STUDY OF HYSTERESIS EFFECT AND COMPENSATION IN PNEUMATIC ACTUATOR SYSTEM

By

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STUDY OF HYSTERESIS EFFECT AND

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This thesis
is dedicated
to

PAPA
&
MAMA

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NOMENCLATURE

A	effective area of the pneumatic spring
A_o	valve orifice area
B	coefficient of friction & damping constant
C_c	coefficient of leakage
C_d	valve restriction
D_c	compressor capacity
e_c	electric potential applied to the compressor
g	gravity acceleration
g_c	unit conservation constant
J	angular inertia
k	specific heat ratio
K_c	proportional constant (compressor parameter)
m	mass of the load
n	polytropic constant
P	air pressure
P_1	air pressure at the junction
P_2	air pressure inside the pneumatic spring
P_3	air pressure inside the accumulator
P_a	ambient atmospheric pressure
P_d	downstream pressure
P_g	gauge pressure
P_u	upstream pressure
Q	mass flow rate

Q_c	mass flow rate of air through the compressor
Q_{in}	inlet mass flow rate
Q_{out}	outlet mass flow rate
R	gas constant
T	absolute temperature
T_c	temperature of the compressor
T_u	upstream temperature
V	volume
v	specific volume
W	weight flow rate
w	load weight
W_a	weight flow rate of air from/to the accumulator
W_c	weight flow rate through the compressor
W_{in}	inlet weight flow rate
W_{out}	outlet weight flow rate
W_s	weight flow rate from/to the solenoid valve
W_v	weight flor rate from/to the vent valve
y	vertical displacement
α	angular acceleration
β	gas compressibility
ω	angular velocity

CHAPTER I

INTRODUCTION

In daily life as well as engineering processes, many usage of a pneumatic spring control can be encountered, such as in car suspension systems and automatic level controls. The objective of an automatic level control is to maintain the height of an object unchanged -- or at least to make it return to initial position quickly -- when the load is changed. A car, for instance, has the base height of about 10 inches from the ground empty. It should measure the same distance from the ground when it is full of passenger and highly loaded.

Pneumatic spring systems typically consist of at least an air compressor or a supply pressure tank, a discharging valve, and pneumatic spring. A pneumatic spring is an elastic container with gas confined within. The gas pressure will balance an external force in the same manner as a spring in mechanical system. The elasticity of the pneumatic spring is dependent upon the compressibility, amount and temperature of the gas, along with the type of compression / expansion the gas experiences: isothermal, isentropic, or polytropic. The advantages of using this kind of spring compared to conventional spring are its low nonlinear spring rate, constant sprung mass frequency, accurate headlamp aiming, and constant body height. In addition, available travel of the pneumatic spring is relatively high and its rate is highly variable compared to the other kind of springs [1].

The compressor in a pneumatic spring system functions to charge the system to bring the controlled object back to its desired position after the position is lowered due to increasing load. On the other hand, if the load is diminished or removed, the air in the system is discharged to compensate for the load change. The problem faced in this effort involves the lag time encountered when the load is increased. In this situation the object will remain in a "wrong" position for some time before returning to the set point height.

The culprit is hysteresis in the pneumatic spring. The control path taken by the object as it rises differs from that as it lowers. Friction and material characteristics of the material in pneumatic spring cause this hysteresis problem because the air pressure needed to pump the spring up is greater than that needed to lower it. Hysteresis causes the pneumatic system to respond slower and decreases efficiency. It also creates a limit cycle problem which increases instability and makes the system difficult to control.

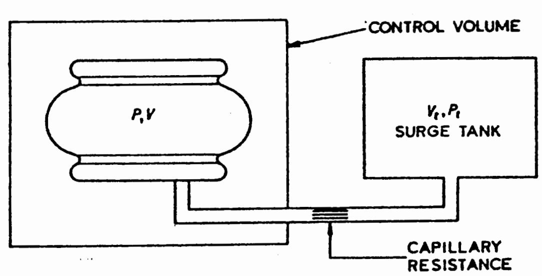
The purpose of this thesis is to study the effect of pneumatic spring hysteresis on the pneumatic actuator system and develop a method to reduce this effect in order to generate a faster response. A computer model has been formulated to simulate the motion of the actuator system. In addition, experimental data has been obtained to provide a comparison. Hysteresis characteristics have been derived from experiment measurement for direct use in the simulation. The usage of an accumulator in the system has proven to be an excellent method of diminishing the response delay caused by hysteresis.

CHAPTER II

LITERATURE REVIEW

Most of the literature on pneumatic springs relate to the air suspension system in automobiles and vibration isolators used in machine tools. Both systems have one thing in common: they protect the machine or the passenger -- in the case of a car -- from fatigue by absorbing the vibration. One condition that must be met, however, is that the system must maintain a constant position regardless of the distrubance frequency or magnitude.

Early in 1966, Soliman and Ardabili [2] worked on a self-damped pneumatic isolator for variable frequency excitation. A bellow air spring was used (figure 1). The



P_t = instantaneous pressure of air in the surge tank; P = instantaneous pressure of air in the isolator; V = instantaneous volume of air in the isolator; V_t = instantaneous volume of air in the surge tank.

Figure 1. Soliman and Ardabili Self-damped Pneumatic Isolator [2]

damping was produced by the transient pressure feedback (the damper) was a surge tank connected through capillary resistance to the pneumatic spring.) The main advantage of this method lies in the fact that the damping is dependent on the disturbing frequency. That is, the system provides a large amount of damping at resonant frequency and a decreasing amount of damping as the frequency of vibration increases.

An active mechano-pneumatic shock isolator (see figure 2) was introduced by Ruzicka [3]. He used integral displacement controls to reduce the steady state relative displacement of the mass to zero and to isolate vibration in the presence of sustained acceleration. A feedback lever was connected to a servo valve which admitted air to the air

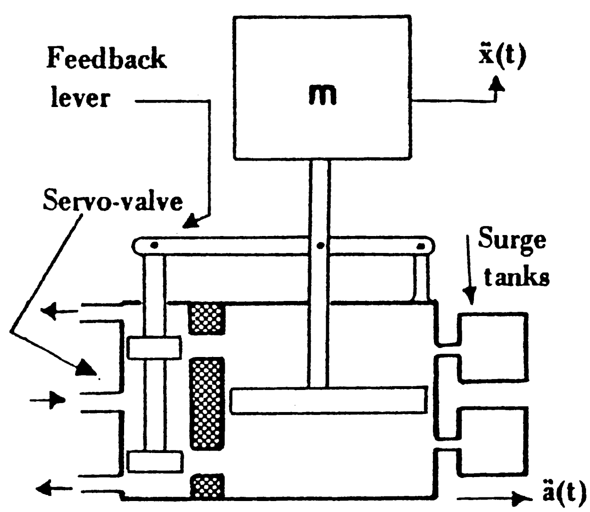


Figure 2. Ruzicka's Mechano-pneumatic Shock Isolator [3]

spring when the height was less than the desired height and to reject air when the height was greater than the set point.

Bachrach and Rivin [4] approached the problem by determining the complex dynamic stiffness of a damped pneumatic spring. The damping is the result of the transient pressure feedback from an auxiliary tank connected to the spring cylinder by a capillary. Also, Bachrach and Rivin examined the behavior of a compound spring, consisting of a damped pneumatic spring in parallel with a stiffer linear spring (figure 3.) These authors discovered that the

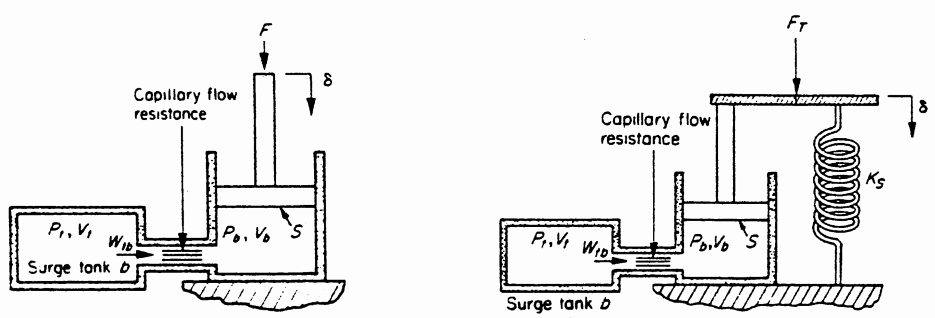


Figure 3. Bachrach and Rivin Model of Pneumatic Spring [4]

damping loss factor depended only on the ratio of the tank and the cylinder volume. (The complex stiffness of a damped pneumatic spring depends on excitation frequency and fundamental component dimensions.)

The generalized analysis of a shock absorber consisting of a pneumatic damper in parallel with a mechanical spring (figure 4) had been presented by Hundal [5]. He non-dimensionalized the nonlinear pneumatic equations in terms

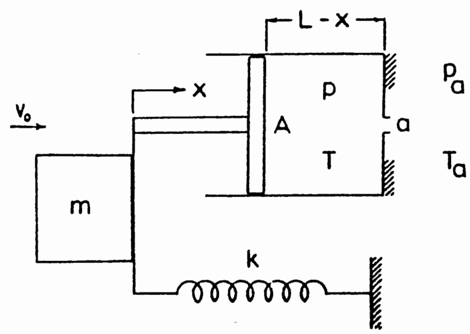


Figure 4. Hundal's Model of Shock Absorber [5]

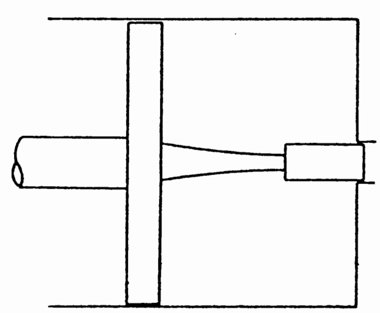


Figure 5. Hundal's Damper with Variable Area Orifice [5]

of the variables displacement, velocity, and pressure along with the parameters of mass, stiffness, and orifice area ratio. Hundal presented the results of the standard orifice with fixed area and then compared the results to those of dampers with variable area orifice (figure 5.)

Hundal [6] also analyzed the response of pneumatic shock isolators to base loading of rectangular and half-sine shape. The isolator consists of a pneumatic damper (a pneumatic cylinder with an orifice in the piston) in parallel with a linear spring (figure 6.)

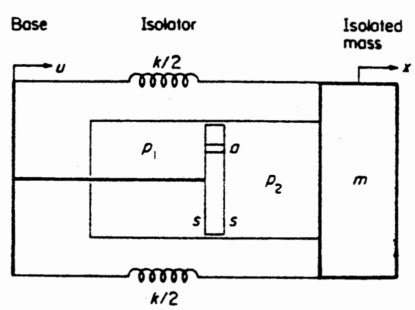


Figure 6. Hundal's Model to Analyze the Response to Certain Acceleration [6]

Wang, Singh, Yu, and Guenther [7] managed to achieve the computer simulation of a shock-absorbing pneumatic cylinder. A mathematical model of a double-acting pneumatic

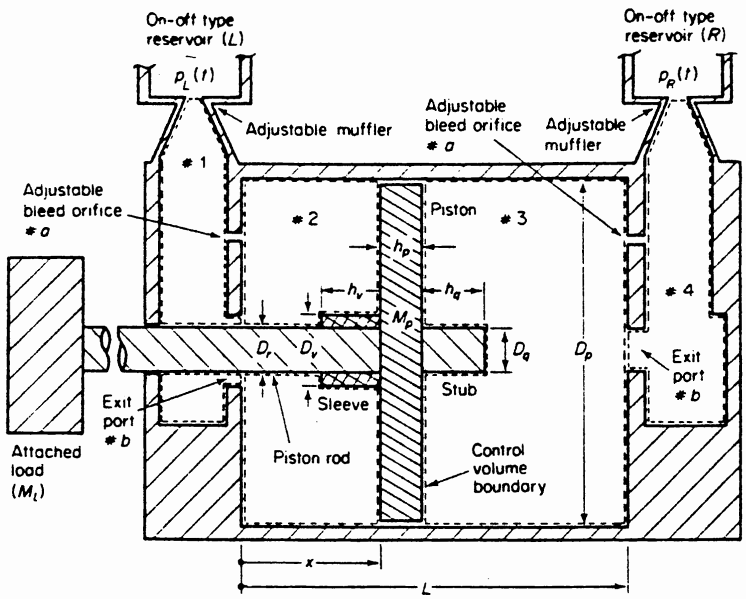


Figure 7. Double-acting Pneumatic Cushioning Cylinder [7]

cushioning cylinder (figure 7,) designed to absorb periodic shock loads was presented.

Wang and Singh [8] also worked together in the study of the nature of the nonlinearities associated with a closed pneumatic chamber coupled to a linear mechanical system (figure 8.) This model could represent passive vibration isolators, shock absorbers, and cushioning type actuators.

A chamber connected to a reservoir through an orifice (figure 9) was examined also by Wang and Singh [9]. They studied the dynamic behavior of the pneumatic chamber. Nonlinearities associated with mass flow rate through the orifice was the focus of the research. Methods of harmonic balance were used.

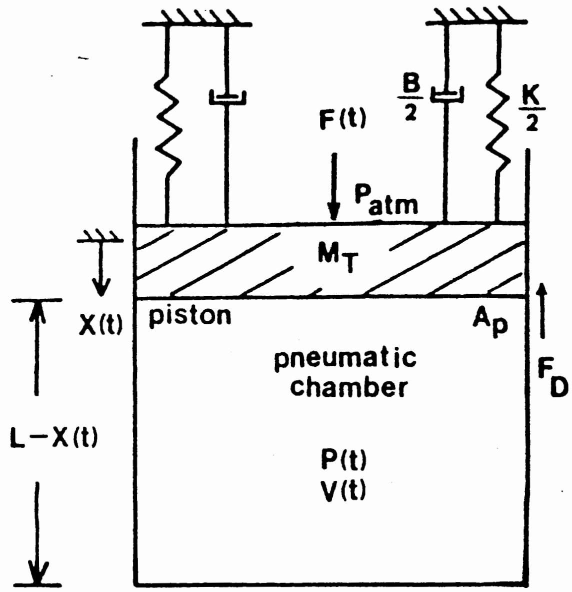


Figure 8. Closed Pneumatic Chamber Coupled to Linear Mechanical System [8]

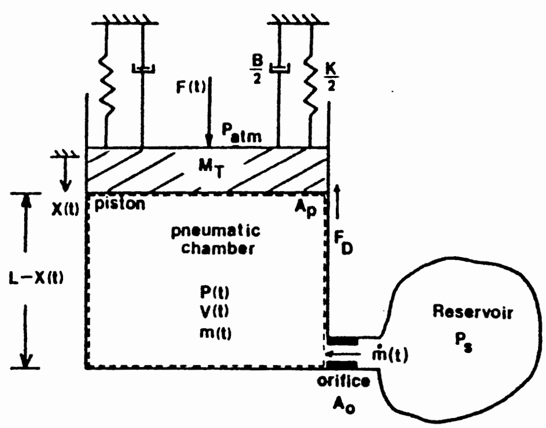


Figure 9. Closed Chamber Connected to Reservoir [9]

In 1988, Sharp and Hassan [10] modelled a pneumatic active car suspension system in a single wheel station form excited by realistic road roughness.

Lai, Menq, and Singh [11] proposed a method to achieve position control of pneumatic system. They used pulse width modulation to simulate the proportional control. Proportional-plus-integral-control was utilized in an inner loop to control the actuator pressure. The load displacement was controlled by implementing an outer loop with displacement and velocity feedbacks. The response of the system to the step input was presented.

All these papers address the dynamic characteristic of the pneumatic spring system. The focus of attention is the disturbance caused by road roughness or mechanical vibration. This kind of disturbance has only a small amplitude. Few paper discusses the static effect of heavy load changes and few have taken hysteresis into consideration. Heavy load were used to depress the spring down due to the relatively small stiffness of the pneumatic spring and in the presence of hysteresis, it would take considerable time to recharge the spring.

This thesis is focused on this static response of pneumatic actuator system with hysteresis to a considerable amount of load change. This research is important in a leveling system where not only small amplitude (high frequency) load is encountered, but the low frequency large load changes also occurs. The air spring used in the activity is of the rolling lobe type. This type of spring has an advantage of much greater axial displacement than the other spring types. Another benefit of using this type is that the effective area of the spring is not constant but changes with respect to displacement. This variation in the area with displacement minimizes the nonlinearity that is common in the load-deflection characteristics of a pneumatic springs with constant area. In addition, this type of spring allows low stiffness to damp the small vibration during small displacement but provides high stiffness when the displacement is large.

This thesis concentrates on the study of the hysteresis effect in pneumatic springs due to the material used in the rolling lobe type spring.

CHAPTER III

THEORETICAL DEVELOPMENT

Pneumatic System

Pneumatic systems are used because of their excellent reliability. Its availability in small size and light weight while still providing great power ranges and high accuracy is also a reason the system is desired. In general a pneumatic system is nonlinear. The nonlinearity is caused by the nature of the gas it used as a transmission medium. The high compressibility along with high sensitivity of the gas to temperature changes, leakage, turbulence, and the saturation of the flow rate (which occurs when the ratio of the upstream and downstream pressure is too large) make the pneumatic system nonlinear.

Basic elements that must exist in a pneumatic system are an air compressor, transmission pipe, valves, and an actuator. The control used in most pneumatic systems is an on-off control type. Other types of control, such as proportional control, are hard to implement due to the high nonlinearity and the slow transmission speed of pneumatic system. On-off control can create a limit cycle problem but the limit cycle is stable and it will be damped out by the system.

<u>Compressor</u>

Piston-type compressors are the power source for many pneumatic systems. It is nonlinear, but in certain ranges (pressure differences not exceeding 200 psi and temperature below 600°K) the flow rate can be considered linearly proportional to the frequency of the compressor stroke. The proportionally constant is the capacity of the compressor established by the size of the compressor. The units used are gallons per stroke. External leakage also contributes to nonlinearity since its value is dependent upon the pressure difference across the compressor. A typical formula for air compressor flow rate is given by:

$$Q_c = D_c \omega - C_c dP \qquad (1)$$

Q_c is the volumetric flow rate of compressor (it is positive if the air flows into the system.) D_c is the capacity of compressor, ω is the frequency of compressor stroke, C_c is coefficient of leakage, dP is pressure difference. In pneumatic systems, however, it is more practical to measure the weight flow rate than the volumetric flow rate. The weight flow rate $W = Q\ g\ /\ (g_c\ v)$. g_c is unit conversion constant -- in S.I., $g_c = 1.0$ (kg m/s^2) / N, while in the British system, $g_c = 32.2$ (lbm ft/s^2) / lbf -- and $v = R\ T\ /\ P$ (valid for air when its temperature is above -10°F and its pressure is below 4000 psi) where Q is volumetric flow rate, g is gravity acceleration, v is specific volume or the inverse of density, R is gas constant, T is absolute temperature, and P is pressure. Hence in term of weight flow rate,

$$W_c = \frac{g\ P\ Q_c}{g_c\ R\ T} = \frac{g\ P}{g_c\ R\ T}\ (D_c\ \omega - C_c\ dP) \tag{2}$$

Since a compressor is the power source of the pneumatic system, another equation that involve power term is needed.

$$T_c = K_c\ e_c = J\ \alpha + B\ \omega + C\ D_c\ dP \tag{3}$$

T_c is the power torque needed to activate the compressor, e_c is the electric potential difference, K_c is the proportional constant, J and B are angular inertia and damping constant respectively, α is angular acceleration, $\alpha = d\omega/dt$, C is a constant.

