necessarily decoupling the system into a one-to-one input-output mapping. The INA technique could be used to design these compensators if a system had a dominance ratio of .99, but the system would not be decoupled.

4.2 Pseudo-Diagonalization of the Series-Bypass Circuit

One of the objectives of the series-bypass circuit is to provide independent leg motion when necessary. Rosenbrock's INA technique appeared promising both as a method to achieve this actuator decoupling and as a tool in designing compensators and establishing the stable gain space for the series-bypass circuit. The method depends on obtaining a diagonally dominant matrix to establish the gain space and specify compensation. As
shown below, this dominant $\hat{Q}$ may not result in a decoupled system. Referring to the expression of the closed loop series-bypass circuit, equation (4.6), to accomplish independent leg control it is necessary that the frequency dependent ratios of $|h_{12}(i\omega)/h_{11}(i\omega)|$ and $|h_{21}(i\omega)/h_{22}(i\omega)|$ approach 0.

The elements of $H(i\omega)Kl$ can be expressed in terms of the elements of $Q$ and $F$ as (For the purposes of brevity the dependence of $G$, $K$ and $Q$ on frequency will be neglected.)

$$\frac{1}{\Delta H} \begin{bmatrix} (q_{12} + f_2(q_{11}q_{22} - q_{12}q_{21}))k_{11}^1 & q_{12}k_{22}^1 \\ q_{21}k_{11}^1 & (q_{22} + f_1(q_{11}q_{22} - q_{12}q_{21}))k_{22}^1 \end{bmatrix}$$

(4.16)

where

$$\Delta H = 1 + q_{11}f_1 + q_{22}f_2 + f_1f_2(q_{11}q_{22} - q_{12}q_{21})$$

(4.17)

The closed loop system becomes increasingly decoupled as the ratios of expressions (4.18) and (4.19) approach 0.

$$\frac{|q_{12}k_{22}^1|}{|(q_{11} + f_2(q_{11}q_{22} - q_{12}q_{21}))k_{11}^1|}$$

(4.18)

$$\frac{|q_{21}k_{11}^1|}{|(q_{22} + f_1(q_{11}q_{22} - q_{12}q_{21}))k_{22}^1|}$$

(4.19)
Mayne [35] has shown (and the statement is easily verified by equations (4.18) and (4.19)) that the degree of closed loop decoupling increases with the proportional gain (assuming relatively equal "high gains") for matrices with dominant determinants, specifically:

\[ |q_{11}(i\omega)q_{22}(i\omega)| > |q_{12}(i\omega)q_{21}(i\omega)| \]  

(4.20)

Efforts in this chapter will concentrate on pseudo-diagonalizing \( \hat{Q} \). \( \hat{Q} \) does not have to be dominant for equation (4.20) to be true, but a dominant \( \hat{Q} \) guarantees that equation (4.20) will be true. High gains are not an easy solution however, since frequency response methods and time response simulations have shown the system to be unstable for the large values of gains which decouple the system.

The INA method is based on a dominant \( \hat{Q} \) matrix which can be written in terms of the elements of \( Q \) as

\[
\frac{1}{\Delta Q} \begin{bmatrix} q_{22} & -q_{12} \\ -q_{21} & q_{11} \end{bmatrix}
\]  

(4.21)

where

\[
\Delta Q = q_{11}q_{22} - q_{12}q_{21}
\]  

(4.22)

As stated in section 4.1, the INA method can be applied
to either row or column dominant matrices. The elements of a row dominant $\hat{Q}$ matrix would satisfy:

$$
|q_{22}| > |q_{12}|
$$

$$
|q_{11}| > |q_{21}|
$$

\hspace{1cm} (4.23)

A column dominant $\hat{Q}$ matrix would satisfy:

$$
|q_{22}| > |q_{21}|
$$

$$
|q_{11}| > |q_{12}|
$$

Comparing expressions (4.16) and (4.21) it can be seen that a dominant $\hat{Q}$ does not guarantee a dominant $H(i\omega)K_1$ or, in other words, a decoupled closed loop system. However, there are many examples [50, 51, 40, 55, 20] in which pseudo-diagonalizing $\hat{Q}$ greatly decreases the coupling of the closed loop system.

Comparing equations (4.18), (4.19), (4.23) and (4.24) it would appear that a column dominant $\hat{Q}$ would be more likely to result in a decoupled closed loop system. However, as shown below, column dominance is more difficult to attain.

Recalling that $GK=\hat{Q}$ and $KG=\hat{Q}$, $\hat{Q}$ can be written as

$$
\frac{1}{\Delta_0 \Delta_K} \begin{bmatrix}
-k_{22}g_{22} + k_{12}g_{21} & -(k_{22}g_{12} + k_{12}g_{11}) \\
-(k_{21}g_{22} + k_{11}g_{21}) & k_{21}g_{12} + k_{11}g_{11}
\end{bmatrix}
$$

\hspace{1cm} (4.25)

where
The advantages of pseudo-diagonalizing the rows instead of the columns of $KG$ can be seen by examining expression (4.25). Notice that the coefficients that are used to obtain dominance in the first row are independent of the coefficients in the second row. Specifically, $k_{22}$ and $k_{12}$ are used to achieve dominance in the first row of $Q$ and $k_{21}$ and $k_{11}$ are used to obtain dominance in the second row. This independence between rows facilitates finding gains which achieve dominance for $\hat{Q}$ since each row can be considered separately. From equation (4.25) it can be seen that the coefficients which establish column dominance for the first and second columns are not independent of each other. Consequently both columns must be considered when attempting to establish column dominance for $\hat{Q}$.

Also, a row dominant $\hat{Q}$ matrix has advantages when designing compensators for the closed loop system. In section 4.1, $F$ was implicitly considered to be constant. It is this author's experience when using Ostrowski plots as an aid in designing single loop compensators, that the single loop compensators are most easily incorporated into the elements of the matrix $K$. $K$ can be
written as KDKC where KC is a diagonal matrix whose elements are the single loop compensators and KD is the matrix whose elements are used to pseudo-diagonalize \( \hat{Q} \). Specifically KDKC can be written:

\[
\begin{bmatrix}
  k_{11}^c & k_{12}^c \\
  k_{12}^c & k_{22}^c
\end{bmatrix}
\]

Q can now be written as

\[
\frac{1}{\Delta_K} \begin{bmatrix}
  k_{22}^c (k_{22} g_{22} + k_{12} g_{21}) & -k_{22}^c (k_{22} g_{12} + k_{12} g_{11}) \\
  -k_{11}^c (k_{21} g_{22} + k_{11} g_{21}) & k_{11}^c (k_{21} g_{12} + k_{11} g_{11})
\end{bmatrix}
\]

(4.28)

where \( \Delta_K \) now equals

\[
k_{11}^c k_{22}^c (k_{11} k_{22} - k_{12} k_{21})
\]

(4.29)

Note that row dominance is maintained using these single loop compensators but column dominance may not be. Therefore the gains which were chosen for stability considerations may not be valid. The Gershgorin envelope would then have to be recomputed to determine if a dominant \( \hat{Q} \) matrix still exists.
4.3 Pseudo-Diagonalization of $\hat{Q}$

Figure 4-11 displays the Nyquist plots of $\hat{G}$. These curves show the perturbation frequency response around the equilibrium conditions in Table 2-4. It can be seen from these plots that the Nyquist diagrams of all four elements are similar, indicating the highly coupled nature of the system. Tables 4-1 and 4-2 list the real and imaginary components and magnitudes of the elements of $\hat{G}$ and also the row dominance ratio. As seen in the table, neither the top nor the bottom row are diagonally row dominant over the frequency range. As discussed previously, these dominance ratios must be decreased if the system is to be decoupled and if the INA method is to be employed.

It is important to realize that both the bypass valves and the variable displacement pump contribute to the frequency response curves of $G$ and $\hat{G}$. Due to the nature of the bypass valve and pump control algorithms (equations (2.40), (2.41) and (2.48)), a change in $A_{2Cp}^\prime$ and $A_{4Cp}^\prime$ will require both the bypass valves and the variable displacement pump to change the actuator velocities. The fact that the commanded bypass valve areas are common to the control algorithms creates difficulty in assigning individual contributions to the
Figure 4-11: Nyquist Plots of $\hat{G}$
Table 4-1: $\hat{g}_{11}(i\omega)$ and $\hat{g}_{12}(i\omega)$, $k_{11}=1.0$, $k_{12}=0.0$

<table>
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<th>IMAG</th>
<th>MAG</th>
<th>REAL</th>
<th>IMAG</th>
<th>MAG</th>
<th>RATIO</th>
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</table>

frequency response curves for the purpose of designing compensators.

The remainder of this chapter is devoted to applying pseudo-diagonalization techniques to the $\hat{Q}$ matrix. For the reasons stated in section 4.2, the initial attempts will concentrate on achieving row dominance. The majority of reported examples in the literature [21, 22, 40, 55, 51] obtain row dominance by utilizing a $\hat{K}$ matrix consisting of constant elements.
Table 4-2: $\hat{g}_{21}(iw)$ and $\hat{g}_{22}(iw)$.

$\hat{k}_{21} = 0.0, \hat{k}_{22} = 1.0$

<table>
<thead>
<tr>
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<th>IMAG 1</th>
<th>REAL 2</th>
<th>IMAG 2</th>
<th>MAG 1</th>
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</table>

Recall that in implementing the control scheme, $\hat{K}$ must be inverted. A constant $\hat{K}$ can be easily inverted to yield a physically realizable $\hat{K}$. But if $\hat{K}$ contains dynamic terms, it may not be possible to invert this matrix to obtain a $K$ with elements that contain poles in the left half plane. Therefore, in the following section, the diagonalization techniques will concentrate on achieving row dominance with a constant $\hat{K}$ matrix.

From Figure 4-11 and Tables 4-1 and 4-2 it can be
seen that the diagonal terms of \( \hat{G} \) intersect the negative real axis at \( \omega = 0 \) rad/sec. This indicates that positive perturbations of \( A'_{2Cp} \) and \( A'_{4Cp} \) result in decreased piston velocities \( \dot{X}_{1p} \) and \( \dot{X}_{2p} \). To ensure that positive velocity errors increased the bypass valve area the sign of the proportional gain in equations (2.40) and (2.41) must be negative. Rosenbrock's theorem states that the Gershgorin bands must exclude and encircle the origin and the point \((-f_i, 0)\) for negative feedback systems. Recall that the \( f_i \)'s are the diagonal terms of \( KIB \). If the point \(-f_i\) is located on the positive real axis then \( f_i \) is a negative number, which is consistent with the sign of the proportional gains in equations (2.40) and (2.41). Consequently, in order to utilize this theorem, the diagonal elements must be dominant from \( \omega = 0 \) rad/sec until their Nyquist plots intersect the positive real axis. From Table 4-1, \( \hat{q}_{11}(i\omega) \) intersects the positive real axis at 150 rad/sec. Open and closed loop frequency response curves show that the system response appears to be significant up to 250 rad/sec. Hereafter, the frequency range of 0 to 250 rad/sec will be referred to as the frequency range of interest. In determining the gain space, it is not clear whether it is necessary for the system to be dominant up to 250 rad/sec, as long as dominance is obtained until the
Nyquist plot intersects the real axis.

The first method proposed by Rosenbrock [50] consists of obtaining dominance by performing elementary column operations on $\hat{G}$. These column operations can then be transformed into a $\hat{K}$ matrix. The frequency response data in Tables 4-1 and 4-2 were examined in order to select coefficients which may pseudo-diagonalize the top row of $\hat{G}$. Pseudo-diagonalization of the top row was selected as the first priority since its dominance ratios were much larger than the bottom row. Generally it was observed that the imaginary terms of $\hat{g}_{22}(i\omega)$ were greater than $\hat{g}_{11}(i\omega)$ over the range of frequencies in which $|\hat{g}_{12}(i\omega)| > |\hat{g}_{11}(i\omega)|$. Also, the imaginary terms of $\hat{q}_{12}(i\omega)$ and $\hat{q}_{22}(i\omega)$ were similar in sign and magnitude while the imaginary terms of $\hat{q}_{11}(i\omega)$ were negligible in comparison to $\hat{q}_{21}(i\omega)$. Therefore, it appeared reasonable to set $k_{11} = 1$ and let $k_{12}$ equal a negative value less than 1 in magnitude so the large imaginary terms of $\hat{q}_{12}(i\omega)$ and $\hat{q}_{22}(i\omega)$ would cancel.

Some results for $k_{12}$ equal to -.5 and -1 are shown in Tables 4-3 and 4-4. From these tables, it can be seen that these coefficients decrease the maximum value of the dominance ratio in direct proportion to their magnitude but the frequency at which $\hat{q}_{11}$ intersects the real axis also increases. Other values of $k_{12}$ were tried
Table 4-3: $q_{11}(i\omega)$ and $q_{12}(i\omega)$, $k_{11}=1.0$, $k_{12}=-0.5$

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but the results were similar.

A method developed by Hawkins [21, 22] minimizes the sum of the squared magnitudes of the off diagonal row elements of a MxM matrix, i.e.

$$\sum_{r=1}^{N} \left( \sum_{k \neq j}^{M} \gamma_r |q_{jk}(i\omega_r)|^2 \right)$$

over a range of selected frequencies, $\omega_r$. The $\gamma_r$ are real and positive, and are weighting values at frequencies $\omega_1$, $\omega_2$, ..., $\omega_N$. Hawkins has shown that the
coefficients obtained by this method are the eigenvectors of $B_j$ corresponding to the smallest eigenvalue of $B_j$. $B_j$ is defined as

$$B_j = \sum_{r=1}^{N} \sum_{k=1}^{M} \frac{C(b_{il})}{\gamma_{ik}} \frac{\alpha_{ik}(r)}{\gamma_{ik} + \lambda_{ik}} \alpha_{lk}(r) \beta_{ik}(r) \gamma_{ik}$$

(4.31)

$\alpha_{ik}(r)$ and $\lambda_{ik}(r)$ are defined by:

$$\hat{q}_{ik}(i\omega) = \alpha_{ik}(r) + i\lambda_{ik}(r)$$

(4.32)

Note that a $B_j$ matrix is computed for each row of

Table 4-4: $\hat{q}_{11}(i\omega)$ and $\hat{q}_{12}(i\omega)$,

$\hat{k}_{11}=1.0, \hat{k}_{12}=-1.0$

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the MxM matrix.

Notice that this procedure can be used to achieve row dominance at a single frequency, a finite range of frequencies, or the total frequency range of interest. For example, if diagonal dominance fails in the low frequencies, this procedure could be used in an attempt to establish dominance over the low frequency range. These coefficients may also establish dominance over the total frequency range of interest.

A SPEAKEASY program was written to compute the Bj matrices and find the smallest eigenvalue and the eigenvector components. This procedure is included in the program MFRSIM in Appendix B.3. The first attempt at pseudo-diagonalization minimized the sum of the squares of the off diagonal terms at one discrete frequency. Tables 4-1 and 4-2 were examined to locate the frequency of the maximum dominance ratios of the top and bottom rows. This occurred at 150 and 100 rad/sec respectively. The coefficients obtained from Hawkins' method and the resulting \( \hat{Q} \) matrix are presented in Tables 4-5 and 4-6. It can be seen from these tables that the dominance ratios at these frequencies have been decreased. But the coefficients which decrease these ratios increase the dominance ratios at other frequencies or increase the frequency at which the
diagonal terms intersect the positive real axis. It is interesting to note that Hawkins' technique yielded coefficients for the top row which were similar to those selected by inspection.

Another attempt at pseudo-diagonalization set the weighting values to unity for those frequencies in the top and bottom rows at which the dominance ratios were greater than 1. Although not shown, the method decreased the dominance ratios across the weighted frequency range but did not yield dominance at other frequencies.
An additional attempt at pseudo-diagonalization using Hawkins' technique set the weighting factors across the entire frequency range equal to 1. Another attempt used weighting values other than 1, placing unequal emphasis at specific frequencies. Dominance was not obtained in either of the two attempts.

Leininger [30] suggested using a non-gradient optimization method to minimize the dominance ratios of the \( \hat{Q} \) matrix. This method would find coefficients (elements of the \( \hat{K} \) matrix) which would minimize either the row or column dominance ratios. For row dominance
the functions to be minimized are

\[
\frac{\hat{k}_{11}\hat{\sigma}_{12} + \hat{k}_{12}\hat{\sigma}_{22}}{|\hat{k}_{11}\hat{\sigma}_{11} + \hat{k}_{12}\hat{\sigma}_{21}|} \quad (4.33)
\]

and

\[
\frac{\hat{k}_{21}\hat{\sigma}_{11} + \hat{k}_{22}\hat{\sigma}_{21}}{|\hat{k}_{21}\hat{\sigma}_{12} + \hat{k}_{22}\hat{\sigma}_{22}|} \quad (4.34)
\]

For column dominance the functions are

\[
\frac{\hat{k}_{21}\hat{\sigma}_{11} + \hat{k}_{22}\hat{\sigma}_{21}}{|\hat{k}_{21}\hat{\sigma}_{12} + \hat{k}_{22}\hat{\sigma}_{22}|} \quad (4.35)
\]

and

\[
\frac{\hat{k}_{11}\hat{\sigma}_{12} + \hat{k}_{12}\hat{\sigma}_{22}}{|\hat{k}_{21}\hat{\sigma}_{12} + \hat{k}_{22}\hat{\sigma}_{22}|} \quad (4.36)
\]

The Leininger optimization procedure locates the maximum dominance ratio over the desired frequency range. If this ratio is greater than one, the selected function is minimized by optimization of the coefficients of the \( \hat{K} \). Using these coefficients the dominance ratios are recalculated over the frequency range. If the maximum value is greater than 1, the optimization routine is called again to minimize the function at this frequency and a new set of coefficients is obtained. If the procedure is successful, for some
value of coefficients, all dominance ratios over the frequency range of interest will be less than 1.

A Fortran program was written to implement Leininger's procedure. This program (OPPMAS) is included in Appendix B.5. A Box-Jenkins non-gradient technique [16, 33, 64] was used to minimize the selected dominance ratio. The user must supply the initial values of the coefficients and the limits on the ranges of the coefficients. For a 2x2 matrix, the values between -1 and 1 form a basis for all coefficients which minimize the functions. It was found that the technique would locate all the function minima in this range, independent of the initial values on the coefficients.

Table 4-7 displays the second row of \( \hat{\Omega} \) using coefficients \( \hat{k}_{21} \) and \( \hat{k}_{22} \) obtained from three iterations of Leininger's procedure. The second row was easily pseudo-diagonalized by the procedure but, despite numerous iterations, the procedure could not find coefficients which resulted in a dominant first row across the frequency range of interest. During these iterations, it was observed from the interactive computer output that the frequency of the maximum dominance ratio would shift from low to high and then high to low on consecutive calls to the optimization routine. This meant that coefficients which minimized
the dominance ratio at low frequencies caused large
dominance ratios at high frequencies. An analogous
condition existed for coefficients which minimized the
function at high frequencies but caused large dominance
ratios at low frequencies. Consequently, another
procedure was instituted which obtained coefficients
which pseudo-diagonalized the first row from 0 rad/sec
up to a frequency at which the dominance ratio was
greater than unity. It was hoped that the Nyquist plot

Table 4-7: \( \hat{q}_{21}(i\omega) \) and \( \hat{q}_{22}(i\omega) \),
\[ \hat{k}_{21} = 0.264, \quad \hat{k}_{22} = 0.765 \]

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of $\hat{q}_{11}$ would intersect the real axis in this range. The procedure uses the initial values of the coefficients to calculate the dominance ratios at each frequency, beginning at 0 rad/sec. When it encounters a dominance ratio greater than one, the optimization routine is called to calculate coefficients which minimize this ratio. Using these new set of coefficients, new dominance ratios are calculated for each consecutive frequency, beginning at 0 rad/sec. The optimization routine is called again at the first frequency in which a dominance ratio greater than one is calculated. If the program fails to pseudo-diagonalize the row after a specified number of iterations, it selects the coefficients which yield a dominant first row across the largest frequency range, starting at 0 rad/sec. 0 rad/sec was chosen as the beginning frequency because it was anticipated that the majority of the inputs during ASV operation would be in the lower frequency ranges and consequently it would be more important to decouple the system at steady state and for low frequencies.

A typical result of this procedure is shown in Table 4-8.

As seen in this table and also in the previous attempt at pseudo-diagonalizing the first row of $\hat{Q}$, the coefficients which pseudo-diagonalize the top row at low
Table 4-8: \(\hat{q}_{11}(iw)\) and \(\hat{q}_{12}(iw)\),

\[k_{11} = 1.0, \quad k_{12} = 0.355\]

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frequencies cause large dominance ratios at high frequencies and shift the frequency at which \(\hat{q}_{11}(iw)\) intersects the real axis.

In the course of using Hawkins' and Leininger's methods, it was observed that in many instances the methods would converge to a local minimum which resulted in a dominance ratio much less than unity. Recall that a dominance ratio less than unity is the only requirement for a dominant matrix. It was hypothesized that the
coefficients obtained from these local minimization routines may cause excessively large dominance ratios at other frequencies. The coefficients obtained by minimizing a function to obtain a dominance ratio slightly less than unity may offer advantages when attempting to pseudo-diagonalize the first row of \( \mathbf{Q} \). It can be shown that there is not a linear relationship between the ratio of \( \frac{k_{i1}}{k_{ij}} \) and the dominance ratio at a particular frequency.

A modification of Hawkins' method could not be found which permitted a selective minimization of the sum of squares of the off diagonal terms such that a desired dominance ratio was achieved. However, using the Box-Jenkins optimization routine, the function to be minimized was changed from equation (4.33) to

\[
\text{ABS}(\hat{k}_{11} \hat{g}_{12} + \hat{k}_{12} \hat{g}_{22})
- \left( \text{Dominance Ratio} \right) \left( \hat{k}_{11} \hat{g}_{11} + \hat{k}_{12} \hat{g}_{21} \right) \tag{4.37}
\]

The dominance ratio was varied from .99 to .1. As the dominance ratio approached .1 it became increasingly difficult to obtain coefficients which pseudo-diagonalized the first row at more than one frequency. Table 4-9 shows the first row of \( \mathbf{Q} \), calculated using coefficients obtained by specifying a dominance ratio of .9. It can be seen from this table that the top row has
been pseudo-diagonalized to 190 rad/sec. However the Nyquist plot of \( \hat{q}_{11} \) does not intersect the real axis in this frequency range. This problem also occurred for several other pairs of coefficients which were obtained by specifying other dominance ratios.

Table 4-9: \( \hat{q}_{11}(i\omega) \) and \( \hat{q}_{12}(i\omega) \).

\[ k_{11} = 0.590, \quad k_{12} = -0.482 \]

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4.4 Alternatives to Row Dominance of $\hat{Q}$

4.4.1 Column Dominance

As stated previously, the INA technique can be applied to either a row or column dominant $\hat{Q}$ matrix. For the reasons stated in the previous section, establishing a row dominant $\hat{Q}$ was given priority over column dominance. Since attempts at pseudo-diagonalizing the first row of $\hat{Q}$ failed, attempts were made to pseudo-diagonalize the columns of $\hat{Q}$. Hawkins' technique was formulated by assuming that the same coefficients which minimize the off diagonal terms also multiply the diagonal term. The matrix in expression (4.28) shows that this is not true for column dominant matrices. Leininger's procedure can be used in attempting to pseudo-diagonalize the columns of a matrix. However, for column dominance expressions (4.35) and (4.36) must be considered simultaneously. The function to be minimized using the Box-Jenkins optimization routine in this case is the higher value of expressions (4.35) and (4.36). Both expressions are computed at each frequency to determine the maximum dominance ratio at that frequency. The array of maximum dominance ratios at a particular frequency are scanned over the frequency
range of interest to determine the maximum dominance ratio. If this value is greater than one, this function (either expression (4.35) or (4.36)) is optimized to obtain coefficients which if successful, minimize its magnitude to less than one. The cycle begins again using this new set of coefficients to compute the dominance ratios. The cycling ends successfully if all dominance ratios over the frequency range of interest will be less than unity.

This program was executed a number of times using different initial conditions and coefficient limits. In all instances, column dominance of $\hat{Q}$ was not achieved. From the interactive computer output it could be seen that the coefficients which pseudo-diagonalized the first column (expression (4.35)) would not pseudo-diagonalize the second column (expression (4.36)). The failure of this method was not surprising if expressions (4.35) and (4.36) are examined. Because $\hat{q}_{11}(i\omega)$, $\hat{q}_{12}(i\omega)$, $\hat{q}_{21}(i\omega)$ and $\hat{q}_{22}(i\omega)$ are similar over the frequency range of interest, the coefficients obtained from the optimization routine which minimize (4.35) will tend to maximize (4.36). The reverse is also true when attempting to pseudo-diagonalize the second column.
4.4.2 Row Dominance Using Nonconstant $\hat{K}$ Matrix

In a further attempt to pseudo-diagonalize the first row of $\hat{Q}$ the top row of the $\hat{K}$ matrix was designated as:

$$\hat{k}_{11s} + \hat{k}_{11}, \quad \hat{k}_{12s} + \hat{k}_{12}$$

As in the case of column dominance, four coefficients must be obtained, $\hat{k}_{11s}, \hat{k}_{11}, \hat{k}_{12s}$ and $\hat{k}_{12}$, but these coefficients are independent of the second row. It was decided to use the Box-Jenkins optimization routine and Leininger's procedure in this attempt. In order to ensure that the $K$ matrix was physically obtainable, the determinant of $\hat{K}$ had to have all its poles in the left half plane. Consequently, constraints were included in the optimization routine, such as

$$\hat{k}_{11s} \hat{k}_{22} > \hat{k}_{12s} \hat{k}_{21} \quad (4.38)$$

and

$$\hat{k}_{11} \hat{k}_{22} > \hat{k}_{12} \hat{k}_{21} \quad (4.39)$$

The program was executed numerous times using various ranges and initial values of the coefficients but Leininger's procedure did not yield coefficients which pseudo-diagonalized the top row. As before, the optimized coefficients which pseudo-diagonalized the top
row for low frequencies produced large nondominant ratios at high frequencies. The reverse was also true.

With a few modifications, Hawkins' technique can also be used to determine the coefficients of a nonconstant precompensation matrix. This pseudo-diagonalization technique was tried by weighting various frequencies. However, these coefficients failed to achieve a dominant first row over the frequency range of interest. In addition, many of the coefficients obtained from this method would not lead to a physically realizable K.

4.4.3 Pseudo-Diagonalization of Q

Rosenbrock also derives stability theorems and design procedures using a dominant Q matrix. However, for the 2-input, 2-output system studied here, it can be shown that the row dominance ratios of \( \hat{Q} \) (expressions (4.33) and (4.34)) are equivalent to the column dominance ratios of Q and the column dominance ratios of \( \hat{Q} \) (expressions (4.35) and (4.36)) are equivalent to the row dominance ratios of Q. Since prior attempts at achieving dominance for the first row or column of \( \hat{Q} \) failed, there would be no reason to expect to obtain dominance of the second column or row of Q.
4.5 Pseudo-Diagonalization at High System Pressures and Low Bypass Valve Areas

To summarize the previous section, it appears that it is not possible to pseudo-diagonalize \( \hat{Q} \) over the frequency range of interest at the conditions listed in Table 2-4. The results of the most successful attempts are listed in Tables 4-9 and 4-7. Using these coefficients, the precompensation matrix \( \hat{K} \) would be:

\[
\begin{bmatrix}
0.590 & -0.482 \\
0.264 & 0.765
\end{bmatrix}
\]

Comparing Table 4-9 to 4-1 and Table 4-7 to 4-2 it can be seen that the dominance ratios for the respective rows are of the same order of magnitude, indicating that the first row dominance ratios of \( \hat{Q} \) have not been appreciably improved with the pseudo-diagonalization precompensation matrix above.

Since attempts to pseudo-diagonalize \( \hat{Q} \) failed at these conditions, it was not possible to use the Gershgorin or Ostrowski plots to determine stability limits or select gain margins for individual loops. However, the method of 2-Hodographs can be used to quantitatively evaluate the gain margin for the series-bypass system. Because there was no apparent reason to do otherwise, the gains in both loops were
chosen to be equal. Various authors [6, 42, 51, 61] recommend different gain margins for single loop servovalve controlled linear hydraulic actuators. These values range from 2 to 10. For the series-bypass configuration with the inherent interaction at these conditions and the feedforward pump control it was necessary to select a high value of gain margin. Figure 4-12 illustrates the system response (no lead filters or integral compensation and using the pseudo-diagonalization matrix above) for proportional gains of .0007 amps/in/sec. Using the 2-Hodograph plots, the gain margin was calculated to be 4, but as seen by this figure the response is highly oscillatory. Using gains of .0005 amps/in/sec, the gain margin is increased to 10 and gives the much more acceptable response shown in Figure 4-13.

4.6 INA Compensation Design at Low Actuator Loads and High Bypass Valve Areas

Using the techniques presented in the previous section, it was not possible to pseudo-diagonalize the $\hat{Q}$ matrix for high actuator loads and small bypass valve areas. The conditions listed in Table 2-4 model the contact phase of the stride during cruise mode when the ASV is traversing uphill and a high degree of energy
Figure 4-12: Velocity of Actuators 1 and 2

Figure 4-13: Velocity of Actuators 1 and 2
efficiency is desired. The analysis of the open loop linearized transfer functions in section 2.6 indicated that the decoupling in the system would decrease as the bypass valve areas increase and the system pressures decrease. This would occur when the legs are lightly loaded, such as during the return phase of the stride, or when the ASV is traversing over straight and level terrain. As explained previously, the equilibrium bypass valve areas could be increased and still maintain a high degree of energy efficiency since the pressure drops across the bypass valves are small under these conditions. The investigations in sections 2.6 and 2.7 have shown that the series-bypass circuit is more stable and less oscillatory as the loads on the actuators decrease. This section will investigate the validity of using the linearized equations and transfer functions to examine the system behavior at low actuator loads and high bypass valve areas.

During the return phase of the stride or for low contact loads, the loads seen at the actuators are estimated to be 100 lbf. To maintain a high degree of energy efficiency (70%) at high velocities (8 in/sec), the equilibrium bypass valve area was chosen to be 75% open or .03 in$^2$. The elements of the G matrix are shown in Figure 4-14. Comparing this figure to Figure 4-2, the
difference in the frequency response due to equilibrium conditions can be seen. At lower actuator pressures and high bypass valve areas, the commanded bypass valve area has less influence (lower magnitude) on the actuator velocities. The feedforward contribution of \( \dot{X}_M \) increases to maintain a zero steady state velocity error.

At low pressures and large bypass valve areas, the \( \hat{Q} \) matrix is diagonally dominant. Leininger's technique was used in an attempt to find coefficients to increase the dominance. The technique failed to find coefficients which increased the dominance in the top row but located coefficients to increase the dominance in the bottom row. The Gershgorin circles at these equilibrium conditions and without precompensation are shown in Figures 4-15 and 4-16 and display the diagonal dominance of the system. The FORTRAN program (OSTROWPLT) which displays these circles or Ostrowski circles is listed in Appendix B.6. It uses the output of MFRSIM to supply the frequency response values. Figures 4-15 and 4-16 predict stability for \( K_{E1} < .015 \) and \( K_{E2} < .006 \) amps/in/sec. Note that these gains are much greater than the gains achieved at higher pressures and decreased bypass valve areas and indicate a more stable system. Also, for this magnitude step (6 inch/sec) and for these relatively high gains, \( A_2 \) closes completely.
Figure 4-14: Nyquist Plots of Matrix G, Low Pressure, High Bypass Valve Areas
Figure 4-15: Gershgorin Circles of $\hat{q}_{11}(i\omega)$

Figure 4-16: Gershgorin Circles of $\hat{q}_{22}(i\omega)$
Figure 4-17: Velocity of Actuator 1, Not Precompensated

Figure 4-18: Velocity of Actuator 2, Not Precompensated
Figure 4-19: Velocity of Actuator 1, Precompensated

Figure 4-20: Velocity of Actuator 2, Precompensated
and \(A_4\) almost closes for a short period of time at the beginning of the step response. After examining the system response at smaller steps, it was decided that the saturation effect which occurs when a valve closes completely for a short period of time does not alter the general transient and steady state characteristics of the response. This is primarily due to the bypass valve control algorithm which attempts to maintain the valve at its non zero equilibrium position. For the remainder of this dissertation saturation effects will be ignored unless otherwise stated. However, valve saturation would definitely effect the system response if the valve was closed for prolonged periods of time.