<u>Transmission Pipe</u>

This section presents the expression which describe the transmission pipes performance. The difference between inlet flow rate and outlet flow rate of a pipe is given by

$$Q_{in} - Q_{out} = \left(\frac{V}{\beta}\right) \frac{dP}{dt} + \frac{dV}{dt} \tag{4}$$

where β is the gas compressibility (β of air can be approximated by the multiplication of polytropic constant n with air pressure P, i.e. $\beta = Z\ n\ P$ -- Z is compressibility constant.) In term of weight flow rate, assuming $P_{in} = P_{out} = P$ and $T_{in} = T_{out} = T$ (which is reasonable for small restriction pipe,) equation (4) can be written as follows:

$$\frac{g_c\ R\ T}{g\ P}\ (W_{in} - W_{out}) = (\frac{V}{\beta})\frac{dP}{dt} + \frac{dV}{dt} \tag{5}$$

$$\frac{dP}{dt} = \frac{\beta}{V}\ \{\frac{g_c\ R\ T}{g\ P}(W_{in} - W_{out}) - \frac{dV}{dt}\} \tag{6}$$

The compressibility of the pipe is neglected since it is too small compared to that of the gas. Hence,

$$\frac{dP}{dt} = \frac{\beta}{V}\ \{\frac{g_c\ R\ T}{g\ P}\ (W_{in} - W_{out})\} \tag{7}$$

<u>Valve</u>

The valve is the component where most of the nonlinearity is found. The flow rate across the valve is a nonlinear function of orifice size, upstream pressure, downstream pressure, and the ratio between them.

$$W = C_d\ A_o\ C_1\ \frac{P_u}{\sqrt{T_u}}\ (\frac{P_d}{P_u})^{\frac{1}{k}} \sqrt{1 - (\frac{P_d}{P_u})^{\frac{k-1}{k}}}$$

$$C_1 = g\sqrt{\frac{2\ k}{R\ g_c\ (k-1)}} \quad for \quad \frac{P_d}{P_u} \geq (\frac{2}{k+1})^{\frac{k}{k-1}} \tag{8}$$

where W is weight flow rate through the valve, C_d is a constant, A_o is orifice area, P_u is stagnation upstream pressure, T_u is stagnatian upstream absolute temperature, P_d is downstream pressure, k is specific heat ratio C_p/C_v.

When $P_d/P_u = 1$ or $P_d = P_u$, there is no flow across the valve. If P_d is decreased ($P_d/P_u < 1$,) the flow rate W will begin to increase. As P_d continues to decrease, while P_u is maintained constant, W will continue increasing till it reaches its maximum value W_{max} when the ratio of $P_d/P_u = (2/(k+1))^{k/(k-1)}$. This ratio where the flow rate is maximum for a certain P_u is abbreviated by (P_d/P_u)crit. If after reaching this point, P_d decreases further -- but P_u stays at the same value -- that is $P_d/P_u < (P_d/P_u)$crit, W will not exceed W_{max}, instead it will stay at that magnitude, no matter how much smaller P_d is. In the range $P_d/P_u < (P_d/P_u)$crit, W is dependent upon P_u only, $W_{max} = W_{max}(P_u)$.

$$For\ \frac{P_d}{P_u} < \left(\frac{P_d}{P_u}\right)_{crit},$$

$$W = W_{max}$$

$$= C_d A_o C_1 \frac{P_u}{\sqrt{T_u}} \left(\frac{P_d}{P_u}\right)^{\frac{1}{k}} \sqrt{1 - \left(\frac{P_d}{P_u}\right)^{\frac{k-1}{k}}} \quad \bigg| \frac{P_d}{P_u} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

$$W = C_d A_o C_1 \frac{P_u}{\sqrt{T_u}} \{(\frac{2}{k+1})^{\frac{k}{k-1}}\}^{\frac{1}{k}} \sqrt{1 - \{(\frac{2}{k+1})^{\frac{k}{k-1}}\}^{\frac{k-1}{k}}}$$

$$= C_d\ A_o\ C_1\ \frac{P_u}{\sqrt{T_u}}\ (\frac{2}{k+1})^{\frac{1}{k-1}} \sqrt{1 - \frac{2}{k+1}}$$

$$= C_d\ A_o\ C_1\ \frac{P_u}{\sqrt{T_u}} \sqrt{\frac{\frac{1}{2}\ (k-1)}{(\frac{k+1}{2})^{\frac{k+1}{k-1}}}}$$

$$= C_d\ A_o\ g\sqrt{\frac{2\ k}{R\ g_c\ (k-1)}}\ \frac{P_u}{\sqrt{T_u}} \sqrt{\frac{\frac{1}{2}\ (k-1)}{(\frac{k+1}{2})^{\frac{k+1}{k-1}}}}$$

$$= C_d\ A_o\ \frac{P_u}{\sqrt{T_u}}\ g\sqrt{\frac{k}{R\ g_c\ (\frac{k+1}{2})^{\frac{k+1}{k-1}}}}$$

$$= C_d\ A_o\ C_2\ \frac{P_u}{\sqrt{T_u}},$$

where

$$C_2 = g\sqrt{\frac{k}{R\ g_c\ (\frac{k+1}{2})^{\frac{k+1}{k-1}}}} \qquad (9)$$

Actuator

The actuator is the termination of the system. It converts pneumatic energy into mechanical energy. Actuators usually are cylinders or motors. The actuator of interest here is an air spring. An air spring is an air cushion. It has the characteristic of the spring because the compressibility of the air inside has the ability to store energy. An air spring is a nonlinear spring, its rate is low for small displacement and high for large displacement. (This phenomenon is a benefit to level system since it makes the spring less sensitive to vibration and small excitation.) In general for an isolated air spring, the spring rate is given by the following expression:

$$Rate = \frac{n\ P\ A^2}{V} + P_g \frac{dA}{dy} \tag{10}$$

P is absolute pressure of air inside the spring, P_g is gauge pressure, V is volume of air inside the spring, y is spring displacement, A is effective area that support the load. The nonlinearity is obvious since P V = C and V = f(y),

$$Rate = \frac{n\ C\ A^2}{\{f(y)\}^2} + P_g \frac{dA}{dy} \tag{11}$$

In this system, however, the air spring is not isolated. It is connected to an air compressor and to a discharge port. It is reasonable to consider this spring as a cylinder, but instead of having a piston and barrel, this pneumatic spring uses an elastic rubber bag. When the bag is filled, it swells and pushes on the load, see figure 10.

Notice the second term of the right hand side of equation (11). The area of the spring is not constant with respect to displacement. In this case the displacement-area curve is nonlinear, but in certain small ranges, it is reasonable to assume a linear curve.

In order to simulate the pneumatic spring used in this system, a set of equations are needed. The first equation relates the pressure inside the spring with the flow rate into the spring. The spring can be approached by modelling it as a large cross section "pipe" with blocked outlet.

$$\frac{dP}{dt} = \frac{\beta}{V} \{\frac{g_c\ R\ T}{g\ P}\ (W_{in} - W_{out}) - \frac{dV}{dt}\} \tag{12}$$

Since the outlet is blocked, $W_{out} = 0$. $dV = A\ dy$, $dV/dt = A\ dy/dt$; V is the volume of the air spring (its increase or decrease should be proportional to the spring displacement.) Therefore equation (12) can be written as:

$$\frac{dP}{dt} = \frac{\beta}{V} \{\frac{g_c\ R\ T}{g\ P}\ W_{in} - A\ \frac{dy}{dt}\} \tag{13}$$

Mechanical consideration provides the second equation,

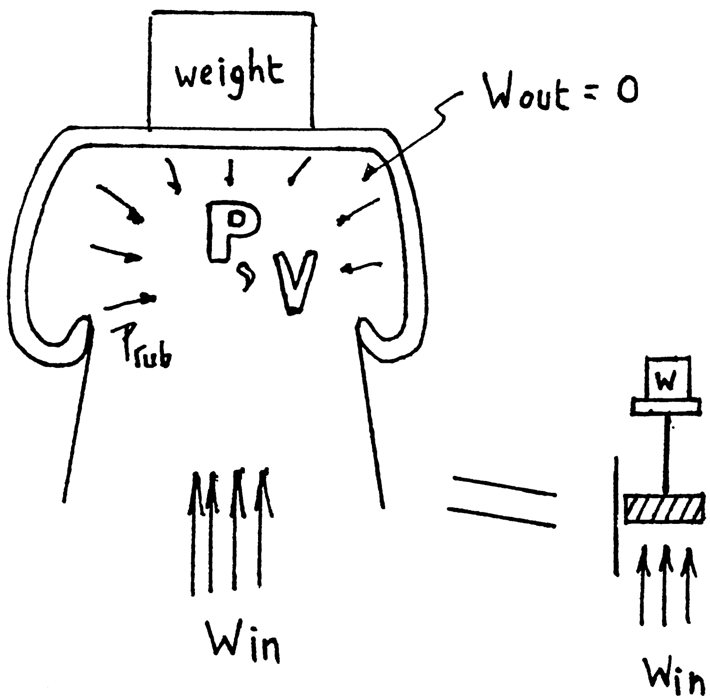


Figure 10. Analogy Between Pneumatic Spring and Cylinder

$$m \frac{\ddot{y}}{g_c} + B \dot{y} = (P_g - P_{rub}) A - w \tag{14}$$

m is the mass of the load, B is the coefficient of friction, $\ddot{y} = d^2y/dt^2$, $\dot{y} = dy/dt$, P_{rub} is the reactive pressure exerted by the rubber bag, w is the load weight.

System Modelling

This system consists of an air spring, a compressor, pipe, and two valves. One valve is linked to the air spring and then connected to the compressor through a pipe. This pipe is attached a vent valve in order to discharge the system. A platform was constructed above the air spring to apply the load. Attached to this platform is a linear variable displacement transducer to measure the height of the platform. Two pressure transducers were installed, one to measure the pressure inside the air spring and another to measure the pressure inside the pipe at the junction as shown in figure 11.

Control Algorithm

The opening and closing of the valves are controlled by on-off controllers. This type of controller is also used to turn the compressor on and off. The control logic is given as follows:

1. Determine a set point height for the platform. This height is the objective height the system must reach

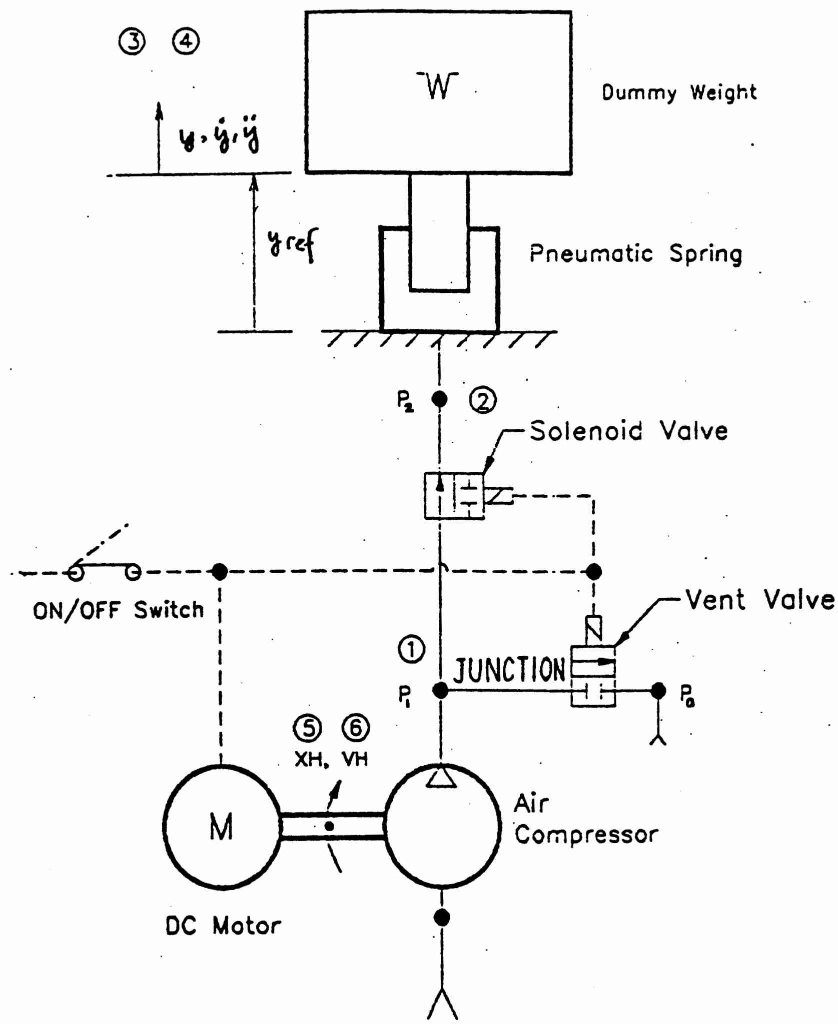


Figure 11. System Diagram

and is referred to as the desired height. This desired height generally includes a small dead band.

2. If the platform height is in this desired range, then both valves are to be closed and the compressor is to be turned off.
3. If the platform height is below this range of desired height, then the system must be charged. To accomplish this, the air spring valve is opened, the discharge valve is closed, and the compressor is turned on.
4. If the platform height is higher than the desired height, then the system must be discharged: the air spring valve is open, and the discharge valve is opened while the compressor is turned off.

The fact that the desired height must have a dead band should be obvious in visualizing rule 2, 3. 4. By setting the height into a range, the on-off controller needs not to be actuated too often. That in turn would prolong the controller life. This range setting recreates a dead-band nonlinearity that causes a stable limit cycle. Fortunately this limit cycle in the real situation would be attenuated by the air in the air spring. Switching the on-off controller often as in the case of the system that is set to reach a certain height would cause the system to oscillate even worse than if the on-off controller is actuated only when the height is out of range. The reason is the slow transmission speed of the air hinders the system from following the switching of the on-off controller. As the result of the closing and opening motion of the valve and the compressor, severe turbulence is induced in the system and create system instability.

Procedure

The procedure to record the response of the system to load changes must be defined for simulation purposes. The first step of this procedure is to subject the system to some base load and permit to the air spring to charge until the desired height is achieved. After the system has stabilized, more weight is added to the load. This extra load would force the platform down. When the height of the platform is lower than the desired height, the compressor is turned on and the system is charged to bring the platform back to its initial position.

In the second step of the procedure, some of the load is removed. The platform would rise since the pressure inside the air spring is higher than the pressure needed to keep the reduced load in that position. This excess air would be discharged, see rule 4, till the platform moves back to the initial position.

State Equations

To simulate this system, all the preceeding expressions must be converted into state equations. Let the pressure at the junction be defined as P_1 and the pressure inside the air spring as P_2, the air flow rate through the solenoid valve to and from the air spring as W_S and the discharged flow rate as W_v. Recall that pressure difference across the compressor dP in equations (1), (2), and (3) is actually the difference between P_1 and ambient atmospheric pressure P_a. Notice also that gauge pressure P_g inside the air spring -- see equations (10), (11), and (14) -- is actually the pressure difference between absolute air spring pressure P_2 and P_a.

$$\frac{d\omega}{dt} = \frac{1}{J} \{T_c - B\ \omega - D_c\ (P_1 - P_a)\} \tag{15}$$

$$\frac{dP_1}{dt} = \frac{\beta}{V} \{\frac{g_c\ R\ T}{g\ P_1}\ (W_c - W_s - W_v)\} \tag{16}$$

$$\frac{dP_2}{dt} = \frac{\beta}{V} \{\frac{g_c\ R\ T}{g\ P_2}\ W_s - A\ \dot{y}\} \tag{17}$$

$$\frac{dV}{dt} = A\ \frac{dy}{dt} \tag{18}$$

$$\frac{d\dot{y}}{dt} = \frac{g_c}{m} \{(P_2 - P_a - P_{rub}) - A - w - B\ \dot{y}\} \tag{19}$$

where

$$W_c = \frac{g\ P_1}{g_c\ R\ T} \{D_c\ \omega - C_c\ (P_1 - P_a)\} \tag{20}$$

For $\frac{P_d}{P_u} \geq (\frac{P_d}{P_u})crit,$

$$W|_{s,v} = C_d A_o C_1 \frac{P_u}{\sqrt{T_u}} (\frac{P_d}{P_u})^{\frac{1}{k}} \sqrt{1 - (\frac{P_d}{P_u})^{\frac{k-1}{k}}}$$

and for $\frac{P_d}{P_u} \leq (\frac{P_d}{P_u})crit,$

$$W|_{s,v} = C_d A_o C_2 \frac{P_u}{\sqrt{T_u}}, \tag{21}$$

with (P_d/P_u)crit = $(2/(k+1))^{k/(k-1)}$ and C_1 and C_2 defined by equations (8) and (9). P_d and P_u in W_S equation correspondent to P_1 and P_2 whichever is greater; the greater pressure is upstream pressure P_u and the less one is downstream pressure P_d. In W_v equation, they stand for P_1 and ambient atmosphere pressure P_a but most of the time P_1 is P_u and P_a is P_d since it is not unreasonable to expect that the pressure inside the system is always greater than atmospheric pressure. W_S is defined positive if the flow is from the compressor into the pneumatic spring; otherwise, W_S is negative. W_v is positive if the air is discharged. To implement the computer program with this sign change, the following must be used: if W_S or W_v is negative, then C_d corresponding to them is redefined as C_d = $-C_d$

To simulate the on-off controller in the system, under any condition requires the following logic: if the controller is off or the solenoid valve is closed, the corresponding flow rate W_c, W_S, or $W_v = 0$. Now $P_1(t)$, $P_2(t)$, y(t) could be calculated using Runge-Kutta iteration method.

Simulation Result (Without Hysteresis)

Using the method of iteration, the simulation results are shown as the time-displacement curve in figure 12. This curve, however, is not realistic. The true curve looks somewhat like the one shown by a dashed-line curve in the same figure.

It should be noticed that when some extra load is added, both curves agree until they proceed to point A. In the dashed-curve, the platform, instead of going back upward immediately to the initial position like that in the simulation curve, it stays in the same lower position for some time before it moves upward at point B.

The temporary lag of the platform at the lower position means the system needs time to charge itself to overcome the additional load. However, the difference is more than only a lack of pressure because if this is were the only problem, the platform should not stay there unmoved, instead it should rise upward immediately at some lower rate. This phenomenon occurs because the force to move the load upward differs from that to move it down. Consider figure 10 and equation (14), P_g has to overcome P_{rub} and w to move the load up or in the other direction, w together with P_{rub} need

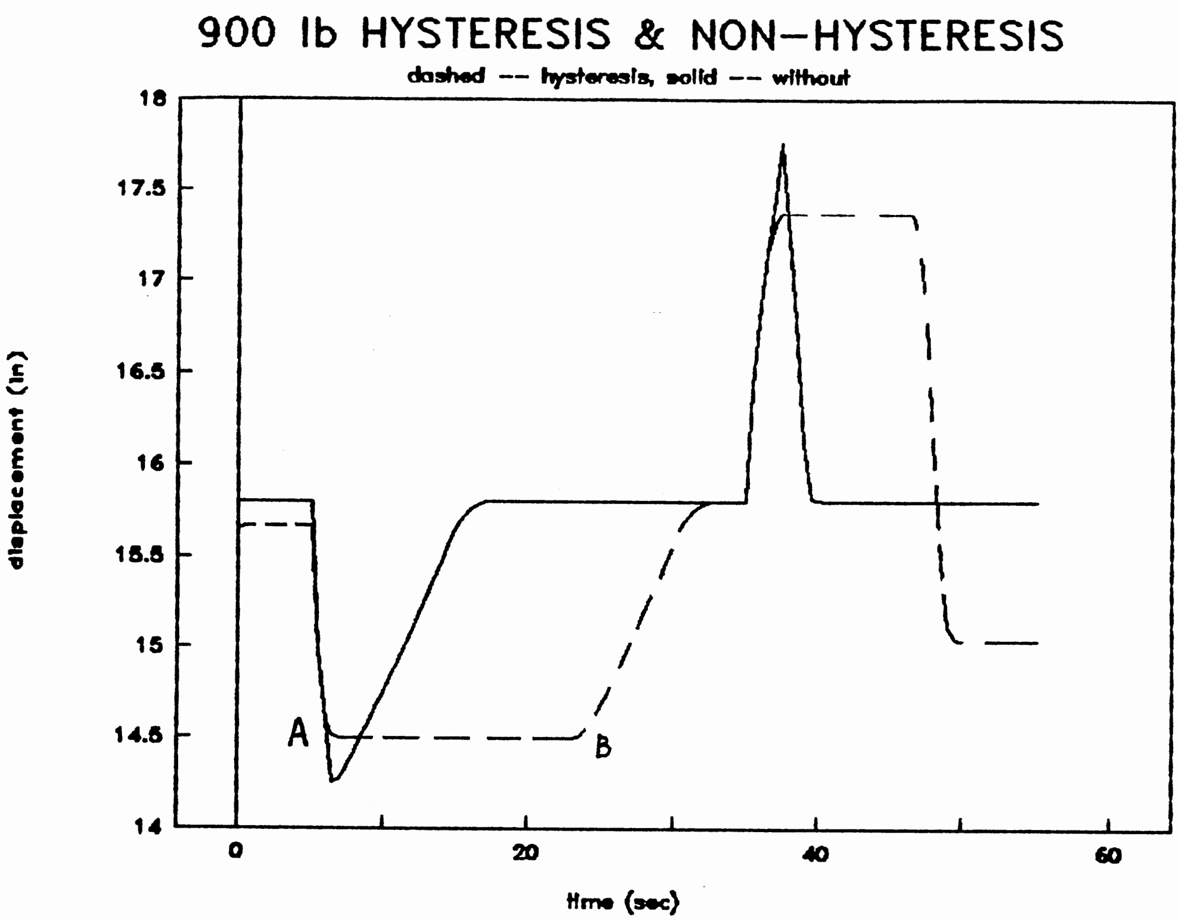


Figure 12. Simulation of the System Without Hysteresis

to exceed P_g to move the load down. In figure 12, it can be seen that the first time the load stops, $\ddot{y}_A = \dot{y}_A = 0$, $P_g = P_A$, $P_{rub} = P_A - w/A$. The load stays for some time in that position. During this time lag, the compressor continues to charge the system until there is enough energy to push the load upward. At point B, $\ddot{y}_B = \dot{y}_B = 0$, $P_g = P_B$. Obviously, $P_B > P_A$ and since w does not change, P_{rub} at point B must be greater than P_{rub} at point A.