Figures 4-17 and 4-18 display the closed velocity loop response of the system to a step in \(\dot{X}^{1D}\) when only the commanded bypass valve areas are used as the input to the bypass valves and variable displacement pump (\(\dot{X}_M = \) equilibrium value) and \(K_{E1}\) and \(K_{E2}\) equal to .017 and .01 amps/in/sec respectively. In Figures 4-17 through 4-20, \(\dot{X}_M\) is a constant (i.e. 3"/sec) in order to exclusively identify the transient and steady state effects of the precompensation matrix. In Comparing these figures to Figures 4-3 and 4-4, it can be seen that the interaction has been decreased since the actual velocities of Figures 4-17 and 4-18 are much closer to
their desired values. Figures 4-19 and 4-20 display the velocity response to conditions identical to Figures 4-17 and 4-18 when the precompensation coefficients obtained by using Leininger's technique were used to increase the decoupling in the system. The precompensation matrix was:

\[
\begin{pmatrix}
1.0 & 0.0 \\
-0.45 & 1.04
\end{pmatrix}
\]  

Comparing these figures to Figures 4-17 and 4-18 it can be seen that the precompensation matrix decreases the velocity errors in the response of the second actuator. Recall that Leininger's technique minimizes the dominance ratio across a range of frequencies. It is important to note that not only is the steady state velocity of the second actuator in Figure 4-20 much closer to its desired value, the response is also less oscillatory when compared to the velocity responses shown in Figures 4-4 and 4-18. The reduced oscillations indicate that the interaction has been reduced across the frequency range of interest.

It is important to realize that much of this decoupling seen by comparing Figures 4-3 and 4-4 to Figures 4-17 through 4-20 results from the larger proportional gains. Section 3.7.2 showed that the influence of the bypass valves on the closed velocity
loops decreased at lower pressures and larger bypass valve areas. However, the system has been shown to be more stable at lower system pressures and larger bypass valve areas. Consequently, the proportional gains can be increased without causing instability. Figure 4-21 shows the closed loop response of the first actuator to a step input in its desired value for conditions identical to Figure 4-19 and $K_{E1} = K_{E2} = 0.0005$ amps/in/sec. Comparing this figure to Figures 4-19 and 4-3, it can be seen that large velocity errors occur under these conditions and when the feedforward loop is omitted.

Comparing the Nyquist diagrams of Figures 4-14 and 4-2 or the time domain response of Figures 4-3, 4-4, 4-17, 4-18, 4-19 and 4-20, it is difficult to quantitatively describe the closed loop coupling characteristics of the system. Davison and Man [11] define an interaction index to assess the closed loop coupling in a multivariable system. This index can be used to compare the effects of different precompensation matrices on the system. Although the index is calculated from analytical methods and was derived by using the equations of system state, it was possible to adapt the index to be used in the time domain simulation. It is felt that this time domain index
yields a better quantitative description of the closed loop coupling than an inspection of the dominance ratios over the frequency range of interest. The interaction index $I_j$ is defined as:

$$I_j = \frac{\tilde{J}_j - J_j}{J_j}$$  \hspace{1cm} (4.42)$$

The subscript $j$ refers to the $j$th control loop and $\tilde{J}_j$ can be defined as:

$$\int (\tilde{v}_j)^2 \, dt$$  \hspace{1cm} (4.43)$$

Figure 4-21: Velocity of Actuator 1 at Low System Pressures, Large Bypass Areas
* denotes that both loops are closed. \( J_j \) refers to the case when only the \( j \)th control loop is closed and is defined as:

\[
\int (y_j^*)^2 \, dt \tag{4.44}
\]

\( y_j^* \) and \( y_j \) are defined as:

\[
X_P^j_D - X_P^j \tag{4.45}
\]

Numerical values of the interaction can be quantitatively described by:

- \( I_j < 0 \), favorable interaction will occur in controlling \( y_j \) when the controllers are all closed.

- \( I_j = 0 \), no interaction. The behavior of the \( j \)th loop is independent of the multiple closed loop control system.

- \( I_j > 0 \), unfavorable interaction will occur in controlling \( y_j \) when the controllers are all closed.

- As \( I_j \to \infty \) the interaction will become more severe.
As an example of using these indices to quantitatively describe the closed loop interaction, $J_1$, $\hat{J}_1$ and $I_1$ values for the conditions of Figures 4-3 and 4-19 were computed and are listed in Table 4-10.

Table 4-10: Comparison of Interaction Indices

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<td>4.60</td>
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It can be seen from this table that the $J$ values for Figure 4-19 are much less than the values obtained from the conditions in Figure 4-3. The $J$ values reflect that the interaction at low pressures and large bypass valve areas are much less but this decreased interaction is not as obvious when comparing the $I$ values. Both $I$ values indicate that closing the second loop increases the interaction in the system. Table 4-11 compares the interaction values for the second loop when a unity precompensation matrix and the precompensation matrix of expression (4.41) were used in the system. This table indicates that the precompensation matrix does decrease the decoupling in the closed loop system.
Table 4-11: Comparisons of Unity and Pseudo-diagonalization Matrices

<table>
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<td>Pseudo-Dia</td>
<td>1.25</td>
<td>1.21</td>
<td>0.032</td>
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Using the gain space obtained from Figures 4-15 and 4-16 as a guide, gains were selected which would produce a stable system, satisfy the requirements for a dominant \( \hat{H} \) matrix and yield narrow Ostrowski bands. It proved to be difficult to accomplish all three tasks simultaneously. Gains which produced a stable system and yielded narrow Ostrowski bands failed to produce a dominant \( \hat{H} \) matrix. Figures 4-22 and 4-23 display the Ostrowski envelopes for feedback gains of \( K_{E1} \) and \( K_{E2} \) equal to .015 and .007 amps/in/sec. respectively. As seen in these figures the envelopes are very broad, making any identification of the INA plots of the diagonal elements very ambiguous. These broad bands also indicate that the velocity outputs of feedback loops are influenced by the adjacent bypass valves.

Realizing that the inverse Nyquist plots of the diagonal elements lie within these bands, a lead filter
Figure 4-22: Ostrowski Circles of $\hat{q}_{11}(i\omega)$

Figure 4-23: Ostrowski Circles of $\hat{q}_{22}(i\omega)$
compensation design was attempted. The filters $k_{11}^c$ and $k_{22}^c$ were designed using a procedure introduced by Rosenbrock [51]. Rosenbrock proposes drawing a line at -55 degrees from the real axis. A -55 degree line is the consequence of choosing a 10 to 1 difference between the numerator and denominator breakpoint frequencies of the lead compensators. The numerator breakpoint frequency of the lead filter is defined as the frequency at which the -55 degree line intersects the inverse Nyquist plot of $\hat{q}_{11}(i\omega)$. Note from Figures 4-22 and 4-23 that it is difficult to identify the frequency of the intersection because of the uncertainty in the exact location of the inverse Nyquist plots. Approximating the limits of the bands, it appears that the -55 degree lines could intersect the inverse Nyquist plot of $\hat{q}_{11}(i\omega)$ between 70 and 140 rad/sec and $\hat{q}_{22}(i\omega)$ between 50 and 80 rad/sec.

The uncompensated system response with the precompensator in (4.41) and $k_{E1}$, $k_{E2}$ equal to .015 and .0007 amps/in/sec respectively is shown in Figures 4-24 and 4-25. The simulation was used to study the effect of various lead filters on the system response using the range of numerator frequency values obtained from the Ostrowski envelopes. For the first closed loop system, the lead filters were inserted in locations $G_{1c}(s)$ and $G_{3c}(s)$ in Figure 3-2. Analogously, the lead
filters for the second closed loop system were inserted at \( G_7(s) \) and \( G_5(s) \). Following a systematic search, the responses in Figures 4-26 and 4-27 were selected as the best. These filters were of the form of equation (3.19) with \( \alpha = .1 \) and time constants of .01 and .02 seconds (numerator breakpoint frequencies of 100 and 50 rad/sec) for the first and second velocity loops respectively. Therefore the lead filter for the first loop can be written as:

\[
\frac{.01s + 1}{.001s + 1}
\]

(4.46)

The lead compensator for the second loop can be written as:

\[
\frac{.02s + 1}{.002s + 1}
\]

(4.47)

Comparing Figures 4-26 and 4-27 to Figures 4-24 and 4-25, it can be seen that the rise time of both actuators has been decreased when the lead filters were used. It is interesting that the selected numerator time constants of the first and second velocity loops were similar to the value selected for the lead filter design at higher system pressures and smaller bypass valve areas (.01 seconds, see section 3.5). The oscillations in the responses in Figures 4-26 and 4-27 were caused by using the commanded bypass valve
area as an input to both the bypass valves and variable displacement pumps. Although the input to these actuators occur at the same instant, the dynamics of the servovalves and variable displacement pumps are different.

Figures 4-28 and 4-29 illustrate the effect of the lead filters on the diagonal elements of the \( \hat{Q} \) matrix. As discussed in section 4.2 these figures show that the row dominance of the matrix has been unaffected by the addition of the lead filters. Note that the Ostrowski envelopes in these figures are narrower than in Figures 4-22 and 4-23.

4.7 Summary

In Chapter 4, the coupled nature of the series-bypass circuit was addressed. Rosenbrock's INA method is a multivariable technique to design compensators and determine the gain space of multiple-input, multiple-output systems. After a search of available techniques, this method was selected because it is a robust frequency response technique which lends itself to using the linearized frequency response curves. The method relies on a frequency response model of the system which may be obtained by empirical testing or from a mathematical description of
Figure 4-24: Velocity of Actuator 1

Figure 4-25: Velocity of Actuator 2
Figure 4-26: Velocity of Actuator 1, Lead Filtered

Figure 4-27: Velocity of Actuator 2, Lead Filtered
Figure 4-28: Ostrowski Circles, \( \hat{q}_{11}(i\omega) \), Lead Filtered

Figure 4-29: Ostrowski Circles, \( \hat{q}_{22}(i\omega) \), Lead Filtered
the system. Also, to apply the technique, the system must be open loop diagonally dominant. It was shown that pseudo-diagonalization of the open loop transfer functions will sometimes decrease the closed loop coupling. While the integral and unequal desired velocity compensation algorithms decreased the closed loop velocity errors due to the coupling in the system, they did not specifically attempt to decrease the coupling in the system. Also, these algorithms did not directly address the coupling displayed at higher frequencies. The pseudo-diagonalization techniques attempt to decrease the coupling across the frequency response of the system. Several pseudo-diagonalization techniques were applied but they were unable to pseudo-diagonalize the series-bypass circuit at high pressures and low equilibrium bypass valve areas. At low pressures and large bypass valve openings, the series-bypass circuit became diagonally dominant without a precompensation matrix. However, simulation results showed that a precompensation matrix designed by using a pseudo-diagonalization technique decreased the closed loop coupling in the system below that which occurred without using the precompensator. Lead compensators for the commanded bypass valve areas of both closed loops were designed from the diagonally dominant inverse open
loop frequency response curves. Simulation studies showed that these filters increased the closed velocity loop response of both actuators.

An important reason for the increased closed loop decoupling at low pressures and large bypass valve areas was the greater system stability at these conditions. It was shown in this chapter that a diagonally dominant \( \hat{Q} \) matrix does not ensure that the closed loop response would display an equivalent amount of decoupling. However, it was shown in Chapter 2 that the magnitude of the open loop transfer functions decrease as the pressures decrease and the bypass valve areas increase. The system stability increases as the magnitude of the resonant peaks decrease. As the stability of the system increases (gain margin increases), higher proportional gains can be used in the feedback control loops. If the gains were kept constant at the stable values for high pressures and low bypass valve areas, simulation results showed that the decoupling of the closed velocity loops would not increase as the pressures decrease and bypass valve areas increase. This occurs because the contribution of the bypass valves to the total closed loop response decreases. As the proportional gains increase, the closed velocity loops become "tighter" and the magnitudes of the diagonal terms of the closed
velocity loop matrix increase, contributing more to the total closed loop response and increasing the magnitude of the total closed loop decoupling. This is not unlike increasing the proportional gains of a SI-SO system in order to "stiffen" the system response to external disturbances. In the series-bypass system, the external disturbances include the interaction from the adjacent closed velocity loop.
Chapter 5

PROTOTYPE SYSTEM BEHAVIOR

5.1 Description of Prototype System Peripherals and Selection of Experimental Operating Conditions

A hydraulic power supply consisting of two manually operated variable displacement pumps, a pressure compensated variable displacement pump and the prototype electronically actuated variable displacement pump, was used to furnish power to the prototype system described in section 2.1 and Appendix A. A schematic of the power supply is presented in Figure 5-1. The two manually operated variable displacement pumps supplied hydraulic power to the load circuits (see Appendix A for a description of the load circuits). The pressure compensated pump supplied a 1000 psi pilot supply to the second stage of the prototype bypass valves (valves 2 and 4 in Figure 2-1) and the proportional valves (valves 1 and 3 in Figure 2-1), the servovalve actuator for the swash plate of the electronically actuated variable displacement pump and also provided the supercharge...
supply to the same pump. The supercharge was necessary because of the closed loop connection between the series-bypass circuit and the variable displacement pump (see Figure 2-1). A 15 hp electric motor was used to rotate the pumps but due to motor, mechanical transmission and pump losses the useful hydraulic power from the power supply was estimated to be 10 hp.

Before acquisition of experimental data it was necessary to determine the conditions which would divide the available power among the four pumps in an optimum manner. Optimum conditions were determined from the operating characteristics of the actuators. These conditions attempted to maximize the output from all transducers while duplicating ASV operating conditions.

Figure 5-2 shows the internal fluid paths of the four way servovalve which was used as the bypass valve. The bypass flow entered the servovalve at port P and exited at port C2. Port C1 was blocked. The external pilot supply entered at the pilot port and returned to the reservoir through port R. Simulation studies revealed that for certain servovalve models and during rapid spool movements around null, it was possible for the flow to exit through R from C2 if the spool overshot the null position. In an attempt to prevent this problem, the servovalves were returned to the factory
Figure 5-1: Hydraulic Power Supply
and retrofitted with spools which had the length of the right land slightly increased to overlap R.

It should also be mentioned that carts carrying weights were attached to the rod ends of the prototype actuators (not shown in any figure) to emulate the leg inertias seen by the main drive actuators of the ASV.

Experimental tests were to be initially conducted at small nominal bypass valve openings. However, preliminary testing revealed that the four port servovalves could not maintain small bypass valve openings. When tests were conducted at small bypass valve openings, the actuators would exhibit irregular movements, at times halting. After eliminating all other
possibilities, it became apparent that despite the overlap, the bypass valves were permitting the flow at C2 to return to the reservoir through R. Consultation with factory representatives supported this hypothesis. The representative stated that a four way servovalve was only calibrated at null and was not designed to accurately maintain a steady state position other than null, especially positions in the highly nonlinear region around null. He further stated that the servovalve operates best in closed loop applications when the spool is continually shifting around null. Another factor affecting the irregular behavior was the nonequal hydraulic forces on the spool caused by using only one side of the servovalve. Considering that a 10% valve opening has an area of .004 in², it was not surprising that the servovalves had difficulty in maintaining this small opening. Consequently, a 50% opening was chosen as the minimum equilibrium bypass valve opening. This allowed operation in a region as far from null as possible while permitting the maximum possible valve movements in either direction.

To determine the empirical static and dynamic characteristics of the series-bypass system, a velocity step of 6 in/sec was introduced from an initial equilibrium velocity of 3 in/sec. These values were
selected on the basis of maximizing the output of the tachometer while minimizing the amount of power supplied to the main drive circuit. A step of 6 in/sec was necessary to clearly observe the dynamic response to a step change in velocity.

Unless otherwise stated, experimental tests were performed with nominal actuator loads of 1300 lbf. This value was selected as a compromise between maximizing actuator loads to simulate loaded ASV conditions and remaining within horsepower limitations. A typical actuator load history is shown in Figure 5-3. The sampling interval for the data acquisition/load control system was 2.9 milliseconds.

As shown by this figure, the loads on the actuators are not constant. The electronically actuated pressure relief valves "chug" as they attempt to maintain the desired load. After an investigation into this behavior, it is believed that this performance can be attributed to valve operation, the volumetric flow rate delivered to the relief valves and the coupling which exists in the hydraulic power supply and return lines. During the step change in velocity or load change, all pumps compete for power from the single electric motor. Since the prototype circuit is using the maximum available power from the hydraulic power supply, changes in one
component of the circuit would sometimes affect the other components. Despite this problem, it was observed that the variations in the actuator loads were sporadic and multiple tests could be performed at an operating condition to obtain a test with minimum load variations. The load variations are not surprising in view of the open loop nature of the load control. A static open loop calibration was conducted on the load circuits. However, the gains obtained from this calibration did not control the load circuits during the actual dynamic testing of
the prototype circuit. Since the load is measured continuously, the load variation does not pose a real obstacle to verifying models of prototype behavior. The measured load can be used as the loading function and input to the prototype simulation.

Figure 5-4 illustrates the computer network which controlled the prototype system, load circuit and data acquisition. The prototype control, load control and data acquisition programs are listed in Appendix C. The prototype and load control programs which reside on the Single Board Computers (SBC’s) were primarily written in PASCAL. The A/D and D/A routines were written in 8086 assembly language to minimize the execution time. During a test, the controller gains, desired velocities, lead filter time constants, integral controller gains and data acquisition interval can be input to the program through impedance buffered potentiometers or included in the software. The prototype and load control/data acquisition SBCs coordinate the load cycle and data acquisition interval through the parallel port. To obtain high data acquisition sampling rates, the prototype control SBC sends parallel signals to the data acquisition/load control SBC. These signals inform the data acquisition/load control SBC of the direction, position and speed of the prototype actuators. This
information was necessary for the functioning of the prototype control but would only decrease the sample rate if performed on the data acquisition/load control SBC.

After a test has been conducted and the experimental data stored on the external RAM board, it is transferred from the data acquisition/load control SBC to a Microcomputer Development Station (MDS) file. The data acquisition system can be user programmed for multiple channels and different sampling rates. A major portion of the SBC data acquisition program and the program to transfer data from the SBC to the MDS was written in PASCAL. The routines that transmit and receive data through the RS-232 link were written in assembler, for speed considerations. This data file can then be transferred to the VAX for analysis and plotting using the higher level facilities available on the VAX. The MDS to VAX data transfer programs were written in PASCAL (MDS) and FORTRAN (VAX). The FORTRAN program used numerous calls to system command procedures. Both programs are also listed in Appendix C and were co-written by the author.

To implement the analog control equations introduced in section 2.4 on a digital computer they must be programmed in their discrete form. The discrete
Figure 5-4: Prototype System Computer Hierarchy
forms of the commanded bypass valve areas (equations (2.44) and (2.45)), the commanded pump flow rate ((2.48)) and the algorithm to correct for unequal desired velocities ((3.27)) are the same as the continuous time equations because these expressions do not contain any dynamic elements. A hold equivalence algorithm [17] was used to program the lead filters and a numerical trapezoidal formula [56] was used to approximate the integral controller. Using the hold equivalence method, the output of a lead filter of the form (see equation (3.19))

\[
\frac{as + 1}{bs + 1}
\]

(5.1)
can be written in the form of a difference equation as:

\[
y(k) = y(k-1)e^{-T/b} + (a/b)x(k) - (e^{-T/b} + (a-b)/b)x(k-1)
\]

(5.2)

If equation (5.1) is compared to (3.19) it can be seen that 'a' is equivalent to \( \tau \) and \( b \) is equivalent to \( \omega \).

The numerical form of the integral controller (equation (3.22)) can be expressed as

\[
y(k) = \{x(k) + x(k-1)}(T/2)A_SK_{\text{Integral}} + y(k-1)
\]

(5.3)

where
- $y(k)$ is the output of the filter/controller at the $k$th sampling instant

- $y(k-1)$ is the output of the filter/controller at the $(k-1)$th sampling instant

- $x(k)$ is the input to the filter/controller at the $k$th sampling instant

- $x(k-1)$ is the input to the filter/controller at the $(k-1)$th sampling instant

- $T$ is the sample period

- $K_{\text{Integral}}$ is the discrete integral gain

The discrete computational sequences used to calculate the commanded bypass valve areas and commanded variable displacement pump flow rate are displayed in Figure 5-5. Although not shown in the figure a precompensation matrix would operate on the values of $A_{2C,P}(k)$ and $A_{4C,P}(k)$.
Figure 5-5: Discrete Control Equations
5.2 Experimental Behavior

Figures 5-6 and 5-7 display the unloaded step response of the prototype system. The control loop was implemented every 7 milliseconds and the proportional gains $K_{E1}$, $K_{E2}$ were .0031 amps/in/sec. Integral and lead filter compensation were not employed in these tests. The system responds to the change in desired velocity but displays high frequency velocity fluctuations that were not observed in the simulation results. However, the actual velocities oscillate around the desired velocity. From the overshoot exhibited by these plots, it was concluded that the inherent system (prototype and load circuit) damping was sufficient and it was not necessary to augment the system damping with acceleration feedback. The damping of the prototype and load circuit system was greater than the damping constant listed in Table 2-3. A measurable value was difficult to obtain because the amount of damping varied with back pressure of load circuit, locations of pistons in cylinders and load circuit performance in general. Also, recall that acceleration feedback was judged necessary in the prototype simulation for 10% bypass valve openings and for actuator loads of 5740 lbf. In these prototype tests, the larger bypass openings
Figure 5-6: Velocity of Actuator 1, No Loads

Figure 5-7: Velocity of Actuator 2, No Loads
dissipate more hydraulic energy, increasing the system damping. The lower pressures decrease the system stiffness, also decreasing the magnitude of the oscillations in the response and eliminating the need for additional damping.

Figures 5-8 and 5-9 show the simulation response at 1300 lbf actuator loads, proportional gains of .0031 amps/in/sec, 50% bypass valve openings, no filters or integral compensation and using acceleration feedback to supplement the system damping in order to match the additional damping of the prototype circuit. Figures 5-10 and 5-11 exhibit the prototype system velocity response at the same loads, gains, mean bypass valve openings, no compensators and a control loop sample time of 7 milliseconds. The low frequency velocity fluctuations in these figures are due to the load circuit behavior. From these two sets of figures, it can be seen that the prototype response displays high frequency oscillations similar to Figures 5-6 and 5-7. The rise times of the two prototype actuators are greater than the simulated response, indicating that the prototype system has a slower speed of response. Comparing the magnitudes of the initial overshoots in the velocity responses, it can be seen that the overshoots in the velocity responses of the prototype
Figure 5-8: Velocity of Actuator 1, Simulation

Figure 5-9: Velocity of Actuator 2, Simulation
actuators are approximately equal and lie between the simulated velocity overshoots of the first and second actuators. Consequently, it is believed that the prototype high frequency oscillations were not caused by poor system damping. Possible reasons for the oscillations are unknown load circuit interactions, control loop sampling effects or inadequacies in the dynamic model implemented by the simulation. Comparing Figures 5-10 and 5-11 to 5-6 and 5-7 it can be seen that the system response at higher pressures is more oscillatory, agreeing with analytical and simulation analyses in sections 2.7 and 4.3.

In an attempt to determine whether additional line dynamics were causing the oscillations in the prototype response the length of segment A-B in Table 2-1 and Figure 2-2 was changed from 118 inches to 66 inches. This length was the only part of the prototype circuit which could be easily modified. However, this alteration had no distinguishable affect on the system response. Due to the small variation in the total fluid volume caused by this change it does not offer conclusive proof that line dynamics are not a factor in the response.

To investigate the effect of sampling rates on the prototype system velocity response, Figures 5-12, 5-13 and 5-14 illustrate the system behavior under
Figure 5-10: Velocity of Actuator 1, Prototype

Figure 5-11: Velocity of Actuator 2, Prototype
proportional control using proportional gains of .0031 amps/in/sec, 1300 lbf actuator loads and control loop sampling intervals (T) of 4.3, 7 and 11 milliseconds respectively. 4.3 milliseconds is the minimum time necessary to implement the bypass valve and pump proportional control algorithms (equations (2.44), (2.45) and (2.48)). 7 milliseconds is the time required to implement the digital algorithms for the lead filters (equation (5.2)), PI control algorithms ((2.44), (2.45) and (5.3)), pump control algorithm ((2.48)) and compensate for unequal desired velocities (equation (3.27)). See Figure 5-5 for the computational sequence. A delay was included in this sequence to obtain a 11 millisecond control interval.

Comparing Figures 5-12, 5-13 and 5-14, it can be seen that the system response becomes more oscillatory as the sampling time increases. There is a definite effect between sampling times of 4.3 and 11 milliseconds but not a large difference between the responses for sampling intervals of 4.3 and 7 milliseconds. Note that the frequencies of oscillation in Figures 5-12, 5-13 and 5-14 are 246, 256 and 282 rad/sec respectively. These frequencies may be related to the second resonant peak observed in Figures 3-79 and 3-82.

Figures 5-15 and 5-16 display the simulated
Figure 5-12: Velocity of Actuator 2, $\dot{x}_2$

Figure 5-13: Velocity of Actuator 2, $\dot{x}_2$
response of the first actuator with a continuous and discrete proportional controller for the same conditions as Figures 5-10 through 5-14. The discrete controller was implemented every 4.3 milliseconds, as in Figure 5-12. Comparing the two figures, it can be seen that when the discrete controller was used, the amplitude of the velocity oscillations increased and their decay rate decreased. At these sample rates, the discrete controller appeared to have little effect on the frequency of the oscillations.
Figure 5-15: Velocity of Actuator 1, Simulation

Figure 5-16: Velocity of Actuator 1, Simulation
The analytical frequency response curves displayed in Figures 2-14 and 2-15 have a flat response to approximately 80 rad/sec and a second resonance peak at 225 rad/sec. Various researchers [17, 29, 43, 56] state that a sampled data servo system may require sample rates as high as 10 times the maximum significant analog frequency for the sampling effects to be negligible. Using this criterion, a system with significant response to 80 rad/sec would require a digital control loop with a sample time of 7.9 milliseconds. Systems with significant frequency components greater than 80 rad/sec would require sampling intervals less than 7.9 milliseconds. Consequently, this criterion and the simulation and experimental data appear to indicate that sampling intervals of 7 milliseconds would cause the velocity response of the system to be more oscillatory than the velocity response of an analog controlled system. A nominal value for the period of oscillations in Figures 5-12, 5-13 and 5-14 is at least twice the highest sampling interval (22 milliseconds). Unless otherwise stated, the sampling interval for all experimental results displayed in this chapter is 7 milliseconds.

Based on the frequency response analysis in section 2.7, it has been stated that the system response to a
bypass valve movement should be faster than the response due to a change in the output of the variable displacement pump. Figures 5-17 and 5-18 show the prototype open loop velocity response to step change in the bypass valve and pump flow rate. Comparing the initial slope of the actual velocities in these figures, it can be seen that the system responds slightly faster to a change in the bypass valve area. This result is consistent with the simulation results presented in Figures 3-12 and 3-13.

Figures 5-19 and 5-20 exhibit the open loop step response of the first actuator to changes in the bypass valve area or the variable displacement pump, one at a time. These figures display the same data as Figures 5-17 and 5-18 from 0 to .3 seconds and show that the velocity response of the bypass valve controlled actuator is much more oscillatory than the pump controlled actuator. This suggests that the oscillations seen in the velocity profiles of the prototype are primarily caused by the response to bypass valve operation. Note that the frequency of oscillations in Figure 5-19 is 285 rad/sec, similar to those observed in Figures 5-12, 5-13 and 5-14.

Figure 5-21 shows the Nyquist plot of the determinant of the linearized system discussed in
Figure 5-17: Velocity of Actuator 1, $\dot{x}_1$

Figure 5-18: Velocity of Actuator 1, $\dot{x}_1$
Figure 5-19: Velocity of Actuator 1, $\dot{x}_1$

Figure 5-20: Velocity of Actuator 1, $\dot{x}_1$
section 3.3 for actuator loads of 1300 lbf, 50% bypass valve opening and proportional gains of .0031 amps/in/sec. The Nyquist plot predicts a stable system, which is in agreement with the experimental results of Figure 5-12. However, the plot passes near the origin, suggesting that a decrease in gain might cause a more desirable response. The velocity response of the simulated system in Figures 5-8, 5-9, 5-15 and 5-16 and the prototype response in Figures 5-10 and 5-11 display the characteristics of a system with a low degree of stability. A decrease in the prototype system gains did decrease the frequency and amplitude of the oscillations in the response. However, the prototype and hydraulic power supply had a tendency to stall at lower values of gain. For smaller proportional gains, the corrective action due to the step change in velocity is less. At lower gains and at random instances, the actuator response was unable to overcome the high load variations and the actuators would stall. A proportional gain of .0031 amps/in/sec resulted in a reasonable and reliable system performance. However, as shown by Figure 5-13, this gain may be too high for desirable system response at these conditions.

As a further comparison between the linearized analysis and prototype system, the proportional gains
Figure 5-21: Nyquist Plot of Expression (3.7), \( K_{E1} K_{E2} = 0.0031 \)

were increased to .0041 amps/in/sec. The characteristic equation plotted in Figure 5-22 clearly indicates an unstable system. This agrees with the prototype response shown in Figure 5-23 and the simulated response in Figure 5-24. Note that the frequencies of the oscillations in Figure 5-23 vary from 300 to 350 rad/sec while the frequency of the oscillations in Figure 5-24 is approximately 120 rad/sec. The Nyquist plot crosses the real negative axis at 120 rad/sec and almost encircles the origin again at 315 rad/sec. The results
presented in Figures 5-21, 5-22, 5-23 and 5-24 suggest that the modeling, linearization and analog approximations of the prototype circuit were valid and the linearized equations and simulation can be used to guide a designer in selecting proportional gains for the series-bypass system. The apparent deficiency in the model is its inability to predict the high frequency oscillations observed experimentally.

Figure 5-22: Nyquist Plot of Expression (3.7)

Simulation studies in section 3.6.2 revealed the effectiveness of integral control in reducing the
Figure 5-23: Velocity of Actuator 1, Prototype

Figure 5-24: Velocity of Actuator 1, Simulation
velocity errors induced by dynamic loading. Figure 5-26 exhibits the prototype behavior as a result of the loading function shown in Figure 5-25. As displayed in Figure 5-27, the errors were substantially reduced when the discrete integral control algorithm, equation (5.3), was included in the program. The sampling interval during these tests was 7 milliseconds. The integral and proportional gains used in the prototype experiments were .0156 amps/in and .0021 amps/in/sec respectively. For comparison, the integral and proportional gains used in the simulation were .04 amps/in and .0005 amps/in/sec respectively. As discussed previously, the lower proportional gains used in the simulation can be attributed to the higher system pressures used in the simulation of the prototype. Also, the output of the PI controller in the simulation was lead filtered (see section 3.6.1).

In section 3.5, the prototype simulation displayed the feasibility of inserting lead filters to improve an actuator's speed of response. Figures 5-28 and 5-29 show the response of the prototype actuators when the commanded pump volumetric flow rate was lead filtered at the location of filter $G_9(s)$ in Figure 3-2, according to equation (5.2). These figures were obtained using a filter with a numerator breakpoint frequency of 66.5
Figure 5-25: Actuator Loads, $F_1$ and $F_2$

rad/sec, a 10 to 1 difference between numerator and denominator breakpoint frequencies, 1300 lbf actuator loads, proportional gains of 0.0023 amps/in/sec and a sampling interval of 7 milliseconds. Comparison of Figure 5-28 with 5-10 and Figure 5-29 with Figures 5-11 and 5-13 indicate an improvement in the speed of response. The lead filter used in the simulation investigations of section 3.5 used a numerator breakpoint frequency of 50 rad/sec, a 10 to 1 difference between numerator and denominator breakpoint frequencies.
Figure 5-26: Velocity Actuator 2, No Integral Control

Figure 5-27: Velocity of Actuator 2, Integral Control
Figure 5-28: Velocity of Actuator 1, Pump Filtered

Figure 5-29: Velocity of Actuator 2, Pump Filtered
and proportional gains of 0.0005 amps/in/sec.

In section 3.5, the commanded bypass valve areas to the servovalves were also lead filtered according to equation (3.19). In practice however, this proved difficult to duplicate in the prototype response. Figure 5-30 shows the prototype system response using the discrete implementation of (3.19), equation (5.2), with a numerator breakpoint frequency of 500 rad/sec, a 10 to 1 difference between the numerator and denominator breakpoint frequencies, sampling interval of 7 milliseconds, 1300 lbf actuator loads and proportional gains of 0.0023 amps/in/sec. This algorithm was implemented at locations $G_1(s)$, $G_3(s)$, $G_5(s)$ and $G_7(s)$ in Figure 3-2. The pump command was not lead filtered during these experiments. For comparison, the lead filter selected for the simulation model had a numerator breakpoint frequency of 100 rad/sec. Suspecting that the 10 to 1 frequency range of the filter was too large, a variable range was introduced into the prototype control program. Figure 5-31 shows the system response using a lead filter with a numerator breakpoint frequency of 275 rad/sec and a 3 to 1 breakpoint frequency ratio. As seen in this figure, the system oscillations have decreased but comparing this figure to Figure 5-13 it can be seen that the system is
more oscillatory and the speed of response has not increased. Figure 5-32 displays the prototype behavior using a slightly higher numerator breakpoint frequency of 325 rad/sec and a 2.5 breakpoint frequency ratio. In this figure and for similar lead filters tested, no increase in the speed of the response to a step input could be seen.