This double-valued force could be caused by hysteresis in air as the transmission medium. But greater still is the hysteresis due to coulomb friction and the material characteristic of the rubber bag of the air spring.

Hysteresis

At room temperature, rubber exhibits a hysteresis. That is, the path taken by the rubber in the strain-stress curve when it is extended is different from the path when it retracts. This hysteresis is caused by the internal friction between the internal network chains. To the rubber itself, hysteresis is an internal energy loss. To other mechanical system that use rubber, hysteresis cause discontinuity and nonlinearity in the system response and makes the system hard to control. Particularly in this system, hysteresis slows the system response.

The difference between extension and retraction path is a function of the characteristics of the rubber material itself. The compound used to make the rubber material determines the curve shape. Artificial, natural and vulcanized rubber all have different hysteresis curves. The history of the rubber together with the physical and heat treatment it experienced, will also affect the curve. External factors that affect the hysteresis curve include temperature and velocity (how quick the rubber is extended or retracted.) The hysteresis loss decreases with increasing temperature. Typical stress-strain curve is showed in figure 13 [12].

Many models for this curve have been proposed [12]. No attempt to create a new model is intended in this thesis. Since it is insufficient to select an arbitrary model to simulate hysteresis in this system, an experiment is necessary to measure the emperical characteristics of the hysteresis curve exhibited by the air spring used in this system.

Emperical Hysteresis Characteristic Measurement

The experimental procedure used to measure hysteresis was as follows. The set up of the experiment is included in detail in chapter IV. In general, however, the implementation consisted of a compressor connected to the air spring through a pipe. A solenoid valve is placed between this compressor and the air spring. A vent valve located by the compressor was installed to discharge the system. The solenoid valve is open at all the times.

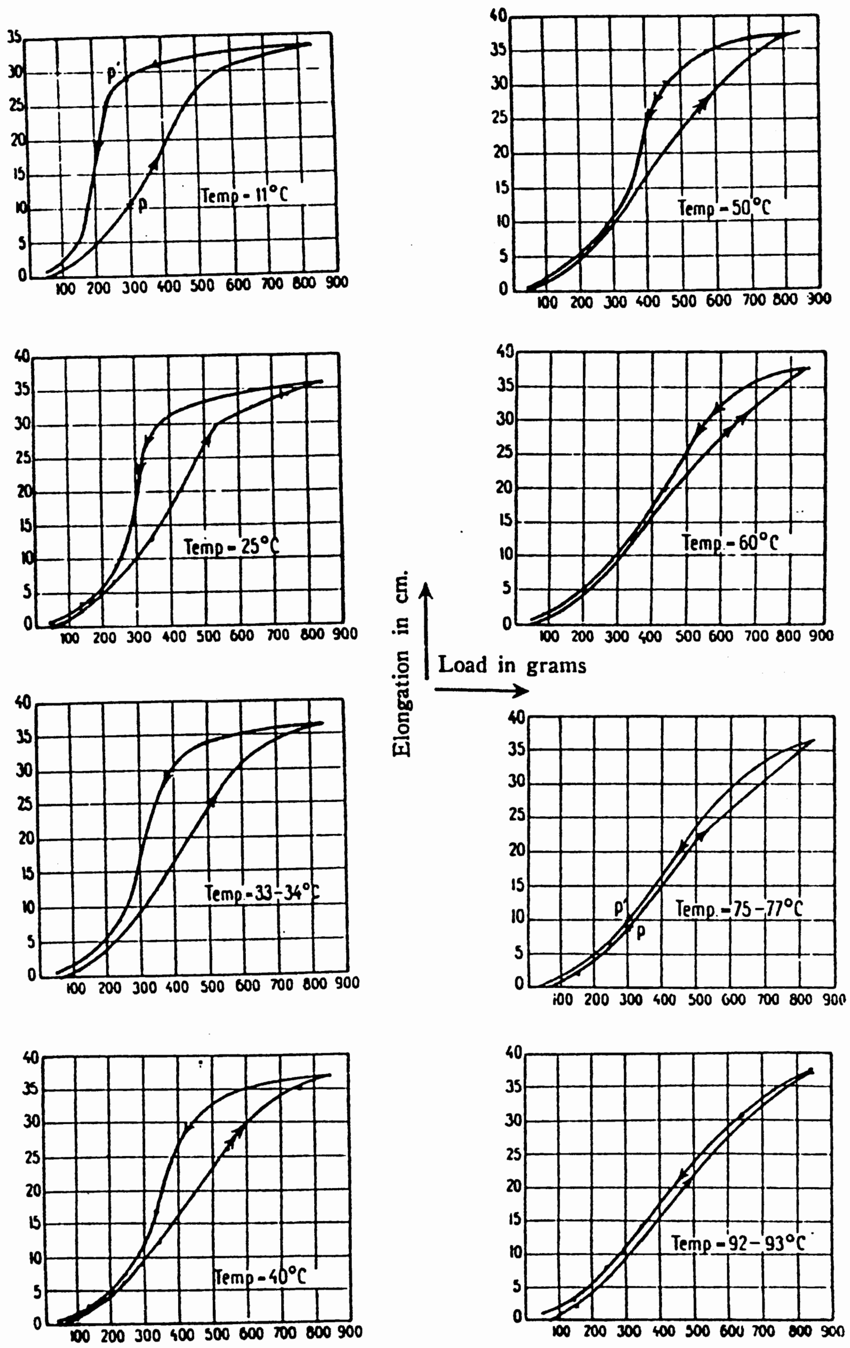
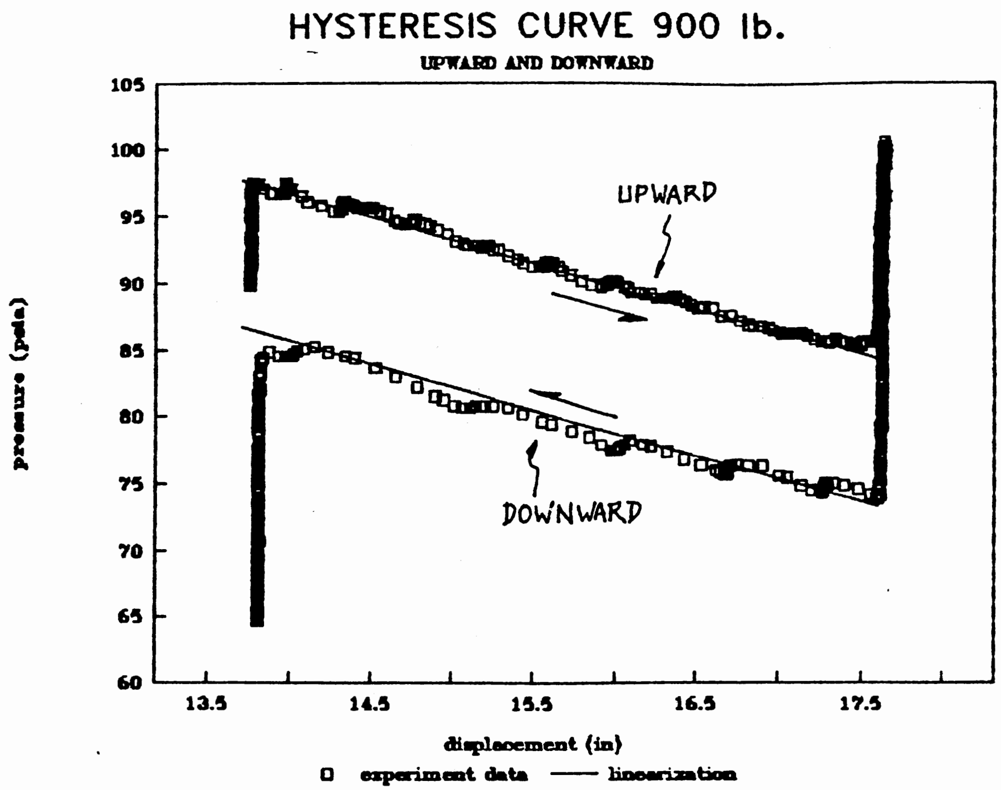


Figure 13. Typical Hysteresis Curve [12]

Initially, the system is totally uncharged. After placing a base weight on the platform, the experiment is started by turning the compressor on with the vent valve closed. The displacement and the pressure inside the air spring were measured by transducers and were recorded into a data file. The system was charged until the platform reached its maximum height (the air spring could not expand any more.) At this point, the second step of this experiment was executed. The compressor was turned off and the system was discharged through the discharge vent valve. The procedure is complete after the platform returns to the uncharged position. These measurements are repeated for various load weights. The simulation involved load weights of 900 lb, 1000 lb, 1100 lb, and 1200 lb, in order to develop the hysteresis curves for these loads.

Emperical Hysteresis Curve

Figures 14 and 15 show the emperically derived hysteresis curve. The initial height of the pneumatic spring is 13.76 inches. Referring to figure 14, with 900 lb load the air spring is charged to a pressure of about 98 psia. During this time, the displacement is unchanged since the system does not have enough pressure to lift the load on the platform. After the air spring pressure reaches 97 psia, first movement was recorded. The platform rose and continued moving until it reached the maximum height about 17.61 inches. During this rise, the pressure decreased



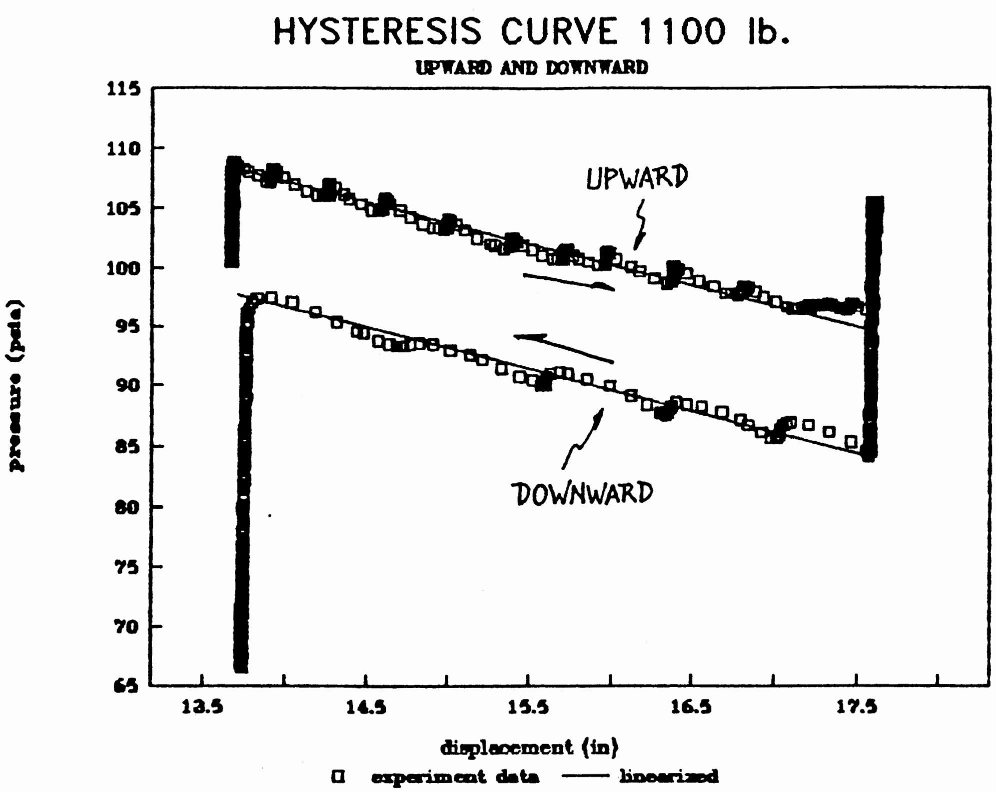
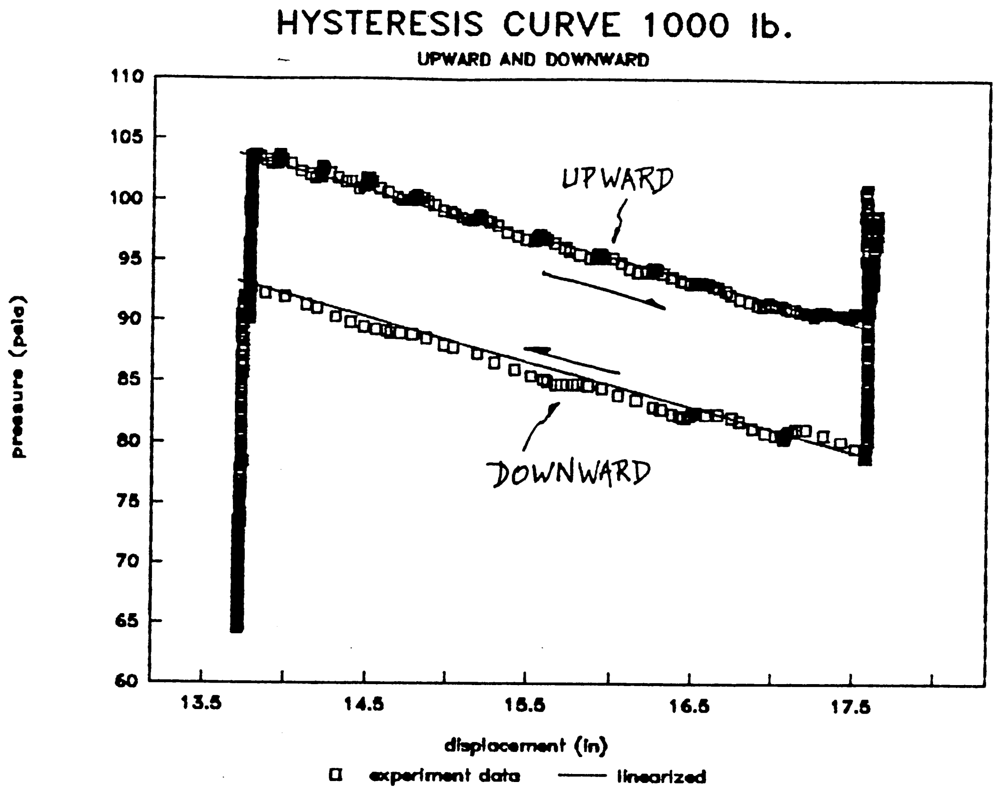


Figure 14. Emperical Hysteresis Curve for 900 and 1100 lb



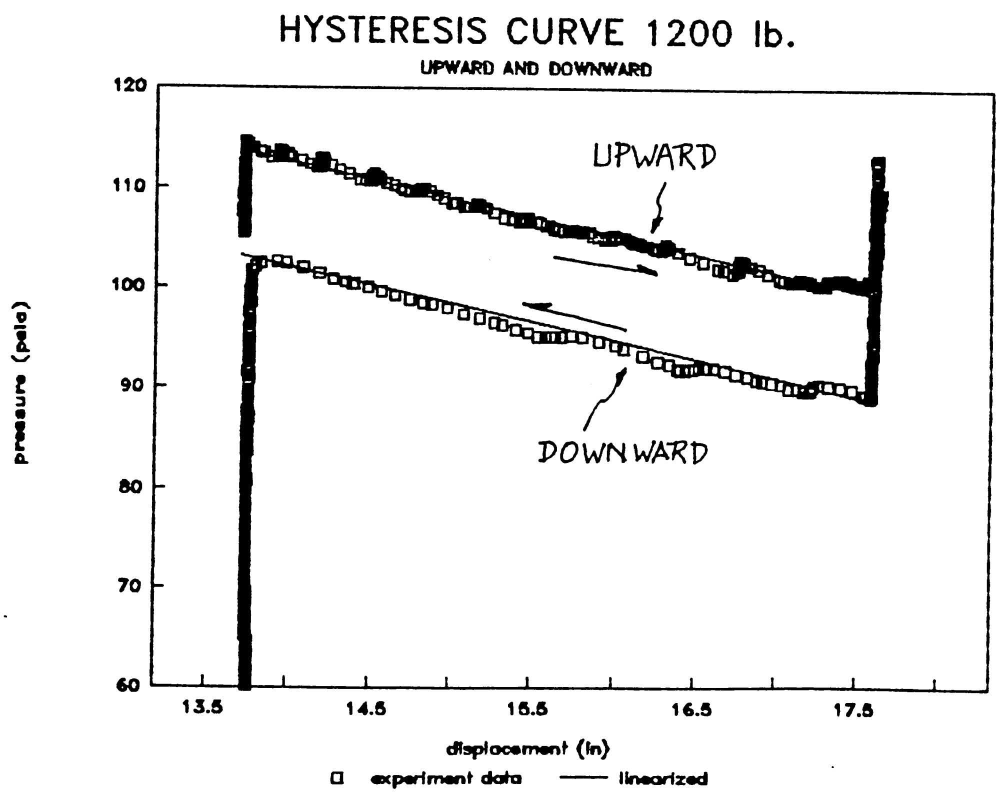


Figure 15. Emperical Hysteresis Curve for 1000 and 1200 lb

linearly with respect to the displacement from 97 psia to 86 psia. This pressure decrease is due to an area variation as the air spring extents. After the platform reached maximum displacement the upward motion stopped, however the pressure continued to increase because the air spring continued to charged. The system was then discharged. The displacement did not change until the pressure reached 74 psia. At this pressure, the platform started moving down (while the pressure is increasing) and makes a new curve located about 12 psi below the first curve. This new curve continued while the system was being discharged until the pressure increased approximately to 84 psia and the platform returned to its original height.

As shown in figure 15 with a 1000 lb load, the pressure increased vertically (without displacement change) to 103 psia and then dropped to 92 psia at the end of the platform travel. The discharge motion started at 79 psia pressure and ended at 90 psia when the platform reached the initial height. The 1100 lb, shown in figure 14, began motion at 109 psia. The pressure continued decreasing to 97 psia while the displacement went to 17.6 inches. The reverse motion direction started at 85 psia, and reached a pressure 96 psia at 13.76 inches. The upward motion curve with the 1200 lb load, shown in figure 15, began at 114 psia and ended at 101 psia while the downward curve started at 90 psia and stopped at a pressure of 102 psia.

The upper curves in the figures are the path taken when the load goes upward and the lower curves are when the load descend. The pressure measured in the experiment is the pressure of air inside the air spring, i.e. P_2 in psig. If no hysteresis existed in the system, P_2 should be a single curve. The curves mean that for the system to be able to let the load move downward at a certain position, air pressure inside the air bag, P_2, has to be less than the lower curve pressure at that corresponding position (displacement). For the system to have a force large enough to lift the load, P_2 has to be greater than the upper curve pressure. These curves of P_2 are different for different loads.

Hysteresis Curve Linearization

In this simulation, the curves are linearized as shown by the straight lines in figures 14 and 15. The following formulae represents the linearized hysteresis characteristic of the air spring.