This discrepancy between the simulation and experimental behavior can probably be attributed to the approximate second order servovalve frequency response model (see Table 2-3) used in the simulation and the servovalve frequency response curves supplied by the manufacturer shown in Figure 5-33. As shown in Figure 5-33, the phase angle curve drops more sharply than for a second order system. The amplitude frequency response curve fits that for a second order system. This feature frustrated the frequency response curve fitting program [13] which was used to fit the manufacturer's curves. Attempts at curve fitting Figure 5-33 with more elaborate models did not yield results which fit the curve considerably better than the second order approximation. Even if a model had been found to fit the curves of Figure 5-33, it is doubtful that these would have been relevant because of the conversion of the four way servovalve into a bypass valve and the spool alterations.
Figure 5-30: Velocity of Actuator 2, Bypass Filtered

Figure 5-31: Velocity of Actuator 2, Bypass Filtered
Figure 5-32: Velocity of Actuator 2, Bypass Filtered

Figure 5-33: Servovalve Frequency Response
Figures 5-34 and 5-35 show the prototype system response when only the first actuator is subjected to step change in velocity. In order to isolate the effects of a difference in desired velocity, no integral or lead filters were used to obtain this response. The steady state errors are consistent with the simulation results discussed in section 3.6.1. When the bypass valve adjustment algorithm, equation (3.27), is used to compensate for the difference in desired velocities, Figures 5-36 and 5-37 are obtained. Again, to isolate the effects of this algorithm, no other form of compensation was used. These figures agree with the simulation results, which showed that this algorithm reduced the steady state errors. Note that this algorithm can be programmed directly without any discrete conversion.

Figures 5-38 through 5-41 display the prototype system behavior to the conditions similar to those described in section 4.6 (100 lbf actuator loads, 0.03 in\(^2\) bypass valve areas, no other forms of compensation). These tests were conducted to experimentally verify the closed velocity loop decoupling observed for simulation studies at low system pressures and high bypass valve openings and predicted by the linearized analysis. Since these predictions only involved the transfer
Figure 5-34: Velocity of Actuator 1, $\dot{x}_1$

Figure 5-35: Velocity of Actuator 2, $\dot{x}_2$
Figure 5-36: Velocity of Actuator 1, Compensated

Figure 5-37: Velocity of Actuator 2, Compensated
functions between the bypass valves and actuator velocities, only the bypass valves were used to control the prototype velocities. $K_{E1}$ and $K_{E2}$ were set at 0.017 and 0.01 amps/in/sec respectively and the precompensation matrix of expression (4.41) was used. However, repeated tests showed that the response appeared to be independent of whether expression (4.41) or a unity matrix was used as precompensation. This may be due to bypass valve saturation. The bypass valve model used in this study predicts that the bypass valves would be fully closed during the initial moments of the step response for these proportional gains and for this velocity step input. For reasons stated previously, it was very difficult to perform valid experiments at smaller step sizes. Also, it was not possible to adequately control the load circuit for loads under 400 lbf. Therefore, the load commands were set to zero to minimize the effects of the load circuit on the system response and the precompensation matrix (4.41) was retained. Typically, the load cells measured actuator loads of only 50 lbf during a test.

Figures 5-38 and 5-39 display the velocity response to a step change in the desired velocity of the first actuator. The characteristics of the steady state response are similar to those exhibited by Figures
4-19 and 4-20. Note that when the first actuator is subjected to a step in its desired velocity, its steady state error is less than that when the second actuator is subjected to a step in its desired velocity (Figures 5-40 and 5-41). This is due to the difference in the values of the proportional gains. Also, the high gains reduced the steady velocity errors in the responses of the actuators not subjected to the step increase in velocity to zero.

The coupling at different actuator loads and bypass valve areas can be compared from Figures 5-38, 5-39, 5-19 and 5-42. Figure 5-42 was obtained during the same test as Figure 5-19 (1300 lbf actuator loads, .02 in$^2$ bypass valve areas and $K_{E1}, K_{E2} = .0031$). Figure 5-21 showed that the system of Figures 5-19 and 5-42 had a low degree of stability (gain margin approximately 1). A conservative gain margin of 4 can be estimated from the Ostrowski plots of Figures 4-22 and 4-23 since the controller gains in these figures are similar to those for the system of Figures 5-38 and 5-39. Consequently, the proportional gains of Figures 5-38 and 5-39 could be increased so that the gain margins of the two systems are equal. The discrepancy in the gain margins was mentioned because previous results (see section 3.6.1) have shown that steady state velocity errors decrease as
Figure 5-38: Velocity of Actuator 1, $\dot{X}_1$

Figure 5-39: Velocity of Actuator 2, $\dot{X}_2$
Figure 5-40: Velocity of Actuator 1, $\dot{X}_1$

Figure 5-41: Velocity of Actuator 2, $\dot{X}_2$
the proportional gains are increased. However even for higher gain margins, the system of Figures 5-38 and 5-39 display smaller velocity errors when compared to Figures 5-19 and 5-42 respectively. This would seem to experimentally verify the statements made regarding decreased velocity errors as the system pressures decrease and the bypass valve areas increase.
5.3 Summary

The empirical data displayed in Chapter 5 agreed quite well with the frequency response predictions and simulation results obtained from the series-bypass model, despite evidence that the sampling interval should have been included in the analysis. The empirical response exhibited high frequency oscillations which were not seen in the simulation studies. The frequency of these oscillations was of the same order of magnitude as the location of the second resonant peak seen in the linearized frequency analyses in Chapters 2 and 3. Using a frequency domain criterion presented in Chapter 3, these frequency response curves were used to successfully predict the stability of the prototype system. Also, empirical data showed that the velocity response of an actuator to a change in bypass valve area was faster than the response to a change in the pump flow rate.

It was shown that the simulation could be used as a guide to select control algorithms and suggest integral and proportional controller gains and pump lead filter time constants for the prototype system. The empirical data from the prototype circuit demonstrated that the algorithm to correct for unequal desired velocities,
integral controller and lead filter for the variable displacement pump could also be successfully implemented. However, the author was unable to effectively lead filter the commanded bypass valve areas or duplicate the closed loop decoupling seen in the simulation studies of section 4.6. There was no obvious change in the response when the matrix of expression (4.41) was inserted in the control loop. As stated previously, this may have been due to some effect of valve saturation which was not predicted by the bypass valve model. The commanded bypass valve area lead filter and precompensation matrix design are related in that they both rely on an accurate frequency response model of the bypass valve. It is unlikely that compensation techniques based on this model can be effectively implemented on the prototype system if this model does not accurately describe the physical valve.

The closed velocity loop coupling of the empirical system did decrease when the proportional gains were increased and the system was operating at relatively low pressures and large bypass valve areas. As mentioned above, this decoupling was independent of the precompensation matrix. It is believed that this decoupling can be attributed to the higher gains which were made possible by the increased system stability at
these operating conditions.
6.1 Conclusions

The previous five chapters have analyzed the series-bypass circuit using:

1. Symbolic transfer functions and frequency response curves obtained by linearizing a mathematical model of the system about an equilibrium condition.

2. A time domain simulation consisting of the nonlinear mathematical descriptions of the system.

3. Experimental testing.

Whenever possible, comparisons have been made between these three techniques. It was shown throughout this dissertation that the linearized methods could be
used to predict the operating characteristics of the series-bypass circuit and suggest methods of control and controller gains for the simulation and prototype system. The effect of these or other controllers could be studied or "tuned" by using the nonlinear time domain series-bypass simulation and then implemented on the prototype system. Based on the results obtained in this dissertation, it can be stated that the series-bypass system was adequately modeled by both the linearized equations and nonlinear simulation. Therefore, both these methods could be used to obtain additional data on the operating characteristics of this hydraulic circuit.

Additionally, these three methods of analyses have shown the series-bypass circuit to be a feasible method to provide controlled hydraulic power to the main drive actuators of the ASV. The circuit was designed to meet the requirements of the ASV during cruise mode. The primary control algorithms for the series-bypass circuit were formulated to achieve equal actuator velocities for equal desired velocity inputs and actuator loads. These objectives were accomplished since simulation studies displayed essentially zero actuator velocity errors during simulated cruise mode conditions. For the same set of conditions, the prototype response was more oscillatory. However, the experimental velocities varied
about the mean desired velocity, suggesting that zero velocity errors could be achieved if the oscillations could be suppressed. At this time, no judgement has been made as to whether these oscillations would be acceptable in the ASV application.

During cruise mode, the series-bypass circuit was designed to be more energy efficient than a servovalve controlled piston. However, time response studies indicate that the bandwidth of the series-bypass circuit is probably less than the servovalve controlled piston. Both the simulation studies and empirical results have shown that the response of the actuators to a step in the bypass valve area is only slightly faster than the response to a step in the variable displacement pump flow rate. Consequently, the addition of a bypass path in order to increase the speed of response of a hydrostatic circuit can not be justified. When compared to the servovalve controlled piston, the lower bandwidth of the series-bypass circuit is probably due to the relatively large fluid lengths between the bypass valve and actuator. Consequently, in relative terms, it could be stated that the bandwidth of the series-bypass circuit is equivalent to the bandwidth of a hydrostatic circuit which used the same variable displacement pump as the series-bypass circuit in this study.
A hydrostatic circuit is currently being used for the ASV application. This hydrostatic circuit is displayed in Figure 6-1. Note that this figure only displays the hydraulic configuration of one leg of the ASV. Review section 1.1 and Figure 1-2 for a description of the leg assembly and actuators. Comparing Figure 6-1 to Figure 1-4, it can be seen that the current ASV hydraulic circuit uses a single variable displacement pump to control the motion of an individual actuator. Because these pumps only supply a single actuator they are smaller and lighter than the pumps evaluated in the hydrostatic circuit of section 1.3.2. Because of their relatively small size and weight, these pumps can be located very close (< 2 feet) to the actuators. As stated in section 1.3.2, locating a pump near the actuator it is controlling increases the bandwidth of a closed loop system.

It was shown in this dissertation that by using relatively simple proportional control algorithms, the series-bypass circuit could accomplish the objectives of the cruise mode. Because the main drive actuators of each leg are independent, it is uncertain whether the current ASV hydraulic design can duplicate the equal actual leg velocities required in cruise mode without a more sophisticated control structure. This control
structure would have to monitor the actual leg velocities and ensure that they are nominally equal. Another example of the inherently simple hydromechanical control (neglect control algorithms) of the series-bypass circuit is as follows. The legs and frame of the ASV form a closed chain kinematic mechanism when the legs are in contact with the ground. Therefore, unlike the model used in this dissertation, the loads on the actuators are not independent. A conflict may occur if one leg attempts to move at a different velocity than another. This can be illustrated by examining Figure 1-1 and imagining that leg 2 is attempting to pull the
vehicle forward faster than leg 6 wants to permit. This will increase the pressure drops across the bypass valves, increasing the flow through the bypass valves and decreasing the hydraulic power to the actuators, thereby averting a potentially crippling situation. In contrast to the series-bypass circuit, the current ASV hydraulic circuit would require a monitor to detect the pressure increase and implement corrective action.

An advantage of the hydrostatic circuit in Figure 6-1 is that it has the potential to be more energy efficient than the series-bypass circuit. Although not studied in this dissertation, it is reasonable to expect that the current ASV design would be superior to the series-bypass circuit in permitting independent leg motions. It was shown in this study that the series-bypass circuit required compensation to achieve unequal desired velocity inputs. However, to obtain these advantages, the current hydrostatic ASV design, requires a variable displacement pump, a servovalve controlled rotary actuator and a pressure controlled hydraulic power supply to the servovalve for each main drive actuator. Comparing the hardware necessary to implement the hydrostatic and series-bypass circuits (see Figure 1-6), it can be said that the series-bypass circuit requires less equipment and is a less complex
The primary impetus for the addition of the bypass path was to facilitate independent leg motions. A large part of this dissertation was concerned with decoupling the actuators of the series-bypass circuit to achieve these independent leg motions. Three approaches were taken:

1. Decreasing the velocity errors caused by the actions of the adjacent actuator by integrating these errors in the feedback loop (integral controller).

2. Decoupling by using an algorithm obtained from the series-bypass model to correct the calculated static velocity errors caused by unequal desired velocity inputs (model matching algorithm).

3. Designing a controller that would modify the command signal to an actuator to compensate for the disturbances caused by the adjacent actuator before or as they occur (INA).

Based on the simulation studies and prototype data,
it can be said that each of these techniques has some merit. The integral controller and model matching algorithm can be used at a variety of operating conditions. However, the integral gains are selected by trial and error techniques, either through simulation studies or adjusting a potentiometer input to the prototype circuit during operation. The INA technique uses a system model to predict quantitative values of the coupling between individual open loop circuits. Based on this model, a precompensation matrix modifies the input to a circuit in an attempt to cancel the coupling disturbances from adjacent circuits. The INA method offers a graphical method and a relatively easier set of guidelines to select controller gains and additional compensators, such as lead filters. In addition, as opposed to a steady state compensation technique such as the model matching algorithm, this technique attempts to decouple the actuators across the frequency range of interest. However, as shown in this dissertation, it is not always possible to meet the requirements of the INA technique at different operating conditions. Therefore, this method is not viewed to be as universal as either the integral controller or model matching algorithm. Also, the INA design attempts to decouple the closed velocity loops. It is not
specifically designed to compensate for unequal actuator loads. A combination of integral, model matching and precompensation may be necessary to effectively decouple the actuators of the series-bypass circuit for the different modes and operating conditions of the vehicle. When this is accomplished, the series-bypass circuit should be viewed as an alternative to the independent leg motion offered by the current ASV hydrostatic circuit.

Two methods were used to lead compensate the actuators of the series-bypass circuit. The method in Chapter 3 ignored the coupling in the system and used the open loop frequency response curves to select lead filter time constants for the commanded bypass valve areas and commanded variable displacement pump flow rate. In Chapter 4, Rosenbrock's INA technique was used to lead compensate the commanded bypass valve areas by using the inverse open loop frequency response curves. However, prior to using the INA technique, open loop diagonal dominance had to be achieved. Because a precompensation matrix was used to obtain diagonal dominance, the open loop frequency response curves used in the INA method were different from those of Chapter 3. The precompensation matrix was calculated to decrease the open loop coupling of the system so a compensation
design based on a dominant open loop system could proceed. Recall that the author was unable to obtain diagonal dominance at certain operating conditions. Also, Rosenbrock's INA technique did not offer a method to directly lead compensate the variable displacement pump. The methods used in Chapters 3 and 4 were "tuned" to provide an optimum time response by using the simulation. The commanded bypass valve area lead filter time constants selected from both methods were similar, suggesting that the coupling in the circuit did not influence the lead filter design.

6.2 Recommendations For Further Study

To gain additional knowledge into the behavior of the series-bypass circuit the following investigations should be conducted.

1. Algorithms should be proposed to control the position of the actuators during the Precision Footing mode. Due to the nature of the series-bypass circuit it is difficult to envision a closed position loop for each actuator which did not utilize the proportional valves as continuous controllers and not merely as solenoid switching valves.
which control actuator direction at the end of the stride. It is believed that the modeling, control design techniques and multivariable methods presented in this dissertation offer a good guide to follow in this application.

2. The pressure drops across the proportional valves should be included in the linearized model. This would expand the applicability of the model and hopefully resolve the questions concerning the dependence of the gain space on the equilibrium actuator velocity. This feature could also be used to determine whether control of the proportional valves could increase the decoupling of the series-bypass circuit. Qualitatively speaking, it is felt that the difficulty in decoupling the series-bypass circuit is the consequence of connecting two identical hydraulic systems in series and desiring the same performance of both systems. As one would expect, it was shown that the diagonal and off diagonal open loop frequency response curves were similar, indicating the
difficulty in decoupling symmetrical systems in series. It is the authors belief that in a practical sense, a difference must exist either in the systems or the desired performances of the systems to effectively achieve a degree of decoupling. Consequently, it is felt that including the proportional valves in the velocity or position control would not enhance the possibilities for closed loop decoupling. In addition, the linearized analysis showed that the open loop system became increasingly decoupled as the bypass valve areas increased. As the bypass valve areas increase they permit more flow through the bypass path and the series-bypass circuit appears less like a series circuit. Including the proportional valves in the velocity control or as position controllers would not alter the bypass flow and therefore not offer a means to increase the decoupling in the circuit.

3. In Chapter 3, the integral compensators, equations (3.22) and (3.23), and the model matching algorithm to correct for unequal
desired velocities, equation (3.27), were used to compensate for velocity errors caused by unequal desired velocities and actuator loads. It would be possible to add these terms to the proportional control algorithms, equations (2.44) and (2.45) and substitute the resulting expressions into both the pump control algorithm (2.48), and the equations modeling the series-bypass circuit, equations (2.20) through (2.23). It should be possible to linearize this set of equations and arrange them in the form of equation (2.71). Numerical frequency methods could then be used to obtain the linearized frequency response curves. It may be possible to identify equilibrium conditions or apply a pseudo-diagonalization technique to obtain a diagonally dominant \( \hat{Q} \) matrix for this new controller design. Since the integral and unequal desired velocity algorithms were formulated to correct for velocity errors caused by the adjacent actuator, it may be possible to obtain greater decoupling than was achieved by only using the proportional control algorithms in this dissertation.
However, as discussed previously, in a strict sense, the integral and model matching controllers do not decouple the actuators. Also, it must be kept in mind that these new controller equations would have to be linearized about a new equilibrium condition, specifically, the difference between an actuators desired velocity and the maximum desired velocity, \( \dot{X}_M \) (see equation (3.27)). This would lead to a seemingly infinite number of equilibrium conditions. A more feasible approach may be to simply combine the integral controllers with the appropriate proportional control algorithms, neglecting the unequal desired velocity algorithm. If a diagonally dominant \( \hat{H} \) matrix can be obtained for either of these new control algorithms, lead filters could be designed from the INA plots as in section 4.6.

4. The actuators' velocity responses in both the simulation and experimental studies exhibited effects due to the sampling interval of the digital controller. Although it would involve a great amount of effort, it may be
possible to obtain the discrete representation of the linearized expressions in equation (2.71) and thereby include the effects of the digital controller in the model. This discrete model could be used to select sampling intervals or predict their effect on system behavior.

5. The effects of valve and pump saturation on the system response should be investigated. In this study, the saturation was included in the nonlinear simulation but not modeled in the linearized expressions. It was shown that precompensation coefficients selected from the linearized model would decouple the simulated prototype system when a valve was completely closed for a short time during the velocity response to a step input. Saturation could be modeled in the linearized expressions by using describing functions \([14]\). It would be interesting to analyze the effects that saturation has on selecting coefficients to pseudo-diagonalize the \( \hat{\mathbf{Q}} \) matrix. Different precompensation matrices may have to be selected by the
digital controller on the basis of whether the valves are saturated at a particular point in the stride.

6. Experiments conducted with the prototype system were restricted to a narrow range of operating conditions because of equipment limitations. If the prototype system is to be used for further experimentation, it would be necessary to find a better method to control the load on the actuators. For ASV implications, the greatest difficulty was the inability of the four-port servovalve to serve as a variable orifice bypass valve. It is felt that a two-port single stage servovalve should be used as the bypass valve. The internal flow paths of the two-port valve eliminates the unequal flow forces on the spool and the single stage eliminates the requirement for the separate pilot flow. However the single stage requires more electrical power, a disadvantage in the ASV application.

7. The velocity responses of the actuators
displayed high frequency oscillations which were not predicted by the simulation or linearized frequency response analysis. On the basis of empirical and simulation studies it is felt that these oscillations were not caused by the discrete controller or system damping. Two other possibilities were suggested in Chapter 5 but it was not possible to positively determine the source. The two possibilities were inadequately modelled bypass valve performance or fluid line dynamics. The bypass valves should be tested under controlled conditions to determine the frequency response characteristics of the transfer function between the servovalve input current and actual orifice area. Doebelin [13] presents a fluid element model which could be used in the simulation or linearized analysis to include the effects of fluid line dynamics on the system response. Doebelin recommends that a fluid line should be divided into lumps whose lengths are defined according to

\[ f_{\text{l}} = \frac{\sqrt{\frac{\beta}{\rho}}}{10f_{\text{max}}} \]  
(6.1)
where

\[ f_l \] = the recommended lump length

\[ f_{\text{max}} \] = maximum frequency of interest in the system

\[ \beta \] = bulk modulus

\[ \rho \] = fluid density

The frequency response curves in Chapter 2 showed that the frequency response of the series-bypass circuit became attenuated past 225 rad/sec. Using the appropriate units, the lump length can be calculated to be 16 inches. Comparing this value to Table 2-1, it is apparent that the fluid line should certainly be divided into elements. Also, it is not surprising that no change was observed in the experimental response when the length A-B was shortened from 110 to 66 inches. This change would have a small effect on the highest natural frequency of the system. According to equation (2.89), the natural frequencies of the series-bypass circuit would increase as the fluid volumes decrease. Consequently, for an accurate model, length
A-B would still be have to be divided into lumps.
Appendix A.

Prototype System Components

Figure A-1: Hydraulic Schematic of Load Circuit

The load circuit is illustrated in Figure A-1. The electronically actuated pressure relief valve controls the relief pressure of the manual pressure relief valve.
The flow displaced from the load actuator during the forward stroke of the prototype actuator and the flow from the variable displacement pump is bled to the reservoir through the manual valve at the relief pressure determined by the load command to the electronic valve. Although a calibration curve for the electronic valve was supplied by the manufacturer, the electronic and manual pressure relief valve combination had to be empirically calibrated because of the addition of the manual valve in the circuit and the dynamic nature of the load commands. The manual valve was necessary because of the flow rate limitations of the electrically actuated valve (maximum flow rate 1.5 GPM). The variable displacement pump supplies flow to the load actuator during the return stroke of the prototype actuator.

The primary hardware used in the series-bypass circuit is listed below.

Equipment: Electronically Actuated Variable Displacement Pump
Purpose: Supply hydraulic power to series-bypass circuit
Pertinent Data: Hydura #PVQ32-LDFY-VVTR-07

Equipment: Bypass Valves
Purpose: Control Actuator Velocity
Pertinent Data: Four port servovalves, Olsen #S-5F,
Valve driver, Moog #121-103, modified to provide 80 ma per valve

Equipment: Proportional Valves
Purpose: Directional Control of Actuators
Pertinent Data: Rexroth #4WRZ10-E85-30/6A24, Valve driver, Rexroth VT3000

Equipment: Double Ended Linear Actuators
Purpose: Provide main drive ASV leg motion
Pertinent Data: Milwaukee Cylinder #1542-10-4-D

Equipment: Manually Operated Pressure Relief Valves
Purpose: Provide load control of prototype actuators
Pertinent Data: Abex Denison #R4V-03-S13

Equipment: Electronically Operated Pressure Relief Valves
Purpose: Control set relief pressure of manually operated relief valves
Pertinent Data: Abex Denison #SE03-21042, Valve driver, Abex Denison #S17-23181

Equipment: Single Ended Linear Actuators
Purpose: Provide load to prototype actuators
Pertinent Data: Milwaukee Cylinder #1520-10-4

Equipment: Differential Pressure Transducers
Purpose: Measure pressure
Pertinent Data: Variable reluctance type, Standard Controls, #1200-38-5
Equipment: Load Cells
Purpose: Measure load
Pertinent Data: Strain gage type, Sensotec #51/1073

Equipment: Series-Bypass Controller and Data Acquisition/Load Controller
Purpose: Control motion of series-bypass circuit, load on series-bypass circuit and data acquisition
Pertinent Data: Intel 86/12A SBC, with 8087 numeric coprocessor

Equipment: A/D and D/A Converters
Purpose: Provide data I/O to SBC boards
Pertinent Data: Data Translation #711 and 732

Equipment: RAM Memory Board
Purpose: Data acquisition data storage, 64K bytes
Pertinent Data: Intel SBC 064
Appendix B.

Source Listings

B.1 Prototype Simulation

INITIAL
CONSTANT INTCON=0., SKIPIN=0., AV20DPN=0., AV40DPN=0., XPEG=3.0
CONSTANT PRTVE1=1.0, PRTVE2=1.0, TAUS2=.042, TAUP2=.042
CONSTANT UNTAU1=.0075, UNTAU2=.01
CONSTANT STARTT=1., ENDTIM=1., VELBUG=9., MINBUG=.05
CONSTANT DELTA=0.
CONSTANT OPEN1=1., OPEN2=1.
CONSTANT CONDI1=1.499, CONDI2=1., 501
CONSTANT SINFDR=2870., SGNFR1=-1., SGNFR2=1.
CONSTANT WVEL1=1.25, WDVEL2=1.25, ACCPUM=1.
CONSTANT STARTR=1.5, ENDR=4.5, INTGRI=0003, INTGR2=0003
CONSTANT FRTPCM=1., TAUSYS=.0044, PRTANA=0
CONSTANT KA=6. E-6, KA2=6. E-6, A=1., TAU=.0044, ALPHA=1., TAUPCM=.01
CONSTANT AMPSER=1.687, TAUNUM=2.9, TAUNMM=568, PHORI1=400.
CONSTANT PHORI2=400.
CONSTANT K11=761., K12=.234, K21=-.45, K22=.45
CONSTANT RH=7.8E-04, BE1=1., E05=2.5E-04, K=97.68, C=97.68
CONSTANT AV1MAX=1., ZETAP=1., WNP=100., TAUPRO=.03, TAUSER=0064
CONSTANT VT=32.8, VP=59.16, VS=26.5, VL=57.0, AP=4.1
CONSTANT PR=120.
CONSTANT CONTR1=.001, CONTR2=.0005
CONSTANT XV1MAX=8.0, XV1MIN=0., XV2MAX=8., XV2MIN=0.
XP1=EX=XV1MAX=.25
XP2=EX=XV2MAX=.25
CONSTANT B=10., M=1., AVMAX=0., AV4MAX=.04
CONSTANT FRCOPN=.5
CONSTANT WNS=345., ZETAS=1.0
INTEGER DIREC1, DIREC2, NLL, UNIT1, UNIT2, UNIT3, UNIT4, UNIT5, UNIT6
LOGICAL PRTDV1, PRTDV2
MAXTERVAL MAXT=.0031
ALGORITHMIALG=5
NSTEPS NSTP=1
CONSTANT TSTOP=2.
PRTDV1=.FALSE.
PRTDV2=.FALSE.
CONSTANT TAU=.0029, TAUS=.0044, TAUP=01, TAUP1=.0029
CONSTANT ONOFFS1=10., ONOFFP1=10., ONOFFS=10., ONOFFP=10.
" INSERT INITIAL CONDITIONS HERE "
UNIT2=UNIT1+1
UNIT3=UNIT1+2
UNIT4=UNIT1+3
UNIT5=UNIT1+4
UNIT6=UNIT1+5

387
UNIT7=UNIT1+6

P=1./((WNP**2))
Q=2.*ETAP/WNP
PSS=1./((WNS**2))
GSS=2.*ETAS/WNS
FOROUT=PHOR11*AP
CONSTANT XP1INT=1.0, XP2INT=1.0
AV1INT=AV1MAX
AV3INT=AV1MAX
READ(1,200)AV2INT, AV4INT, XP1VEL, XP2VEL
200. FORMAT(1X,2(F6.4,2X)),2(F6.2,X))
READ(2,201)P1INT, P11INT, P12INT, PLINT, P21INT, P22INT, QPINT
201. FORMAT(1X,6(F8.3,2X),F8.4)
AV2INP=AV2INT
AV4INP=AV4INT
IF(FRTPCM EQ. 0.)GO TO 720
QPINT2= (-1./FRTPCM+1.)*QPINT
QPINT3= (-1./FRTPCM+1.)*QPINT
720. CONTINUE
AV2 EQ= FRCPN*AV2 MAX
AV4EQ=FRCPN*AV4MAX
AV2 EQ= FRCPN*AV2MAX*AMPSER
AV4EQ=FRCPN*AV4MAX*AMPSER
IF(ONDFS1.EQ.0.)GO TO 700
AV2 EQ=AV2EQ*(1.-ONDFS1)
AV4EQ=AV4EQ*(1.-ONDFS1)
700. CONTINUE
IF(ONDPF1.EQ.0.)GO TO 701
AV2 EQ=AV2EQ*(1.-ONDPF1)
AV4EQ=AV4EQ*(1.-ONDPF1)
701. CONTINUE
703. CONTINUE
AV2EQ L=AV2EQ 1
AV4EQ L=AV4EQ 1
END $ " OF INITIAL "

DYNAMIC

DERIVATIVE

" COMPUTE EFFICIENCY "
EFF= ((P1-PL)*XP1DOT*AP+(PL-PR)*AP*XP2DOT)/(GP*(P1-PR))*100.

" DETERMINE DIRECTION OF PISTON "

PROCEDURAL(DIREC1, DIREC2=XP1, XP2)
IF(T.LT.0.5)DIREC1=1
IF(T.LT.0.5)DIREC2=1
IF(XP1.LE.XV1MIN)DIREC1=1
IF(XP2.LE.XV2MIN)DIREC2=1
IF(XP1.GE.XV1MAX)DIREC1=0
IF(XP2.GE.XV2MAX)DIREC2=0
END

" INSERT DESIRED VELOCITY "

PROCEDURAL(IERR1, IERR2, DEVEL1, DEVEL2=DIREC1, DIREC2, XP1, XP2)
IF(XP1.GT.STARTR)PRTVD1=.TRUE.
IF(XP1.GT. ENDR. AND. WVEL1.EQ.0.)PRTVD1=.FALSE.
IF(WVEL1.EQ.0.)GO TO SKIP1
DEVEL1=RSW(PRTDV1, (XP1VEL*PRTVE1*Sin(WDVEL1*(T-.0125))), XP1VEL)
GO TO SKIP2

SKIP1. DEVEL1=RSW(PRTDV1, (XP1VEL*PRTVE1), XP1VEL)

SKIP2. CONTINUE
IF (XP2 GT STARR) PRTDV2=. TRUE.
IF (XP2 GT ENDR AND WDVEL2 EQ 0.) PRTDV2=. FALSE.
IF (WDVEL2 EQ 0.) GO TO SKIP3
DEVEL2=RSW(PRTDV2, (XP2VEL*PRTVE2*Sin(WDVEL2*(T-.0125))), XP2VEL)
GO TO SKIP4

SKIP3. DEVEL2=RSW(PRTDV2, (XP2VEL*PRTVE2), XP2VEL)

SKIP4. CONTINUE
IF (XP1 LT STARR) GO TO SKPIN1
IF (ERR1=INTEG((XP1DOT-DEVEL1)**2), 0, 0)
GO TO INTOIN1

SKPIN1. IERR1=0.0

INTON1. CONTINUE
IF (XP1 LT STARR) GO TO SKPIN2
IF (ERR2=INTEG((XP2DOT-DEVEL2)**2), 0, 0)
GO TO INTOIN2

SKPIN2. IERR2=0.0

INTON2. CONTINUE
IF (DIREC1 EQ 0) DEVEL1=-XP1VEL
IF (DIREC2 EQ 0) DEVEL2=-XP2VEL
END

" PROPORTIONAL VALVE CONTROL BELOW "

PROCEDURAL(AV1COM, AV3COM=DIREC1, DIREC2)

" FOR A PRACTICAL D-A DEVICE OUTPUT SHOULD BE LIMITED."
" OPTIMUM CONDITION WOULD BE FOR MAXIMUM VOLTAGE OUTPUT ...
TO CORRESPOND TO MAXIMUM VALVE SETTINGS."

AV1COM=AV1MAX
AV3COM=AV1MAX
IF (DIREC1 EQ 0) AV1COM=-AV1MAX
IF (DIREC2 EQ 0) AV3COM=-AV1MAX
END

" BYPASS VALVE CONTROL BELOW "

PROCEDURAL(AV2COM, AV4COM, AV2EQ, AV4EQ, AV2PER, AV4PER, XPMAX=P1, P2, ...
DEVEL1, DEVEL2, ...
XP1DOT, XP2DOT)

AV2PRT=-CONTR1*(ABS(DEVEL1)-(OPEN1*ABS(XP1DOT))
IF (XP1 GE STARR) AV2PRT=AV2PRT+AV2DPN
AV2ING=-INTOR1*INTEG((ABS(DEVEL1)-ABS(XP1DOT)), 0, 0)
AV2PRT=AV2PRT+AV2ING
AV4PRT=-CONTR2*(ABS(DEVEL2)-(OPEN2*ABS(XP2DOT))
IF (XP2 GE STARR) AV4PRT=AV4PRT+AV4DPN
AV4ING=-INTOR2*INTEG((ABS(DEVEL2)-ABS(XP2DOT)), 0, 0)
AV4PRT=AV4PRT+AV4ING
AV2PER=KP11*AV2PRT+KP12*AV4PRT

AV2COM=AV2EQ+AV2PER+KA1*XP1ACC
AV4PER=KP21*AV2PRT+KP22*AV4PRT

AV4COM=AV4EG+AV4PER+KA2*XP2ACC

IF (DIREC1.EQ.0 .AND. F1.LT.0.0) AV2COM=0.0
IF (DIREC2.EQ.0 .AND. F2.LT.0.0) AV4COM=0.0

" SEE ABOVE NOTE UNDER PROPORTIONAL VALVE CONTROL "
IF (CONDNO.1.EQ.0.0) GO TO 600

" F1=F2 "
" F7=F8 "

AV2SER=LEDLAG((ONOF1*TAUS1),TAUS1,(AV2COM-INTCON*AV2ING),AV2EG)
AV4SER=LEDLAG((ONOF1*TAUS2),TAUS2,(AV4COM-INTCON*AV4ING),AV4EG)
AV2SER=AV2SER+INTCON*AV2ING
AV4SER=AV4SER+INTCON*AV4ING
GO TO 602

600. CONTINUE
AV2SER=AV2COM
AV4SER=AV4COM

602. CONTINUE
IF (CONDNO.1.EQ.0.0) GO TO 601
" F3=F4 "
" F5=F6 "

IF (ACCPOM.EQ.1.0) GO TO 1688

AV2PUM=LEDLAG(TAUP1,(ONOF1*TAUP1),...)
(AV2COM-INTCON*AV2ING-KA1*XP1ACC),AV2EG)
AV4PUM=LEDLAG(TAUP1,(ONOF1*TAUP1),...)
(AV4COM-INTCON*AV4ING-KA2*XP2ACC),AV4EG)
GO TO 1689

1688. CONTINUE
AV2PUM=LEDLAG((ONOF1*TAUP1),TAUP1,(AV2COM-INTCON*AV2ING),AV2EG)
AV4PUM=LEDLAG((ONOF1*TAUP2),TAUP2,(AV4COM-INTCON*AV4ING),AV4EG)

1689. CONTINUE
AV2PUM=AV2PUM+INTCON*AV2ING
AV4PUM=AV4PUM+INTCON*AV4ING
GO TO 603

601. CONTINUE
IF (ACCPOM.EQ.1.0) GO TO 1690
AV2PUM=AV2COM-KA1*XP1ACC
AV4PUM=AV4COM-KA2*XP2ACC
GO TO 1691

1690. CONTINUE
AV2PUM=AV2COM
AV4PUM=AV4COM

1691. CONTINUE

603. CONTINUE
XPMAX=ABS(DEVEL1)
IF (ABS(DEVEL2).GT.ABS(DEVEL1)) XPMAX=ABS(DEVEL2)
XPIERR=XPMAX-DEVEL1
XP2ERR=XPMAX-DEVEL2
AV2DEL=XPIERR*AP/(K*(PI-PL)**.5)
AV4DEL=XP2ERR*AP/(K*(PL-PR)**.5)
AV2PUM=(AV2PUM)/AMPSER
AV4PUM=(AV4PUM)/AMPSER
AV2DLL=LEDLAG(0.,UNTAU1,AV2DEL,0.0)
AV4DLL=LEDLAG(0.,UNTAU2,AV4DEL,0.0)
AV2SER = (AV2SER) / AMPSER + AV2DLL * DELTAC
AV4SER = (AV4SER) / AMPSER + AV4DLL * DELTAC
IF (AV2SER GT AV2MAX) AV2SER = AV2MAX
IF (AV4SER GT AV2MAX) AV4SER = AV2MAX
IF (AV2SER LT 0.) AV2SER = 0.
IF (AV4SER LT 0.) AV4SER = 0.
IF (AV2PUM GT AV2MAX) AV2PUM = AV2MAX
IF (AV4PUM GT AV2MAX) AV4PUM = AV2MAX
IF (AV2PUM LT 0.) AV2PUM = 0.
IF (AV4PUM LT 0.) AV4PUM = 0.
"AV2PUM = AV2COM"
"AV4PUM = AV4COM"
"PUMP CONTROL ONLY"
"AV2COM = 0. 0... AV4COM = 0. 0"
"PUMP CONTROL ONLY"
END

"PROCEDURAL (GPCOM = AV2PUM, AV4PUM,...
DEVEL1, DEVEL2, P1, PL, AV2EQ, AV4EQ, AV2PER, AV4PER, XPMAX)"

GPCOM = XPEQ * AP + ((2. * FRCOPN * AV2MAX - AV2PUM) * C * ABS (P1 - PL) ** 5) / 2.
GPCOM = ((2. * FRCOPN * AV4MAX - AV4PUM) * C * ABS (PL - PR) ** 5) / 2. * GPCOM
IF (FRTPCM EQ 0.) GO TO 620
"--"
GPCOM = LEDLAG (TAUPCM, (TAUPCM * FRTPCM), GPCOM, QPINT3)
"GPCOM = LEDLAG (TAUPCM, (TAUPCM * FRTPCM), GPCOM, QPINT2)"
620. CONTINUE
"SEE ABOVE NOTE UNDER PROPORTIONAL VALVE CONTROL"
IF ((DIREC1 EQ 0. AND. F1 LT 0.0) OR (DIREC2 EQ 0. AND. F2 LT 0.0)) GPCOM = XPMAX * AP

"FOR PUMP CONTROL ONLY"
"GPCOM = XPMAX * AP"
"FOR PUMP CONTROL ONLY"

IF (GPCOM LT 0.) GPCOM = 0.
IF (GPCOM GT 115.) GPCOM = 115.
END

"SIMULATED FORCE BELOW"
PROCEDURAL(F1, F2=DIREC1, DIREC2, XP1, XP2)
IF(DIREC1. EQ. 0. . OR. XP1. LT. 0.0)GO TO 20
IF(XP1. LT. 25)F1=615. +(PHORI1*AP-615. )/.25*XP1
IF(XP1. GE. 25. AND. XP1. LE. XP1EXD)F1=PHORI1*AP
IF(XP1. GE. STARTR. AND. XP1. LE. 7.75)F1=F1+SIN((XP1-STARTR)/")
IF(XP1. GT. XP1EXD)F1=PHORI1*AP+(XP1-XP1EXD)*(-PHORI1*AP+615. )/ 25
GO TO 11
20. CONTINUE
F1=615.
11. CONTINUE
IF(DIREC2. EQ. 0. . OR. XP2. LT. 0.0)GO TO 12
IF(XP2. LT. 25)F2=615. +(PHORI2*AP-615. )/.25*XP2
IF(XP2. GE. 25. AND. XP2. LE. XP2EXD)F2=PHORI2*AP
IF(XP2. GE. STARTR. AND. XP2. LE. 7.75)F2=F2+SIN((XP2-STARTR)/")
IF(XP2. GT. XP2EXD)F2=PHORI2*AP+(XP2-XP2EXD)*(-PHORI2*AP+615. )/ 25
GO TO 13
12. CONTINUE
F2=615.
13. CONTINUE
END

PROCEDURAL(P1DOT, P11DOT, P12DOT, P13DOT, XP1ACC, P21DOT, P22DOT, ...
XP2ACC, DAV1, DAV3=P1, P11, P12, PL, P21, P22, XP1, XP2, XP1DOT, ...
XP2DOT, F1, F2, AV1, AV2, AV3, AV4, AV1COM, AV2SER, AV3COM, AV4SER, GP)

IF(AV1. LT. 0. . AND. AV3. GT. 0. )GO TO 6
IF(AV1. GT. 0. . AND. AV3. LT. 0. )GO TO 7
IF(AV1. LT. 0. . AND. AV3. LT. 0. )GO TO 8

" PROPORTIONAL VALVES ARE IN THE FORWARD DIRECTION "

GA1=AV1*K*(ABS(P1-P11))**.5
GA2=AV2*K*(ABS(P1-PL))**.5
IF(AV2. LT. 0.0)GA2=0.0
GA3=AV1*K*(ABS(P12-PL))**.5
GB1=AV3*K*(ABS(PL-P21))**.5
GB2=AV4*K*(ABS(PL-PR))**.5
IF(AV4. LT. 0.0)GB2=0.0
GB3=AV3*K*(ABS(P22-PR))**.5

IF(P11. GT. P1)GA1=-GA1
IF(PL. GT. P1)GA2=-GA2
IF(PL. GT. P12)GA3=-GA3
IF(P21. GT. PL)GB1=-GB1
IF(PL. GT. PL)GB2=-GB2
IF (PR.GT.P22) GB3=-QB3
GS=GA1+QA2
IF (P1.LT.3500.0) GO TO 100
GRELEF=QP-GS
CD TO 101
100. CONTINUE
GRELEF=0.0
101. CONTINUE

P1DOT=(QP-GS)*BE/VP
P1DOT=BE*((GA1-XP1DOT*AP)/(VS+XP1*AP))
P12DOT=BE*((-QA3+XP1DOT*AP)/(VS+VT-XP1*AP))
P1DOT=((GA2+QA3-QB2-QB1)/(VL/BE))
XP1ACC=(-B*XP1DOT+F1+(P11-P12)*AP)/M
P21DOT=BE*((QB1-XP2DOT*AP)/(VS+XP2*AP))
P22DOT=BE*((-QB3+XP2DOT*AP)/(VS+VT-XP2*AP))
XP2ACC=(-B*XP2DOT+F2+(P21-P22)*AP)/M

GO TO 10
6. CONTINUE

"FIRST PROPORTIONAL VALVE IS REVERSED,
SECOND IS IN FORWARD POSITION."

QA1=ABS(AV1)*K*(ABS(P11-PL))**.5
QA2=AV2*K*(ABS(P1-PL))**.5
IF (AV2.LT.0.0) QA2=0.0
QA3=ABS(AV1)*K*(ABS(P1-P12))**.5
QB1=AV3*K*(ABS(PL-P21))**.5
QB2=AV4*K*(ABS(PL-PR))**.5
IF (AV4.LT.0.0) QB2=0.0
QB3=AV3*K*(ABS(P22-PR))**.5

IF (P12.GT.P1) QA3=-QA3
IF (PL.GT.P1) QA2=-QA2
IF (PL.GT.P11) QA1=-QA1
IF (P21.GT.PL) GB1=QB1
IF (PR.GT.PL) GB2=QB2
IF (PR.GT.P22) GB3=QB3
GS=GA3+QA2
IF (P1.LT.3500.0) GO TO 103
GRELEF=QP-GS
GO TO 104
103. CONTINUE
GRELEF=0.0
104. CONTINUE

P1DOT=(QP-GS)/VP*BE
P1DOT=BE*(-QA1-XP1DOT*AP)/(VS+XP1*AP)
\[ p_{12dot} = B E \frac{(Q A_3 \times X P_{1dot} + A P)}{(V S + V T - X P_1 \times A P)} \]
\[ p_{Ldot} = \frac{(Q A_1 \times Q A_2 - Q B_3 - Q B_1)}{(V L / B E)} \]
\[ X P_{1 acc} = \frac{(-B \times X P_{1dot} + F_1 + (P_{11} - P_{12}) \times A P)}{M} \]
\[ p_{21dot} = B E \frac{(-Q B_1 - X P_{2dot} + A P)}{(V S + X P_2 \times A P)} \]
\[ p_{22dot} = B E \frac{(-Q B_3 + X P_{2dot} + A P)}{(V S + V T - X P_2 \times A P)} \]
\[ X P_{2 acc} = \frac{(-B \times X P_{2dot} + F_2 + (P_{21} - P_{22}) \times A P)}{M} \]

GO TO 10
CONTINUE

"FIRST PROPORTIONAL VALVE IS FORWARD, SECOND IS REVERSED"

\[ Q A_1 = A V_1 \times K \times (A B S (P_1 - P_{11})) \times 0.5 \]
\[ Q A_2 = A V_2 \times K \times (A B S (P_1 - P_L)) \times 0.5 \]
\[ Q A_3 = A V_1 \times K \times (A B S (P_{12} - P_L)) \times 0.5 \]
\[ Q B_1 = A B S (A V_3) \times K \times (A B S (P_{21} - P_{21})) \times 0.5 \]
\[ Q B_2 = A V_4 \times K \times (A B S (P_L - P_{21})) \times 0.5 \]
\[ Q B_3 = A B S (A V_3) \times K \times (A B S (P_L - P_{22})) \times 0.5 \]

IF (P_{11} \times P_1) Q A_1 = Q A_1
IF (P_L \times P_1) Q A_2 = -Q A_2
IF (P_L \times P_{12}) Q A_3 = -Q A_3
IF (P_{21} \times P_{21}) Q B_1 = -Q B_1
IF (P_{21} \times P_L) Q B_2 = -Q B_2
IF (P_{22} \times P_{21}) Q B_3 = -Q B_3
Q S = Q A_1 + Q A_2
IF (P_L \times 3500) 0 \times 105
Q REL = Q P - Q S
GO TO 106
CONTINUE
Q REL = 0.0
GO TO 106

CONTINUE

\[ p_{1 dot} = \frac{(Q P - Q S)}{V P \times B E} \]
\[ p_{12dot} = B E \frac{(Q A_1 - X P_{1dot} + A P)}{(V S + X P_1 \times A P)} \]
\[ p_{12dot} = B E \frac{(-Q A_3 \times X P_{1dot} + A P)}{(V S + V T - X P_1)} \]
\[ X P_{1 acc} = \frac{(-B \times X P_{1dot} + F_1 + (P_{11} - P_{12}) \times A P)}{M} \]
\[ p_{21dot} = B E \frac{(-Q B_1 - X P_{2dot} + A P)}{(V S + X P_2 \times A P)} \]
\[ p_{22dot} = B E \frac{(-Q B_3 + X P_{2dot} + A P)}{(V S + V T - X P_2)} \]
\[ X P_{2 acc} = \frac{(-B \times X P_{2dot} + F_2 + (P_{21} - P_{22}) \times A P)}{M} \]
"BOTH PROPORTIONAL VALVES ARE REVERSED"

\[ Q_{A1} = |A_{V1}| \times K \times (|P_{11} - P_L|)^{*0.5} \]
\[ Q_{A2} = A_{V2} \times K \times (|P_{11} - P_L|)^{*0.5} \]
\[ Q_{A3} = |A_{V3}| \times K \times (|P_{11} - P_{12}|)^{*0.5} \]
\[ Q_{B1} = |A_{V3}| \times K \times (|P_L - P_R|)^{*0.5} \]
\[ Q_{B2} = A_{V4} \times K \times (|P_L - P_R|)^{*0.5} \]
\[ Q_{B3} = |A_{V3}| \times K \times (|P_L - P_{22}|)^{*0.5} \]

IF (\( P_{11} \) LT. 0.0) \( Q_{A1} = 0.0 \)
IF (\( P_{12} \) LT. 0.0) \( Q_{A2} = 0.0 \)
IF (\( P_{12} \) GT. 0.0) \( Q_{A3} = 0.0 \)
IF (\( P_{21} \) LT. 0.0) \( Q_{B1} = 0.0 \)
IF (\( P_{22} \) LT. 0.0) \( Q_{B2} = 0.0 \)
IF (\( P_{22} \) GT. 0.0) \( Q_{B3} = 0.0 \)

\[ QS = Q_{A3} + Q_{A2} \]
IF (\( P_{11} \) LT. 3500.0) GO TO 107
GO TO 109

107. CONTINUE
GO TO 107
GRELEF = 0.0
108. CONTINUE

\[ P_{1D0T} = (Q_P - Q_S) / V_P * B_E \]
\[ P_{11D0T} = B_E * (P_{GA1} - X_P_{1D0T} * A_P) / (V_S + X_P_{1D0T} * A_P) \]
\[ P_{12D0T} = B_E * (P_{GA3} + X_P_{1D0T} * A_P) / (V_S + V_T - X_P_{1D0T} * A_P) \]
\[ P_{L0D0T} = (P_{GA1} - X_B - X_B - P_{GB3}) / (V_L / B_E) \]
\[ X_P_{1ACC} = (-B * X_P_{1D0T} + F_1 + (P_{11} - P_{12}) * A_P) / M \]
\[ P_{21D0T} = B_E * (-P_{GB1} - X_P_{2D0T} * A_P) / (V_S + X_P_{2D0T} * A_P) \]
\[ P_{22D0T} = B_E * (P_{GB3} + X_P_{2D0T} * A_P) / (V_S + V_T - X_P_{2D0T} * A_P) \]
\[ X_P_{2ACC} = (-B * X_P_{2D0T} + F_2 + (P_{21} - P_{22}) * A_P) / M \]

10. CONTINUE
INTEGRAL SECTION OF MODEL

DAV1 = 1./TAUPRO*(AV1COM-AV1)
DAV3 = 1./TAUPRO*(AV3COM-AV3)
END " OF PROCEDURAL"
P1 = LIMIT(P1DOT, P1INT, PR, 3500.0)
P11 = INTEG(P1DOT, P1INT)
P12 = INTEG(P12DOT, P12INT)
PL = INTEG(PLDOT, PLINT)
P21 = LIMIT(P21DOT, P21INT, PR, 3500.0)
P22 = LIMIT(P22DOT, P22INT, PR, 3500.0)
XP1DOT = INTEG(XP1ACC, XP1VEL)
XP2DOT = INTEG(XP2ACC, XP2VEL)
XP1 = INTEG(XP1DOT, XP1INT)
XP2 = INTEG(XP2DOT, XP2INT)
"AV2CP=LEDLAG(ITTOP, TBOT, AV2COM, AV2INP)"
"AV4CP=LEDLAG(ITTOP, TBOT, AV4COM, AV4INP)"

"THE LIMIT INTEGRATION ON THE AREAS IS NOT REALLY NECESSARY...
FOR A FIRST ORDER SYSTEM, THE INPUT IS LIMITED AND THERE CANNOT...
BE ANY OVERSHOOT."

AV1 = LIMIT(DAV1, AV1INT, -AV1MAX, AV1MAX)
AV2 = CMPXPL(PSS, GSS, AV2SER, 0.0, AV2INT)
AV3 = LIMIT(DAV3, AV3INT, -AV3MAX, AV3MAX)
AV4 = CMPXPL(PSS, GSS, AV4SER, 0.0, AV4INT)

"MODEL FOR PUMP IS CRITICALLY DAMPED, IF UNDERDAMPED MODEL...
IS INSERTED BE SURE TO LIMIT OUTPUT IF SATURATION OCCURS."

QP = CMPXPL(P, 0, QPCOM, 0.0, QPINT)

IF(T_GE.STARTT. AND. T_LT.ENDTIM)CALL DEBUG
IF(DEVEL1.EQ.VELOBUG. AND. (ABS(DEVEL1-XP1DOT)).LT.MINBUG)CALL DEBUG
IF(XP1.GT.CONDI1. AND. XP1.LT.CONDI2)CALL DEBUG
END " OF DERIVATIVE"
CALL PAGE(-1, NULL)
CALL LINES(1)
CALL PAGE(-1, NULL)
CALL LINES(1)
WRITE(UNIT1, 300), T, P1, P11, P12, PL, P21, P22
  300. FORMAT(’ ’, F6.4, F6.4, F8.1, 2X)
CALL PAGE(-1, NULL)
CALL LINES(1)
WRITE(UNIT2, 301), DEVEL1, XP1DOT, DEVEL2, XP2DOT, XP1, XP2, XP1ACC, XP2ACC
  301. FORMAT(’ ’, F6.4, 2X, 6(F8.1, 2X))
CALL PAGE(-1, NULL)
CALL LINES(1)
WRITE(UNIT3, 302), IERR1, AV2SER, AV2, IERR2, AV4SER, AV4
  302. FORMAT(’ ’, F8.6, 2X, 2(F8.6, 1X), F8.6, 1X, 2(F8.6, 1X))
CALL PAGE(-1, NULL)
CALL LINES(1)
WRITE(UNIT4, 303), QPCOM, GP, GRELIF, GA1, GA2, GA3, EFF
  303. FORMAT(’ ’, 7(F7.2, 2X))
CALL PAGE(-1, NULL)
CALL LINES(1)
WRITE(UNIT5,304)Q1, Q2, Q3, F1, F2, AV2DEL, AV4DEL, AV2PUM, AV4PUM
304. FORMAT(' ', 3(F7.2, 2X), 2(F8.1, 2X), 4(F8.5, 1X))
TERM(T.GE.TSTOP)
END "OF DYNAMIC"
TERMINAL
WRITE(UNIT6,305)CONTR1, CONTR2, KA1, KA2, WNP, ZETAP, WNS, ZETAS
305. FORMAT(' ', 4(F9.7, 1X), 1X, F5.1, 1X, F5.3, 1X, F5.1, 1X, F5.3)
WRITE(UNIT7,306)KP11, KP12, KP21, KP22, INTG1, INTG2, AV2ED, AV4ED, FOROUT
306. FORMAT(' ', 4(F7.3, 1X), 2(F9.7, 1X), 2(F5.3, 1X), F7.1)
END "OF TERMINAL"
END
B.2 HYDSIM

IMPLICIT REAL K
REAL KC1,KC2,KC3,KC4,KQ1,KQ2,Kc21,Kc22,KQ2,ICa3,Ka4,Kc4,KQ4,KC31,KC32,rC33
DIMENSION X(15),F(15)

THIS PROGRAM SOLVES THE SIMULTANEOUS NON-LINEAR EQUATIONS
DESCRIBING THE FLOW AND PRESSURE RELATIONSHIPS FOR THE
MAIN DRIVE HYDRAULIC CIRCUIT OF THE A.S.V. THE PROGRAM
USES THE NEHTOH-RAPHSON TECHNIQUE TO SOLVE THE EQUATIONS
DESCRIBED IN THE SUBROUTINE FCN. THE EQUATION SOLVING ROUTINE
IS CALLED NLSYS AND IT CALLS UPON A LINEAR EQUATION SOLVER
CALLED SOLEQ.

EXTERNAL FCN,FCN3,FCSHP
COMMON/ARR/A(15,15),LPC(15),LPR(15),LTEMP(15),XSAVE(15),FSAVE(15)
DOUBLE PRECISION A,FSAVE
COMMON/PARAM/QP, n, F2, XPl, XP2, AV, CIP, RHO, VT, AP, CD, B, AV2, AV4, PR,
CCONTRL, AVMAX ,ŒVELI,  IEVEL2, XPMAX.  FRCOPN, ICNTRL,  EFCNCY
CD=.61
WRITE(5,1018)
1018 FORMAT( '  IF AN OUTPUT FILE IS DESIRED, ENTER A DIGIT',/',
1 SPECIFYING THE FILE; ELSE ENTER 5')
READ(5,*)I0
I0DF=I0+1
I0DC=I0+2
WRITE(5,4000)
4000 FORMAT( '  REXROTH 22.4 GPM VALVE USED')
WRITE(IO,4000)
AV=22.5/SQRT(75.52)*3.944E-2
WRITE(IO,4001)AV
4001 FORMAT( '  AV= ',F7.4)
1081 CONTINUE
WRITE(5,1082)
WRITE(IO,1082)
1082 FORMAT( '  ENTER THE DAMPING COEFFICIENT, B')
READ(5,*)B
WRITE(5,1084)
WRITE(IO,1084)
1084 FORMAT( '  ENTER THE FORCING FUNCTIONS FOR CYLINDERS 1 AND 2')
READ(5,*)F1,F2
WRITE(5,9000)
9000 FORMAT( '  ENTER 1 FOR CSPM STEADY STATE VALUES')
READ(5,*)ICSMP
IF(ICSMP.EQ.1)GO TO 9001
WRITE(5,1085)
1085 FORMAT( '  ENTER THE PUMP FLOW (GPM) ')
WRITE(IO,1085)
READ(5,*)QP
WRITE(IO,*)QP
HRITE(5,1086)
1086 FORMAT( ' ENTER POSITIONS OF CYLINDERS 1 AND 2')

WRITE(5,1086)
READ(5,*)XP1,XP2
WRITE(5,4005)
WRITE(IO,4005)

4005 FORMAT( ' DEFAULT VALUES USED ARE .75 INCH DIA LINE THROUGH,'/
1 ' 5 FEET BETWEEN PUMP AND SERVO MANIFOLD, 4 FEET BETWEEN SERVO,'/
1 ' 6 MANIFOLD AND ACTUATOR AND 8 FEET BETWEEN MANIFOLDS. ALSO PR',/
1 ' 120.')
VP=5.
DIAP=.75
SL=4.
DIAS=.75
SSL=8.
PR=120.
GO TO 4002
WRITE(5,2020)
WRITE(IO,2020)

2020 FORMAT( ' ENTER LINE LENGTH AND DIAMETER BETWEEN PUMP',/
2 ' AND SERVO MANIFOLD. LINE LENGTH IN FEET, DIAMETER',/
2 ' IN INCHES')
READ(5,*)VP,DIAP
WRITE(IO,*)VP,DIAP
WRITE(5,5000)

5000 FORMAT( ' ENTER LINE LENGTH AND DIAMETER BETWEEN SERVO',/
3 ' MANIFOLD AND ACTUATOR. LINE LENGTH IN FEET, DIAMETER',/
3 ' IN INCHES')
READ(5,*) SL,DIAS
WRITE(5,5001)

5001 FORMAT( ' ENTER LINE LENGTH AND DIAMETER BETWEEN SERVO',/
3 ' MANIFOLDS. LINE LENGTH IN FEET, DIAMETER IN INCHES.')
READ(5,*) SSL,DIASL
GO TO 4002

V11=VP*12*3.14/4*DIAP**2+SL*12*3.14/4*DIAS**2+XP1*AP
V22=VT-XPl*AP+SL*3*DIAS**2*3.14+.5*331*3*DIASL**2*3.14
V33=xp2*AP+.5*331*3*DIASL**2*3.14)+3*3*DIAS**3*3.14
V44=VT-xp2*AP+SL*3*DIAS**2*3.14
C C WRITE(5,1087)
C 1087 FORMAT( ' ENTER RESERVOIR PRESSURE')
C READ5,*)PR
C WRITE(IO,*)PR
C WRITE(5,1090)
C WRITE(IO,1090)

1090 FORMAT( ' ENTER 1 FOR SPECIFICATION OF ACTUATOR VELOCITY',/
3 ' OR ENTER 2 FOR BYPASS AREA SPECIFICATION')
READ(5,*) ISPEC
WRITE(IO,*)ISPEC
IP(ISPEC,EQ.2)GO TO 1091
C C C C C
C THE FOLLOWING CALCULATE COEFFICIENTS WHEN ACTUATOR VELOCITY
C IS SPECIFIED.
WRITE(5,2010)
WRITE(10,2010)
2010 FORMAT(' ENTER ACTUATOR VELOCITIES (IN/SEC)')
READ(5,*)XPIDOT,XP2DOT
WRITE(10,*)XPIDOT,XP2DOT
WRITE (5,6000)
WRITE(IO,6000)
6000 FORMAT(' ENTER 1 FOR SIMPLIFIED COEFFICIENTS OR 2 FOR C COMPLICATED MODEL')
READ(5,*)ICOMPL
WRITE(IO,*)ICOMPL
IF(ICOMPL.EQ.1)GO TO 6001
THE FOLLOWING COMPUTES COEFFICIENTS FOR HIGHER ORDER MODEL

FACTOR=CD/ AP*(2./RHO)**.5
P22=PR+(XP2DOT/ (AV*FACTOR))**2
P21=-(F2-B*XP2DOT)/AP+P22
PL=P21+(XPIDOT/(AV*FACTOR))**2
P12=PL+(XPIDOT/(  AV*FACTOR))**2
Pll=P12+-(F1-B*XP1DOT)/AP
P=PL+-(F1-B*XP1DOT)/AP
AV2=(QP-CD*AV*(2.*(P1-P11)/RHO)**.5)/(CD*(2*(P1-PL)/RHO)**.5)
AV4=FACTOR*AV* (PL-PR)**.5
WRITE(5,2001)FACTOR,P22,P21,PL,P12,Pli,PI,AV2, AV4,XP1, XP2
WRITE(10,2001)FACTOR,P22,P21,PL,P12,P11,PI,AV2,AV4,XP1,XP2
2001 FORMAT( '  FACTOR= ',F10.2,/,' P22= ',F10.5,/,' P21= '.F10.S,/,' P
CL= '.F10.5,/,' P12= ', F10.5,/,' P11=', F10.5,/,' P1=', F10.5,/,' P
AV2= ',F10.5,/,' AV4= ',F10.5,/,' XP1= ',F5.2,/,' XP2= ',F5.2)
FACTOR=FACTOR/2.*AP
KC11=FACTOR+AV*(P1-P11)**-.5
KC22=KC11
KC21=FACTOR+AV*(P1-P1)**-.5
KC22=KC21
KC31=FACTOR+AV*(P1-P2)**-.5
KC32=KC21
KC33=FACTOR+AV*(P1-P2)**-.5
KC34=KC33
KC41=FACTOR+AV*(P1-P3)**-.5
KC42=KC41
KC43=FACTOR+AV*(P1-P3)**-.5
KC44=KC43
WRITE(5,1088)KC11,KC12,KC13,KC14,KQ2,KC21,KC22,KC23,KC24,KC31,KC32,KC33,KC34,KC41,KC42,KC43,KC44
WRITE(10,1088)KC11,KC12,KC13,KC14,KQ2,KC21,KC22,KC23,KC24,KC31,KC32,KC33,KC34,KC41,KC42,KC43,KC44
1088 FORMAT('  KC11= ',F10.5,/,' KC12= ',F10.5,/,' KC13= ',F10.5,/,' K
KC14= ',F10.5,/,' KQ2= ',F10.5,/,' KC21= ',F10.5,/,' KC22= ',F10.5,/ C',
KC4= ',F10.5,/,' KQ4= ',F10.5,/,' KC31= ',F10.5,/,' KC32= ',F10.5,/ C',
KC33= ',F10.5,/,' V11= ',F10.5,/,' V22= ',F10.5,/,' C
V33= ',F10.5,/,' V44= ',F10.5)
THE FOLLOWING COMPUTES COEFFICIENTS FOR THE LOW ORDER MODEL

\[ V_5 = V_1 \]
\[ V_6 = V_{22} + V_{33} \]
\[ P_L = \left( F_2 - B \cdot P_{IDOT} / A_P H \right) \]
\[ P_1 = P_L - 0.5 \cdot (P_{IDOT} / A_P H) \]
\[ A_{V2} = \left( Q_P - X_{P1DOT} / A_P H \right) \]
\[ A_{V4} = \left( Q_P - X_{P2DOT} / A_P H \right) \]
\[ K_{C21} = \text{FACTOR} \cdot A_{V2} \cdot \sqrt{P_1 - P_L} \]
\[ K_{Q2} = \text{FACTOR} \cdot (P_1 - P_L) \]
\[ K_{C4} = \text{FACTOR} \cdot A_{V4} \cdot \sqrt{P_L - P_R} \]
\[ K_{Q4} = \text{FACTOR} \cdot (P_L - P_R) \]

WRITE (5, 6002)
WRITE (10, 6002)

6002 FORMAT ("THE SIMPLIFIED COEFFICIENTS ARE")
WRITE (5, 6003) V5, V6, PL, P1, AV2, AV4, KC21, KQ2, KC4, KQ4, XP1, XP2,
XP1DOT, XP2DOT
6003 FORMAT (V5 = ', F10.5, /, V6 = ', F10.5, /, PL = ', F10.5, /
C: P1 = ', F10.5, /, AV2 = ', F10.5, /, AV4 = ', F10.5, /
C: KC21 = ', F10.5, /, KQ2 = ', F10.5, /, KC4 = ', F10.5, /
C: KQ4 = ', F10.5, /, XP1 = ', F5.2, /, XP2 = ', F5.2, /
C: XP1DOT = ', F6.2, /, XP2DOT = ', F6.2)
WRITE (10, 6003) V5, V6, PL, P1, AV2, AV4, KC21, KQ2, KC4, KQ4, XP1, XP2,
XP1DOT, XP2DOT
GO TO 2000

1091 CONTINUE
WRITE (5, 1091)
WRITE (10, 1091)

2011 FORMAT ("ENTER AV2 AND AV4 IN INCHES")
READ (5, *) AV2, AV4
WRITE (10, *) AV2, AV4
WRITE (5, 1092)
WRITE (10, 1092)

1092 FORMAT ("ENTER 1 FOR SIMPLE MODEL OR 2 FOR HIGHER ORDER")
READ (5, *) ISIMP
WRITE (10, *) ISIMP
IF (ISIMP.EQ.1) GO TO 2030

THE FOLLOWING COMPUTES COEFFICIENTS FOR THE HIGHER ORDER MODEL WHEN BYPASS VALVE POSITION SPECIFIED.

CALL INIT3 (FCN3, X, P)
WRITE (5, 1093) (I(I), I=1, 8), XP1, XP2
WRITE (10, 1093) (I(I), I=1, 8), XP1, XP2
1093 FORMAT (P1 = ', F10.5, /, P11 = ', F10.5, /, P12 = ', F10.5, /
C: P1 = ', F10.5, /, XP1DOT = ', F10.5, /, P11 = ', F10.5, /
C: P22 = ', F10.5, /, XP2DOT = ', F10.5, /, XP1 = ', F5.2, /
C: XP2 = ', F5.2, /)
FACTOR = (CD \cdot (2./RHO) \cdot 0.5) / 2
P1 = X(1)
P11 = X(2)
P12 = X(3)
THE FOLLOWING COMPUTES COEFFICIENTS FOR THE SIMPLIFIED MODEL
WHEN THE BYPASS VALVE AREA IS SPECIFIED.