Load = 900 lb

UPlimit = -3.43 y + 144.7

DPlimit = -3.43 y + 133.7

Load = 1000 lb

UPlimit = -3.67 y + 154.03

DPlimit = -3.67 y + 143.53

Load = 1100 lb

UPlimit = -3.5 y + 156.2

DPlimit = -3.5 y + 145.7

Load = 1200 lb

UPlimit = -3.67 y + 164.03

DPlimit = -3.67 y + 153.53

UPlimit is the pressure corresponding to the displacement, y, on the upward curve while DPlimit refers to the downward curve. The upward curve, corresponds to the motion as the platform moves upward. The downward curve, on the other hand, is the lower curve relating to the downward movement traced by the platform. Both UPlimit and DPlimit are in psia while y is in inch.

System Modelling with Hysteresis

To model the pneumatic system with hysteresis, equations (14) and (19) must be modified. Equation (14) after substituting in $P_g = P_2 - P_a$ becomes:

$$m \frac{\ddot{y}}{g_c} + B\dot{y} = (P_2 - P_a - P_{rub})A - w. \qquad (22)$$

In the measurement experiment, the hysteresis characteristic was recorded by measuring the value of P_2 (in psig) or $P_2 - P_a$, while the platform was moving up and down. The value of $P_2 - P_a$ corresponding to the upper curve or the pressure that was necessary to lift up the platform was defined by UPlimit. The pressure that allowed the load to push the platform down (lower curve) was defined by DPlimit. The condition at the equilibrium point was defined by $\ddot{y}$ = 0.0 and $\dot{y}$ = 0.0. During the charging of the air into the pneumatic spring, the pressure P_2 increased. The platform did not move till $P_2 - P_a$ reached UPlimit. At this point $\ddot{y}$ = 0.0 and $\dot{y}$ = 0.0 . Substitution into equation (22) produced

$$0.0 = (UPlimit - P_{rub})\, A - w$$

$$P_{rub} = UPlimit - \frac{w}{A}. \tag{23}$$

On the other hand, when the air was discharged, the platform was ready to move downward when $\ddot{y}$ = 0.0 and $\dot{y}$ = 0.0 . Therefore the equations become for this condition

$$P_{rub} = DPlimit - \frac{w}{A} \tag{24}$$

$$P_{rub} = Plimit - \frac{w}{A} \tag{25}$$

where Plimit = UPlimit during upward motion and Plimit = DPlimit for downward movement. At some arbitrary value of P_2, $\ddot{y}$, and $\dot{y}$ (same load w),

$$m\,\frac{\ddot{y}}{g_c} + B\,\dot{y} = \{P_2 - P_a - (Plimit - \frac{w}{A})\}\,A - w$$

$$m \frac{\ddot{y}}{g_c} + B\,\dot{y} = (P_2 - P_a - Plimit)\,A \tag{26}$$

Therefore equation (19) becomes

$$\frac{d\dot{y}}{dt} = \frac{g_c}{m}\,\{(P_2 - P_a - Plimit)\,A - B\,\dot{y}\} \tag{27}$$

Simulation

Simulation Parameters

The response of the system to increased and decreased weight on the platform can be shown by simulation. At initial weights of 900 lb and 1000 lb the platform height was about 15.65 inches. The load is varied from the base load by 200 lbs. Simulation was conducted by imposing the initial weight for 5 seconds. At the 5th second a 200 lb load was added to the system and remains until the 35th second. Finally the 200 lb load was removed and the simulation was completed at the 55th second.

Some of the parameters used were obtained from information provided by the manufacturer while others were derived from direct measurement and experimentation. The remaining parameters were difficult to measure and were chosen from the possible range. (Study about sensitivity of chosing polytropic constant, n, is presented in appendix A.) These parameters were not equal between the 900 and the 1000 lb. For example, the area and the volume of the air bag were different for each case. Most of the parameters were simplified. Nonlinear (time dependent) variables were linearized and assumed uniform throughout the simulation. These simplifications and assumptions lead to slight differences between the simulation and the experiment result.

The parameters used are:

Pipe	: volume	12.5 cuin.
Solenoid valve	: orifice diameter	2.0 mm.
	C_d to the air spring	0.8
	to the intersection	0.4
Vent valve	: orifice diameter	2.2 mm.
	C_d	0.12
Motor	: K_a	0.926 lb in / volt
	power source e_a	13.5 volts
	moment of inertia J	0.25 lb in sec^2
	Damper B_a	0.1 lb in sec
Compressor	: capacity D_a	0.0161 cuin / rad
	leakage C_a	0.0001 cuin psi/sec
	Torque coefficient	2.0
Air spring	: initial volume (900)	270 cuin.
	(1000)	210 cuin.
	Area	is given below.
Platform	: initial height	15.7 in.
	minimum range	15.6 in.
	maximum range	15.8 in.
	damper up (900)	62.0 lb sec / in

	down (900)	10.0 lb sec / in
	damper up (1000)	58.0 lb sec / in
	down (1000)	160.0 lb sec / in
	mass	50.0 lbm.
Atmosphere	: Compressibility Z	0.95
	specific heat ratio k	1.4
	polytropic constant n	1.1
	temperature	535 Rankine
	gas constant R	640.08 in lbf / lbm R
	gravity g	386.4 in / sec^2

The effective area of the air spring was given by the following linearized equations:

DL = Load / 100,

A1 = 6.711826 * DL,

A2 = -0.644472 * DL * DL,

A3 = 0.026514 * DL * DL * DL,

A4 = -0.000393 * DL * DL * DL * DL,

CC1 = -12.176065 + A1 + A2 + A3 + A4;

CC2 = 0.1271 + 0.0121 * DL,

area A = CC1 + CC2 * y, where y is displacement.

Simulation Result

Figure 16 shows the simulation result of the system with 900 lb initial load. After the 200 lb load is added, the platform height drops from an initial 15.65 inches to 14.53 inches and stays there for 16 seconds before increasing again to initial height. After stabilizing at

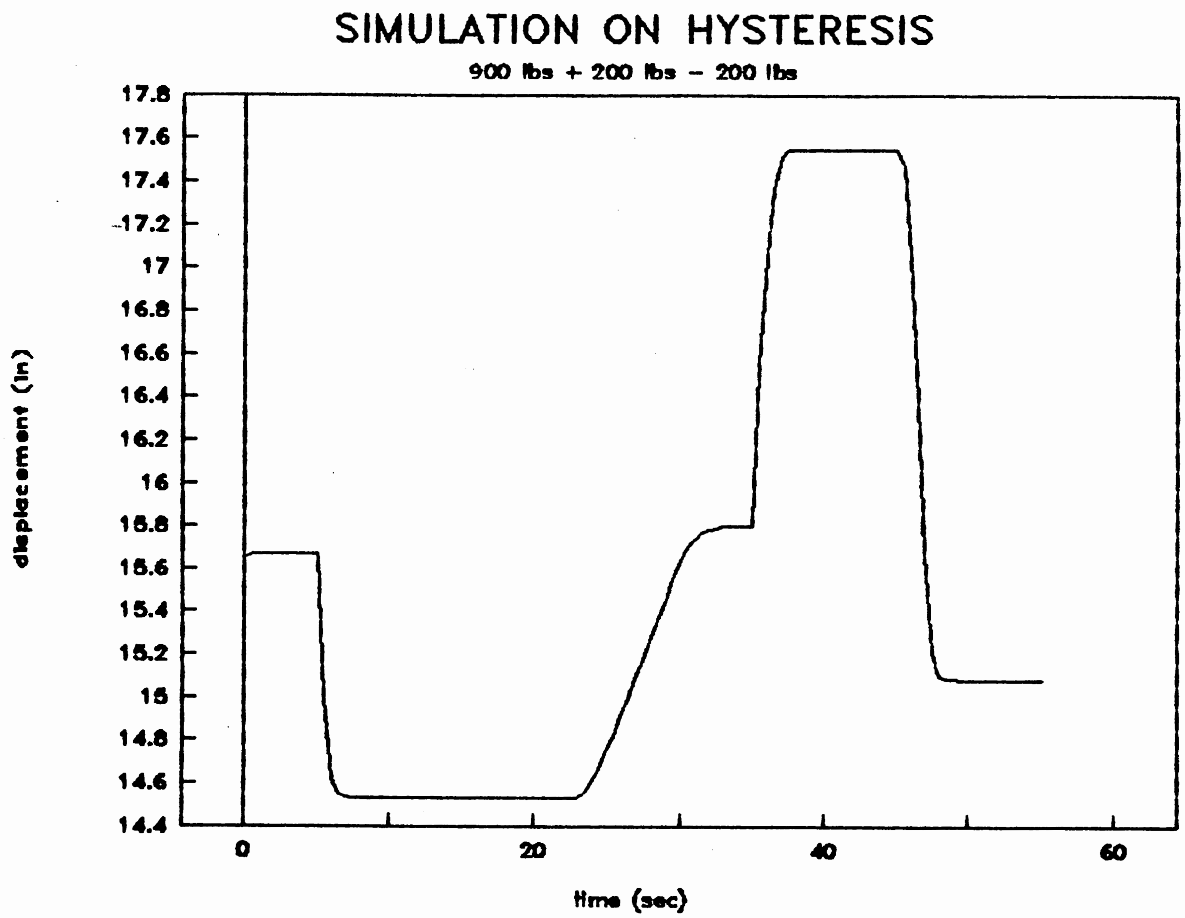


Figure 16. Simulation of the 900 lb Load

the set point height, the load is removed and the platform jumps to 17.54 inches height and remains for 7.5 seconds before it returns.

A graph of pressure versus time for this simulation will give a better idea of what is occuring. (See figure 17). Before the input, the pressure is 89 psia. At the 5th second when the 200 lb load is added, the pressure jumps to 96 psia almost instantly (that is the starting pressure when the platform stops dropping.) In the next 16 seconds while the platform is in rest, the pressure increases to 105 psia. This is the pressure that overcomes the hysteresis to lift the platform toward its desired height. The pressure change rate is different between first increase from 89 to 96 psia and later increase from 96 to 105 psia. The first increase is caused by the sudden reduction of the volume in the air spring due to the load addition while the second increase is due to the compressor as it charges the air spring. The pressure then drops to 101 psia while the platform climbs its way upward. This pressure decrease occurs after overcoming the hysteresis (and the static friction) because the area changes as the air spring extends. This can be seen in the hysteresis curves. The pressure continues dropping to 84 psia as the load is removed. The pressure decreases more slowly to 74 psia, while the platform maintains a constant height 17.55 in. The explanation for the difference in pressure change rate is analogous to that for the increasing pressure portion. The first drop in

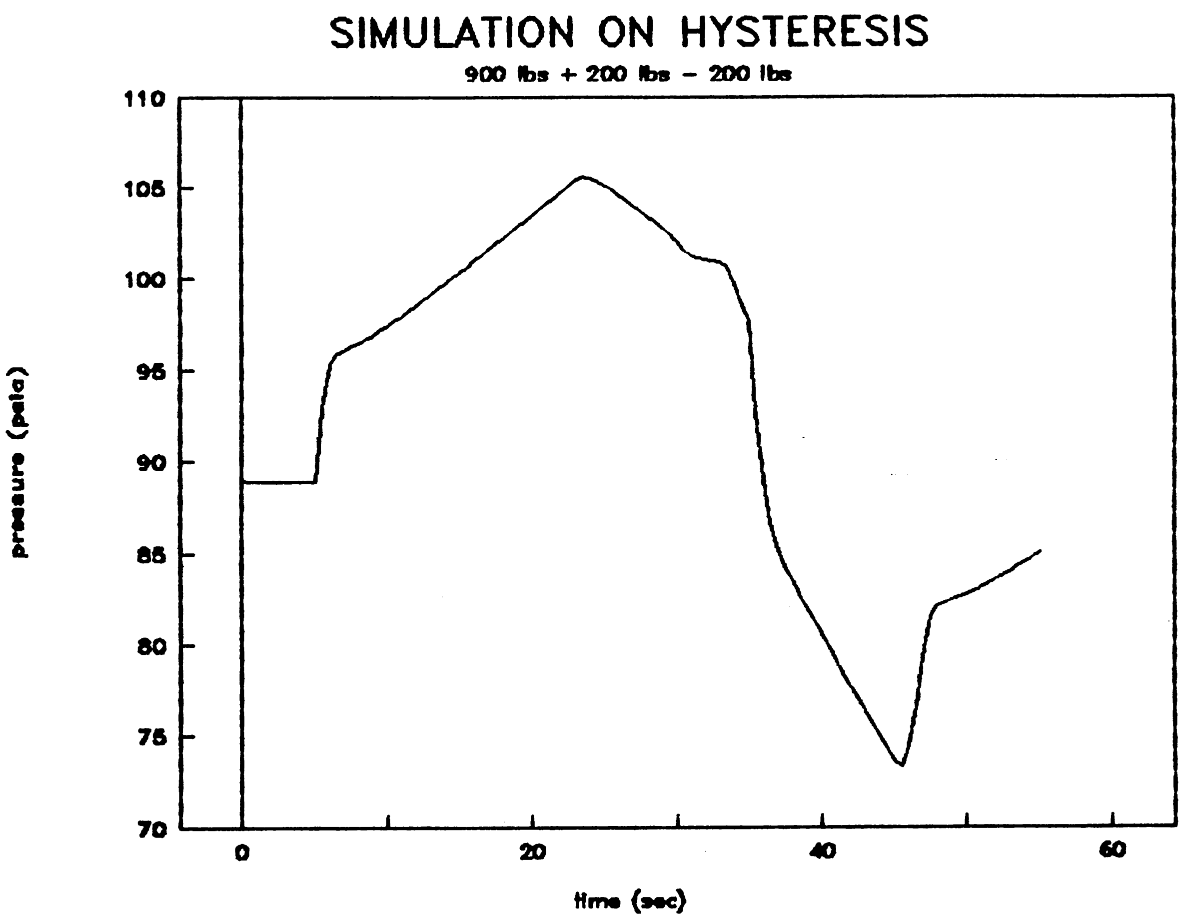


Figure 17. Pressure Graph of 900 lb Simulation

pressure is caused by the sudden expansion of volume as the 200 lb load is removed and the air spring extends while the latter is produced as the system is discharged. The next increase of the pressure corresponds to the return of the platform toward its allowable range.

The simulation with a 1000 lb initial load produced similar output (figures 18 and 19.) The 200 lb additional load moves the platform to 14.41 inches at an air spring pressure of 101 psia. The displacement remained unchanged until the system reached a pressure to 111 psia 13 seconds later. About 10 seconds was needed to bring the platform back to its initial height of 15.65 inches at which point the pressure gradually decreases to 107 psia. The next 3 seconds the system is at rest. The platform jumped the moment the 200 lb load is removed. The maximum position was 17.43 inches and the pressure was 90 psia. Then the pressure dropped to 78 psia allowing the platform to return after pausing for 7 seconds.

For a better understanding of the process, consider figure 20 which is for the simulation with an initial load of 900 lbs. In this figure, the hysteresis curves of the 900 lb and 1100 lb initial loads have been combined. The beginning point A 15.65 makes displacement with a pressure of 89 psia. During the first five seconds, since nothing is happening, the system maintains its position and pressure, hence during this time, the system stays at point A. At the fifth second, the load is added, the platform sinks to 14.5