2030 CONTINUE
V5=V11
V6=V22+V33
CALL INITCN(FCN,X,F)
P1=X(1)
XP1DOT=X(2)
PL=X(3)
XP2DOT=X(4)
FACTOR=CD*(2./RHO)**.5/2.
KC21=FACTOR*AV2*(Pl-PL)**-.5
KQ2=FACTOR*(Pl-PL)**.5*2
KC4=FACTOR*AV4*(PL-PR)**-.5
KQ4=FACTOR*(PL-PR)**.5*2.
HRITE(5,6002)
HRITE(IO,6002)
WRITE(5,6003)V5, V6, PL,PI, AV2, AV4,KC21,KQ2,KC4,KQ4,XPl,XP2,
CXXP1DOT,XP2DOT
GO TO 2000

THIS SECTION COMPUTES THE STEADY STATE VALUES FOR THE
CSMP SIMULATION. IMPORTANT | | | | CYLINDERS ARE ADVANCING
OR THE VELOCITIES ARE POSITIVE | | | |
FORMAT( ' ENTER 0 FOR FRCOPN OR 1 FOR EFCNCY AND RESPECTIVE VALUE ')$

READ(5,*) ICNTRL, DUMMY
IF (ICNTRL.EQ.0) FRCOPN=DUM1Y
IF (ICNTRL.EQ.1) EFCN= DUM1Y
WRITE(I0,9014) ICNTRL, DUMMY
AVMAX=.04
PR=120.
WRITE(I0,9050) PR, AVMAX
WRITE(I0,9003)

CONTROLLER COEFFICIENT DEPENDS ON TYPE OF CONTROL USED

FORMAT( ' ENTER BYPASS AREA CONTROLLER COEFFICIENT')
WRITE(I0,9003)
READ(5,*) CONTAL
WRITE(I0,9004)
WRITE(I0,9003)

FORMAT( ' ENTER DESIRED VELOCITIES FOR PISTONS 1 AND 2')
READ(5,*) DEVEL1, DEVEL2
WRITE(I0,9004)
WRITE(I0,9003)

XPMAX=ABS(DEVEL1)
IF (ABS(DEVEL)) .GT. (ABS(DEVEL2)) XPMAX=ABS(DEVEL2)
CALL INCSPC(FCSMP, X, P)
QP=XC1)
P1=XC2)
AV2=XC3)
P11=XC4)
P12=XC5)
XP1DOT=XC6)
PI=XC7)
AV4=XC8)
P21=XC9)
P22=XC10)
XP2DOT=XC11)
WRITE(I0,9012) AV2, AV4, XP1DOT, XP2DOT

2000 CONTINUE
STOP
END
SUBROUTINE INITCN(FCN,X,F)
C
C
DIMENSION X(15),F(15)
EXTERNAL FCN
COMMON/ARR/A(15,15),LPC(15),LPR(15),LTEMP(15),XSAVE(15),FSAVE(15)
DOUBLE PRECISION A,FSAVE
COMMON/PARAM/QP,F1,F2,XP1,XP2,AV1,AV2,AV3,AV4,PR,
cCTRL,AMAX,DEVEL,DEVEL2,IPMAX,FROPN,ICNTRL,EPNCY
C
INITIAL GUESSES OF VARIABLES
C
X(1)=P1
C
X(2)=XP1DOT
X(3)=P2
C
X(4)=XP2DOT
X(5)=PR-F2/2/FP
X(6)=X(3)-P1/2/FP
X(7)=QP/2/FP
X(8)=QP/FP
CALL NLSYS(FCN,4,100,X,F,.01,.1E-8,.1,.1)
WRITE(5,1070)
1070 FORMAT(’I= ’,I3)
RETURN
END

SUBROUTINE INIT3 (FCN3,X,F)
DIMENSION X(15),F(15)
EXTERNAL FCN3
COMMON/ARR/A(15,15),LPC(15),LPR(15),LTEMP(15),XSAVE(15),FSAVE(15)
DOUBLE PRECISION A,FSAVE
COMMON/PARAM/QP,F1,F2,XP1,XP2,AV1,AV2,AV3,AV4,PR,
cCTRL,AMAX,DEVEL,DEVEL2,IPMAX,FROPN,ICNTRL,EPNCY
C
INITIAL GUESSES OF VARIABLES
C
X(1)=P1
C
X(2)=P1
X(3)=P12
C
X(4)=P12
X(5)=XP1DOT
X(6)=P21
X(7)=P22
C
X(8)=XP2DOT
FLOW=(QF/CDFP)+*2*(RHO/2.)
X(1)=PR-F2/2/FP-F1/2/FP+F1/2/FP
X(2)=PR-F2/2/FP-F1/2/FP+F1/2/FP
X(3)=PR-F2/2/FP-F1/2/FP+F1/2/FP
X(4)=PR-F2/2/FP-F1/2/FP+F1/2/FP
X(5)=PR-F2/2/FP-F1/2/FP+F1/2/FP
X(6)=PR-F2/2/FP-F1/2/FP+F1/2/FP
X(7)=PR-F2/2/FP-F1/2/FP+F1/2/FP
X(8)=PR-F2/2/FP-F1/2/FP+F1/2/FP
CALL NLSYS(FCN3,8,100,X,F,.01,.1E-8,.1,.1)
WRITE(5,2070)
2070 FORMAT(’I= ’,I3)
RETURN
END

SUBROUTINE INCN (FCN3,X,F)
DIMENSION X(15),F(15)
EXTERNAL FCN3
COMMON/ARR/A(15,15),LPC(15),LPR(15),LTEMP(15),XSAVE(15),FSAVE(15)
DOUBLE PRECISION A,FSAVE
COMMON/PARAM/QP,F1,F2,XP1,XP2,AV1,AV2,AV3,AV4,PR,
INITIAL GUESSES OF VARIABLES

\[ X(1) = QP \]

\[ X(2) = P1 \]
\[ X(3) = AV2 \]
\[ X(4) = P11 \]
\[ X(5) = P12 \]
\[ X(6) = X'P1DOT \]
\[ X(7) = PL \]
\[ X(8) = AV4 \]
\[ X(9) = P21 \]
\[ X(10) = P22 \]

\[ X(11) = X'P2DOT \]

\[ \text{FACTOR} = CD \times (2 / \text{RHO})^{0.5} \]

\[ X(1) = (\text{DEVEL1} + \text{DEVEL2}) / 2 \times \text{AP} \]
\[ X(11) = \text{DEVEL2} \]
\[ X(6) = \text{AVEX} / 2 \]
\[ X(10) = X(11) / (\text{FACTOR} \times \text{AV})^{2 + \text{PR}} \]
\[ X(9) = X(10) + (F2 + B \times X(11)) / \text{AP} \]
\[ X(7) = (X(11) \times \text{AP} / (\text{FACTOR} \times \text{AV}))^{2 + X(9)} \]
\[ X(6) = \text{DEVEL2} \]
\[ X(5) = (X(1) / (\text{FACTOR} \times \text{AV}))^{2 + X(7)} \]
\[ X(4) = X(5) + (F2 + B \times X(6)) / \text{AP} \]
\[ X(3) = \text{AVEX} / 2 \]
\[ X(2) = (X(6) \times \text{AP} / (\text{FACTOR} \times \text{AV}))^{2 + X(4)} \]

CALL NLSYS(FCS, P, 11, 100, X, F, .01, 1.E-8, 1.E-4, I)
WRITE(5, 9005) I

9005 FORMAT(' I= ', I3)
WRITE(10, 9005) I
RETURN
END
B.3 MFRSIM

PROGRAM INA
DOMAIN COMPLEX
D=0.0;MXXCAL=0.0
V1=MATRIX(2,1);V2=V1
M=MATRIX(4,4);KBAR=M;Q=MATRIX(2,2);B1=Q;B2=Q;KE=1.0
FREQ=1D1;RX1AV2=FREQ;IX1AV2=FREQ
HEIGHT1=FREQ;HEIGHT2=FREQ
RATIO1=FREQ;RATIO2=FREQ
TEM1=FREQ;TEM2=FREQ;TEM3=FREQ;TEM4=FREQ
RX1AV4=FREQ;IX1AV4=FREQ;RX2AV3=FREQ
IX2AV2=FREQ;RX2AV4=FREQ;IX2AV4=FREQ
MAG1=FREQ;MAG2=FREQ;MAG3=FREQ;MAG4=FREQ
MASS=1.;WNS=354.;ZETAS=1.0;WNP=100.;ZETAP=1.
B=10.;AP=4.1;BE=1.5
KC21=.0052;KC4=.0052
KQ2=3669.;KQ4=3669.
V5=102.;V6=143.
KE=4.6;KE2=1.6;AS=.53
INCR=10
DELT2W=2.*INCR
MAXW=35
FOR L=1,MAXW,1;WEIGHT1(L)=1.0;ENDLOOP L
FOR L=1,MAXW,1;WEIGHT2(L)=0.0;ENDLOOP L
$FOR L=12,MAXW,1;WEIGHT1(L)=1.0;ENDLOOP L
$FOR L=10,MAXW,1;WEIGHT2(L)=1.0;ENDLOOP L
$FOR L=10,14,L;WEIGHT2(L)=1.0;ENDLOOP L
$WEIGHT1(16)=1.0;HEIGHT2(12)=1.0
FOR L=1,MAXW,1;S=(L-1)*INCR*11
J=IMAG(S)
FREQ(L)=J
SV = 0.53/((S/WNS)**2+2.*ZETAS*S/WNS+1.)
PD=1./((S/WNP)**2+2.*ZETAP*S/WNP+1.)
$PD=0.0
M(1,1)=S*V5/BE+KC21-KC21/2.*PD;M(1,2)=-KC21+KC21/2.*PD-KC4/2.*AP
M(1,3)=AP+KQ2ASVAKE1AS+KQ2/2.*AP
M(1,4)=KQ4ASVAKE2ASASA
M(2,1)=-KC21,SAV6/BE+KC21+KC4,-AP-KQ2ASVAKE1AS,AP+KQ4ASVAKE2ASASA
M(3)=AP,-AP,-MASSAS-B
M(4)=0.0,AP,0.0,-MASSAS-B
VAV2=VECTOR(4:-KQ2A(SV+PDAS/2.,-KQ2ASV)
VAV4=VECTOR(4:-KQ4APDAAS/2.,-KQ4ASV)
INUM=INVERSE(M)
PAV2=INUM*VAV2
PAV4=INUM*VAV4
X1AV2=PAV2(3)
X1AV4=PAV4(3)
X2AV2=PAV2(4)
X2AV4=PAV4(4)
Q(1)=X1AV2,X1AV4
Q(2)=X2AV2,X2AV4
QBAR=INVERSE(Q)
$QBAR=QBAR
QBAR11=QBAR(1,1)
QBAR12=QBAR(1,2)
QBAR21=QBAR(2,1)
QBAR22=QBAR(2,2)
RX1AV2(L)=REAL(QBAR11);IX1AV2(L)=IMAG(QBAR11)
RX1AV4(L)=REAL(QBAR12);IX1AV4(L)=IMAG(QBAR12)
RX2AV2(L)=REAL(QBAR21);IX2AV2(L)=IMAG(QBAR21)
RX2AV4(L)=REAL(QBAR22);IX2AV4(L)=IMAG(QBAR22)
B2(1,1)=(RX1AV2(L)*X2+IX1AV2(L)*X2)*HEIGHT2(L)+B2(1,1)
B2(1,2)=(RX1AV4(L)*X2+IX1AV4(L)*X2)*HEIGHT2(L)+B2(1,2)
B2(1,2)=(RX1AV2(L)*RX2AV2(L)+IX1AV2(L)*IX2AV2(L))*HEIGHT2(L)+B2(2,1)
\[ B2(2,2) = (RX2AV2(L) \times RX2AV2(L) + IX2AV2(L) \times IX2AV2(L)) \times WEIGHT2(L) + B2(2,2) \]

\[ B1(1,1) = (RX1AV4(L) \times RX1AV4(L) + IX1AV4(L) \times IX1AV4(L)) \times WEIGHT1(L) + B1(1,1) \]

\[ B1(1,2) = (RX1AV4(L) \times RX2AV4(L) + IX1AV4(L) \times IX2AV4(L)) \times WEIGHT1(L) + B1(2,1) \]

\[ B1(2,1) = (RX2AV4(L) \times RX1AV4(L) + IX2AV4(L) \times IX1AV4(L)) \times WEIGHT1(L) + B1(1,2) \]

\[ B1(2,2) = (RX2AV4(L) \times RX2AV4(L) + IX2AV4(L) \times IX2AV4(L)) \times WEIGHT1(L) + B1(2,2) \]

\[ ENDLOOP \ L \]

\[ VALUE1 = EIGENSYSTEM(B1,V1:-1) \]

\[ VALUE2 = EIGENSYSTEM(B2,V2:-1) \]

\[ P11 = V1(1,1) \]

\[ P12 = V1(2,1) \]

\[ P21 = V2(1,1) \]

\[ P22 = V2(2,1) \]

\[ P11 = 1.0; P12 = 0.0; P21 = 0.0; P22 = 1.0 \]

\[ FOR \ L = 1, \text{MAXW}, 1 \]

\[ TEMP1(L) = RX1AV2(L) \]

\[ TEMP2(L) = IX1AV2(L) \]

\[ RX1AV2(L) = (RX1AV2(L) \times P11 + P12 \times RX2AV2(L)) \]

\[ IX1AV2(L) = (IX1AV2(L) \times P11 + P12 \times IX2AV2(L)) \]

\[ MAG1(L) = (RX1AV2(L) \times RX1AV2(L) + IX1AV2(L) \times IX1AV2(L)) \]

\[ RX1AV4(L) = (RX1AV4(L) \times P11 + P12 \times RX2AV4(L)) \]

\[ IX1AV4(L) = (IX1AV4(L) \times P11 + P12 \times IX2AV4(L)) \]

\[ MAG2(L) = (RX1AV4(L) \times RX1AV4(L) + IX1AV4(L) \times IX1AV4(L)) \]

\[ RX2AV2(L) = (RX2AV2(L) \times P22 + TEMP1(L) \times P21) \]

\[ IX2AV2(L) = (IX2AV2(L) \times P22 + TEMP2(L) \times P21) \]

\[ MAG3(L) = (RX2AV2(L) \times RX2AV2(L) + IX2AV2(L) \times IX2AV2(L)) \]

\[ RX2AV4(L) = (RX2AV4(L) \times P22 + TEMP3(L) \times P21) \]

\[ IX2AV4(L) = (IX2AV4(L) \times P22 + TEMP4(L) \times P21) \]

\[ MAG4(L) = (RX2AV4(L) \times RX2AV4(L) + IX2AV4(L) \times IX2AV4(L)) \]

\[ RATIO1(L) = MAG2(L) / MAG1(L) \]

\[ RATIO2(L) = MAG4(L) / MAG3(L) \]

\[ ENDLOOP \ L \]

\[ NEWPAGE \]

\[ FMTC = "(' THE FOLLOWING ARE VALUES FOR QBAR ' )" \]

\[ WRITE(FMTC:) \]

\[ FMTH1 = "(')" \]

\[ WRITE(FMTH1:) \]

\[ FMTK1 = "('KEl =',F7.6,' KE2 =',F7.6,' V5 =',F4.0,' V6 =',F4.0)"
\]

\[ FMTK2 = "('Kc21 =',F5.4,' KC4 =',F5.4,' KQ2 =',F5.0,' KQ4 =',F5.0)"
\]

\[ FMTK3 = "('WNP =',F5.0,' ZETAP =',F3.1,' WN5 =',F5.0,' ZETAS =',F3.1)"
\]

\[ WRITE(FMTK1:KE1,KE2,V5, V6) \]

\[ WRITE(FMTK2:KC21,KC4,KQ2,KQ4) \]

\[ WRITE(FMTK3:WNP,ZETAP,WN5,ZETAS) \]

\[ ENDLOOP \]

\[ NEWPAGE \]

\[ FMTC = "(')" \]

\[ WRITE(FMTC:) \]

\[ FMTH1 = "(')" \]

\[ WRITE(FMTH1:) \]

\[ FMTK1 = "('KBARll =',F8.4,' KBAR12 =',F8.4,' KBAR21 =',F8.4,' KBAR22 =',F8.4)"
\]

\[ WRITE(FMTK1:P11,P12,P21,P22) \]

\[ WRITE(FMTK1:) \]

\[ FMTH2 = "('30X, 'QBARll', 33X, 'QBAR12')"
\]

\[ WRITE(FMTH2:) \]

\[ FMTH2 = "('3X,F0.4X,WT, 8X,RL, 11X,IM, 11X, MG, 11X, RL, 11X,IM, 9X, RATIO')"
\]

\[ WRITE(FMTH2:) \]

\[ FMTH2 = "('3X,FREQ, 4X,WEIGHT, 9X, REAL, 13X, IMAG, 13X, MG, 14X, REAL')"
\]

\[ NEWPAGE \]

\[ FMTC = "(')"
\]

\[ WRITE(FMTC:) \]

\[ FMTH1 = "(')"
\]

\[ WRITE(FMTH1:) \]

\[ FMTK1 = "('FREQ, 4X,WEIGHT, 9X, REAL, 13X, IMAG, 13X, MG, 14X, REAL')"
\]

\[ WRITE(FMTH1:) \]

\[ FMTH2 = "(')"
\]

\[ WRITE(FMTH2:) \]

\[ FMTH2 = "(')"
\]

\[ WRITE(FMTH2:) \]

\[ FMTH2 = "(')"
\]

\[ WRITE(FMTH2:) \]

\[ FMTH2 = "(')"
\]

\[ WRITE(FMTH2:) \]

\[ FMTH2 = "(')"
\]

\[ WRITE(FMTH2:) \]
FMTH2="(30X,'QBAR21',33X,'QBAR22')"
WRITE(FMTH2:)
WRITE(F:)
WRITE(FMTD:FREQ,WEIGHT2,RX2AV2,IX2AV2,MAG3,RX2AV4,IX2AV4,MAG4,RATIO2)
ENDWRITE
END
EXECUTE INA
/*
//FT06F001 DD SYSOUT=A
B.4 FORMAC

INPUT TO KFA FORMAC PREPROCESSOR — VERSION 2A — SIU, AUGUST 1973

1. TEST PROCEDURE OPTIONS

2. FORMAC OPTIONS:

3. LETK1-CODEM(D+V5/BE+K21):

4. K2=CODEM(B0/BE+K21+K41):

5. K3=CODEM(K1+K2*K21+K11):

6. K4=CODEM(K2+K1+K21+K11):

7. K5=CODEM(K1+K2*K21+K3*K11):

8. K6=CODEM(K2+K1+K21+K11):

9. K7=CODEM(K2+K1+K21+K11):

10. K8=CODEM(K2+K1+K21+K11):

11. K9=CODEM(K2+K1+K21+K11):

12. K10=CODEM(K2+K1+K21+K11):

13. K11=CODEM(K2+K1+K21+K11):

14. K12=CODEM(K2+K1+K21+K11):

15. SAVETF(1):

16. LET(TF1)=CODEM(K1):

17. LET(TF2)=CODEM(K1):

18. LET(TF3)=CODEM(K1):

19. LET(TF4)=CODEM(K1):

20. LET(TF5)=CODEM(K1):

21. LET(TF6)=CODEM(K1):

22. LET(TF7)=CODEM(K1):

23. LET(TF8)=CODEM(K1):

24. LET(TF9)=CODEM(K1):

25. LET(TF10)=CODEM(K1):

26. LET(TF11)=CODEM(K1):

27. LET(TF12)=CODEM(K1):

28. LET(TF13)=CODEM(K1):

29. LET(TF14)=CODEM(K1):

30. LET(TF15)=CODEM(K1):

31. LET(TF16)=CODEM(K1):

32. LET(TF17)=CODEM(K1):

33. LET(TF18)=CODEM(K1):

34. LET(TF19)=CODEM(K1):

35. LET(TF20)=CODEM(K1):

36. LET(TF21)=CODEM(K1):

37. LET(TF22)=CODEM(K1):

38. LET(TF23)=CODEM(K1):

39. LET(TF24)=CODEM(K1):

40. LET(TF25)=CODEM(K1):

41. LET(TF26)=CODEM(K1):

42. LET(TF27)=CODEM(K1):

43. LET(TF28)=CODEM(K1):

44. LET(TF29)=CODEM(K1):

45. LET(TF30)=CODEM(K1):

46. LET(TF31)=CODEM(K1):

47. LET(TF32)=CODEM(K1):

48. LET(TF33)=CODEM(K1):

49. LET(TF34)=CODEM(K1):

50. LET(TF35)=CODEM(K1):

51. LET(TF36)=CODEM(K1):

52. LET(TF37)=CODEM(K1):

53. LET(TF38)=CODEM(K1):

54. LET(TF39)=CODEM(K1):

55. LET(TF40)=CODEM(K1):

56. LET(TF41)=CODEM(K1):

57. LET(TF42)=CODEM(K1):

58. LET(TF43)=CODEM(K1):

59. LET(TF44)=CODEM(K1):
60  PUT SKIP(2):
61  PUT LIST(*COEFFICIENT OF D**4);
62  PRINT OUT(1):
63  PUT SKIP(2):
64  PUT LIST(*COEFFICIENT OF D**5);
65  PRINT OUT(1):
66  PUT SKIP(2):
67  PUT LIST(*COEFFICIENT OF D**6);
68  PRINT OUT(1):
69  PUT SKIP(2):
70  PUT LIST(*COEFFICIENT OF D**7):
71  PRINT OUT(1):
72  PRINT PAGE;
73  END;
74
75  PUT SKIP(2):
76  PUT LIST(*DENOMINATOR OF ABOVE TRANSFER FUNCTIONS*);
77  PUT SKIP(2):
78  PRINT OUT(0);
79  PUT SKIP(2):
80  PUT LIST(*EXPANDED DENOMINATOR*);
81  LET IDE1 = EXPAND(IDE1);
82  PUT SKIP(2):
83  PRINT OUT(0):
84  LET IDE1 = COEFF(IDE1, D**1); IDE2 = COEFF(IDE1, D**2); IDE3 = COEFF(IDE1, D**3);
85  IDE4 = COEFF(IDE1, D**4); IDE5 = COEFF(IDE1, D**5); IDE6 = COEFF(IDE1, D**6);
87  LET IDE1 = EXPAND(IDE1, D**1); IDE2 = EXPAND(IDE1, D**2);
88  IDE3 = EXPAND(IDE1, D**3); IDE4 = EXPAND(IDE1, D**4); IDE5 = EXPAND(IDE1, D**5);
89  IDE6 = EXPAND(IDE1, D**6);
90  PUT SKIP(2):
91  PUT LIST(*COEFFICIENT OF D**0*);
92  PRINT OUT(0);
93  PUT SKIP(2):
94  PUT LIST(*COEFFICIENT OF D**1*);
95  PUT SKIP(2):
96  PRINT OUT(0);
97  PUT SKIP(2):
98  PUT LIST(*COEFFICIENT OF D**2*);
99  PUT SKIP(2):
100  PRINT OUT(0);
101  PUT SKIP(2):
102  PUT LIST(*COEFFICIENT OF D**3*);
103  PUT SKIP(2):
104  PRINT OUT(0);
105  PUT SKIP(2):
106  PUT LIST(*COEFFICIENT OF D**4*);
107  PUT SKIP(2):
108  PRINT OUT(0);
109  PUT SKIP(2):
110  PUT LIST(*COEFFICIENT OF D**5*);
111  PUT SKIP(2):
112  PRINT OUT(0);
113  PUT SKIP(2):
114  PUT LIST(*COEFFICIENT OF D**6*);
115  PUT SKIP(2):
116  PRINT OUT(0);
117  PUT SKIP(2):
118  PUT LIST(*COEFFICIENT OF D**7*);
119  END TEST;
120  END TEST;
B.5 OPPMAS

COMMON/GPT,XU(18),XL(18),J(20,20),ACC,XX(18),Y,IRON,DOMRAT
DIMENSION XM(12),Z(26),XNU(12)
DIMENSION GBJ(26),P(26),ICHG(12)
DIMENSION NPONT(1)

CHARACTER *5 LINE(1)/'SOLID'/,CHAR(1)*1/' ' /
COMMON/VAR/ FREQ(100),RX1AV2(100,1),GX1AV2(100,1),RX1AV4(100,1),
C GX1AV4(100,1),RX2AV2(100,1),GX2AV2(100,1),RX2AV4(100,1),
C RX2AV4(100,1),RMMAGI(100,1),RMMAGI(100,1),RMMAGI(100,1),
C RMMAGI(100,1),
CHARACTER *8 TITAX1,TITAX2
CHARACTER *20 FILEIN
REAL MAG1(100),MAG2(100),MAG3(100),MAG4(100),K2,KC21,KC4,KQ2,KQ4
REAL K21,K22
INTEGER BEFREQ,EFDORQ

1 FORMAT(1X,F6.1,2(F14.8))
20 FORMAT(2(F12.8))
CALL PLOTS
WRITE(5,42)
42 FORMAT( '  ENTER INFILE')
READ(5,43)FILEIN
43 FORMAT(A)
WRITE(5,44)
44 FORMAT( '  ENTER DOMRAT')
READ(5,* ,DOMRAT)
INTEGER BEFREQ,EFDORQ
OPEN(UNIT=29,NAME=FILEIN,TYPE='OLD',READONLY)
WRITE(5,3)
3  FORMAT( '  ENTER NO. OF DATA POINTS')
READ(5,* ,NDPTS)
WRITE(5,9000)
9000 FORMAT( '  ENTER 1 FOR TOP ROW OR 2 FOR BOTTOM ROW')
READ(5,* ,IROW)
CONTINUE
READ(29,1,ERR=30,FREQ(I),RX1AV2(I,1),GX1AV2(I,1))
IF(FREQ(I).EQ.0.0.AND.RX1AV2(I,1).EQ.0.0.AND.GX1AV2(I,1).EQ.0.0)
CGO TO 30
DO 6 I=2,NDPTS
2  CONTINUE
READ(29,1,ERR=2)FREQ(I),RX1AV2(I,1),GX1AV2(I,1)
6  CONTINUE
86 CONTINUE
READ(29,85,ERR=86)K21,K22,V5,V6,KC21,KC4,KQ2,KQ4,WNP,ZP,WNS,
C CZS)
85 FORMAT(F7.6,F7.6,F4.0,F4.0,F5.4,F5.4,F5.0,F5.0,F4.0,F3.1,F4.0,
CF3.1)
DO 7 I=1,NDPTS
8  CONTINUE
READ(29,1,ERR=8)FREQ(I),RX1AV4(I,1),GX1AV4(I,1)
7  CONTINUE
DO 10 I=1,NDPTS
10 CONTINUE
READ(29,1,ERR=10)FREQ(I),RX2AV2(I,1),GX2AV2(I,1)
9  CONTINUE
DO 12 I=1,NDPTS
12 CONTINUE
READ(29,1,ERR=12)FREQ(I),RX2AV4(I,1),GX2AV4(I,1)
11 CONTINUE
71 CONTINUE
WRITE(5,1005)
1005 FORMAT( 'K21=',F7.6,' K22=',F7.6,' V5=',F4.0,' V6=',F4.0,
V5=',F4.0,' V6=',F4.0,
V5=',F4.0,
C K21=',F5.4,
C K21=',F5.4,' K22=',F5.4,' V5=',F5.0,' V6=',F5.0,
V5=',F5.0,' V6=',F5.0,
V5=',F5.0,' V6=',F5.0,' WNP=',F4.0.
412

1005 FORMAT( ' ENTER XI(1),XI(2),X(1,1),X(2,1),XX(1),
CXX(2) ' )
READ(5,*)XI(1),XI(2),X(1,1),X(2,1),XX(1),XX(2)
WRITE(16,1005)

1006 FORMAT( ' ENTER XI(3),XI(4),X(3,1),X(4,1),XX(3),
CXX(4) ' )
READ(5,*)XI(3),XI(4),X(3,1),X(4,1),XX(3),XX(4)
WRITE(16,1006)

1007 FORMAT( ' ENTER XI(5),XI(6),X(5,1),X(6,1),XX(5),
CXX(6) ' )
READ(5,*)XI(5),XI(6),X(5,1),X(6,1),XX(5),XX(6)
WRITE(16,1007)

ACC=.05
IIX=1.0
IIY=0.
NPAR=2
L1=2
QUIT=0
ITYPE=0
WRITE(5,*)(' ENTER BEGINNING FREQ AND ENDFREQ ','
BEFREQ,ENDFRQ)
WRITE(5,*)(' BEGINNING FREQ IS ',F5.1,' ENDFREQ IS ',F5.1)

IFLAG=0
DO 1000 L1=BEFREQ,ENDFRQ
    L=L1
    IFLAG=IFLAG+1
    RMAG1=(XI(1,1)*RX1AV2(L,1)+X(3,1)*RX2AV2(L,1)-X(2,1))**2
    RMAG2=(X(1,1)*RX1AV4(L,1)+X(3,1)*RX2AV4(L,1)-X(2,1))**2
    RMAG3=(X(4,1)*GX1AV2(L,1)+X(5,1)*GX2AV2(L,1)+X(6,1))**2
    RMAG4=(X(2,1)*RX1AV4(L,1)+X(4,1)*RX2AV4(L,1)+X(6,1))**2
    R1MAG1=L1
    R1MAG2=LI
    R1MAG3=LI
    R1MAG4=LI
    CRX1AV2(L,1)=...
WRITE(5,*)L1,RMAG1,RMAG2,X(1,1),X(2,1)
C IF(RMAG2.LT.RMAG2)ITYPE=0
IF(IROW.EQ.2)GO TO 9001
IF(RMAG1.LT.RMAG2)GO TO 2000
GO TO 9002
9001 CONTINUE
C IF(RMAG2.LT.RMAG1)GO TO 2000
9002 CONTINUE
C IF(RMAG4.LT.RMAG3)ITYPE=1
C IF(RMAG4.LT.RMAG3)GO TO 2000
1000 CONTINUE
1001 CONTINUE
C WRITE(5,*)X(1,1),X(2,1),X(3,1),X(4,1),X(5,1),X(6,1)
C WRITE(16,*)X(1,1),X(2,1),X(3,1),X(4,1),X(5,1),X(6,1)
call pause(1,1)
WRITE(5,*)IROW,X(1,1),X(2,1)
9003 FORMAT( 'IL,--- 1 FOR TOP ROW, 2 FOR BOTTOM',//',', KBAR1 = ',', CF7.4',', KBAR2 = ',',CF7.4)
4000 FORMAT( ' THE FOLLOWING ARE VALUES FOR QBAR ',//)
WRITE(16,4000)
IF(IROW.EQ.1)WRITE(16,4001)
IF(IROW.EQ.2)WRITE(16,4002)
IF(IROW.EQ.1)WRITE(5,4001)
IF(IROW.EQ.2)WRITE(5,4002)
4001 FORMAT(21X,QBAR1',27X,QBAR12',//)
4002 FORMAT(21X,QBAR21',27X,QBAR22',//)
WRITE(16,4006)
WRITE(5,4006)
4006 FORMAT( ' FREQ',5X,'REAL',7X,'IMAG',7X,'MAG',8X,'REAL',7X,
C 'IMAG',7X,'MAG',9X,'RATIO')
4005 FORMAT(1X,F6.1,2X,F9.6,5(2X,F9.6),2X,F9.4)
DO 1002 L=1,NDPTS
FREQ(L)=L-1*10.
RX1AV2(L,1)=X(1,1)*RX1AV2(L,1)
GX1AV2(L,1)=X(1,1)*GX1AV2(L,1)
RX2AV2(L,1)=X(2,1)*RX2AV2(L,1)
GX2AV2(L,1)=X(2,1)*GX2AV2(L,1)
RX1AV4(L,1)=X(1,1)*RX1AV4(L,1)
GX1AV4(L,1)=X(1,1)*GX1AV4(L,1)
RX2AV4(L,1)=X(2,1)*RX2AV4(L,1)
GX2AV4(L,1)=X(2,1)*GX2AV4(L,1)
RMAG1(L,1)=(RX1AV2(L,1)+RX2AV2(L,1))
RMAG1(L,1)=(GX1AV2(L,1)+GX2AV2(L,1))
RMAG2(L,1)=(RX1AV4(L,1)+RX2AV4(L,1))
RMAG2(L,1)=(GX1AV4(L,1)+GX2AV4(L,1))
RMAG(RMAG1(L,1)**2+RMAG2(L,1)**2)**.5
RMAG2(L,1)=(RX1AV4(L,1)+RX2AV4(L,1))
RMAG2(L,1)=(GX1AV4(L,1)+GX2AV4(L,1))
RMAG2(RMAG2(L,1)**2+RMAG2(L,1)**2)**.5
IF(IROW.EQ.1)RATIO1=RMAG1/RMAG2
IF(IROW.EQ.2)RATIO1=RMAG2/RMAG1
WRITE(16,4005)FREQ(L),RMAG1(L,1),
CRMAG1(L,1),RMAG2(L,1),RMAG2(L,1),RIMAG2(L,1),RIMAG2(L,1),RIMAG2(L,1)
WRITE(5,4005)FREQ(L),RMAG1(L,1),
CRMAG1(L,1),RMAG2(L,1),RIMAG2(L,1),RIMAG2(L,1),RIMAG2(L,1)
1002 CONTINUE
WRITE(5,900)L
900 FORMAT( ' PROCEDURE HAS BEEN SUCCESSFUL, HOORAY',//',I4)
call pause(1,1)
SUBROUTINE COHPLX(M,IIY,NPAR,U)
DIMENSION XH(12),Z(26),XNU(12)
DIMENSION OBJ(26),P(26),ICHG(12)
COMMON/0PT/XU(18),XL(18),X(20,20),ACC,XX(18),Y,IRCH,DOMAT
COMMON/VAR/PRQ(100),RXAV2(100,1),GXAV2(100,1),RXAV4(100,1)
COMMON/RAV2(100,1),GXAV2(100,1),RXAV4(100,1),GXAV4(100,1)
INTEGER P,PR,SHRINK,CHECK

THIS SUBROUTINE SEARCHES FOR A LOCAL MINIMUM USING THE COMPLEX
METHOD AS MODIFIED BY GUIN TO COPE WITH NON-CONVEX BOUNDARIES.