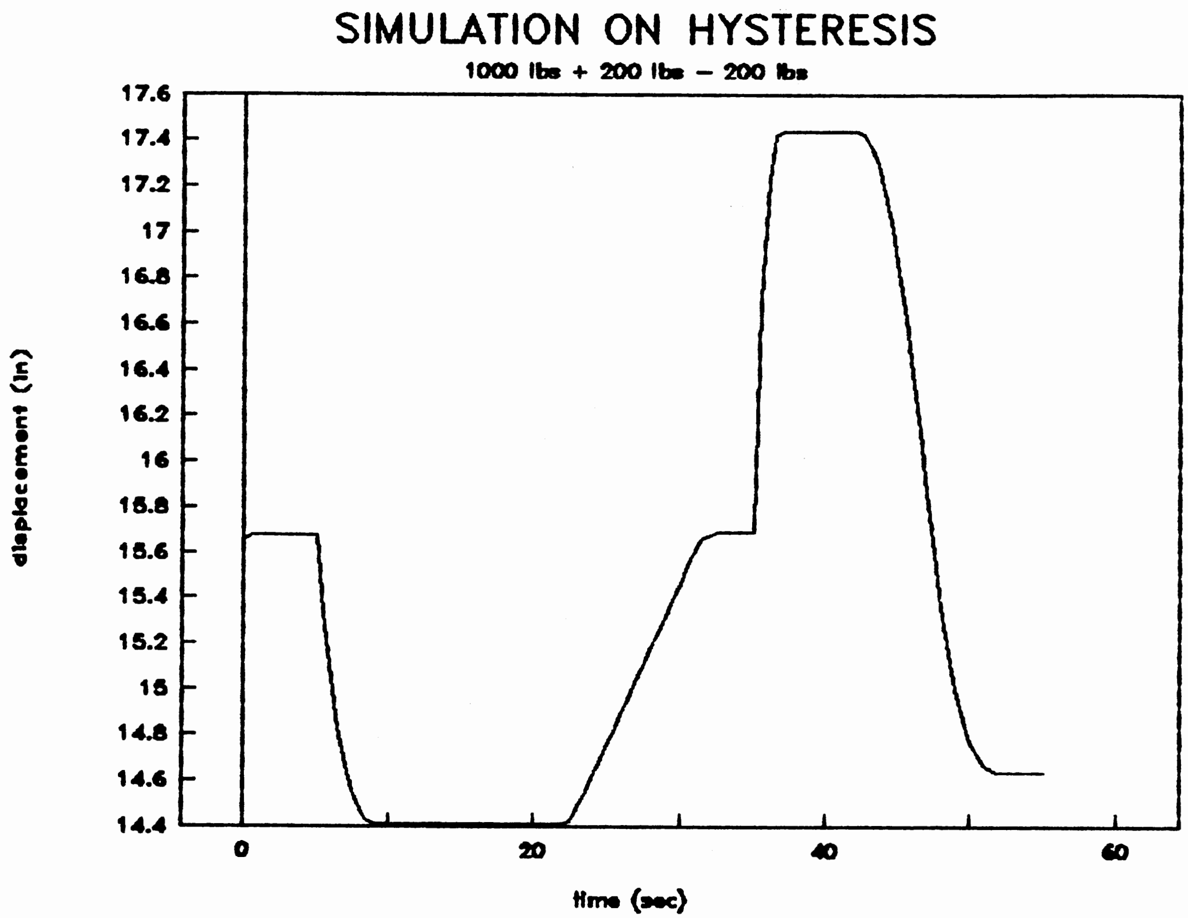


Figure 18. Simulation of the 1000 lb Load

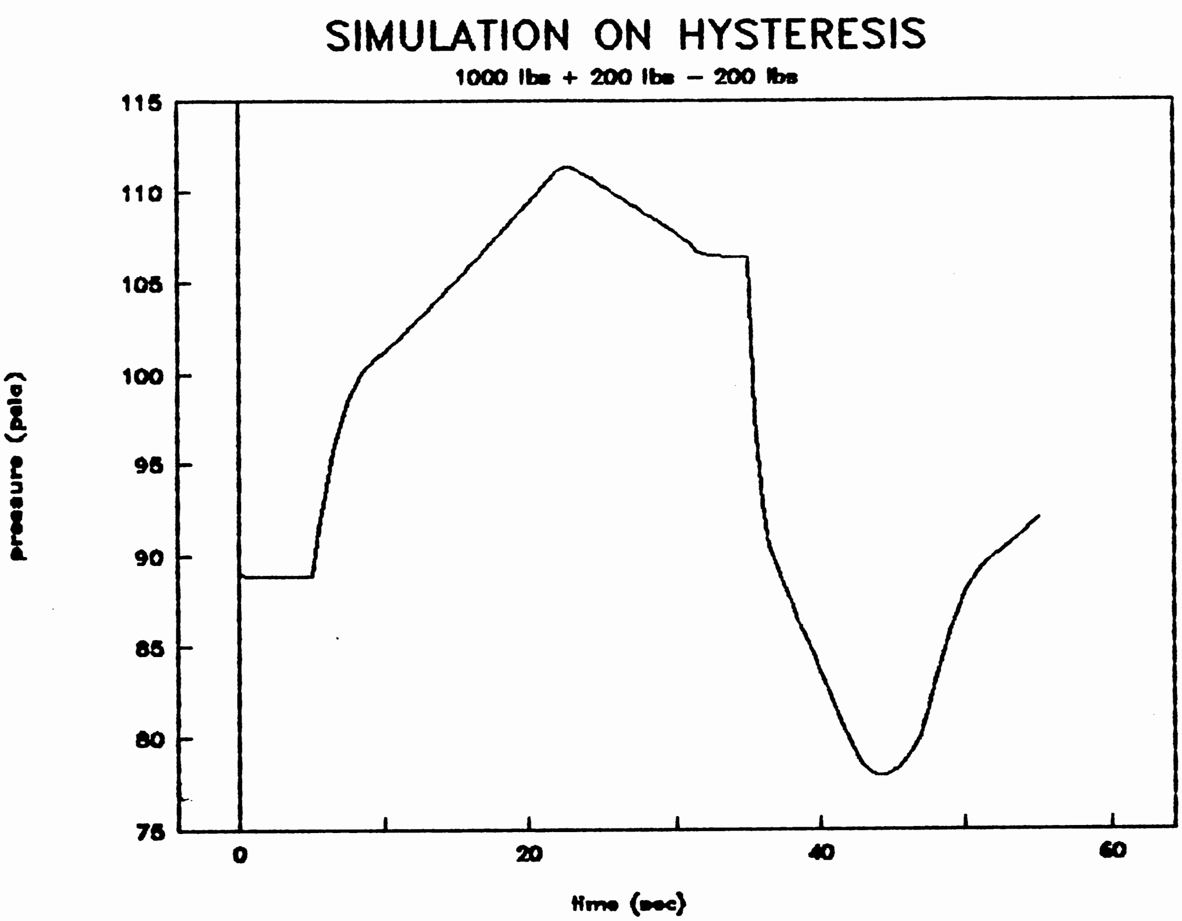


Figure 19. Pressure Graph of 1000 lb Simulation

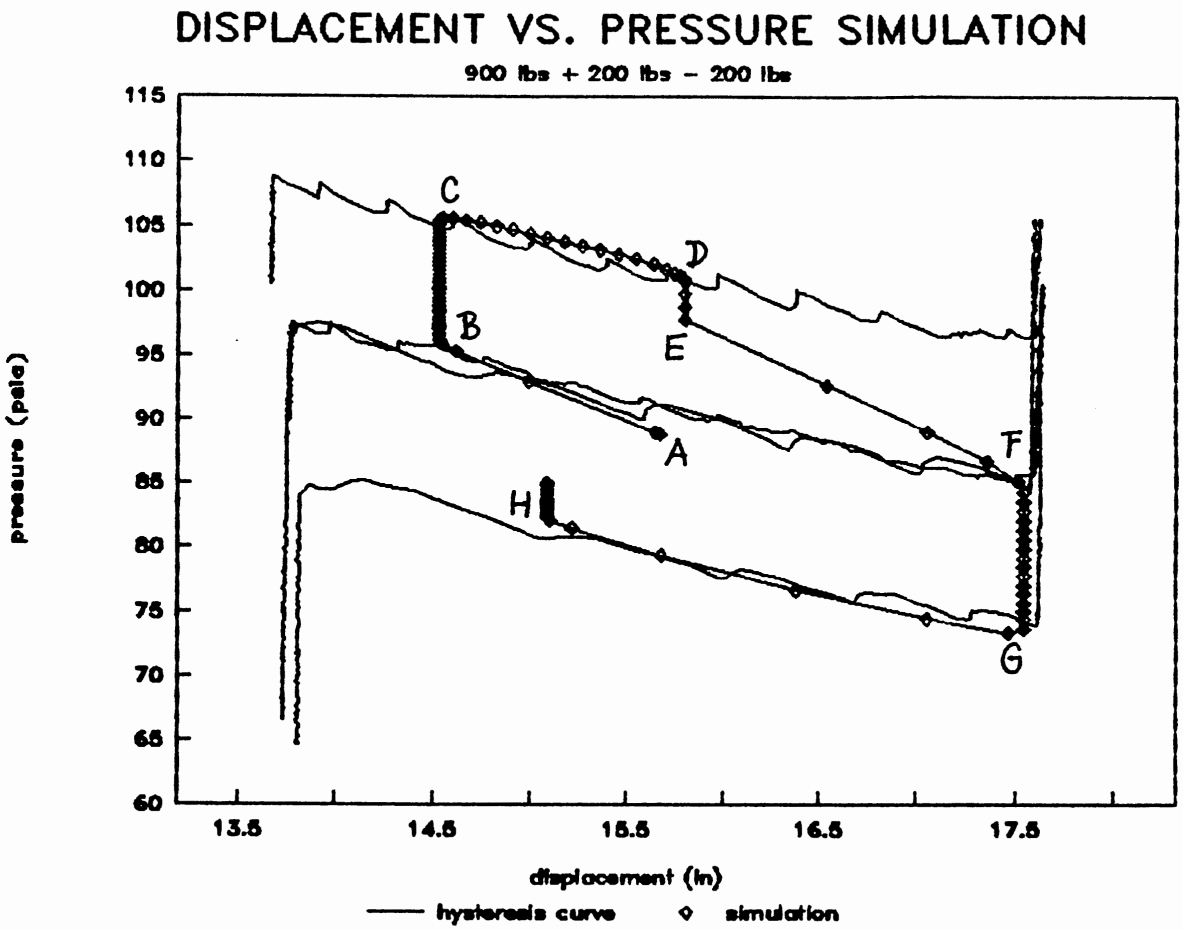


Figure 20. Displacement-pressure Graph of 900 lb Simulation

inches and the pressure increases to 96 psia, shown as point B. At this point, the platform stops for 16 seconds and the pressure continues to increase to 105 psia. Point C defines the point where the pressure in the air spring reached a value which overcomes the hysteresis. Notice that point B is located on the lower curve of the 1100 lb hysteresis curve. Between B and C, the displacement does not change, however the system is still charging until the pressure reaches the upper curve of the 1100 lb hysteresis curve. The pressure at this point is sufficient to overcome the hysteresis and lift the platform upward. The system moves from C to D along this upper curve while the platform is moving upward. When the system reaches point D, the platform is in the desired range. The platform stops at point D while the pressure in this simulation continues to decrease. This pressure is assuming its steady state equilibrium pressure. When the time is 35 seconds, the 200 lb load is removed. The platform jumps, the pressure drops, and the system moves to F. Notice that F is lying on the upper curve of the 900 lb hysteresis since the load is now only 900 lb and the platform has moved upward. At point F the platform remains stationary and the pressure is decreased to point G. The system moves from G to H along the curve while the platform is lowering to the desired height. In 1000 lb simulation, shown in figure 21, the system does not quite follow the curve because the selection of the resistance of the discharging valve is too small

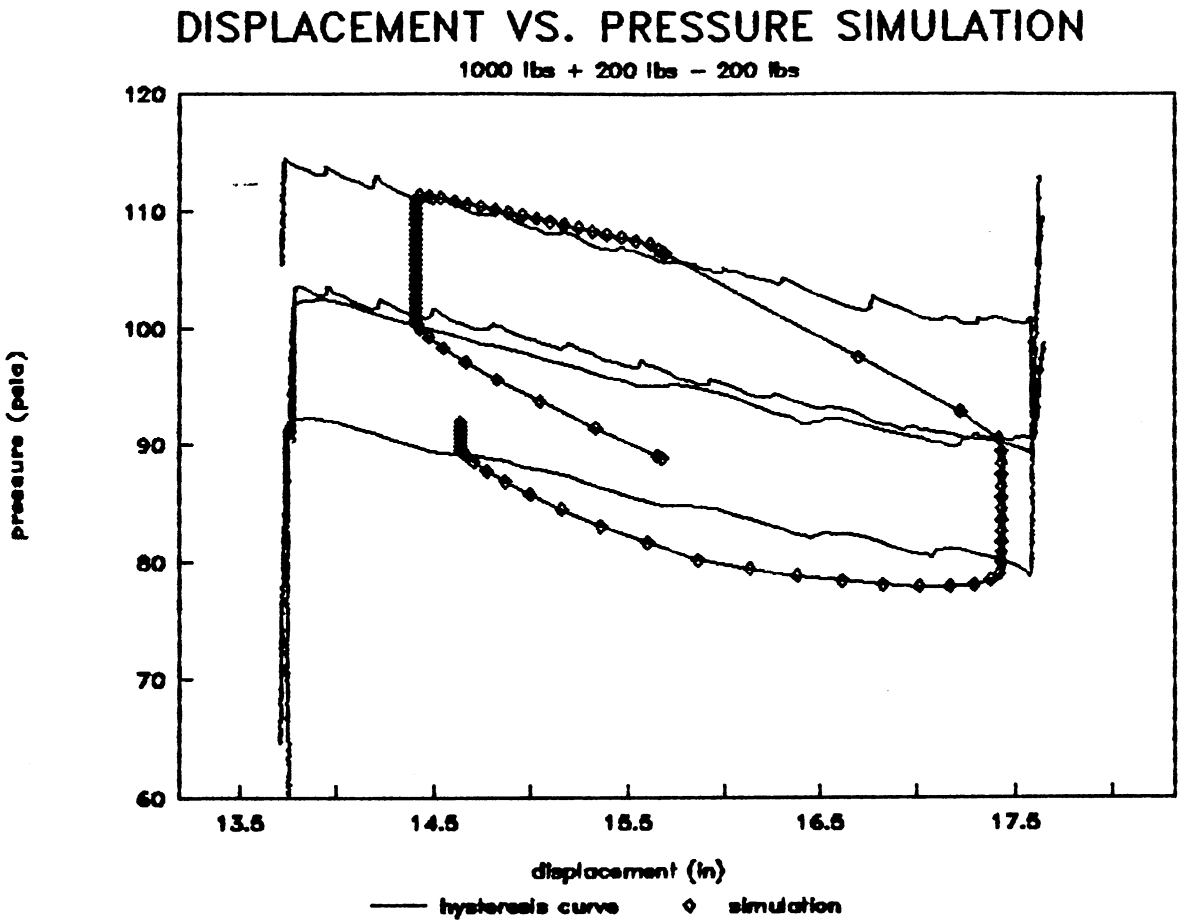


Figure 21. Displacement-pressure Graph of 1000 lb Simulation

producing a lack of pressure.

To understand the effect of hysteresis on the system, visualize what has been really happening when the air is pumped into the system. Start at point A (referring back to figure 16.) The condition that exists when the platform stops at A reveals that at that time the system is in equilibrium. This means that the force exerted by the air pressure in the air bag is equal to the force exerted by the weight, the ambient pressure and the reactive force of the bag.

This reactive force is due to the internal rubber force when the rubber is retracted. (The fact that platform drops down from its initial height implies the air bag has shrunk in size or that the rubber retracts. Please recall that the rubber is always in stretched condition.) The force needed to keep the rubber under retraction movement at a certain elongation is less than that needed under extension: P_{rub}(retract) < P_{rub}(extend). To push the load upward, the system needs extra pressure. That is what causes the platform to delay at that same location till P_{rub}(A) = P_{rub}(retract) reached P_{rub}(A) = Prub(extend). It is also true in the case where the load is removed. In the figure 16, notice the time needed by the system to return to its initial position is somewhat less during weight removal than during its addition. This is because it is much faster to discharge air than to charge the system. (In order to discharge the system quicker, an effective way would be to use a valve with less resistance.

The Remedy

To overcome the problem of hysteresis, it is necessary to add more air flow to the system so that the system builds enough pressure to overcome the hysteresis more quicker and reduce the time needed to move from B to C. The best remedy would be a compressor with larger capacity. However, such a compressor would be hard to find. Moreover, purchasing a larger compressor greatly increases system cost. In addition, using the large capacity compressor would not be efficient since most of the time the system is in a stable state (when there is no load change) and also during the load removal, the compressor is in off condition.

Therefore, instead of using larger compressor, it would be more feasible to solve this problem by adding another branch to the system and connecting it to the accumulator.

Accumulator

The working principle of an accumulator would provide extra volume to the system in the case where more flow rate is needed. An accumulator more or less works like a capacitor in electric circuit to store the energy. Statically, it does not change anything, but dynamically it reduces the shock of pressure increase and compensates for pressure decrease. Installing an accumulator into the system changes the damping ratio and natural frequency, but since stability is not a critical issue in this system, it is omitted in the calculation.

There are various types and shapes of accumulator. In this simulation a simple tube type accumulator and a diaphragm type accumulator are used. The tube type accumulator as the name implies is simply an empty tube. This tube is an accumulator since it can satisfy the function to provide the system more volume. The propagation of the pressure with respect to time, hence, becomes as follows

$$\frac{dP}{dt} = \frac{\beta}{V} \left\{ \frac{g_c\ R\ T}{g\ P} \left(W_{in} - W_{out} \right) - \frac{dV}{dt} \right\} \tag{28}$$

W_{out} = 0 since the other end is blocked, dV/dt = 0.

$$\frac{dP}{dt} = \frac{\beta}{V} \frac{g_c\ R\ T}{g\ P} W_{in} \tag{29}$$

W_{in} is the weight flow rate into the accumulator. In case the air leaving the accumulator to the system, a minus sign is used to imply opposite direction. W_{in}(out) = $-W_{in}$. Look back to the equation (29), if W_{in} positive, the accumulator pressure increase with respect to time, on the other hand, when W_{in} negative, i.e. the air flows from the accumulator to the system and the accumulator pressure drops.

W_{in} was determined by the pressure of the accumulator as compared to that of the system. Whichever pressure is less determines the direction the fluid flows. The magnitude is dependent upon the restriction between them. Letting the restriction to be a valve, the flow:

for $\frac{P_d}{P_u} \geq (\frac{P_d}{P_u})_{crit}$,

$$W_a = C_d A_o C_1 \frac{P_u}{\sqrt{T_u}} (\frac{P_d}{P_u})^{\frac{1}{k}} \sqrt{1 - (\frac{P_d}{P_u})^{\frac{k-1}{k}}}$$

and for $\frac{P_d}{P_u} \leq (\frac{P_d}{P_u})_{crit}$,

$$W_a = C_d A_o C_2 \frac{P_u}{\sqrt{T_u}}. \qquad (30)$$

Then rewriting equations (29) and (16) respectively,

$$\frac{dP_3}{dt} = \frac{\beta}{V} \frac{g_c R T}{g P_3} W_a \qquad (31)$$

$$\frac{dP_1}{dt} = \frac{\beta}{V} \{\frac{g_c R T}{g P_1} (W_c - W_s - W_v - W_a)\} \qquad (32)$$

P_3 is the pressure inside the accumulator and W_a is W_{in} to and from the accumulator.

The parameters of the accumulator and the valve are as follows:

Accumulator : volume (tube) 50 cuin.

Accumulator

valve: orifice diameter 2.2 mm.

C_d 0.3

A diaphragm accumulator is a tube type accumulator with additional diaphragm inside of it. This diaphragm functions to increase pressure to the air inside the accumulator for any given volume. The high elasticity of the diaphragm makes it possible to let the accumulator function as a small volume accumulator while its pressure is small and as a large volume one when the pressure inside is high.

The relationship between the pressure and the flow rate of the diaphragm accumulator is similar to that of the tube type accumulator. However, the accumulator volume is now a function of pressure instead of a constant value. $V = f(P)$ and dV/dt is not zero. The characteristic of the accumulator used in this simulation is depicted in figure 22.

Using all the above formulae and conducting a simulation reveals that placing an accumulator into the system does not improve the performance. In fact, it might even make the result worse. The time the platform takes to move back to its initial position when the load is added is still long. This result is not surprising considering that the compressor will now need more time to fill the larger volume due to the accumulator. The response is similar in the analogy of a capacitor in electric circuit. A Capacitor

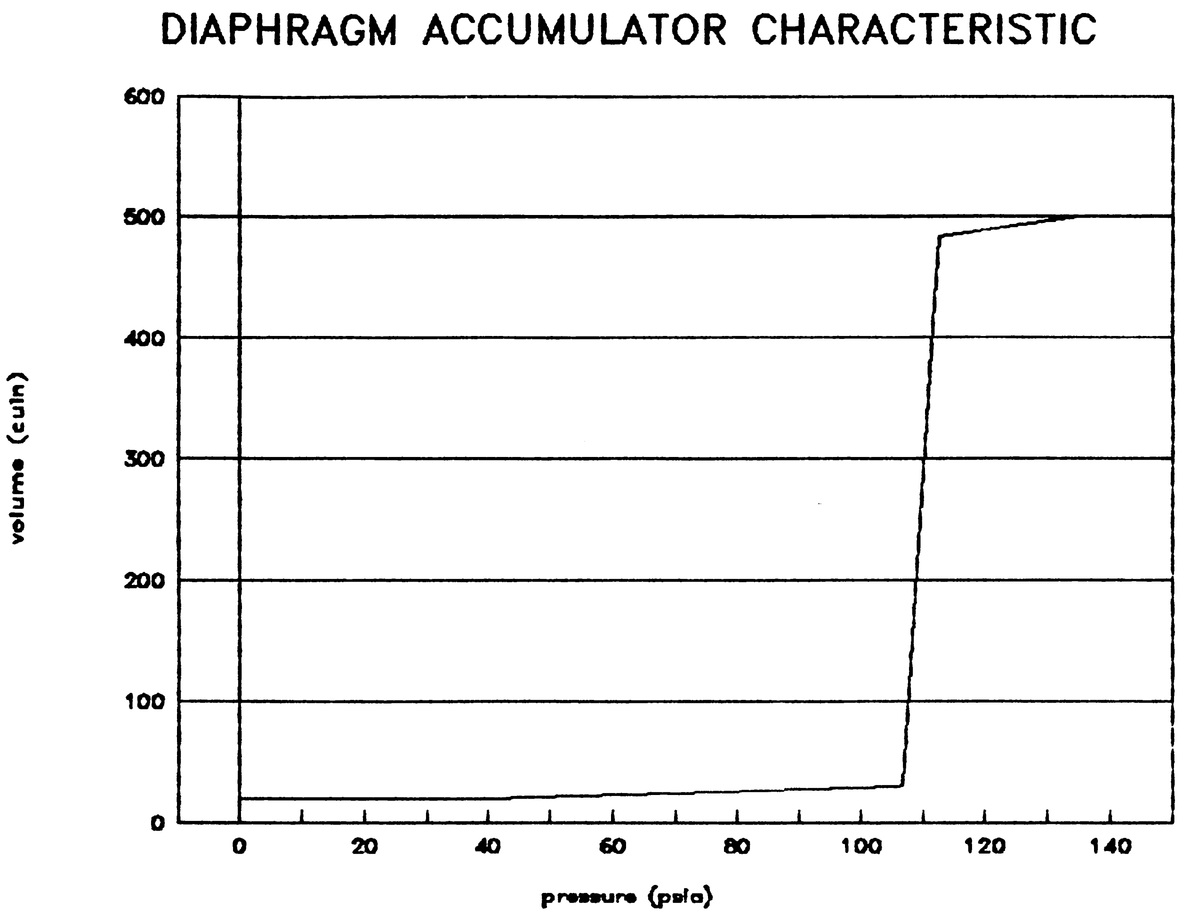


Figure 22. Characteristic of the Diaphragm Accumulator Used

slows the response of the system since it uses the energy to charge itself before letting the whole energy go to the system.

Therefore an addition to the accumulator circuit must be made. On-off control valve must be placed between the accumulator and the system. Placing an accumulator in the system is bad when it comes to system charging, but highly a charged accumulator gives quite a push to the system. Hence, to improve system performance, the accumulator should be charged beforehand in order to be used effectively when the system needs extra air flow to force the load upward.

Following is the new controlling algorithms:

1. If the height of the platform is lower than the desired height -- remember this height is a range, not a single value, then the compressor is on, the solenoid valve is open, the discharge valve is closed.

   If at this position, the accumulator pressure is greater than the system pressure, then the accumulator valve is open, otherwise it is closed.

2. If the platform height is over the desired height, the compressor is off, the solenoid valve is open, the discharged valve is also open.

   At this point, if the accumulator pressure is lower than the system pressure, the accumulator valve is open, otherwise it is closed.

3. If the platform height is in the range of the desired height, the solenoid valve is closed, the discharged

valve is closed.

In this range, if the accumulator pressure is less than the system pressure and it does not exceed 120 psig, the accumulator valve is open, otherwise it is closed.

The compressor is on if the accumulator pressure is not over 120 psig.

In algorithm 1, when the height is lower than the desired height, the system is just simply being charged. During this process care must be taken with the accumulator. Since the system is being charged, the accumulator should be opened only when its pressure is greater than the system pressure. The system pressure refers to the intersection pressure P_1 where the accumulator circuit is attached. Otherwise, the air will be devoured by the accumulator since its natural for fluid to flow from the greater pressure to the lower pressure. Opening the accumulator valve while its pressure is greater than that of the system gives energy to the system.

In algorithm 2, the system is discharged while the platform height is above the desired height. The system has too much volume and it must be discharged or the better alternative is to release the surplus air volume to charge the accumulator while the accumulator pressure is less than the system pressure. However, it should be noticed that the accumulator valve should not be opened while its pressure is greater than that of the system pressure for then the accumulator will add unwanted volume to the system. In addition, opening accumulator at this condition will only waste the volume that has already been accumulated.

Algorithm 3 relates to the system when it is in the desired range of height. The system does not need to be charged or discharged so that the solenoid valve in the line to air spring can now be closed to separate the air spring from the main system. The whole system will be in rest except for the accumulator and the compressor. This permits the time to charge the accumulator. Assume the working pressure of the air spring does not exceed 120 psig and therefore the accumulator is to be charged to maximum pressure 120 psig.

For these algorithms to be effective, the accumulator pressure should be much much higher than the system working pressure. The accumulator is charged mainly at algorithm number 3. Therefore, for the accumulator to be sufficiently charged, there should be enough time for the system to stay at algorithm 3 condition, namely at equilibrium in the desired range. Fortunately, once the system gets equilibrium at the desired height, the platform will remain there with relatively slight movement due to mechanical vibration. The other oscillation factor are readily damped out by the hysteresis. The selection of the tolerable height range effects the stability of this system. A selection of a range which is too wide causes inaccuracies, however if the range is selected too narrow, the system will oscillate. The oscillation movement will reduce the life of the system components. In realistic situation, the time between each excitation must be long enough to permit the accumulator be charged.

Simulation Using Accumulator

Figures 23 and 24 show the simulation result after using 50 cubic inch tube type accumulator. The accumulator is charged to 90, 100, and 112 psig in the system initially loaded at 900 lb. In addition, the accumulator is charged to 100, 110, 120 psig for the system initially loaded with a 1000 weight. In both cases, the usage of an accumulator does improve the performance. It reduced the displacement change and the time required to return to its initial height. The higher the accumulator charge pressure the better the result. In these figures, it appears that the accumulator is more effective in the 1000 lb system than in the 900 lb. The 1000 lb system sank less than the 900 lb because a greater friction value is used in the 1000 lb simulation. The friction value is hard to define in the test fixture because the roughness of the platform pillar is not uniform. In addition, the pillar might not exactly perpendicular to the platform that in turn would make the friction value hard to predict since it would be function of load, position and velocity. The linear assumption used in this simulation hardly matches the real thing.

Figures 25 and 26 show the simulation result of the system with diaphragm accumulator. Due to the

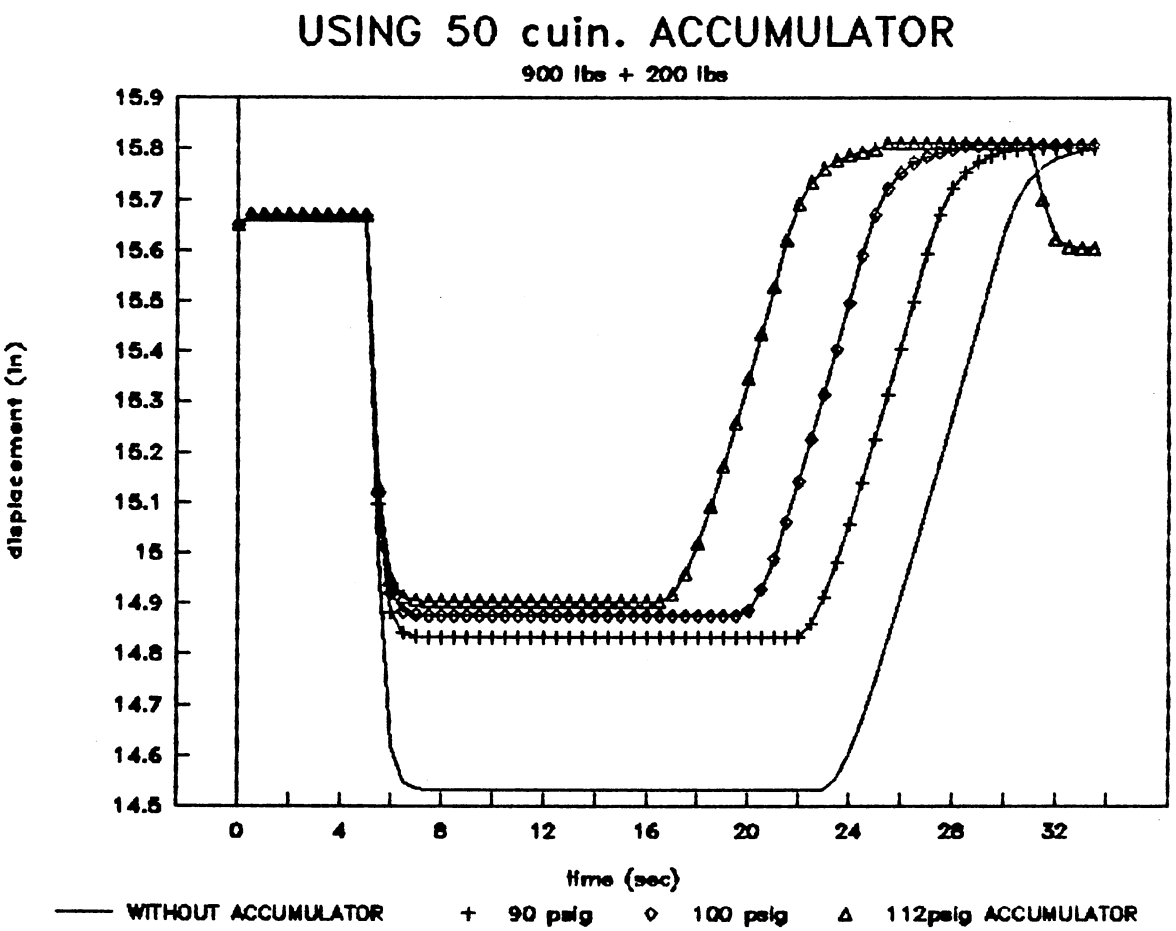


Figure 23. Simulation of 900 lb System + 50 cuin. Tube Accumulator

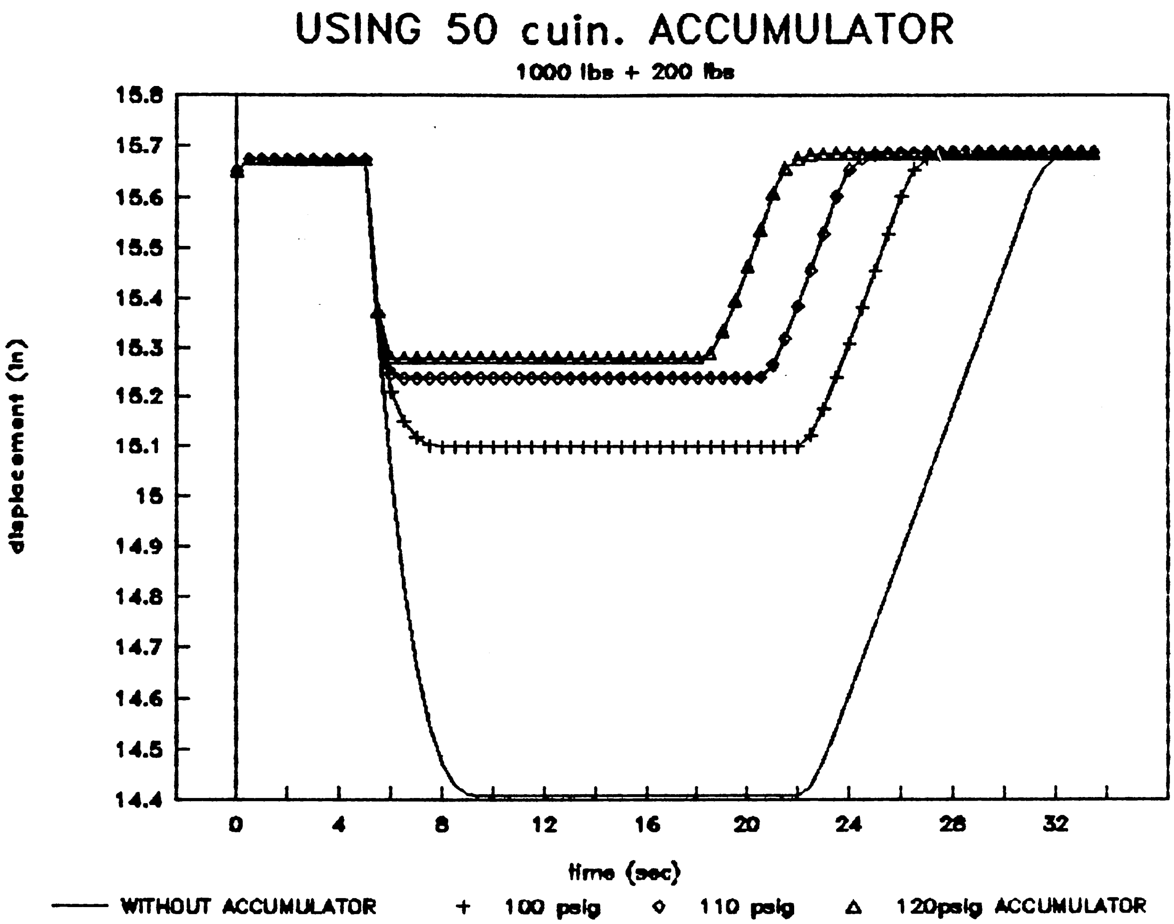


Figure 24. Simulation of 1000 lb System + 50 cuin. Tube Accumulator

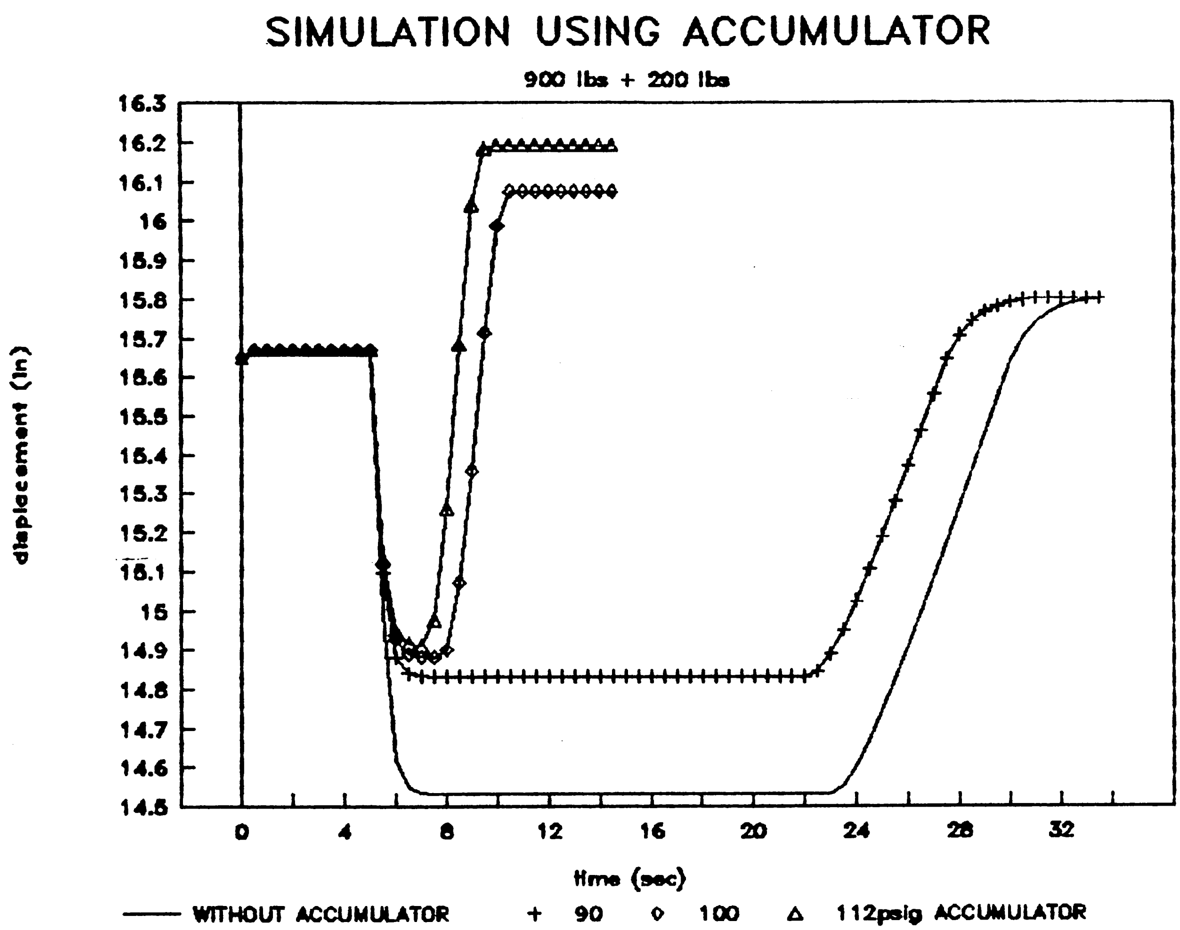


Figure 25. Simulation of 900 lb System + Diaphragm Accumulator

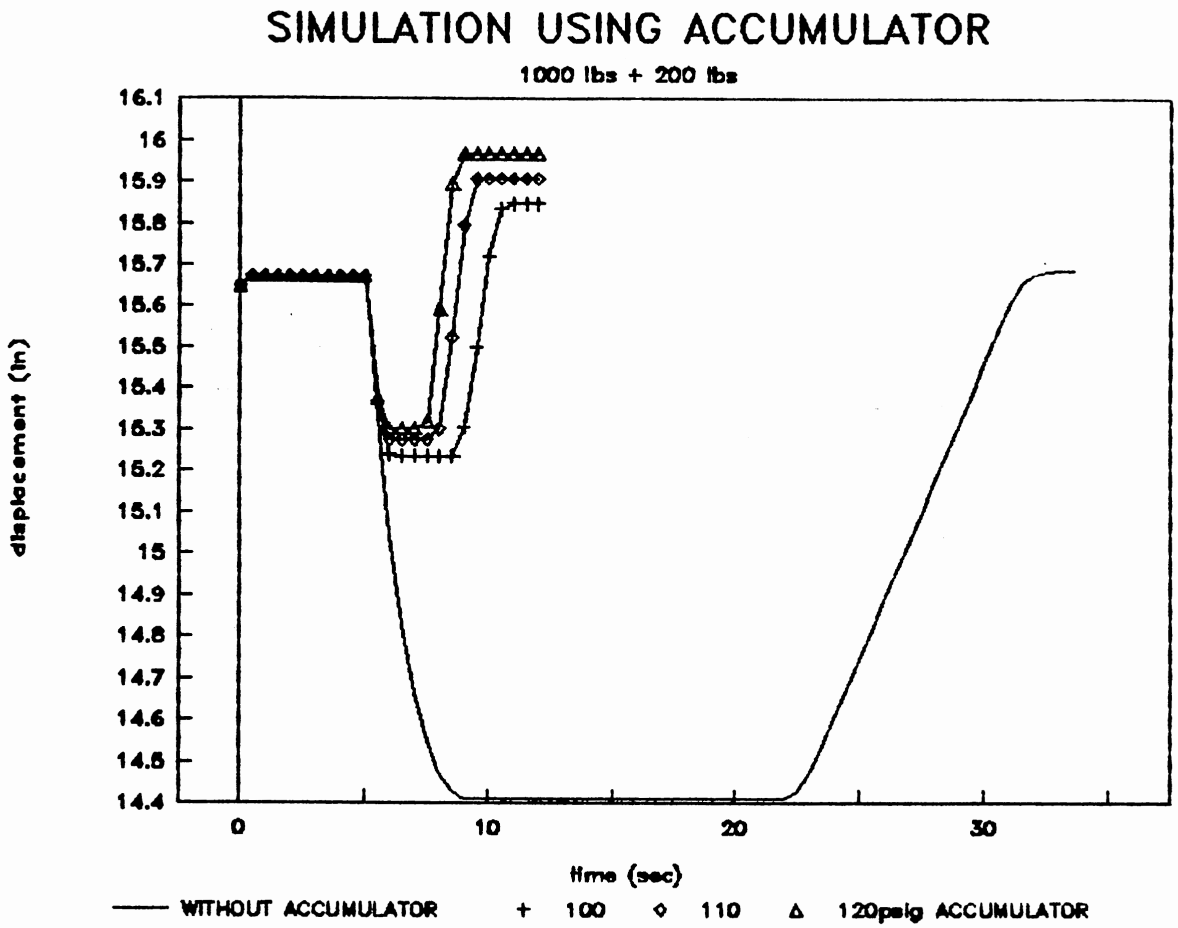
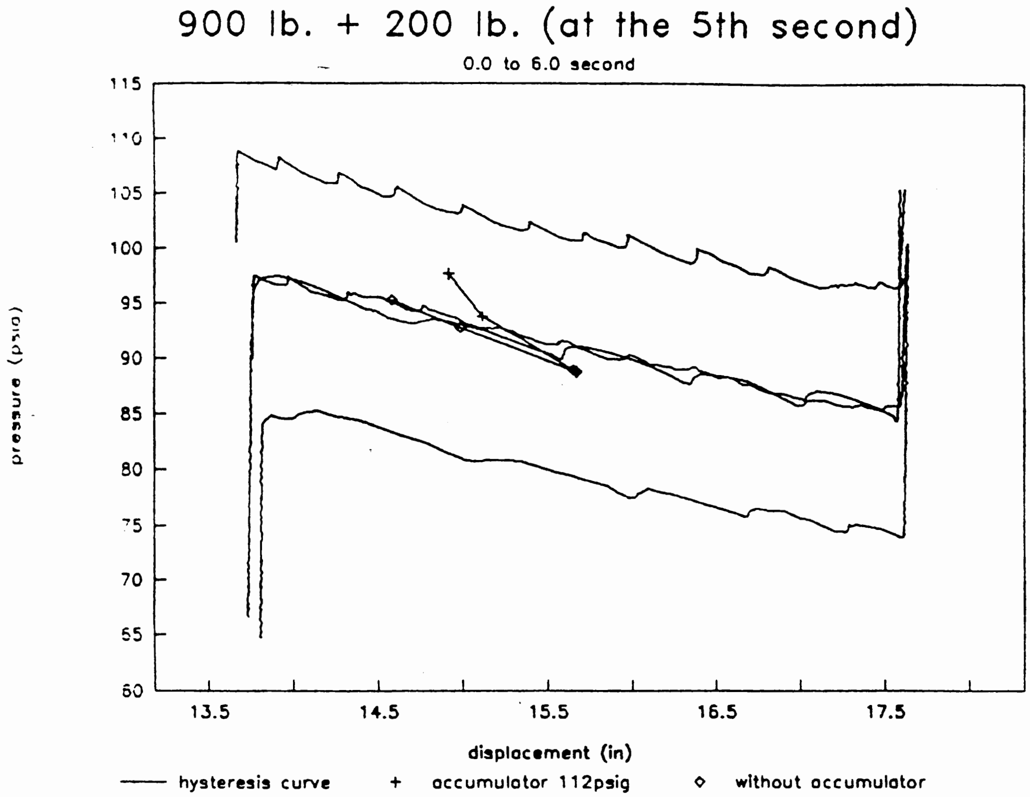


Figure 26. Simulation of 1000 lb System + Diaphragm Accumulator

characteristic of the diaphragm accumulator used (see figure 22,) the difference between 90 psig and 100 psig charge is striking. When the pressure is only 90 psig, the accumulator volume is only 30 cubic inches, while after the accumulator pressure exceeds 93 psig, the diaphragm resistance -- caused by friction and hysteresis -- becomes much smaller and the diaphragm greatly expands to reach much larger volume of stored air. In both 900 lb and 1000 lb cases, it could be seen that the appending of the accumulator is able to reduce the effect of hysteresis.