REFERENCES-
1. M.J. BOX, A NEW METHOD OF UNCONSTRAINED OPTIMIZATION AND A
   COMPARISON WITH OTHER METHODS, THE COMPUTER JOURNAL, VOL.8,
2. J.A. GUIN, MODIFICATION OF THE COMPLEX METHOD OF
   CONSTRAINED OPTIMIZATION, THE COMPUTER JOURNAL, VOL. 10,
3. G.S.C. BEVERIDGE AND R.S. SCHECHTER, OPTIMIZATION THEORY AND

SOME OF THE PARAMETERS USED IN THE SUBROUTINE ARE GIVEN BELOW.
THE REST OF THE IMPORTANT PARAMETERS ARE DEFINED IN THE MAIN
PROGRAM.

N = NUMBER OF COMPONENTS IN X.
X = INDEPENDENT VARIABLES.
   DIM N+1,K+2
OBJ = FUNCTION TO BE OPTIMIZED.
   DIM K+2
X = NUMBER OF VERTICES IN COMPLEX.
XH = COORDINATES OF CENTROID OF COMPLEX.
XU = UPPER LIMITS FOR COMPONENTS OF X.
XL = LOWER LIMITS FOR COMPONENTS OF X.
IP = IDENTIFICATION FOR VERTICES USED IN COMPLEX.
NVTX = NUMBER OF VERTICES WHICH DO NOT HAVE TO BE GENERATED.
BETA = CONTRACTION FACTOR FOR COMPLEX.
ALPHA = EXPANSION FACTOR FOR THE COMPLEX.
ACC = ACCURACY REQUIRED FOR THE FUNCTION IN THE OPTIMIZATION
       PROCEDURE.
CHECK = CONSTRAINT VIOLATION INDICATOR. CHECK = 1.0 WHEN
       CONSTRAINTS ARE VIOLATED.
KMAX = MAXIMUM NUMBER OF FUNCTION EVALUATIONS PERMITTED
       BEFORE THE OPTIMIZATION PROCEDURE IS TERMINATED.
KOUNT = NUMBER OF FUNCTION EVALUATIONS.
NTST = NUMBER OF TIMES TEST SUBROUTINE HAS CALLED.
ICNG = FIXED VARIABLE ID NUMBER
NII = NUMBER OF TIMES THE RANDOM NUMBER GENERATOR CALLED
      FOR STARTING COMPLEX SEED

AT LEAST ONE INITIAL FEASIBLE POINT I(N,1) IS ASSUMED TO BE GIVEN
AS AN INPUT TO THE SUBROUTINE. THE CONSTRAINTS ARE TESTED IN THE
SUBROUTINE TEST AND THE FUNCTION IS EVALUATED BY THE
SUBROUTINE FUN.
MAX=4000

NTST=2000
IPRINT=1
NVR=1
K=2*NPAB
M1=100
ALPHA=1.3
BETA=0.8
IN=15
IOUT=6
DO 8000 I=1,10
8000 ICHG(I)=0
N=NPAB
N2=NVTI+1
MCY=0
MTST1=0
IDAD=0
KOUNT=0
KKK=K+1
KK2=K+2
K1=K-1
DO 1 I=1,K
1 P(I)=I

ESTABLISH AN UPPER BOUNDARY FOR THE OBJECTIVE FUNCTION

DO 3 I=1,N
3 XX(I)=1.
WRITE(5,*)N,XX(1),XX(2)
CALL FNO1(L1)
KOUNT=KOUNT+1
FUN0=FNO

TEST INITIAL POINT FOR DATA FEASIBILITY. IF THE POINT IS
NOT FEASIBLE, THE ROUTINE IS TERMINATED.

DO 2 I=1,N
2 XX(I)=X(I,1)

SET INITIAL GUESS SO A STARTING IS ALWAYS GENERATED

CALL TESTI (XX,CHECK,XU,XL,NPAR)
IF (CHECK.NE.0) WRITE (IOUT,55)
IF (CHECK.NE.0) GO TO 51

DEFINE THE STARTING COMPLEX.
SET UP FIX OPTIMIZATION VARIABLE COMPLEX

DO 120 J=1,K
DO 120 I=1,N
X(I,J)=X(I,1)
120 CONTINUE

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C X(1,1)=-.0004
C X(2,1)=-.00005
C X(3,1)=-.0003
C X(4,1)=-.0004
C X(1,2)=-.00035
C X(2,2)=-.000045
C X(3,2)=-.0003
X(2,3)=-.00004

X(3,3)=-.0002
X(4,3)=-.0003
X(1,4)=-.0004
X(2,4)=-.0004
X(3,4)=-.0003
X(4,4)=-.0004

GO TO 1999

6 DO 7 M=1,N
7 XM(M)=X(M,1)

CALL RAN1X(IIX,IIY,Y)

DO 130 1=1,Nil
CALL RAN1X(IIX,IIY,Y)
IF (ICHIG(LL).NE.0) GO TO 11
X(LL,J)=Y*(XU(LL)-XL(LL))+XL(LL)
11 CONTINUE

XX(I)=X(I,J)
IV0PT=1
IIIOPT=1
CALL TEST (XX, CHECK, XU, XL, NPAR, IV0PT, IIIOPT)
NTST1=NTST1+1
IF (NTST1.GT.NTST) WRITE(IOUT,63) NTST
IF (CHECK.NE.0.AND.I.EQ.1) GO TO 8

IF (CHECK.NE.0) GO TO 14
GO TO 16

IF (CHECK.NE.0) GO TO 14
GO TO 16

IF (J.GT.1.OR.MY.GT.0) GOTO 160

WRITE(IOUT,80)
80 FORMAT(I X , A THEN ORIGINAL OBJECTIVE FUNCTION COULD NO
**'/,IX,'** BE FOUND AT RANDOM. THE ROUTINE IS TERMINATED. **')
DO 18 KK=1,M
  16 DO 18 JJ=1,J
  17 SUM=SUM+MX(KK,JJ)
  18 CONTINUE
  CALL TEST (XM,CHECK,XU,XL,NPAR,IVOPT,IIIOPT)
  NTST=NTST+1
  IF (NTST.GT.NTST) WRITE(IOUT,63)NTST
  IF (NTST.GT.NTST) GO TO 51
  IF (CHECK.NE.0) WRITE (IOUT,52)
  IF (CHECK.NE.0) GO TO 6
  19 CONTINUE
C THE VERTICES OF THE COMPLEX ARE NOW DEFINED. CALCULATE
C THE OBJECTIVE FUNCTION AT EACH FEASIBLE VERTEX.
C CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C 1999 CONTINUE
  WRITE(IOUT,64)
  DO 22 J=1,K
     DO 21 1=1,N
        XX(I)=X(I,J)
     21 CONTINUE
     KOUNT=KOUNT+1
     CALL FUNO(Ll)
     OBJ(J)=Y
  22 CONTINUE
C PRINT OUT INITIAL COMPLEX DATA IF IPRINT = 1
C IF (IPRINT.EQ.0) GO TO 26
WRITE (IN,53)
WRITE (IN,57)
DO 24 1=1,K
   II=P(I)
   IF (N.LT.20) X(N+1,II)=0.0
   WRITE (IN,56) II,II,OBJ(II)
   DO 23 J=1,N,2
      J1=J+1
   23 WRITE (IN,54) J.X(J,II),J1,X(J1,II)
  24 CONTINUE
C ORDER THE COMPLEX POINTS FROM BEST TO WORST. P(1) GIVES THE BEST
C POINT AND P(K) GIVES THE WORST. THE WORST OBJECTIVE FUNCTION WILL
C BE DEFINED AS OBJ(PR).
C 26 SHRINK=0
  L=0
  DL 27 1=1,K
     IP=P(I)
  27 Z(I)=OBJ(IP)
     DO 28 J=1,K
         II=I+1
         DO 28 J=1,K
             IF (Z(J).GT.Z(I)) GO TO 28
             SAVE=Z(I)
             Z(I)=Z(J)
             Z(J)=SAVE
             ISAVE=P(I)
CONTINUE

DO 29 J=1,K
   JJ=P(J)
29 OBJ(JJ)=Z(J)
   RBT=P(I)
   PR=P(K)
   IF(ABS((OBJ(PR)-OBJ(IBT)).LE.(OBJ(IBT)-ACC))) GO TO 51
   NCY=NCY+1
   IF (ITPRINT.NE.0) WRITE (IN,59) NCY,PR,OBJ(PR)
C DETERMINE A NEW VERTEX BY EXPANDING THE COMPLEX AWAY FROM THE
C POOREST VERTEX AND THROUGH THE CENTROID OF THE COMPLEX.
C CALCULATE THE NEW VERTEX FROM THE COMPLEX CENTROID AND EXPANSION
C FACTOR ALPHA. FIRST FIND THE COORDINATES OF THE CENTROID Xm WHEN
C THE POOREST VERTEX IS EXCLUDED.

30 DO 32 I=1,N
   SUM=0.0
   DO 31 J=1,K
      JP=P(J)
31 SUM=SUM+X(I,JP)
   IF(ICHG(I).LE.0) GO TO 32
   XH(I)=(SUM-X(I,PR))/FLOAT(NL)
   CONTINUE
C CHECK THE FEASIBILITY OF THE NEW CENTROID. IF THE CENTROID IS NOT
C FEASIBLE, GO TO THE NEXT WORST VERTEX.
C CALL TEST (XM,CHECK,XU,XL,NPAR,IVOPT,IIIOPT)
NTST1=NTST1+1
   IF(NST1.GT.NST) WRITE(IOUT,63) NTST
   IF(NST1.GT.NTST) GO TO 51
   IF (CHECK.EQ.0) GO TO 35
   WRITE (IOUT,52)
   WRITE (IOUT,54) (J,XM(J),J=1,N)
   IF (L.LT.K-1) GO TO 46
C THE CENTROID OF EACH OF THE K-1 POINTS HAS FALLEN INTO A
C NONFEASIBLE REGION DUE TO THE CONVEX BOUNDARY SO USE GUIN
C MODIFICATION TO BOXES PROCEDURE. DISCARD THE ENTIRE COMPLEX EXCEPT
C FOR THE BEST POINT P(1) AND RANDOMLY GENERATE A NEW COMPLEX.
C IF THE CENTROID FALLS INTO A NONFEASIBLE REGION MORE THAN 6
C CONSECUTIVE TIMES, THE PROGRAM IS TERMINATED.
C THE BEST POINT IS NOT SAVED IN THIS VERSION
33 IBAD=IBAD+1
NEW=1
M=P(1)
NV2=2
DO 34 I=1,N
   XX(I)=X(I,M)
34 XX(I)=X(I,M)
   OBJ(I)=OBJ(M)
   CALL TEST (XX,CHECK,XU,XL,NPAR,IVOPT,IIIOPT)
   NST1=NST1+1
   IF(NST1.GT.NST) WRITE(IOUT,63) NST1
   IF(NST1.GT.NST) GO TO 51
   WRITE (IOUT,60)
   IF (IBAD.GT.6) WRITE (IOUT,61)
   IF (IBAD.GT.6) WRITE (IOUT,54) (XM(I),I=1,N)
   IF (IBAD.GT.6) GO TO 51
THE CENTROID IS A FEASIBLE POINT SO COMPUTE THE NEW VERTEX.

35 DO 36 I=1,N
36 XNU(I)=(1.0+ALPHA)*XH(I)-ALPHA*X(I,PR)
   I=0

CHECK PROXIMITY OF NEW VERTEX TO THE CENTROID
BY COMPARING DISTANCE BETWEEN THE CENTROID AND NEW VERTEX
TO SOME EPSILON VALUE, EPS, WHICH IS A CERTAIN PERCENTAGE
OF THE DISTANCE BETWEEN THE WORST AND BEST VERTEXES.
IF THE DISTANCE IS SMALL AS OCCURS NEAR THE END OF A SEARCH,
THEN MOVING TOWARD THE CENTROID WOULD BE INEFFECTIVE.
THEN USE THE CENTROID AS THE NEW VERTEX. IF THE CENTROID DOES NOT
GIVE A BETTER VALUE FOR THE OBJECTIVE FUNCTION, GO TO THE NEXT
WORST VERTEX.

ICK=0
DSUM=0.0
ESUM=0.0
IBEST=P(I)
DO 37 I=1,N
   ESUM=ESUM+(X(I,IBEST)-X(I,PR))**2
37 DSUM=DSUM+(XNU(I)-XH(I))**2
DIST=SQR(DSUM)
EPS=0.05*SQR(ESUM)
IF (DIST.GT.EPS) GO TO 39
DO 38 KK=1,N
38 XNU(KK)=XM(KK)
ICK=1
GO TO 40

TEST TO SEE IF NEW VERTEX IS A FEASIBLE DESIGN POINT.

39 CALL TEST (XNU,XCHEC,XU,XL,NPAR,IVOPT,IIIOPT)
NTST1=NTST1+1
IF (NTST1.GT.NTST) WRITE (LOUT,63)NTST
IF (NTST1.GT.NTST) GO TO 45

CALL THE OBJECTIVE FUNCTION AT THE NEW VERTEX.

40 DO 8888 III=1,N
8888 XX(III)=XNU(III)
CALL FUNO(L1)
OBJ(XXX)=Y
KOUNT=KOUNT+1

CHECK STOPPING CRITERION BY TESTING WHETHER THE VALUES FOR
OBJ(I), I=1,K, ARE WITHIN A CERTAIN PERCENTAGE OF EACH OTHER.

IU=XXX+1
DO 41 I=1,K
   II=IU+1
   IF (ABS(OBJ(IU)-OBJ(II))/OBJ(II)).LT.AC2) GO TO 41
   GO TO 42
41 CONTINUE

ITOL=0
DO 43 I=1,K
   IF (ABS(OBJ(I)-OBJ(IU))/OBJ(IU)).LT.AC2) ITOL=ITOL+1
43 CONTINUE
IF (ITOL.LT.K) GO TO 51
420

IF (OBJ(KKK).GE.OBJ(PR)) GO TO 44

GO TO 49

THE NEW POINT YIELDS A POORER VALUE OF THE OBJECTIVE FUNCTION.
MOVE IN TOWARD THE CENTROID BY A FACTOR BETA UNLESS THE CENTROID
HAS BEEN USED AS THE NEW POINT. IN THE LATTER CASE, GO TO THE SECOND
WORST POINT AND TRY AGAIN. ALSO IF THE NUMBER OF SUCCESSIVE
CONTRACTIONS EXCEEDS 10, GO TO THE NEXT WORST VERTEX.

44 IF (IPRINT.NE.0) WRITE (IN,62) KOUNT,OBJ(KKK)
45 CONTINUE

46 IF (ICK.EQ.1) GO TO 46

47 DO 48 KJ=1,N
48 XNU(KJ)=BETA*(XNU(KJ)-XH(KJ))+XH(KJ)
GO TO 39

PLACE THE NEW VERTEX (IF ACCEPTABLE) INTO THE COMPLEX
AND DROP THE POOREST VERTEX.

49 DO 50 I=1,N
50 CONTINUE

IF (NTST1*0) WRITE (IN,63) NTST
IF (NTST1.GT.NTST) GO TO 51

51 CONTINUE

IF (IBET.EQ.0) RETURN

FUNCTION
DO 600 I=1,N
600 X(I,I)=X(I,IBET)
IF (NTST1.GT.NTST.OR.KOUNT.GT.KMAX) GO TO 8002
RETURN
8002 DO 8001 I=1,N
8001 XX(I)=X(I,IBET)
52 FORMAT (11X,5SH** THE CENTROID HAS FALLEN INTO A NONFEASIBLE REGI
ION **',/10X,'** THE ROUTINE IS REINITIATED **',/)
53 FORMAT (1H1)
54 FORMAT (20X,2HI(,I2,3H) =,E13.4,5X,2HX(,I2,3H) =,E13.4)
55 FORMAT (15X.47H** THE INITIAL POINT IS NOT A FEASIBLE POINT **',/
115X,'** THE ROUTINE IS TERMINATED **',/)
56 FORMAT (/10I1.10HVERTEX NO.,I3,6H, OBJ(,I2,3H) =,E14.5,/)  
57 FORMAT (///10X,55HTHE INITIAL COMPLEX VERTICES ARE GIVEN IN THE FO
LLLOWING,/)  
58 FORMAT (10I1,26HFUNCTION EVALUATION NUMBER,IS,5X,7HIF(X) = ,E14.5)  
59 FORMAT (/10I1,33HTHE REJECTED VERTEX FOR CYCLE NO.,IS,5H IS,13,I4H 
1 AND THE CORRESPONDING OBJECTIVE FUNCTION IS,E14.5,/)  
60 FORMAT (10I1, ** THE COMPLEX CENTROID HAS FALLEN INTO A NONFEAS 
IBLE REGION **',/10I1, ** THE COMPLEX IS BEING REGENERATED **',/)
61 FORMAT (10I1, ** THE CENTROID HAS FALLEN INTO A NONFEASIBLE REGI 
ON 6 TIMES **',/10I1, ** THE ROUTINE IS TERMINATED **',/10I1, ** THE C 
ENTROID LOCATION IS GIVEN IN THE FOLLOWING:**/)
62 FORMAT (10I1,26HFUNCTION EVALUATION NUMBER,IS,5X,7HIF(X) = ,E14.5,23 
1H ** NOT SATISFACTORY **)
63 FORMAT (10I1,** THE TEST FOR A FEASIBLE VERTEX HAS BEEN CALLED**', 
1I5, ** TIMES **',/10X,** THE ROUTINE IS TERMINATED WITH THE LAST C 
OMPLEX AS THE OPTIMUM **')
64 FORMAT (10I1,** A FEASIBLE COMPLEX HAS BEEN FOUND **')
END
SUBROUTINE TEST (X,ICK,XU,XL,NPAR,IVOPT,IIIOPT)
DIMENSION X(12), XU(12), XL(12)
C
C THIS SUBROUTINE TESTS WHETHER THE CONSTRAINTS ARE VIOLATED BY 
THE VECTOR X. IF CONSTRAINTS ARE VIOLATED, ICK = 1, OTHERWISE, 
ICK = 0.
ICK=0
C
CHECK THE INEQUALITY CONSTRAINTS.

DO 2 I=1,NPAR
IF (X(I).GT.XU(I)) GO TO 3
IF (X(I).LT.XL(I)) GO TO 3
2 CONTINUE
IF(IVOPT.EQ.1)RETURN
C IF(IIIOPT.EQ.0) GO TO 4
IF(ABS(X(I)).LT.0.0) GO TO 3
C IF(ABS(X(3)).LT.ABS(X(4))) GO TO 3
RETURN
4 IF(ABS(X(1)).LT.ABS(X(2))) GO TO 3
IF(ABS(X(4)).LT.ABS(X(3))) GO TO 3
IF(ABS(X(5)).LT.ABS(X(6))) GO TO 3
IF(ABS(X(8)).LT.ABS(X(7))) GO TO 3
RETURN
3 ICK=1
RETURN
END
SUBROUTINE TESTX (X,ICK,XU,XL,NPAR)
DIMENSION X(12), XU(12), XL(12)
C
C THE SUBROUTINE TESTS WHETHER THE CONSTRAINTS ARE VIOLATED 
BY THE VECTOR X. IF CONSTRAINTS ARE VIOLATED ICK=1.
C
1 ICK=0
DO 2 I=1,NPAR
B.6 OSTROWPLT

DIMENSION NP0NT1(2), NP0NT2(2)
CHARACTER *6 LINE(2) /'DASHED', 'SOLID'/
CHARACTER *1 CHAR(2) /'Q'/
DIMENSION FREQ(100), RX1AV2(100,2), GX1AV2(100,2), RX1AV4(100,1)
DIMENSION GX1AV4(100,1), RX2AV2(100,1), GX2AV2(100,1), RX2AV4(100,2)
DIMENSION GX2AV4(100,2), R1(100), R2(100), RATIO1(100), RATIO2(100)
REAL MAG1(100), MAG2, MAG3, MAG4(100), KC21, KC4, KQ2, KQ4, K2, MAGXY
CHARACTER * 20 SPINFILE
REAL K21, K22

1 FORMAT(1X, F6.1, 2(F15.8))
20 FORMAT(2(F12.8))
CALL PLOTS
NP0NT1(2)=2
NP0NT2(2)=2
WRITE(5,71)
71 FORMAT( 'ENTER INPUT FILE *.OUT')
READ(5,70) SPINFILE
70 FORMAT(A)
OPEN(UNIT=29, NAME=SPINFILE, TYPE='OLD')
WRITE(5, 3)
3 FORMAT( 'ENTER NO. OF DATA POINTS')
READ(5,*) NDPTS
30 CONTINUE
READ(29,1, ERR=30) FREQ(1), RX1AV2(1,1), GX1AV2(1,1)
IF(FREQ(1).EQ.0.0 .AND. RX1AV2(1,1).EQ.0.0 .AND. GX1AV2(1,1).EQ.0.0) GO TO 30
DO 6 I=2, NDPTS
READ(29,1, ERR=30) FREQ(I), RX1AV2(I,1), GX1AV2(I,1)
6 CONTINUE
READ(29,1, ERR=8) FREQ(I), RX1AV4(I,1), GX1AV4(I,1)
7 CONTINUE
READ(29,1, ERR=10) FREQ(I), RX2AV2(I,1), GX2AV2(I,1)
9 CONTINUE
READ(29,1, ERR=12) FREQ(I), RX2AV4(I,1), GX2AV4(I,1)
11 CONTINUE
READ(29,85, ERR=86) (K21, K22, V5, V6, KC21, KC4, KQ2, KQ4, WNP, WNS, ZP, ZS)
WRITE(5,*) (K21, K22, V5, V6, KC21, KC4, KQ2, KQ4, WNP, WNS, ZP, ZS)
186 CONTINUE
READ(29,187, ERR=186) (P11S, P11, P12S, P12, P21, P22, TAUS1, TAUS2)
187 FORMAT(8(F9.6))
WRITE(5,*) (P11S, P11, P12S, P12, P21, P22, TAUS1, TAUS2)
5002 CONTINUE
WRITE(5,60)
60 FORMAT('ENTER OUTPUT FILE NUMBER')
READ(5,*) IFILE
CALL VWP0RT(0., 1., 0., 0.76171875)
CALL WIND0W(0., 1., 0., 0.76171875)
IDOM1=0
IDOM2=0
LSAV1=0
LSAV2=0
SCLXM1=0.0
SCLAN1=0.0
SCLXM2=0.0
SCLXN2=0.0
ISCXM1=0
ISCXH2=0
WRITE(5,1000)
1000 FORMAT(' ENTER 1 FOR GERSHGORIN AND 2 FOR OSTROWPLT')
READ(5,*)IIPLOT
WRITE(5,40)
40 FORMAT(' ENTER F1 AND F2 ') READ(5,*)F1,F2
WRITE(5,101)
101 FORMAT(' ENTER NFLTPTM') READ(5,*)NFLTPTM
WRITE(5,106)
106 FORMAT(' ENTER NO. OF POINTS ABOVE ICOUNT ') READ(5,*)IMORPTS
NPLTPNT=NPLTPNT+IMORPTS
IF (NPLTPNT.GT.NDPTS) NPLTPNT=NDPTS
DO 41 I=1,NDPTS
MAGI(I)=RX1AV2(I,1)**2+GX1AV2(I,1)**2**.5
MAG2=(RX1AV4(I,1)**2+GX1AV4(I,1)**2)**.5
MAG3=(RX2AV2(I,1)**2+GX2AV2(I,1)**2)**.5
MAG4=(RX2AV4(I,1)+F2)**2+GX2AV4(I,1)**2)**.5
C FOR OSTROWSKI BANDS
C IF(IIPLOT.EQ.2)R1(I)=MAG2*MAG3/MAG4(I)
IF(IIPLOT.EQ.2)R2(I)=MAG3*MAG2/MAGI(I)
IF(IIPLOT.EQ.2.AND.IDOM1.EQ.0.AND.IDOM2.EQ.0)GO TO 45
IF(IIPLOT.EQ.2.AND.IDOM1.EQ.0.AND.IDOM2.EQ.0)GO TO 46
C FOR GERSHGORIN BANDS
C IF(IIPLOT.EQ.1)R1(I)=MAG2
IF(IIPLOT.EQ.1)R2(I)=MAG3
C RATIO1(I)=MAG2/MAG1(I)
RATIO2(I)=MAG3/MAG4(I)
41 CONTINUE
WRITE(5,50)
50 FORMAT(' ENTER A 1 IF PLOTS DESIRED ') READ(5,*)IIPLOT
IF(IIPLOT.NE.1)GO TO 51
ICOUNT1=0
ICOUNT2=0
DO 44 I=1,NDPTS
ICOUNT1=ICOUNT1+1
ICOUNT2=ICOUNT2+1
IF(GX1AV2(I,1).GT.0.0)GO TO 45
DO 46 I=1,NDPTS
ICOUNT2=ICOUNT2+1
IF(GX2AV4(I,1).GT.0.0)GO TO 47
CONTINUE
44 CONTINUE
45 CONTINUE
46 CONTINUE
47 CONTINUE
ICOUNT1=ICOUNT1+IMORPTS
IF(ICOUNT1.GT.NDPTS)ICOUNT1=NDPTS
ICOUNT2 = ICOUNT2 + IMPORTS
IF (ICOUNT2 .GT. NDPTS) ICOUNT2 = NDPTS

FOR GERSHGORIN BANDS
ICOUNT1 = NDPTS
ICOUNT2 = NDPTS

SCLXM1 = 0.0
SCLXM2 = 0.0
SCLXN1 = 0.0
SCLXN2 = 0.0
DO 48 I = 1, ICOUNT1
SCLXM1 = AMAX1 (SCLXM1, RX1AV2 (1, I) + R1 (I)), (GX1AV2 (1, I) + R1 (I))
SCLXN1 = AMIN1 (SCLXN1, RX1AV2 (1, I) - R1 (I)), (GX1AV2 (1, I) - R1 (I))
48 CONTINUE
DO 49 I = 1, ICOUNT2
SCLXM2 = AMAX1 (SCLXM2, RX2AV4 (1, I) + R2 (I)), (GX2AV4 (1, I) + R2 (I))
SCLXN2 = AMIN1 (SCLXN2, RX2AV4 (1, I) - R2 (I)), (GX2AV4 (1, I) - R2 (I))
49 CONTINUE
NPONT1 = ICOUNT1
NPONT2 = ICOUNT2
SCLFA1 = AMAX1 (ABS (SCLXN1), ABS (SCLXM1))
SCLFA2 = AMAX1 (ABS (SCLXN2), ABS (SCLXM2))
WRITE (5, A) SCLXM1, SCLXN1
WRITE (5, A) SCLXM2, SCLXN2
91 CONTINUE
C IF (SCLFA1 .GT. 0.0) GO TO 1911
ISCXM1 = ISCXM1 + 1
SCLFA1 = SCLFA1 * 10.
IF (ABS (SCLFA1) .LT. 1.0) GO TO 91
1911 CONTINUE
ISCXM2 = ISCXM2 + 1
SCLFA2 = SCLFA2 * 10.
IF (ABS (SCLFA2) .LT. 1.0) GO TO 91
C ISCXM1 = ISCXM1 .LT. 1.0 .GO TO 1911
C SCLFA1 = SCLFA1 .LT. .5
C SCLFA2 = SCLFA2 .LT. .5
C SCLFA1 = ANINT (SCLFA1)
C SCLFA2 = ANINT (SCLFA2)
C ISCXM1 = ISCXM1 / 10.
C ISCXM2 = ISCXM2 / 10.
C ISCXM1 = ISCXM1 * 10.
C ISCXM2 = ISCXM2 * 10.
C ISCLFC = JNINT (SCLFAC)
LOGX1 = - ISCXM1
LOGY1 = - ISCXM1
LOGX2 = - ISCXM2
LOGY2 = - ISCXM2
XMINS1 = - SCLFA1
XMINS1 = - SCLFA1
YMAXS1 = SCLFA1
YMAXS1 = SCLFA1
XMINS2 = SCLFA2
XMINS2 = SCLFA2
YMAXS2 = SCLFA2
YMAXS2 = SCLFA2
RXIAV2 (1, 2) = 0.0
CXIAV2 (1, 2) = SCLFA1 * 10. ** LOGX1
RXIAV2 (2, 2) = 0.0
GX1AV2(2,2)=-SCLFA1*10.**L0GX1
RX2AV4(1,2)=0.0
GX2AV4(1,2)=SCLFA2*10.**L0GX2
RX2AV4(2,2)=0.0
GX2AV4(2,2)=-SCLFA2*10.**L0GX2
CALL PAUSE(0,1)
LSAV1=(LSAV1-1)*10
LSAV2=(LSAV2-1)*10
IF(IDOM1.EQ.1)WRITE(5,5000)LSAV1
5000 FORMAT(' HBAR IS NONDOMINANT IN THE FIRST ROW AT ', I4)
IF(IDOM2.EQ.1)WRITE(5,5001)LSAV2
5001 FORMAT(' HBAR IS NONDOMINANT IN THE SECOND ROW', I4)
LSAV1=(LSAV1-1)*10
LSAV2=(LSAV2-1)*10
IF(IDOM1.EQ.1)WRITE(5,5000)LSAV1
IF(IDOM2.EQ.1)WRITE(5,5001)LSAV2
WRITE(5,84)(K21,K22,V5,V6,KC21,KC4,KQ2,KQ4,WNP,ZP,WNS,ZS,
CF1,F2)
84   FORMAT(' K21=',F7.6,' K22=',F7.6,
     C ' V5=',F4.0,' V6=',F4.0,' KC21=',F5.4,
     C ' KC4=',F5.4,' KQ2=',F5.0,' KQ4=',F5.0,' WNP=',F4.0,
     C ' ZP=',F3.1,' ZS=',F3.1,' CF1=',F9.6,
     C ' F2=',F9.6,/) 
WRITE(5,189)(P11S,P11,P12S,P12,P21,P22,TAUS1,TAUS2) 
189 FORMAT(' P11S=',F8.5,' P11=',F8.5,' P12S=',F8.5,
     C ' P12=',F8.5,' P21=',F8.5,' P22=',F8.5,
     C ' TAUS1=',F8.5,' TAUS2=',F8.5)
CALL PAUSE(0,0)
C CALL GRAPHS(0,-1,11.,11.,.65,.4154,RX1AV4,GX1AV4,100,1,1,
C  ' ', ' X P 1 /AV4 ',1,XMINS1,XMAXS1,YMINS1,YMAXS1,LOGX1,LOGY1,
C  CNFONT1, CLINE,CHAR,IERR)
C CALL VWPORT(0.,1.,0.,0.76171875)
C CALL WINDOW(XMINS1*10.**LOGX1,XMAXS1*10.**LOGX1,
C  CYMINS1*10.**LOGY1, CYMAXS1*10.**LOGY1)
C CALL PAUSE(0,0)
C CALL GRAPHS(0,-1,11.,11.,.15,.0346,RX2AV2,GX2AV2,100,2,2,
C  ' ', ' X P 2 /AV2 ',1,XMINS2,XMAXS2,YMINS2,YMAXS2,LOGX2,LOGY2,
C  CNFONT2, CLINE,CHAR,IERR)
C CALL PAUSE(0,0)
C IF(IDOM1.EQ.1)WRITE(5,5000)LSAV1
C IF(IDOM2.EQ.1)WRITE(5,5001)LSAV2
IF (NPLTPTN.EQ.0) GO TO 201
DO 111 I=1,NPLTPTN
J=I-1
IF ((ICOUNT2-J).EQ.0) GO TO 201
CALL POLYGM(X2AV4((ICOUNT2-J),1),Y2AV4((ICOUNT2-J),1),
CR2(ICOUNT2-J),0.0,360.,30)
111 CONTINUE
201 CALL PAUSE(0,1)
51 CONTINUE
WRITE(IFILE,84)(K21,K22,V6,KC21,KC4,KQ2,KQ4,WNP,ZP,WNS,ZS,
CF1,F2)
WRITE(5,84)(K21,K22,V6,KC21,KC4,KQ2,KQ4,WNP,ZP,WNS,ZS,
CF1,F2)
WRITE(IFILE,189)(P11S,P11,P12S,P12,P21,P21,P22TAUS1,TAUS2)
WRITE(5,52)
WRITE(IFILE,53)
WRITE(5,53)
52 FORMAT(' THE FOLLOWING IS QBAR11 ','/
WRITE(IFILE,53)
WRITE(5,53)
DO 54 1=1,NDPTS
WRITE(IFILE,55)FREQ(I),RX1A72(I,1),GX1A72(I,1),MAG1(I),RATIO1(I),
CR1(I)
WRITE(5,55)FREQ(I),RX1A72(I,1),GX1A72(I,1),MAG1(I),RATIO1(I),
CR1(I)
54 CONTINUE
55 FORMAT(1X,F6.1,3(1X,F14.8),3X,F7.3,2X,F14.8)
CALL PAUSE(0,1)
WRITE(IFILE,56)
WRITE(5,56)
56 FORMAT('1')
WRITE(IFILE,84)(K21,K22,V6,KC21,KC4,KQ2,KQ4,WNP,ZP,WNS,ZS,
CF1,F2)
WRITE(5,84)(K21,K22,V6,KC21,KC4,KQ2,KQ4,WNP,ZP,WNS,ZS,
CF1,F2)
WRITE(IFILE,57)
WRITE(5,57)
57 FORMAT(' THE FOLLOWING IS QBAR22 ','/
WRITE(IFILE,53)
WRITE(5,53)
DO 59 I=1,NDPTS
WRITE(IFILE,55)FREQ(I),RX2A74(I,1),GX2A74(I,1),MAG4(I),RATIO2(I),
CR2(I)
WRITE(5,55)FREQ(I),RX2A74(I,1),GX2A74(I,1),MAG4(I),RATIO2(I),
CR2(I)
59 CONTINUE
CALL PAUSE(0,1)
WRITE(5,5005)
5005 FORMAT(' ENTER 2 TO CONTINUE AND CLEAR PAGE')
READ(5,*)ICONT
IF (ICONT.EQ.2) GO TO 5002
CLOSE(UNIT=29)
STOP
END
Appendix C.
Prototype Control and Data
Acquisition/Load Control Program