Accumulator's Contribution to the System

The accumulator reduces the time needed by the system to move from point B in figure 20 to point C by producing extra air flow. Figures 27, 28, and 29 present the comparison between the simulation of the system without an accumulator and with an accumulator charged to 112 psig. Also shown in these figures the pressure/displacement path at various time intervals. Notice that at the 6th second while the nonaccumulator system is still at the bottom of the 1100 lb hysteresis dead band (the original emperical curve is showed instead of the linearized one), the system with accumulator has already began to climb it. And by the time 7.5 seconds has passed the accumulator system has finished crossing the boundary while the other system only just begins to climb. During the next few seconds, the nonaccumulator system moves very slowly to the



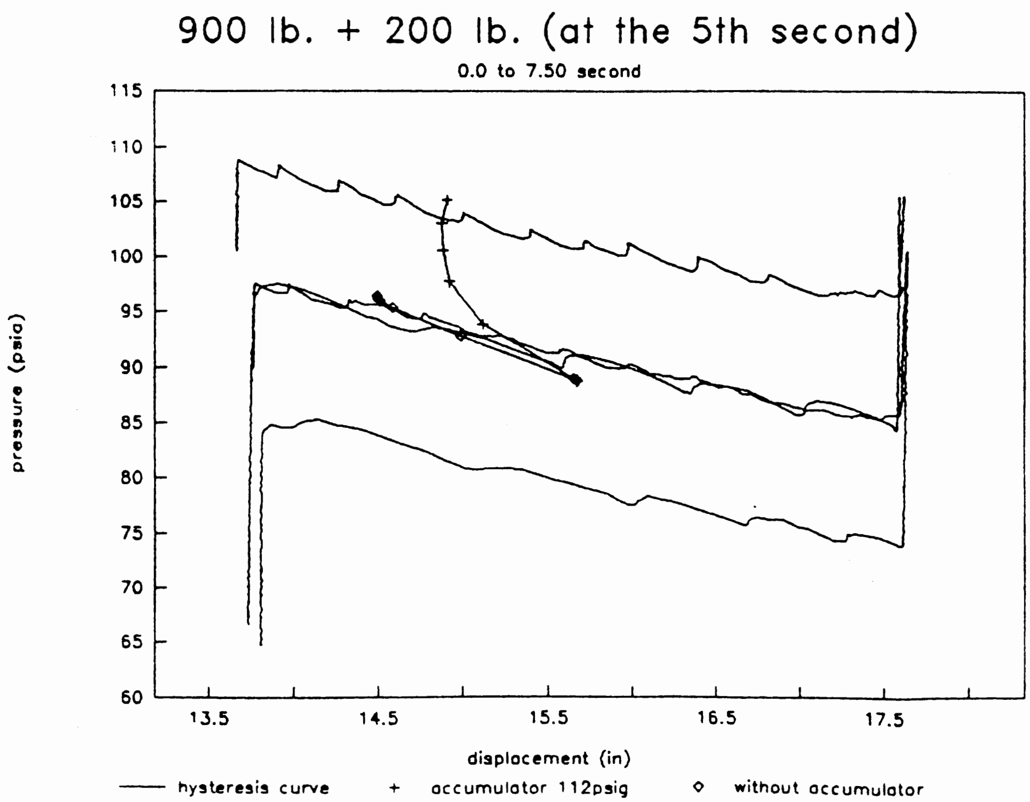
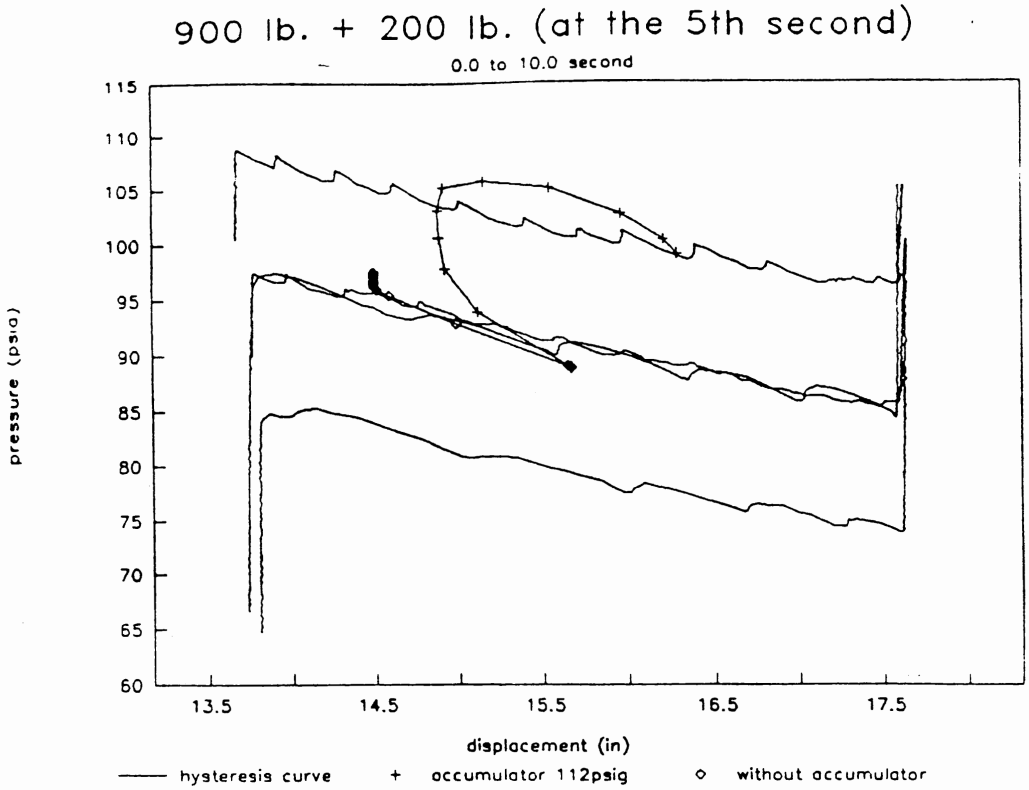


Figure 27. Comparison Between Non-accumulator and with Accumulator I



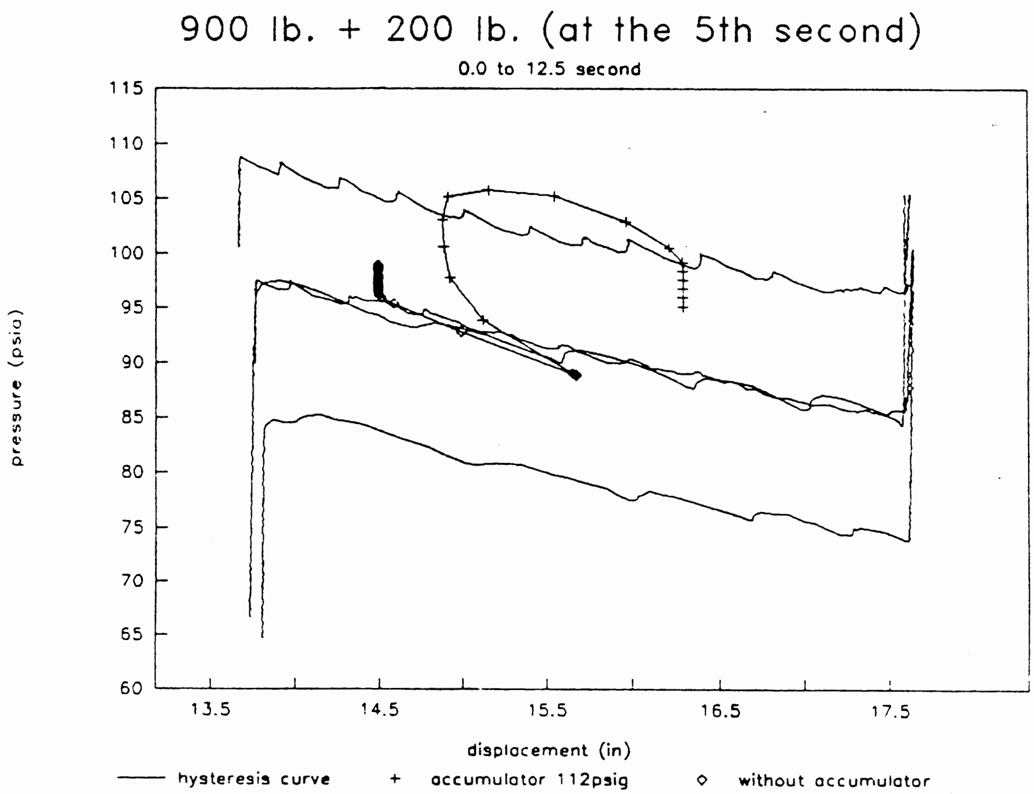


Figure 28. Comparison Between Non-accumulator and with Accumulator II

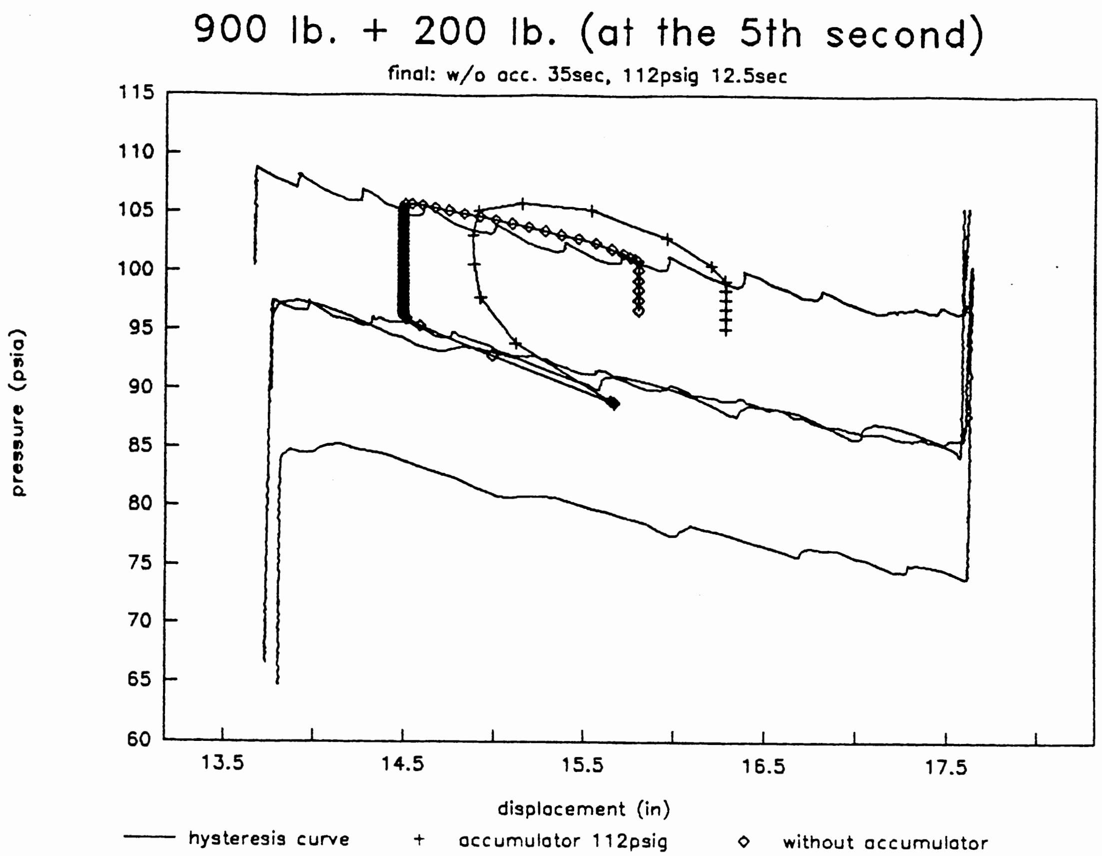


Figure 29. Comparison Between Non-accumulator and with Accumulator III

upper curve. In this same time the system with the accumulator has already reached the stable condition. The final record of the system with the accumulator is at the 12.5 second point.

CHAPTER IV

EXPERIMENT VERIFICATION

Experiment Purpose

Experiments have been conducted to verify the simulation results shown in the preceeding chapters. These experiments were conducted with a system which did not include an accumulator as well as one with an accumulator. The procedure of each experiment is given in the next two sections.

In this thesis, only a diaphragm type accumulator is used in the system. The verification of the tube accumulator simulation would be similar since the diaphragm accumulator used here has characteristics very closely approaching the tube type accumulator during small pressure changes (a volume between 30 - 50 cuin for a pressure about 80 - 90 psig.)

Experiment Set Up

The system consisted of a compressor and a blower, a discharge vent valve, transmission piping, two solenoid valves, one for the accumulator, one for the pneumatic spring, an accumulator, and a rolling lobe type pneumatic spring. The accumulator used is of diaphragm type. The compressor pumped air to the pneumatic spring through the transmission pipe. A solenoid valve was placed by the pneumatic spring to serve as the gate. A branch line was attached to the transmission pipe to connect the accumulator and the valve to the system. A sliding platform was constructed above the pneumatic spring to stabilize the load. A linear variable displacement transducer was attached to the platform to measure its displacement, the height measurement was transmitted to ADALAB board and was processed by a microcomputer. A pressure transducer was used to measure the air pressure inside of the pneumatic spring. All the valves and the compressor were controlled by on-off signals from the microcomputer. Figures 30 and 31 show the experimental set up used. The algorithm that processed the signals from the transducers and controled the compressor and valves was located in the microcomputer.

Details of instruments used are as follows:

1. BOURNS linear variable displacement transducer.
2. BOURGWARNER AUTOMOTIVE pressure transducer.
3. GOLDSTAR GP-233 power supply to the transducers.
4. General purpose air compressor and dryer and vent valve.
5. SORENSEN DCR 20-50B power supply to air compressor and valves.
6. DAYTON 2 way general purpose valve (solenoid valve.)
7. ADALAB SN 611244H board.
8. IBM microcomputer.

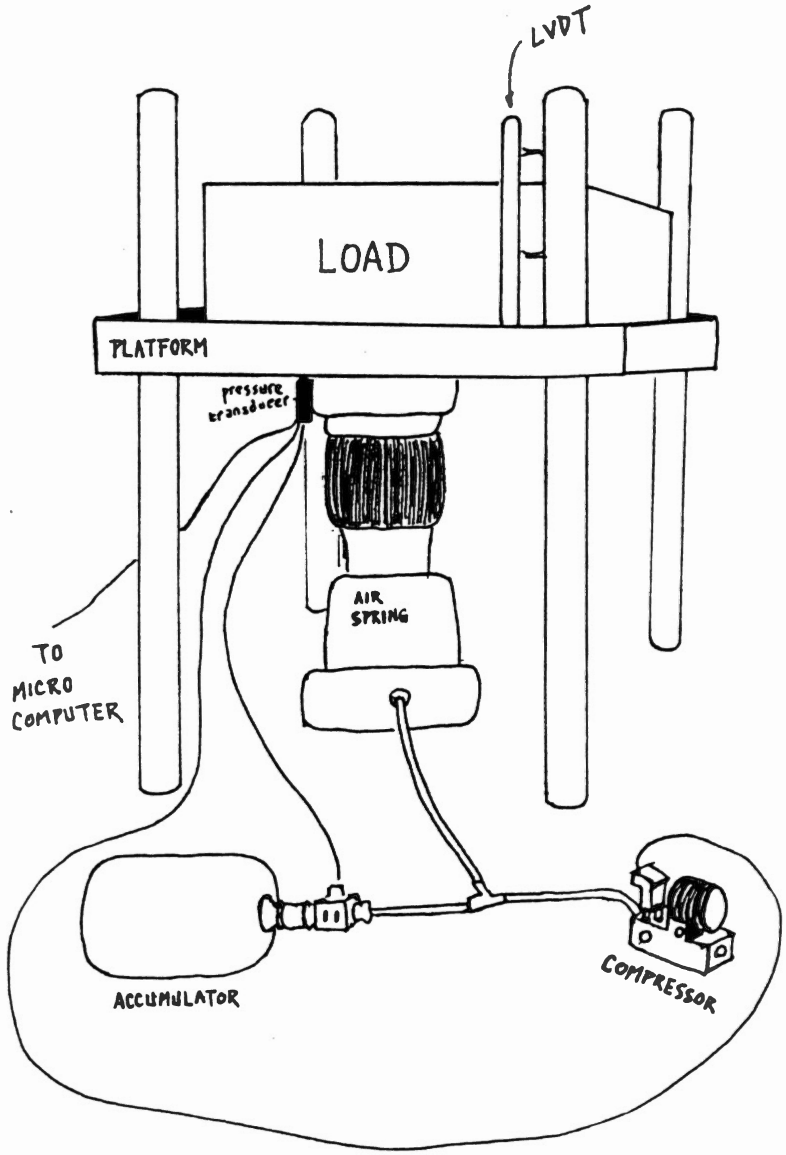


Figure 30. Experiment Set Up I

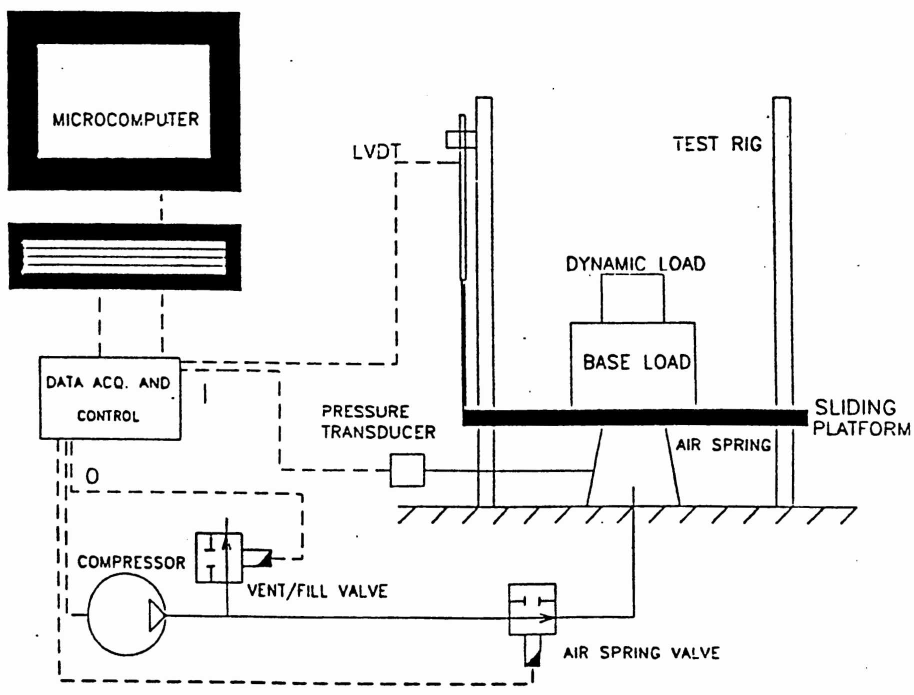


Figure 31. Experiment Set Up II

9. Rolling lobe type pneumatic spring.
10. GREER hydraulic bladder accumulator 2½ gal. (3000 psi.)
11. 5 ft plastic tubing (air spring to compressor.)
12. Load platform.
13. 900 lb, 1000 lb, and 200 lb weight.

Experiment Procedure

The experiment was conducted in two parts. Part one included the experiment without an accumulator. In the second part the system was tested with the accumulator. In part one, the accumulator valve is simply closed for the duration. The accumulator used in this research was a diaphragm type accumulator with characteristics as described in chapter III. The computer was used to turn the compressor on and off and to open and close the valves as a function of height and pressure.