C.1 Prototype Control Program-SBC 86/12A

MODULE CALIBRATE:

*INCLUDE("F4:FACTOR.SRC")
PROGRAM CALIBRATE:
LABEL 1:
CONST ENLINK=0B7H:

*THIS PROCEDURE CONTROLS THE FORWARD MOTION OF*
*THE ACTUATORS. IT IS INTERRUPT DRIVEN AT PRESET*
*SAMPLE TIMES*

PROCEDURE FORWARDCONTROL;
BEGIN
<LOAD INTERRUPT COUNTER AGAIN>

OUTBYT(0D0H,CHR(LOWCOUNTBYTE));
OUTBYT(0D0H,CHR(HIGHCOUNTBYTE));

<IF READOUT IS TRUE THEN READ DESIRED VELOCITY POSITION FROM A/D
CONVERTERS. IF READOUT FALSE, DESIRED VELOCITY OR POSITION IS
SUPPLIED BY PROGRAM.>
IF READOUT THEN
BEGIN
OUTBYTE(0F1H,CHR(OOH));
POSITION(IDEVEL1);
DEVEL1:=IDEVEL1/CONVERT; {CONVERT TO IN OR IN/SEC}
OUTBYTE(0F1H,CHR(01H));
POSITION(IDEVEL2);
DEVEL2:=IDEVEL2/CONVERT;
AV20PN:=REXZERO;
AV40PN:=REXZERO
END
ELSE BEGIN
IF((LOCATION1R >= STARTR) OR (LOCATION2R >= STARTR)) THEN
BEGIN
DEVEL1:=XP1VEL+PRTDV1;
DEVEL2:=XP2VEL+PRTDV2;
END
ELSE BEGIN
DEVEL1:=XP1VEL;
DEVEL2:=XP2VEL;
END
END:
IF DEVEL1 >= DEVEL2 THEN VELOUT:=DEVEL1*CONVERT1
ELSE VELOUT:=DEVEL2*CONVERT2;
PUMPOUT(OUTVOLT, VELOUT);

{OBTAIN ACTUAL VELOCITY OR POSITION}

OUTBYTE(0F1H,CHR(OOH));
POSITION(IXP1DOT);
XP1DOT:=IXP1DOT*VELCON1; {CONVERT TO IN/SEC OR INCHES}
OUTBYTE(0F1H,CHR(04H));
POSITION(IXP2DOT);
XP2DOT:=IXP2DOT*VELCON2; {CONVERT TO IN/SEC OR INCHES}
VELERR1 := DEVEL1 * XP1DOT;  
AV2PRT := -CONTR1 * VELERR1;  
VELERR2 := DEVEL2 * XP2DOT;  
AV4PRT := -CONTR2 * VELERR2;

(BRKFRQS := 1.0: FOR TIMING PURPOSES)

(LEAD FILTER SERVOVALVE)

IF BRKFRQS <> 0.0 THEN
BEGIN
  AV2SER := AV2SAV * ETB + AV2PRT * AB - ETBAUG * AV2SERSAV;
  AV2SAV := AV2SER;
  AV2SERSAV := AV2PRT;
  AV4SER := AV4SAV * ETB + AV4PRT * AB - ETBAUG * AV4SERSAV;
  AV4SAV := AV4SER;
  AV4SERSAV := AV4PRT;
END
ELSE
BEGIN
  AV2SER := AV2PRT;
  AV4SER := AV4PRT;
END;

(INTEGRAL CONTROLLER)

(TSAMPL2 := 0.0: FOR TIMING PURPOSES)

IF (TSAMPL2 <> 0.0) THEN
BEGIN
  AV2ING := (VELERR1 + VELERR1OLD) * TSAMPL2 + AV2ING;
  VELERR1OLD := VELERR1;
  AV4ING := (VELERR2 + VELERR2OLD) * TSAMPL2 + AV4ING;
  VELERR2OLD := VELERR2;
  IF (AV2ING < AV2EGNEG) THEN AV2ING := AV2EGNEG;
  IF (AV4ING < AV4EGNEG) THEN AV4ING := AV4EGNEG;
  IF (AV2ING > AV2EQ) THEN AV2ING := AV2EQ;
  IF (AV4ING > AV4EQ) THEN AV4ING := AV4EQ;
END
ELSE
BEGIN
  AV2ING := 0.0;
  AV4ING := 0.0;
END;

AV2SER := (AV2EG + AV2SER + AV2ING) / AMPSER;
AV4SER := (AV4EG + AV4SER + AV4ING) / AMPSER;

(OBTAIN DIFFERENTIAL CYLINDER PRESSURES)

OUTBYT(OFI.H.CHRI(08H));
POSITION(IDELTAPl);
DELTAP1 := (ABS(IDELTAPl) - OFFSETPRES1) * CONPRESS1;
OUTBYT(OFI.H.CHRI(09H));
POSITION(IDELTAP2);
DELTAP2 := (ABS(IDELTAP2) - OFFSETPRES2) * CONPRESS2;

(SELECT MAXIMUM DESIRED VELOCITY)

XPDMAX := DEVEL1;
IF (DEVEL2 <=)
{ALGORITHM TO CORRECT FOR DISSIMILAR DESIRED VELOCITIES}

IF DELTAVEL THEN BEGIN
  IF ((XP1ERR <> 0.0) AND (DELTAP1 > 0.0)) THEN
    AV2SER:=AV2SER+XP1ERR*APIS/(K*SQRT(DELTAP1)):
  IF ((XP2ERR <> 0.0) AND (DELTAP2 > 0.0)) THEN
    AV4SER:=AV4SER+XP2ERR*APIS/(K*SQRT(DELTAP2))
END;

{LIMIT SERVOVALVE OUTPUT COMMANDS TO PREVENT OVERRUNNING 12 BIT D/A}

IF((AV2SER < 0.0) OR (AV2SER > AVMAX)) THEN BEGIN
  IF AV2SER > AVMAX THEN AV2SER:=AVMAX;
  IF AV2SER < 0.0 THEN AV2SER:=0.0
END;
IF((AV4SER < 0.0) OR (AV4SER > AVMAX)) THEN BEGIN
  IF AV4SER > AVMAX THEN AV4SER:=AVMAX;
  IF AV4SER < 0.0 THEN AV4SER:=0.0
END;

IF((AV2PUM < 0.0) OR (AV2PUM > AVMAX)) THEN BEGIN
  IF AV2PUM > AVMAX THEN AV2PUM:=AVMAX;
  IF AV2PUM < 0.0 THEN AV2PUM:=0.0
END;
IF((AV4PUM < 0.0) OR (AV4PUM > AVMAX)) THEN BEGIN
  IF AV4PUM > AVMAX THEN AV4PUM:=AVMAX;
  IF AV4PUM < 0.0 THEN AV4PUM:=0.0
END;

{PUMP COMMAND}

QPCOM:=XPDMAK*APIS;
IF (DELTAP1 > 0.0) THEN QPCOM:=QPCOM+(1.0B7*AV2EQ-AV2PUM)*K*SQRT(DELTAP1)/2.0;
IF (DELTAP2 > 0.0) THEN QPCOM:=QPCOM+(1.0B7*AV4EQ-AV4PUM)*K*SQRT(DELTAP2)/2.0;

{FILTER PUMP COMMAND}

IF BRKFRQ <> 0.0 THEN BEGIN
  QPCOMFIL:=GPSAV*ETP+QPCOM*AP-ETPAUG*QPCOMSAV;
  GPSAV:=QPCOMFIL;
OPCOMSAV:=QPCOM;
END

ELSE
QPCOMFIL:=QPCOM;

QPCOMFIL:=QPCOMFIL*CONPUMP;

<LIMIT PUMP COMMAND>
IF QPCOMFIL < REXZERO THEN QPCOMFIL:= REXZERO;
IF QPCOMFIL > PUMPMAK THEN QPCOMFIL:= PUMPMAK;
AV2SER:=AV2SER*CONSERVO1;
AV4SER:=AV4SER*CONSERVO2;
PUMPOUT(PUMP,QPCOMFIL);
PUMPOUT(SERVO1,AV2SER);
PUMPOUT(SERVO2,AV4SER);
OUTBYT(OC0H,CHR(20H))<RESET INTERRUPT>
END<OF PROCEDURE>

BEGIN

<MAIN PROGRAM>
ENABLEINTERRUPTS;
SIOOUT(ENDLINK):
OUTBYT(OC0H,CHR(13H))<ICW1 TO PPI>
OUTBYT(OC2H,CHR(20H))<ICW2 TO PPI>
OUTBYT(OC2H,CHR(ODH))<ICW4 TO PPI>
OUTBYT(OC2H,CHR(OFDH))<ENABLE INTERRUPT 0 (OR 1 ? )>
OUTBYT(OC0H,CHR(BBH)): <TO PARALLEL PORTS>
READOUT:=FALSE: <READOUT RECEIVES INPUT FROM EXTERNAL SOURCES>
DELTAVEL:=FALSE: <DELTAVEL IS FOR DISSIMILAR VELOCITY INPUTS>
FORIFLAG:=FALSE: <STARTS ACTUATORS IN FORWARD STRIDE>
FOR2FLAG:=FALSE;
FIRSTTIME:=FALSE;
OUTBYT(OF0H,CHR(01H)): <ENABLE A/D CONVERSION>

<DETERMINE ABSOLUTE POSITION OF PISTONS>
I:= DISABLEINTERRUPTS;
OUTBYT(OFH,CHR(02H));
POSITION<LOCATION1>: <READ ACTUATOR 1 POSITION>
LOCATION1:=<LOCATION1-OFFSET1>*CONFAC1: <CONVERT TO INCHES>
IF LOCATION1 >= MAXPOS1 THEN FORIFLAG:=FALSE: <SETS DIRECTION>
IF LOCATION1 <= MINPOS1 THEN FORIFLAG:=TRUE;

OUTBYT(OF1H,CHR(03H));
POSITION<LOCATION2>: <READ POSITION OF ACTUATOR 2>
LOCATION2:=<LOCATION2-OFFSET2>*CONFAC2: <CONVERT TO INCHES>
IF LOCATION2 >= MAXPOS2 THEN FOR2FLAG:=FALSE: <SETS DIRECTION>
IF LOCATION2 <= MINPOS2 THEN FOR2FLAG:=TRUE;

<SEND COORDINATING SIGNALS THROUGH PARALLEL PORT TO DATA ACQUISITION LOAD CONTROLLER COMPUTER>

FORIFLAG = TRUE) AND (FOR2FLAG = TRUE) THEN
OUTBYT(OC0H,CHR(03H));
IF (FORIFLAG = FALSE) AND (FOR2FLAG = FALSE) THEN
OUTBYT(OC0H,CHR(00H));
IF (FORIFLAG =TRUE) AND (FOR2FLAG = FALSE) THEN

IF (FORFLAG = FALSE) AND (FOR2FLAG = TRUE) THEN

OUTBYT(OCBH, CHR(02H));

<IF BOTH PISTONS ARE NOT TRAVELING IN THE SAME DIRECTION>

IF NOT(FORFLAG AND FOR2FLAG) THEN

BEGIN

OUTBYT(OC2H, CHR(0FFH));

<RESET ALL INTERRUPT VECTORS>
OUTBYT(OCOH, CHR(20H));

DISABLE INTERRUPTS;
PUMPOUT(OUTVOLT, REXZERO);

<RESET PIC OF ANY RESIDUAL INTERRUPTS>

FIRSTTIME:= TRUE;
PUMPOUT(PUMP, PUMPEG): <SET PUMP AT A SLOW PACE>

IF FORFLAG THEN

BEGIN
IF ((LOCATION2R >= MAXPOS29) AND (LOCATION1R <= MINPOS111)) THEN

BEGIN
PUMPOUT(PROP1, REXZERO);
PUMPOUT(PROP2, MAXNEG);
PUMPOUT(SERVO1, AVBYEQ1);
PUMPOUT(SERVO2, AVBYEQ2);
END
ELSE
BEGIN
IF LOCATION2R >= MAXPOS29 THEN

BEGIN
PUMPOUT(PROP1, MAXPOS);
PUMPOUT(PROP2, REXZERO);
PUMPOUT(SERVO2, AVBYEQ2);
END
POSITION(ICONTR1);
CONTR1:= ICONTR1*VELCALIB;
OUTBYT(OF1H, CHR(05H));
POSITION(IXPIDOT);
XPIDOT:= IXPIDOT*VELCON1: <CONVERT TO IN/SEC OR INCHES>
VELERR1:= DEVEL1-XPIDOT;
AV2PRT:= CONTR1*VELERR1;
AVZSER:= AV2PRT+AVZSER;
AVZSER:= AVZSER/AMPSER;

IF (AVZSER > AVMAX) THEN AVZSER:= AVMAX;
IF (AVZSER < REXZERO) THEN AVZSER:= REXZERO;
AVZSER:= AVZSER*CONSERVO;

PUMPOUT(SERVO1, AVZSER)
END;
IF LOCATION1R <= MINPOS111 THEN

BEGIN
PUMPOUT(PROP1, REXZERO);
PUMPOUT(PROP2, MAXNEG);
PUMPOUT(SERVO1, AVBYEQ1);
PUMPOUT(SERVO2, AVBYEQ2)
END
END
END

IF FOR2FLAG THEN

BEGIN
IF ((LOCATION1R >= MAXPOS19) AND (LOCATION2R <= MINPOS211)) THEN

BEGIN
PUMPOUT(PROP2, REXZERO);

IF BOTH PISTONS ARE IN RETURN PHASE

{IF BOTH PISTONS ARE IN RETURN PHASE}

IF ((NOT FOR1FLAG) AND (NOT FOR2FLAG)) THEN
BEGIN
  PUMPOUT(PROP1,MAXNEG);
  PUMPOUT(PROP2,MAXNEG);
  PUMPOUT(SERVO1,AVBYEQ1);
  PUMPOUT(SERVO2,AVBYEQ2);
END;

{BOTH ACTUATORS ARE IN FORWARD STRIDE}

IF (FOR1FLAG AND FOR2FLAG) THEN
BEGIN
  PUMPOUT(PROP1,MAXPOS);
  PUMPOUT(PROP2,MAXPOS);
  ENABLEINTERRUPTS;

  IF FIRSTTIME THEN
  BEGIN
    ... (FIRST SAMPLE OF FORWARD STROKE READ GAINS FOR ALL CONTROLLERS AND RESET INTEGRAL CONTROLLER AND SET INITIAL CONDITIONS FOR FILTERS)...
  END;

END;

... (remaining code) ...
OUTBYTE(OCZH.CHROFDH)); {ENABLE INTERRUPT 1 AGAIN}
AV2SAV:=AV2EQ;
AV4SAV:=AV4EQ;
AV2SERSAV:=AV2EQ;
AV4SERSAV:=AV4EQ;
VELERRZOLD:=0.0;
VELERRZOLD:=0.0;
AV2ING:=0.0;
AV4ING:=0.0;
QPSAV:=PUMPEQ*CONVERTPUMP;
QCPMSAV:=PUMPEQ*CONVERTPUMP;
OUTBYTE(OF1H.CHROBH));
POSITION(INP1);
CONTR1:=IC0NTR1*VELCALIB;
OUTBYTE(OF1H.CHROBH));
POSITION(INP1);
CONTR2:=IC0NTR2*VELCALIB;
OUTBYTE(OF1H.CHROBH));
POSITION(INP1);
CONTR2:=IC0NTR2*VELCALIB;
OUTBYTE(OF1H.CHROBH));
POSITION(INP1);
CONTR2:=IC0NTR2*VELCALIB;
OUTBYTE(OF1H.CHROBH));
POSITION(INP1);
CONTR2:=IC0NTR2*VELCALIB;
END;

PUBLIC CALIBRATE;
CONST
PROP1=0E0H: {ADDRESSES OF REXROTH AND BYPASS VALVES}
PROP2=0E2H;
SERVO1=0E4H;
SERVO2=0E6H;
OUTVOLT=0FAH;
PUMP=0FBH: {ADDRESS OF PUMP}

{LIMITING VALUES}
PUMPMAX=4094.0: {MAXIMUM PUMP OUTPUT IN A/D UNITS}
MAXPOS=2046.0: {PLUS 10 VOLTS TO REXROTH}
MAXNEG=-2046.0: {-10 VOLTS TO REXROTH}
REXZERO=0.0;
AVMAX=0.04: {MAXIMUM SERVOVALUE AREA IN AMPS---.04*1.887}
CONVERTPUMP=0.03:(123/4096, FOR PUMP FILTER}

{POSITION INFORMATION--- ALL UNITS IN INCHES}
MAXPOS1=6.5;
MAXPOS2=6.5;  
OFFSET1=1120.0;  
OFFSET2=1120.0;  
CONFAC1=0.007816;  
CONFAC2=0.007816;  
MINPOS1=0.0;  
MINPOS2=0.0;  
STARTR=3.0;  
MAXPOS2=5.85;  
MINPOS2=0.0;  
MAXPOS1=5.85;  
MINPOS1=0.0;  
STARTR=3.0;  
MINPOS1=0.0;  
MINPOS2=0.0;  
MAXPOS1=5.85;  
MAXPOS2=5.85;  
ENDR=9.0;  
END POSITION INFORMATION

XP1VEL=3.0;  
XP2VEL=3.0;  
PRTDV1=6.0;  
PRTDV2=6.0;  
VELCON1=0.00835;  
VELCON2=0.00635;  
AV2EQ=0.0308;  
AV2EQNEG=-0.0308;  
AV4EQ=0.03774;  
AV4EQNEG=-0.03774;

< DELTAVEL=TRUE! DELTAVEL TRUE INCLUDES BYPASS VALVE CHANGE FOR DISSIMILAR VELOCITIES>

TSAMPL=0.007;  
CONVERT_TIME=-0.000001;  
CONVERT_TAU=0.0000027;  
VELCAL1=0.000005;  
CHANNEL 6,7

< TSAMPL2=1.0; SAMPLING TIME/2 ** INTEGRAL GAIN>

CONPRES1=3.04;  
OFFSETPRES1=1431.0;  
CONPRES2=3.068;  
OFFSETPRES2=1441.0;  
AMPSER=1.887;  
APIS=4.1;

CONPUMP=33.3;  
CONSERVO1=70086.4;  
CONSERVO2=74926.0;  
AVBYEQ1=1184.7;  
AVBYEQ2=1498.5;

PUMPEG=818.0;  
K=97.67;  
CONVERT=204.7;  
converts a/d into volts--1 volt=1 inch desired position or 1 inch/sec desired velocity.
PROCEDURE forward control:

PUBLIC POSITZ:
PROCEDURE POSITION(VAR DUMMY:INTEGER):
PUBLIC PPC0N2:
PROCEDURE PUMPOUT(OUTPORT:INTEGER:DUMMYR:REAL):
PUBLIC ASSROU:
PROCEDURE S I O O U t d : INTEGER):
PROCEDURE S INCHK(VAR ICHAR:INTEGER):NAME POSITZ

DSA STRUCT
OLD_BP DU ?
RETURN DD ?
DATAPTR DD ?

DSA ENDS
CODE SEGMENT PUBLIC 'CODE'
ASSUME CS:CODE
PUBLIC POSITION
POSITION PROC FAR
PUSH BP
MOV BP,SP
LES BX.[BP],DATAPTR
IN AX,0F5H
AGAIN:IN AL,0FOH
AND AL,08H
JZ AGAIN
MOV AL,0F4H
MOV CL,AL
IN AL,0F5H
MOV CH,AL
MOV AX,CX
MOV CL,4
SAR AX,CL
MOV ES:[BX],AX
POP BP
; ASSROU
; INCLUDE 'WIN.COM'

LOCAL_DATA SEGMENT PUBLIC 'DATA'
TEMPSTORE DW ?
LOCAL_DATA ENDS

CODE SEGMENT PUBLIC 'CODE'
ASSUME CS:CODE, DS:LOCAL_DATA
PUBLIC PUMPOUT
PUMPOUT PROC FAR
PUSH DS
MOV AX, LOCAL_DATA
MOV DS, AX
PUSH BP
MOV BP, SP
MOV DX,[BP].VALUE
FRNDINT
FWAIT
FISTP TEMPSTORE
FWAIT
MOV AX, TEMPSTORE
MOV CL, 4
SAL AX, CL
OUT DX, AL
INC DX
MOV AL, AH
OUT DX, AL
POP BP
POP DS
RET 2
PUMPOUT ENDP
CODE ENDS
END

NAME ASSROU
ASSUME CS:SIODE, CODE
STATUS_PORT EQU ODAH
DATA_PORT EQU ODBH
TXRDY EQU 0IH
RXRDY EQU 02H

DSA STRUC
OLD_BP DW ?
RETURN_ADDR DD ?
ICHAR_VAR DD ?
DSA ENDS

SIODE SEGMENT PUBLIC 'CODE'
PUBLIC SIODE, SINCHK
SIODE PROC FAR
POP AX
POP BX
POP CX
PUSH BX
PUSH AX
TEST _TXRDY; IN AL, STATUS_PORT
TEST AL, TXRDY
J2 TEST_TXRDY
MOV AL, CL
OUT DATA_PORT, AL

LOCAL_DATA SEGMENT PUBLIC 'DATA'
TEMPSTORE DW ?
LOCAL_DATA ENDS

CODE SEGMENT PUBLIC 'CODE'
ASSUME CS:CODE, DS:LOCAL_DATA
PUBLIC PUMPOUT
PUMPOUT PROC FAR
PUSH DS
MOV AX, LOCAL_DATA
MOV DS, AX
PUSH BP
MOV BP, SP
MOV DX,[BP].VALUE
FRNDINT
FWAIT
FISTP TEMPSTORE
FWAIT
MOV AX, TEMPSTORE
MOV CL, 4
SAL AX, CL
OUT DX, AL
INC DX
MOV AL, AH
OUT DX, AL
POP BP
POP DS
RET 2
PUMPOUT ENDP
CODE ENDS
END

NAME ASSROU
ASSUME CS:SIODE, CODE
STATUS_PORT EQU ODAH
DATA_PORT EQU ODBH
TXRDY EQU 0IH
RXRDY EQU 02H

DSA STRUC
OLD_BP DW ?
RETURN_ADDR DD ?
ICHAR_VAR DD ?
DSA ENDS

SIODE SEGMENT PUBLIC 'CODE'
PUBLIC SIODE, SINCHK
SIODE PROC FAR
POP AX
POP BX
POP CX
PUSH BX
PUSH AX
TEST _TXRDY; IN AL, STATUS_PORT
TEST AL, TXRDY
J2 TEST_TXRDY
MOV AL, CL
OUT DATA_PORT, AL
RET
SIOOUT ENDP

SINCHK PROC FAR
PUSH BP
MOV BP,SP
LES BX,[BP].ICHAR_VAR
IN AL,STATUS_PORT
TEST AL,RXRDY
JZ TEST.RXRDY
IN AL,DATA_PORT
MOV AH,01H
JMP TE1
TEST.RXRDY:MOV AX,00H
TE1:MOV ES:[BX].AX
POP BP
RET 4
SINCHK ENDP
SIO.CODE ENDS
END
C.2 Data Acquisition/Load Control Program-SBC 86/12A

MODULE INIT2;
*INCLUDE(:F4:DATA2.SRC)
PROGRAM INIT2;
CONST ENDLINK=0B7H:
PROCEDURE MAINCONTROL:
{ THIS PROCEDURE CONTROLS LOAD BEFORE DATA ACQUISITION}

LABEL 1;
CONST STARTDATA=1472.0:
VAR STARTDATATRUE:BOOLEAN;
STARTDATASIGN:INTEGER;
BEGIN
LOADSHAR:=FALSE:
STOPDATATRUE:=TRUE:
STARTDATATRUE:=FALSE:
OUTBYT(0C0H,CHR(13H));
OUTBYT(0C2H,CHR(20H));
OUTBYT(0C2H,CHR(0DH));
OUTBYT(0C2H,CHR(0FH));
OUTBYT(0CEH,CHR(9BH));
1: DISABLEINTERRUPTS:
{DETERMINE POSITIONS OF THE PISTONS}
OUTBYT(0F1H,CHR(POSICHAN));
OUTBYT(0FH,CHR(01H));
POSITION(POSITION1);
OUTBYT(0F1H,CHR(POS2CHAN));
OUTBYT(0FH,CHR(01H));
POSITION(POSITION2);
OUTBYT(0F1H,CHR(0DH));
OUTBYT(0FH,CHR(01H));
<READ DESIRED LOADS>

POSITION(FORCE1):
LOADSTATIC1:=FORCE1*4.0;
OUTBYTE(OF1H::CHR(10H)));
OUTBYTE(OF0H::CHR(01H));
POSITION(FORCE2):
LOADSTATIC2:=FORCE2*4.0;
OUTBYTE(OF1H::CHR(0EH)));
OUTBYTE(OF0H::CHR(01H));
POSITION(STARTDATASIGN):
IF (STARTDATASIGN > 204) THEN STARTDATATRUE:=TRUE;
OUTBYTE(OF1H::CHR(OFH)));
OUTBYTE(OF0H::CHR(01H));
POSITION(IDIVISOR):
DIVISOR:=IDIVISOR/204.7;
IF (IDIVISOR > 150) THEN LOADSHARE:=TRUE;
ENABLEINTERRUPTS:
INBYTE(OCBH::DIREC1):
INBYTE(OCCH::DIREC2):
IF ((DIREC1=TRUE) AND (DIREC2=FALSE)) THEN
ENDIF
BEGIN
IF (((POSITION1 > STARTDATA) OR (POSITION2 > STARTDATA)) AND
STARTDATATRUE) THEN CAUSEinterrupt(32):
IF (POSITION1 < STARTLOAD) THEN LOADCOM1:=(POSITIONI-OFFSET)/(STARTLOAD-OFFSET)*LOAD:
ELSE
IF ((LOADSHARE) AND (POSITION1 > STARTSIN)) THEN LOADCOM1:=LOADSTATIC1+SIN((POSITION1
-STARTSIN)/(MAXPOS1-STARTSIN)*3.14)*LOADSTATIC1/DIV
ELSE
IF (POSITION1 > STARTLOAD) THEN LOADCOM1:=LOADSTATI
ELSE LOADCOM1:=0.0;


LOADCOM2:=(POSITION2-OFFSET)/(STARTLOAD-OFFSET)*LOADSTATIC2
ELSE
    IF (LOADSHAR) AND (POSITION2 > STARTSIN) THEN
      LOADCOM2:=LOADSTATIC2+sin((POSITION2-STARTSIN)/(MAXPOS2-STARTSIN)*3.14)*LOADSTATIC2/DIV
    ELSE
      LOADCOM2:=LOADSTATIC2
    END
ELSE
    IF (POSITION2 > STARTLOAD) THEN LOADCOM2:=LOADSTATIC2
    ELSE LOADCOM2:=0.0
END
ELSE
    BEGIN
      LOADCOM1:=0.0;
      LOADCOM2:=0.0
    END;
    IF (POSITION2 > STARTLOAD) THEN LOADCOM2:=LOADSTATIC2
    ELSE LOADCOM2:=0.0
    IF LOADCOM1 > LIMITN THEN LOADCOM1 := LIMITN;
    IF LOADCOM1 < ZERO THEN LOADCOM1 := ZERO;
    IF LOADCOM2 > LIMITN THEN LOADCOM2 := LIMITN;
    IF LOADCOM2 < ZERO THEN LOADCOM2 := ZERO;
    PUMPOUT(FORCEPORT1,LOADCOM1);
    PUMPOUT(FORCEPORT2,LOADCOM2);
    GOTO 1;
END; {OF PROCEDURE}
BEGIN {MAIN PROGRAM}
  SIOOUT(ENDLINK); {END LINK TO MDS}
  MAINCONTROL
END.
PUBLIC INITZ:
{ THIS FILE HOLDS THE GLOBAL VARIABLES FOR THE PROGRAM }
  VAR LOADSHAR,STOPDATATRUE:BOOLEAN;
  PROCEDURE MAINCONTROL;
PUBLIC DATAG:3
CONST FORCEPORT1=OFBH;
FORCEPORT2=OFAH;
ZEROCOUNTBYTE=00H;
LOADCOUNTBYTE=006H; {70H}
HIGHCOUNTBYTE=001H; {03H}
NUMCHANNELS=13;
MAXNUM=2420;
MAXINTS=30600;
POSICHAN=2;
POSICHAN=3;
LIMITN=4095.0;ZERO=0.0;
STARTLOAD=1218.0;(STARTLOAD(IN)*VOLTS/IN*INTEG/VOLT+OFFSET)
STARTSIN=1504.0,(START OF SINUSIODAL LOAD)
OFFSET=1120.0;
{ LOADSTATIC1=1392.0;}{(FORCE(LBF)*VOLTS/LBF*INTEG/VOLTS)
MAXPOS1=1951.6;
{ SINF0R1=2048.0;}{(FORCE(LBF)*VOLTS/LBF*INTEG/VOLTS)
SINF0R1=1.0;
{ LOADSTATIC2=2785.4;}
MAXPOS2=1951.6;
{ SINF0R2=2048.0;}
SINF0R2=-0.5; {sinf0r2=sinf0r2(-1)*.5 for divisor}
STOPDATAP0S=1351.5;{8.5 INCHES FROM BEGINNING OF STROKE}
VAR POSITION1,POSITION2,SCANNUM,FORCE1,FORCE2,LOADSHARTRUE,
DIVISOR:INTEGER;
{POSITION1=OFFSET A/D *INCHES*VOLTS/IN*INTEG/VOLT}
{DIVISOR:LOADCOM1,LOADCOM2,LOADSTATIC1,LOADSTATIC2:REAL;
DIREC1,DIREC2:BOOLEAN,
PUBLIC TAKE2:
VAR DUMMY:INTEGER;
PUBLIC PPCT2:
PROCEDURE PUMPOUT(PORT:INTEGER; DUMMY:REAL):
PUBLIC POSI2:
PROCEDURE POSITION(VAR DUMMY:INTEGER):
PUBLIC DAT2:
PROCEDURE DATTRAN:
PUBLIC ASSRO:
PROCEDURE SIOOUT(I:INTEGER):
PROCEDURE SINCHK(VAR I:CHAR; INTEGER):

{THIS PROGRAM CONTROLS LOADS AND ACQUIRES DATA}

PRIVATE DATAQZ:
PROCEDURE STOPDATA:
BEGIN
OUTBYTE(OC2H, CHR(0FH));
OUTBYTE(OD8H, CHR(38H));
OUTBYTE(ODOH, CHR(ZEROCOUNTBYTE));
LOADCOM:=0.01;
PUMPOUT(FORCEPORT1, LOADCOM1);
PUMPOUT(FORCEPORT2, LOADCOM1);
DATTRAN;
END;
PROCEDURE FIRSTTIME:
LABEL 2:
BEGIN
OUTBYTE(OC2H, CHR(0FH));
OUTBYTE(OCOH, CHR(20H));
ENABLEINTERRUPTS;
SCANNUM:=0;
OUTBYTE(OD8H, CHR(34H));
OUTBYTE(OD0H, CHR(LOWCOUNTBYTE));
OUTBYTE(OD8H, CHR(HIGHCOUNTBYTE));
CAUSEINTERRUPT(33):
{33 IS DATA ACQUISITION ROUTINE--TAKEQ2A}
2: IF SCANNUM >= MAXNUM THEN STOPDATA;
DISABLEINTERRUPTS;
OUTBYTE(OF1H, CHR(POS1CHAN));
OUTBYTE(OF0H, CHR(01H));
POSITION(POSITION1);
OUTBYTE(OF1H, CHR(POS2CHAN));
OUTBYTE(OF0H, CHR(01H));
POSITION(POSITION2);
OUTBYTE(OF1H, CHR(0DH));
OUTBYTE(OF0H, CHR(01H));
POSITION(FORCE1);
OUTBYTE(OF1H, CHR(0DH));
OUTBYTE(OF0H, CHR(01H));
POSITION(FORCE2);
OUTBYTE(OF1H, CHR(0EH));
OUTBYTE(OF0H, CHR(01H));
POSITION(LoadSHARETRUE);
IF (LoadSHARETRUE > 204) THEN LOADSHARE:=TRUE;
OUTBYTE(OF1H, CHR(OFH));
OUTBYTE(OF0H, CHR(01H));
POSITION(IDIVISOR);
LOADSTATIC1 := FORCE1 * 4.0;
LOADSTATIC2 := FORCE2 * 4.0;
DIVISOR := IDIVISOR / 204.7;
IF (IDIVISOR > 150) THEN LOADSHAR := TRUE;

INBYT(OC8H, DIREC1) := INPUT BIT FROM FIRST CYLINDER;
INBYT(OC8H, DIREC2) := INPUT BIT FROM SECOND CYLINDER;
IF ((DIREC1 = TRUE) AND (DIREC2 = FALSE)) THEN
  < PORT CC READS OPPOSITE-A HIGH IS FALSE >
BEGIN
  IF (POSITION1 < STARTLOAD) THEN
    LOADCOM1 := (POSITION1 - OFFSET) / (STARTLOAD - OFFSET) * LOAD
  1817.6
  ELSE
    IF ((LOADSHAR) AND (POSITION1 > STARTSIN)) THEN
      LOADCOM1 := LOADSTATIC1 + SIN((POSITION1
- STARTSIN) / (MAXPOS1 - STARTSIN) * 3.14) * LOADSTATIC1 / DIV
    ELSE
      IF (POSITION1 > STARTLOAD) THEN LOADCOM1 := LOADSTATIC1
      ELSE LOADCOM1 := 0.0;
  END
  IF (POSITION2 < STARTLOAD) THEN
    LOADCOM2 := (POSITION2 - OFFSET) / (STARTLOAD - OFFSET) * LOAD
  1818.6
  ELSE
    IF ((LOADSHAR) AND (POSITION2 > STARTSIN)) THEN
      LOADCOM2 := LOADSTATIC2 + SIN((POSITION2
- STARTSIN) / (MAXPOS2 - STARTSIN) * 3.14) * LOADSTATIC2 / DIV
    ELSE
      IF (POSITION2 > STARTLOAD) THEN LOADCOM2 := LOADSTATIC2
      ELSE LOADCOM2 := 0.0;
  END
  IF ((POSITION1 > STOPDATAPOS) OR (POSITION2 > STOPDATAPOS))
    AND STOPDATATRUE) THEN STOPDATA;
END
ELSE
BEGIN
  IF STOPDATATRUE THEN
    BEGIN
      OUTBYT(OCZH, CHR(OFFH));
      STOPDATA
    END;
  LOADCOM1 := 0.0;
  LOADCOM2 := 0.0;
END:
ENDIF LOADCOM1 > LIMIT THEN LOADCOM1 := LIMIT;
ENDIF LOADCOM1 < ZERO THEN LOADCOM1 := ZERO;
ENDIF LOADCOM2 > LIMIT THEN LOADCOM2 := LIMIT;
ENDIF LOADCOM2 < ZERO THEN LOADCOM2 := ZERO;
PUMPOUT(FORCEPORT1, LOADCOM1);
PUMPOUT(FORCEPORT2, LOADCOM2);
GOTO 2
END:

MODULE TAKE2:
{ THIS PROCEDURE ACQUIRES THE DATA }
PRIVATE TAKEZ;
PROCEDURE TAKEDATAZ;
VAR j, i: INTEGER;
BEGIN
OUTBYT(OF1H, CHR(OOH));
OUTBYT(OF2H, CHR(NUMCHANNELS));
OUTBYT(OF6H, CHR(03H));
J := SCANNUM*NUMCHANNELS;
FOR I := 1 TO NUMCHANNELS DO BEGIN
  J := J + 1;
  POSITION(DUMMY);
  DATAWORD(J) := DUMMY;
END;
SCANNUM := SCANNUM + 1;
OUTBYT(OC0H, CHR(61H))
END;
MODULE DATATZ;
{ THIS PROCEDURE SHIPS BINARY DATA TO MDS }
*INCLUDE (:F4:DATAZ.SRC)
PRIVATE DATATZ:
PROCEDURE DATATRAN;
CONST iRS = n;
VAR
INCHAN: BOOLEAN;
INDATA, HIGHBYTE, LOWBYTE, I1, IZ: INTEGER;
PROCEDURE KEYCH(VAR INCHA: BOOLEAN; VAR DATA: INTEGER);
VAR
I1: INTEGER;
BEGIN
SINCHK(I1);
INCHA := FALSE;
DATA := 0;
IF I1 > 255 THEN BEGIN
  DATA := I1 - 255;
  INCHA := TRUE;
END;
BEGIN
  KEYCH(INCHAN, INDATA);
  WHILE NOT (INCHAN OR (INDATA <> iRS)) DO
    WHILE NOT INCHAN DO
      KEYCH(INCHAN, INDATA);
      HIGHBYTE := SCANNUM DIV 256;
      LOWBYTE := SCANNUM - HIGHBYTE*256;
      SIOOUT(LOWBYTE);
      SIOOUT(HIGHBYTE);
      FOR I1 := 1 TO SCANNUM DO
        BEGIN
          J := (I1 - 1)*NUMCHANNELS;
          KEYCH(INCHAN, INDATA);
          WHILE NOT (INCHAN OR (INDATA <> iRS)) DO
            KEYCH(INCHAN, INDATA);
            FOR IZ := 1 TO NUMCHANNELS DO
              BEGIN
                DATAINT := DATAWORD(J + IZ);
                HIGHBYTE := ABS(DATAINT) DIV 256;
                LOWBYTE := ABS(DATAINT) - HIGHBYTE*256;
                IF DATAINT < 0 THEN HIGHBYTE := HIGHBYTE + 128;
                SIOOUT(LOWBYTE);
                SIOOUT(HIGHBYTE);
              END;
            END;
          END;
        END;
      END;
      MAINCONTROL (CAUSE INTERRUPT(3))
    END;
  END;}
NAME ASSPOU
ASSUME CS:SIO_CODE
STATUS_PORT EQU ODAH
DATA_PORT EQU ODBH
TXRDY EQU O1H
RXRDY EQU O2H

DSA STRUC
OLD_BP DW ?
RETURN_ADDR DD ?
ICHAR_VAR DD ?
DSA ENDS

SIO_CODE SEGMENT PUBLIC 'CODE'
PUBLIC SIOOUT,SINCHK
SIOOUT PROC FAR
POP AX
POP BX
POP CX
PUSH BX
PUSH AX
PUSH AX
TEST_TXRDY:IN AL,STATUS_PORT
TEST_AL_TXRDY
JZ TEST_TXRDY
MOV AL,CL
OUT DATA_PORT,AL
RET
SIOOUT ENDP

SINCHK PROC FAR
PUSH BP
MOV BP,SP
LES BX,[BP].ICHAR_VAR
IN AL,STATUS_PORT
TEST_AL_RXRDY
JZ TEST_RXRDY
IN AL,DATA_PORT
MOV AH,01H
JMP TE1
TEST_RXRDY:MOV AX,00H
TE1:MOV ES:[BX].AX
POP BP
RET 4
SINCHK ENDP
SIO_CODE ENDS
END
NAME POSIZ2
DSA STRUC
OLD_BP DW ?
RETURN DD ?
DATAPTR DD ?
DSA ENDS
CODE SEGMENT PUBLIC 'CODE'
ASSUME CS:CODE
PUBLIC POSITION
POSITION PROC FAR
PUSH BP
MOV BP,SP
LES BX.[BP].DATAPTR
AGAIN:IN AL,0FOH
AND AL,0B0H
JZ AGAIN
MOV AL,0FAH
MOV CL,AL
IN AL,0FOH
MOV CH,AL
MOV CL,4
SAR AX,CL
MOV ES:[BX].AX
POP BP
RET 4
POSITION ENDP
CODE ENDS
END
NAME PPC0N2
DSA STRUC
OLD_BP DW ?
RETURN_ADDR DD ?
ICHAR_VAR DD ?
DSA ENDS
LOCAL_DATA SEGMENT PUBLIC 'DATA'
TEMPSTORE DW ?
LOCAL_DATA ENDS
CODE SEGMENT PUBLIC 'CODE'
ASSUME CS:CODE, DS:LOCAL_DATA
PUBLIC PUMPOUT
PUMPOUT PROC FAR
PUSH DS
PUSH BP
MOV AX,LOCAL_DATA
MOV DS:AX
PUSH BP
MOV BP,SP
MOV DX:[BP].VALUE
FANDINT
FWAIT
FISTP TEMPSTORE
FWAIT
MOV AX,TEMPSTORE
MOV CL,4
SAL AX,CL
OUT DX,AL
INC DX
MOV AL,0AH
OUT DX,AL
POP BP
POP DS
RET 2
PUMPOUT ENDP
CODE ENDS
END
NAME PPC0N2
DSA STRUC
OLD_BP DW ?
OLD_DS DW ?
RETURN DD ?
VALUE DW ?
DSA ENDS
LOCAL_DATA SEGMENT PUBLIC 'DATA'
TEMPSTORE DW ?
LOCAL_DATA ENDS
CODE SEGMENT PUBLIC 'CODE'
ASSUME CS:CODE, DS:LOCAL_DATA
PUBLIC PUMPOUT
PUMPOUT PROC FAR
PUSH DS
PUSH BP
MOV AX,LOCAL_DATA
MOV DS:AX
PUSH BP
MOV BP,SP
MOV DX:[BP].VALUE
FANDINT
FWAIT
FISTP TEMPSTORE
FWAIT
MOV AX,TEMPSTORE
MOV CL,4
SAL AX,CL
OUT DX,AL
INC DX
MOV AL,0AH
OUT DX,AL
POP BP
POP DS
RET 2
PUMPOUT ENDP
CODE ENDS
END
C.3 Data Transfer Programs, SBC to MDS -- MDS

MODULE SBC_TO_MDS:
* INCLUDE('F4;SGD10B.SRC');

PROGRAM SBC_TO_MDS(INPUT,OUTPUT):
< MAXRECORD= NUMBER OF SCANS
  MAXCHANNELS = NUMBER OF CHANNELS IN ONE SCAN >

LABEL 2:
VAR IDAT.IP, MAXRECORD, MAXCHANNEL, IOPINTEP:INTEGER;
FILENAME:PARY14;
AGAIN.ICH:BOOLEAN;
HALTCHAR:CHAR;

FUNCTION X(IMAX:INTEGER):DATAPINTER;
< FUNCTION X OBTAINS A SCAN AND PACKS THE CHANNEL DATA
IN PROPER NUMERIC FORM--THE TWO EIGHT BIT DATA FROM THE A/D
CONVERSION ARE READ IN AS
16 BIT INTEGERS SO TWO BYTES FROM ISBC ARE TEMP STORED AS TWO 16
BIT INTEGERS AND MUST BE COMBINED INTO ONE 16 BIT INTEGER>

VAR I, II, INDATA, ITEMI:INTEGER;
VALID-CHAR:BOOLEAN;
BUF:ARRAY[1..401] OF INTEGER:
BEGIN
  FOR I:=1 TO IMAX*2 DO BEGIN
    READ IN 2*NUMBER OF CHANNELS BYTES
    KEYCK(VALID-CHAR,INDATA);
    WHILE (NOT(VALID-CHAR)) DO KEYCK(VALID-CHAR,INDATA):
      BUF[I]::=INDATA:
    READ IN A WORD OF WHICH THE LOWER EIGHT BITS IS DATA FROM A/D CONVERSION.
    DATA IS STORED AS LOW_BYTE, HIGH_BYTE 
    END:
    FOR I:=1 TO IMAX DO BEGIN
      ITEMI :=BUF[I*2];
      <GET HIGH BYTE>
      <CHECK SIGN BIT TO SEE IF NEGATIVE>
      <IF NEGATIVE SUBTRACT OFF SIGN BIT AND CORRECT LATER>
      IF BUF[I*2] >= 128 THEN ITEMI :=BUF[I*2]-128;
      <ADD LOW BYTE TO HIGH BYTE (HIGH BYTE = BYTE * 256>
      X^\_A20[I] :=BUF[I*2-1]+ITEMI*256;
      <CORRECT FOR SIGN>

IF BUF[I*2] >= 128 THEN X^.A20[I]:=-X^.A20[I];
END;

BEGIN
REPEAT
CLRCRT:CSETXY(0,10);
FOR [POINTER]=1 TO 20 DO TEMP^$.A20[POINTER]:=0;
WRITE(' THE NUMBER OF CHANNELS PER RECORD:');
READLN(MAXCHANNEL);
WRITE(' FILE NAME:');
READNAME(FILENAME);
REWRI TE(INFILE, FILENAME);
INFILE^:=TEMP^:
PUT(INFILE);
< CLEAR AS 232 FROM RESIDUAL DATA TRANSMISSIONS >
KEYCK(ICH, IDAT);
WHILE ICH DO KE CK(I CH, IDAT);
< SEND HANDSHAKE TO SBC TO BEGIN DATA TRANSMISSION >
READY:
< FIRST DATA SENT SHOULD BE NUMBER OF SCANS TAKEN >
RCVINTEGER(MAXRECORD);
WRITELN(' THE NUM. OF REC.:', MAXRECORD);
< PUT NUMBER OF SCANS IN FILE >
FOR [POINTER]=1 TO MAXCHANNEL DO TEMP^$.A20[POINTER]:=0;
INFILE^:=TEMP^:
PUT(INFILE);
WRITE(' THE DATA:');
FOR [POINTER]=1 TO 20 DO WRITE(' ', TEMP^$.A20[POINTER]);
WRITELN;
< COLLECT DATA FROM ONE SCAN AND STORE IN FILE >
FOR [POINTER]=1 TO MAXRECORD DO
BEGIN
READY:
TEMP:=X(MAXCHANNEL);
INFILE^:=TEMP^:
PUT(INFILE);
END;
WRITELN(' TRANFORMATION IS FINISHED');
Z: WRITELN(' ENTER C TO CONTINUE OR H TO HALT');
READLN(HALTCHAR);
IF (HALTCHAR = 'C') THEN AGAIN:=FALSE;
IF (HALTCHAR = 'H') THEN AGAIN:=TRUE;
IF ((HALTCHAR <= 'C') AND (HALTCHAR <= 'H')) THEN GOTO Z;
UNTIL AGAIN
END.

MODULE PASCON:
$INCLUDE('F1:SGLOBL.SRC');

PRIVATE PASCON:

PROCEDURE CLRCRT:
VAR X,Y:CHAR;
BEGIN
X:=CHR(1BH);
Y:=CHR(45H);
WRITE(X,Y);
END:
PROCEDURE CLRSS:
VAR X:CHAR;
BEGIN
X:=CHR(1BH);
Y:=CHR(4AH);
WRITE(X,Y);
END;

PROCEDURE CLRLN:
VAR X,Y:CHAR;
BEGIN
X:=CHR(1BH);
Y:=CHR(4BH);
WRITE(X,Y);
END;

PROCEDURE CRET:
VAR X,Y:CHAR;
BEGIN
X:=CHR(00H);
Y:=CHR(0AH);
WRITE(X,Y);
END;

PROCEDURE CURMD:
VAR X,Y:CHAR;
BEGIN
X:=CHR(1BH);
Y:=CHR(42H);
WRITE(X,Y);
END;

PROCEDURE CURMU:
VAR X,Y:CHAR;
BEGIN
X:=CHR(1BH);
Y:=CHR(41H);
WRITE(X,Y);
END;

PROCEDURE CURML:
VAR X,Y:CHAR;
BEGIN
X:=CHR(1BH);
Y:=CHR(44H);
WRITE(X,Y);
END;

PROCEDURE CURMR:
VAR X,Y:CHAR;
BEGIN
X:=CHR(1BH);
Y:=CHR(43H);
WRITE(X,Y);
END;

PROCEDURE CURMH:
VAR X,Y:CHAR;
BEGIN
X:=CHR(1BH);
Y:=CHR(48H);
WRITE(X,Y);
END;
BEGIN
CSETRY(0,23):
WRITE('**** THERE IS AN ERROR DUE TO DISPLAY LOCATION ****');
CURMH;
END;

PROCEDURE CSETRY(IX,IY:INTEGER):
CONST IXMAX=75:IYMAX=23:
VAR I:INTEGER:
BEGIN
  IF((IX > IXMAX) OR (IY > IYMAX)) THEN BEBE
  ELSE BEGIN
    CURMH:
    FOR I:=1 TO IY DO CURMD:
    FOR I:=1 TO IX DO CURMR:
  END:
END:

PROCEDURE READNAME(VAR FILENAME:PARY14):
VAR
  CHARBUFFER:ARRAY[1..14] OF CHAR:
  I:INTEGER:
BEGIN
  FOR I:=1 TO 14 DO CHARBUFFER[I] := ' ':
  FOR I:=1 TO 14 DO IF NOT EOLN THEN READ(CHARBUFFER[I]):
  PACK(CHARBUFFER,1..FILENAME):
  READLN:
END:

PROCEDURE CUT-INTEGER(VAR ILOW,IHIGH:INTEGER;INX:INTEGER):
BEGIN
< SEPARATE 16 BIT INTEGERS INTO TWO INTEGERS WHICH CONTAIN
INFORMATION IN ONLY THE LOW ORDER EIGHT BITS >
  IHIGH:=ABS(INX) DIV 256:
  ILOW:=ABS(INX) - IHIGH*256:
  IF INX<0 THEN IHIGH:=IHIGH+128:
END:

PROCEDURE SENTINTEGER(INX:INTEGER):
VAR ILOW,IHIGH:INTEGER:
BEGIN
  CUT-INTEGER(ILOW,IHIGH,INX):
  SIOOUT(ILOW):
  SIOOUT(IHIGH):
END:

PROCEDURE HANDSHAKING:
CONST IRS=11:
VAR
  VALID_CHAR:BOOLEAN:
  INDATA:INTEGER:
BEGIN
< WAIT FOR HANDSHAKING SIGNAL FROM VAX >
  KEYCK(VALID_CHAR,INDATA):
  WHILE((NOT(VALID_CHAR)) OR (INDATA <> IRS)) DO
    KEYCK(VALID_CHAR,INDATA):
END:

PROCEDURE KEYCK(VAR INCHA:BOOLEAN;VAR DATA:INTEGER):
VAR II:INTEGER:

------ ----
BEGIN
CSETRY(0,23):
WRITE('**** THERE IS AN ERROR DUE TO DISPLAY LOCATION ****');
CURMH;
END:
BEGIN
  PROCEDURE COM_INTEGER(ILOW,IHIGH:INTEGER;VAR INX:INTEGER);
  BEGIN
    INX:=ILOW+IHIGH*256:
  END;
  PROCEDURE RCVINTEGER(VAR INX:INTEGER);
  VAR VALID-CHAR: BOOLEAN;
  INDATA:INTEGER:
  BEGIN
    KEYCK(VALID.CHAR,INDATA):
    WHILE (NOT(VALID.CHAR)) DO KEYCK(VALID.CHAR,INDATA):
    INX:=INDATA:
    KEYCK(VALID.CHAR,INDATA):
    WHILE (NOT(VALID.CHAR)) DO KEYCK(VALID.CHAR,INDATA):
    INX:=INX+INDATA*256:
  END;
  PROCEDURE READY:
  CONST IRS=11:
  BEGIN
    SIOOUT(IRS):
  END;
  PUBLIC MDS.TO.VAX:
  TYPE DATAPOINTER="DATA:
  DATA= RECORD
    A20:ARRAY [1..20] OF INTEGER
  END:
  DATAFILE=FILE OF DATA:
  PARY14=PACKED ARRAY[1..14] OF CHAR:
  VAR TEMP:DATAPOINTER:
  INFILE:DATAFILE:
  PUBLIC MDSOK:
  PROCEDURE SIOOUT(I:INTEGER):
  PROCEDURE SINCHK(VAR ICHAR:INTEGER):
  PROCEDURE COM_INTEGER(ILOW,IHIGH:INTEGER;VAR INX:INTEGER):
  PROCEDURE RCVINTEGER(VAR INX:INTEGER):
  PROCEDURE KEYCK(VAR INCHA:BOOLEAN;VAR DATA:INTEGER):
  PROCEDURE COM_INTEGER(ILOW,IHIGH:INTEGER;VAR INX:INTEGER):
  PROCEDURE RCVINTEGER(VAR INX:INTEGER):
  PROCEDURE KEYCK(VAR INCHA:BOOLEAN;VAR DATA:INTEGER):
  PROCEDURE COM_INTEGER(ILOW,IHIGH:INTEGER;VAR INX:INTEGER):
  PROCEDURE RCVINTEGER(VAR INX:INTEGER):
C.4 Data Transfer Programs, MDS to VAX -- MDS

MODULE MDS_TO_VAX;
$INCLUDE(\"F4:GGLOBL.SRC\")$
PROGRAM MDS_TO_VAX(INPUT,OUTPUT);
LABEL 2;
VAR
  IIX,MIXRECORD,MIXCHANNEL,IPINTER:INTEGER;
  FILENAME:PAR14;
  HALTCHAR:CHAR;
  AGAIN:BOOLEAN;
{ sends one scan to VAX }
PROCEDURE SENTRECORD(X:DATAPINTER;IMAX:INTEGER);
VAR I11:INTEGER;
BEGIN
  FOR I:=1 TO IMAX DO
    BEGIN
      II:=X''.A20CII;
      SENTINTEGER(II);
    END;
END;
BEGIN
BEGIN
REPEAT
  CLEARTEXTXY(0,10);
  WRITELN(' THE NUMBER OF CHANNELS PER RECORD= 13');
  { read in number of channels }
  READLN(MAXCHANNEL);
  WRITE('MAXCHANNEL='MAXCHANNEL);
  { read file name of the data to be sent }
  READNAME(FILENAME);
  NEW(TEMP);
  RESET(INFILE,FILENAME);
  UNGET(INFILE);
  TEMP'':=INFILE'';
  { read number of scans from file }
  MAXRECORD:=TEMP''.A20C1 ;
  WRITELN('MAXRECORD='MAXRECORD);
  writeln('before hand');
  { wait until VAX sends down handshaking signal }
  HANDSHAKING;
  WRITELN(' AFTER HAND');
  { send number of scans to VAX }
  SENTINTEGER(MAXRECORD);
  WRITELN(' AFTER SENT INTEGER');
  FOR IPINTER :=1 TO MAXRECORD DO
    BEGIN
      { wait until VAX sends down handshaking byte }
      HANDSHAKING;
      GET(INFILE);
      TEMP'':=INFILE'';
      { send one scan to VAX }
      SENTRECORD(TEMP,MIXCHANNEL);
      (*
      WRITE(' THE DATA=');
      FOR IIX:=1 TO 20 DO WRITE(' ',TEMP''.A20CIX);
      WRITELN:*)
    END;
  END;
END;
2: WRITELN(' ENTER C TO CONTINUE OR H TO HALT');
READLN(HALTCHAR);
IF (HALTCHAR = 'C') THEN AGAIN:=FALSE;
IF (HALTCHAR = 'H') THEN AGAIN:=TRUE;
IF ((HALTCHAR <> 'C') AND (HALTCHAR <> 'H')) THEN GOTO 2;
UNTIL AGAIN
END.
C.5 Data Transfer Program, MDS to VAX -- VAX

```
c
character*1 outbuffer, CONT_HALT
CHARACTER*12 DATE
CHARACTER*80 IMPORDATA
c
character*4 termname
c
character*10 inbuffer, chan_data, pointer, maxchannel
integer maxtwo, inbuffer, chan_data, pointer, maxchannel
integer in(8002)
integer sys$assign, sys$alloc, sys$qio, sys$qflor, sys$r
integer sys$cancel, sys$dalloc, sys$dassign, sys$qiou
real scale(20), offset(20), bias(20)
c
c parameter read='7a'x
c parameter writeln='70'x
c
c write(5,998)
c format('enter terminal name = ttd?')
c read(5,997) termname
c termname='ttd7'
c
997 format(a)
c 1001 CONTINUE
30 icode=sys$assign(termname, i chan,,)
c if(icode .ne. 1) go to 30
C if(.NOT. ICODE)CALL SYS$EXIT(%VAL(ICODE))
c
311 write(5,311)
c format('transfer a data file from MDS ***',/
c 1 ' Transfer a data file from MDS **',/
c 1 ' with real, fixed record format. **',/
c 1 ' ****************************************',/
c 1 ' ****************************************',/
c
11 write(5,11)
c format('The default number of channels per scan = 13')
c read(5,12) maxchannel
C maxchannel=13
c 12 format(i5)
c
6000 write(5,6000)
c format('enter date and time - 10/05/10:00 - 12 char')
c read(5,6001) date
c write(5,6005)
c format('enter important information')
c read(5,6006) impordata
6006 format(a)
c 6001 format(a)
c call setname(fn ame, maxchannel)
c call set scale fac tor(scale, offset, bias)
c
C C
write(5,15)
c format('The default unit of sampling time (s) = .002957')
c read(5,16) samplingperiod
```
```
samplingperiod=0.002957

16  format(5,3)
C
C OPEN FILES TO WRITE DATA IN DATA FILE MUST BE READ WITH
C AN OPEN STATEMENT NOT FORMATTED READ/WRITE
C
do 50 i=1,maxchannel+1
   open(unit=10+i,file=fname(i),status='new',access='direct',
      form='unformatted',recl=1,recordtype='fixed')
50  continue
C FOR FORMAT ('5.3')
C OPEN(UNIT=10+MAXCHANNEL+2,FILE=FNAME(MAXCHANNEL+2),TYPE='NEW')
C
C HANDSHAKING CHARACTER SENT TO MDS
outbuffer=char(11)
C SEND HANDSHAKING CHARACTER
   i=code=sys$qiow(/%val(ichan),%val(writtenow),...,
      %ref(outbuffer),%val(1),...)
WRITE(5,999)
999 format(' IT WORKED TO HERE')
C
C READ NUMBER OF DATA SCANS ON FILE
   do 60 i=1,2
      i=code=sys$qiow(/%val(ichan),%val(read),%ref(inbuffer),
      %val(1),...)
60 in(i)=inbuffer
C
C COMBINE TWO BYTES SENT INTO 16 BIT INTEGER
C   call combine(in(1),in(2),maxrecord)
write(5,33)maxrecord
33 format( 'The number of record = '/i5)
WRITE(3,34)MAXRECORD,SAMPLINGPERIOD
34 FORMAT('15,F7.5')
C
C maxtwo=maxchannel*2
C do 70 pointer=1,maxrecord
C SEND HANDSHAKING CHARACTER TO MDS, MDS WILL RESPOND BY SENDING A
C WHICH IS ONE SCAN (OF MAXCHANNELS)
   i=code=sys$qiow(/%val(ichan),%val(writtenow),...,
      %ref(outbuffer),%val(1),...)
C
do 80 i=1,maxchannel+2
C READ IN DATA FROM ONE SCAN
C
   i=code=sys$qiow(/%val(ichan),%val(read),%ref(inbu
      %val(1),...)
80 in(i)=inbuffer
C
exec time=(pointer-1)*samplingperiod
write(11,'pointer',exec time)
C THE TWO BYTES SENT FOR EACH CHANNEL MUST BE COMBINED TO A 16 BIT
C INTEGER REPRESENTATION
   do 90 i=1,maxchannel
C EXAMINE HIGH BYTE FOR SIGN
   item=in(i*2)
   if( item>=128) item=in(i*2)-128
C
C COMBINE THE TWO BYTES SENT INTO VAX INTEGER
```
chan_data = in(i*2-1)+item*256
if(in(i*2) .ge. 128) chan_data = -chan_data

write(5,101)in(i*2-1),in(i*2),chan_data
format(1x,3(15.2x))
ran_data = (chan_data+offset(i))*scale(i)+bias(i)
write(11+i,'pointer')ran_data
write(5,102)
format(3x).
c
continue
do 100 i = 1, maxchannel+1
close(unit=10+i)
c
icod=0
ocode=sys$cancell(ichan)
icod=sys$dassign(ichan)
icod=sys$alloc(TERNAME)
write(11+MAXCHANNEL+1,6002)DATE, MAXRECORD, IMPORTDATA
6002 FORMAT(′DATE IS ′,A/, ′NUMBER OF RECORDS = ′,I4,/, ′IMPORTANT INFORMATION ′,/,′′)
close(unit=10+MAXCHANNEL+2)
write(5,1000)
1000 FORMAT(′ENTER C TO CONTINUE OR H TO HALT′)
read(5,997)CONT_HALT
if(cont_halt.eq.′C′)go to 1001
if(cont_halt.eq.′H′)go to 1002
call exit
end

1 subroutine setname(name,maxchannel)
character*10 name(20)
character fnametemp(4)
write(5,1)format(′The first four characters for all file names = ′)$
read(5,2)(fnametemp(i),i=1,4)
2 format(4a1)
do 20 i=1,20
name(i)(5:10)=′00.dat′
do 20 i=1,4
20 name(i)(i1:i1)=fnametemp(i1)
do 30 i=1,6
name(i)(6:i6)=char(i+46)
do 40 i=10,19
30 name(i)(i6:i6)=char(i+48)
do 40 i=10,19
40 name(i)(5:5)=char(49)
name(20)(5:5)=char(30)
c ind_frame=name(i)
c do 41 j=2,maxchannel
c dep_frame(j)=name(j)
c41 continue
c write(5,41)(name(j),j=1,maxchannel)
c41 format(20(′,A10,′))
return end
subroutine combine(ilo, ihigh, in)
    in = ilow + ihigh * 256
    return
end

subroutine setscalefactor(sscale, soffset, sbias)
    real sscale(20), soffset(20), sbias(20)

C ENTER SCALE FACTORS AND BIAS FACTORS IF NECESSARY
C
SSCALE(1) = 0.004885
SOFFSET(1) = 0.0
SBIAS(1) = 0.0
SSCALE(2) = 0.004885
SOFFSET(2) = 0.07916
SBIAS(2) = 0.0
SSCALE(3) = 0.07916
SOFFSET(3) = -1120.0
SBIAS(3) = 0.0
SSCALE(4) = 0.07916
SOFFSET(4) = -1120.0
SBIAS(4) = 0.0
SSCALE(5) = 0.0641
SOFFSET(5) = 0.0
SBIAS(5) = 0.0
SSCALE(6) = 0.0635
SOFFSET(6) = 0.0
SBIAS(6) = 0.0
SSCALE(7) = -3.04
SOFFSET(7) = 1431.0
SBIAS(7) = 0.0
SSCALE(8) = -3.068
SOFFSET(8) = 1441.0
SBIAS(8) = 0.0
SSCALE(9) = -9.139
SOFFSET(9) = 0.0
SBIAS(9) = 0.0
SSCALE(10) = -9.282
SOFFSET(10) = 0.0
SBIAS(10) = 0.0
SSCALE(11) = 0.000028
SOFFSET(11) = 0.0
SBIAS(11) = 0.0
SSCALE(12) = 0.0
SOFFSET(12) = 0.0
SBIAS(12) = 0.0
SSCALE(13) = 0.0000267
SOFFSET(13) = 0.0
SBIAS(13) = 0.0
C ENTER VALUES HERE
SSCALE(11) = 1.204
SOFFSET(11) = 0.0
SBIAS(11) = 0.0
RETURN
SSCALE(i) = 0.0
SOFFSET(i) = 0.0
SBIAS(i) = 0.0
RETURN
end

210 sbias(i) = 0.0
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