The algorithm for part one was included in chapter III. The following is a summary:

1. If platform height < desired height then

compressor	on
solenoid valve	open
vent valve	closed

2. Else if platform height > desired height then

compressor	off
solenoid valve	open
vent valve	open

3. Else (none of the above condition)

compressor	off
solenoid valve	closed
vent valve	closed

The procedure for the experiment was as follows:

1. Put 900 lb base weight on the platform.
2. Turn on the compressor till the platform reach 15.65 in height and 89 psia pressure. Start the timer.
3. Let the system stabilize itself for 5 seconds.
4. Put additional 200 lb load on the platform.
5. Record time parameters for 35 seconds, remove the 200 lb load from the platform and record for 20 seconds.
6. Repeat procedure 2 - 5 after replacing the base load with 1000 lb weight.
7. Analyze data.

Summary of the algorithm for part two:

1. If platform height < desired height then
   a. if accumulator pressure > system pressure then

compressor	on
solenoid valve	open
vent valve	closed
accumulator valve	open

   b. else

compressor	on
solenoid valve	open
vent valve	closed
accumulator valve	closed

2. Else if platform height > desired height then

   a. If accumulator pressure > system pressure then

      compressor off

      solenoid valve open

      vent valve open

      accumulator valve closed

   b. else

      compressor off

      solenoid valve open

      vent valve open

      accumulator valve open

3. Else (if platform height in the desired range)

   a. If accumulator pressure > system pressure then

      *. If accumulator pressure < maximum pressure then

         compressor on

         solenoid valve closed

         vent valve closed

         accumulator valve closed

      *. else

         compressor off

         solenoid valve closed

         vent valve closed

         accumulator valve closed

   b. Else (if accumulator pressure < system pressure)

      compressor on

      solenoid valve closed

      vent valve closed

accumulator valve open

Maximum pressure is the charge pressure desired for the accumulator.

Experimental procedure for this part is the same with the procedure of part one.

Experiment Result and Discussion

The output of the experiment included two parts: one without accumulator and the other one with accumulator. The Starting point was 15.65 inches displacement and air spring pressure 89 psia.

Starting with part one, using 900 lb base weight. Figures 32 and 33 show the data recorded. During first 5 seconds, the system was in a stable condition. The additional 200 lb load was put on the platform at the 5 second point and the platform moved to 14.6 inches while the pressure jumped to 94 psia. During next 19 seconds the displacement was virtually unchanged and the pressure climbed to 105 psia. This pressure was enough to overcome the hysteresis and move the platform back to the desired range. During this period, the pressure decreased to 100.5 psia. After reaching this point, the system was at rest because the platform had reached its desired height. At 35 second, the 200 lb load was removed. The existing pressure in the air spring moved the platform up to 17.5 inches, hence lowering the pressure to 84 psia. During the following 8 seconds, the platform paused while the air was

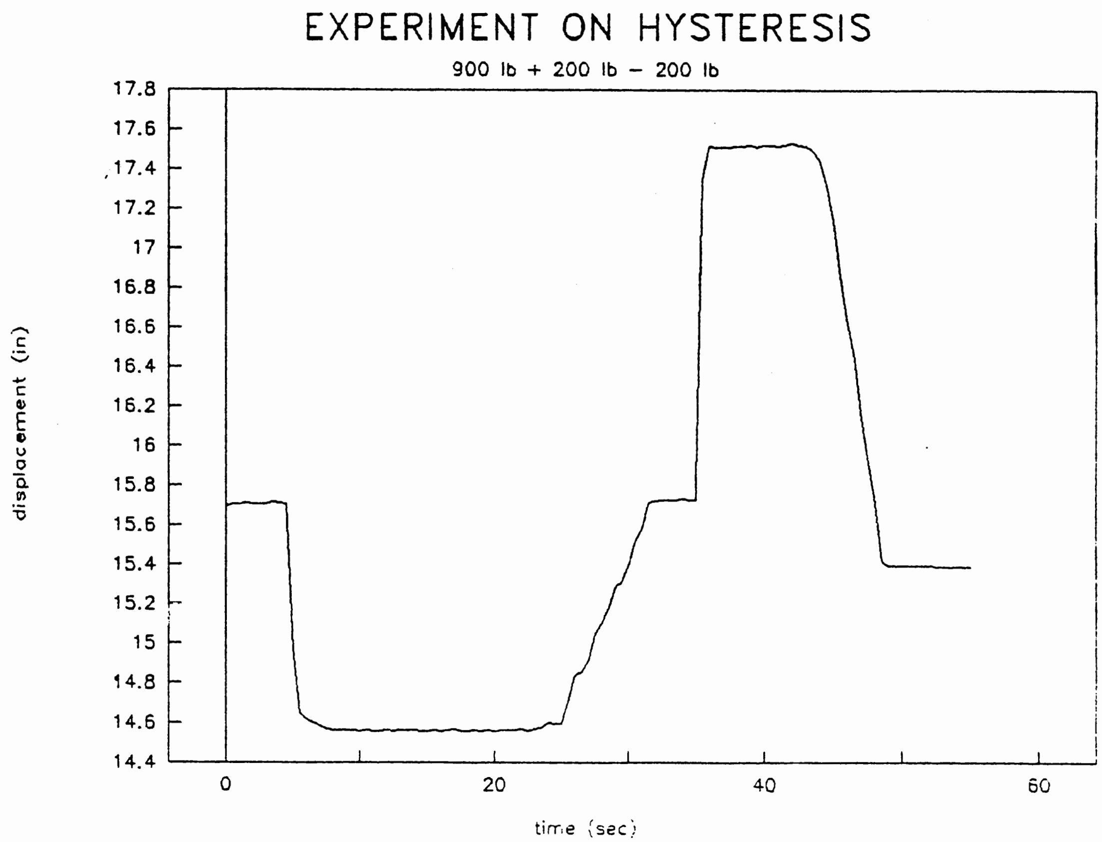


Figure 32. Displacement Curve from Experiment on 900 lb Load

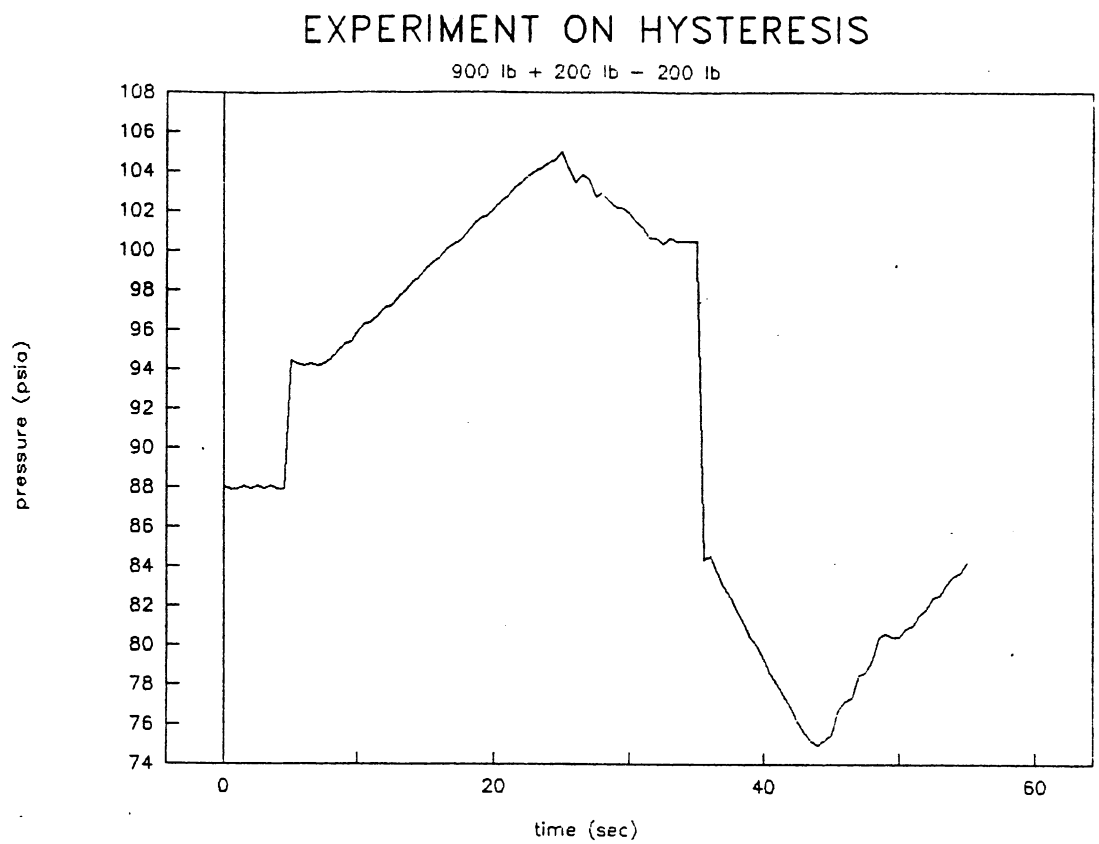


Figure 33. Pressure Curve from Experiment on 900 lb Load

discharged until the pressure reaches 75 psia. Then, the platform returned back to the desired height.

In the 1000 lb base weight experiment (figures 34 and 35), the additional 200 lb load pushed the air spring to 14.5 inches height and forced the platform to stay at that position for about 17 seconds. The pressure needed to overcome this additional load was 111 psia. Removal of the added load from the platform after the system was stabilized brought the system to its initial pressure, but by the time the load was removed, the surplus pressure gave a sudden lift to the platform to 17.5 inches. The platform stayed in this position for about 10 seconds.

The second part of the experiment showed the improvement made by the accumulator. In this thesis the data recorded when the 200 lb load was added to the base weight is shown since the advantage of the accumulator is most obvious during this process. The time delay when the load was removed would be handled effectively by using a discharge valve with smaller resistance.

Figure 36 shows that a 90 psig charged accumulator reduces the displacement of the platform with a 900 lb base weight by about 30.43% or 0.35 inches. Without the accumulator, the platform dropped 1.15 inches to 14.55 inches. Therefore with the accumulator, it only dropped 0.8 inches to 14.9 inches. The time needed to return to the desired height was shortened by 3 seconds. Charging the accumulator to 100 psig gives much better results. Although

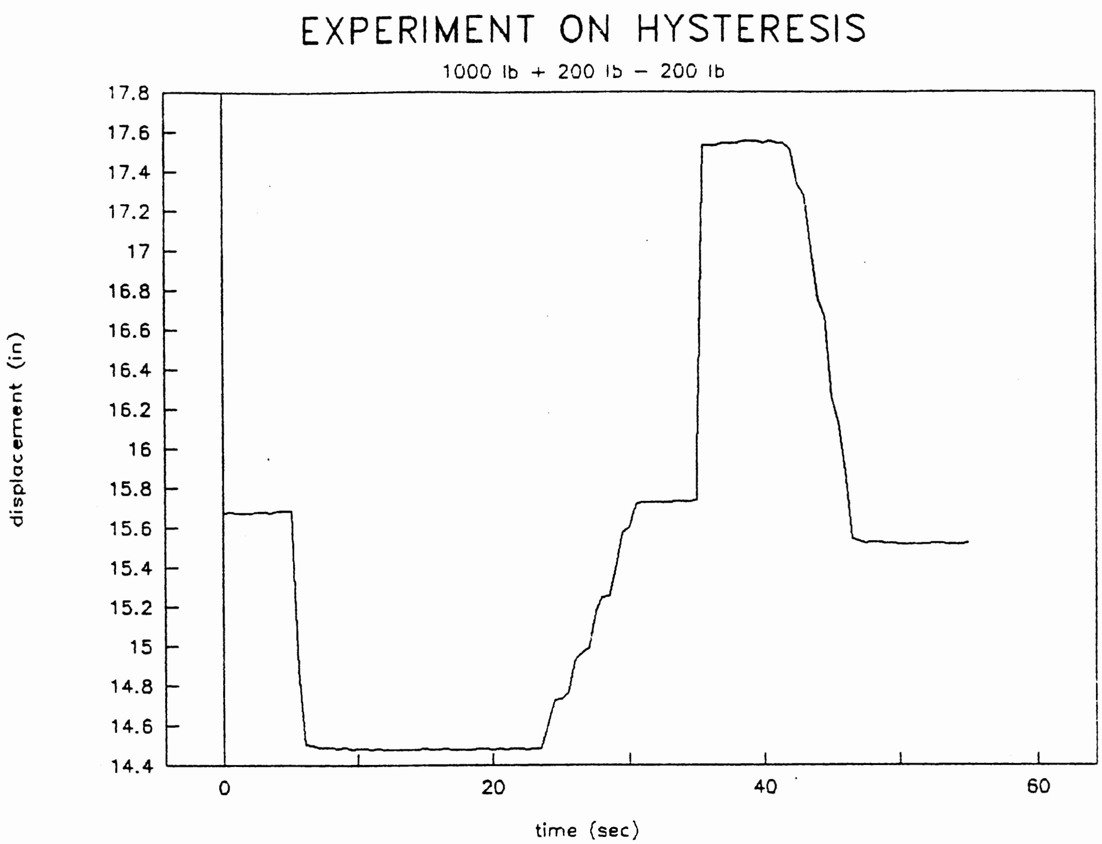


Figure 34. Displacement Curve from Experiment on 1000 lb Load

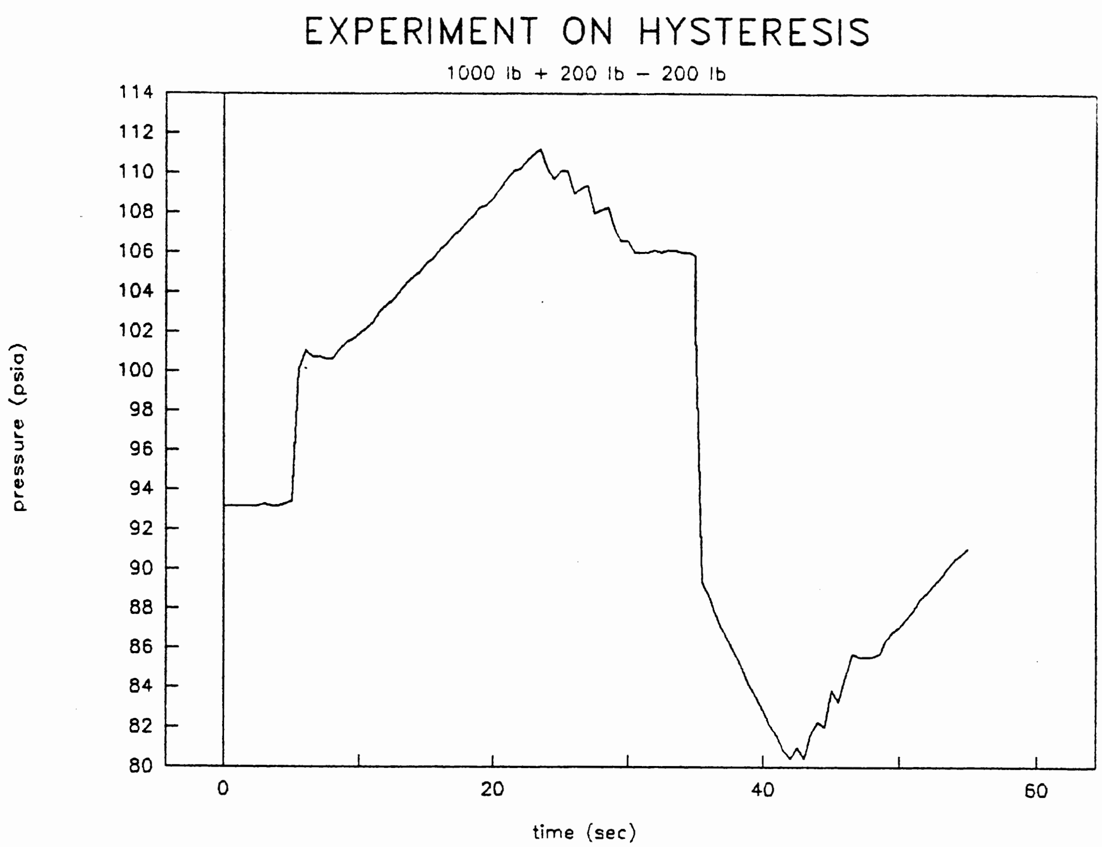


Figure 35. Pressure Curve from Experiment on 1000 lb Load

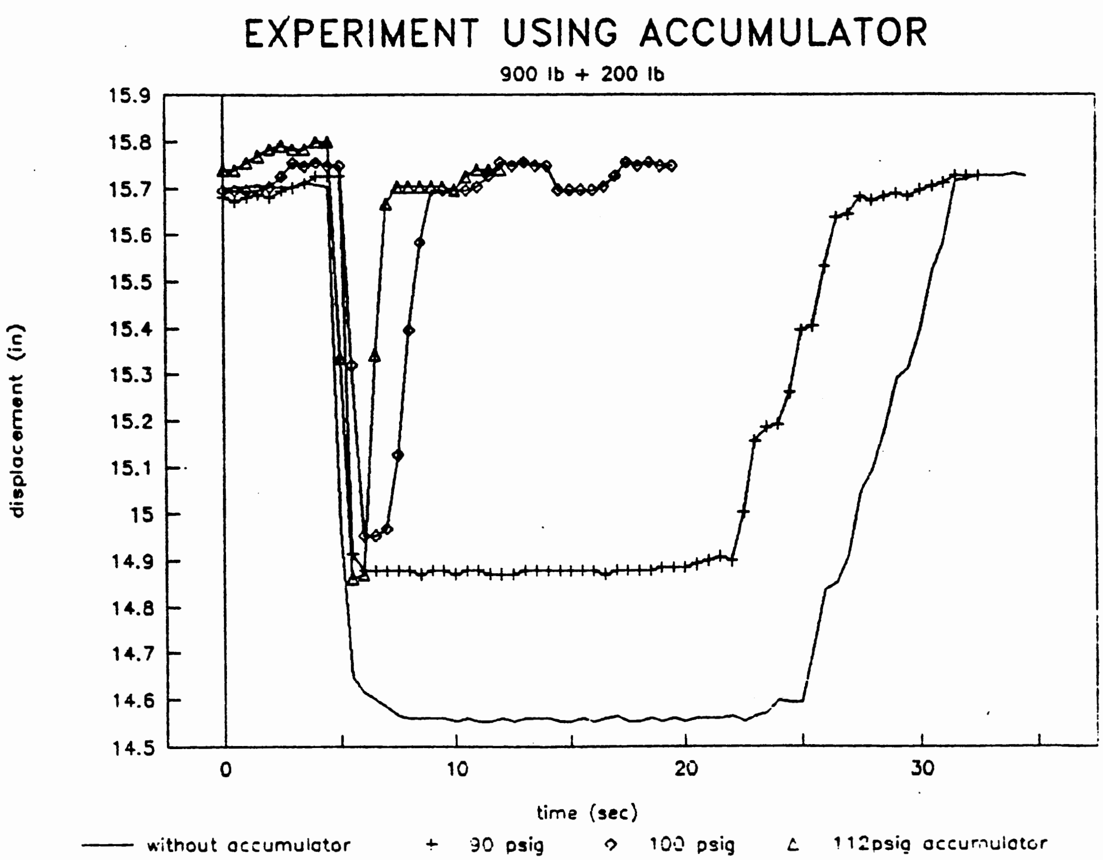


Figure 36. Experiment Using Accumulator on the 900 lb Load System

the displacement change was not improved, the time necessary to return to the desired height was reduced to about 2.5 seconds. Charging the accumulator to 112 psig reduced the time to 2 seconds. The difference in the results of changing the accumulator charge from 90 psig to 100 psig lies in the fact that at 90 psig (104.7 psia) the accumulator stores only 30 cuin of air while the accumulator charged to 100 psig (114.7 psia) stores 487.50 cuin of air for use by the system.

In the 1000 lb base weight system (figure 37), the installing of the 100 psig accumulator reduced the drop by 0.34 inches (28.33%). Before installing the accumulator into the system, the platform fell to 14.48 inches. However, after the accumulator valve was opened, the load only moved to 14.82 inches. The time taken by the system to return was only 2.5 seconds which was 15 seconds (81.08%) less than without an accumulator. Charging the accumulator to 110 psig accelerated the platform return time by about 0.1 second. The 120 psig charge produced no time delay. There is an interesting phenomenon in the 1000 lb weight experiment result. The greater the charge pressure in the accumulator the deeper the platform dropped. The reason is that the greater the pressure the accumulator is holding, the greater the volume occupied. This large volume gives the platform more flexibility to move due to the compressibility of the medium fluid (see equation (10).) This diaphragm type accumulator is powerful in supporting

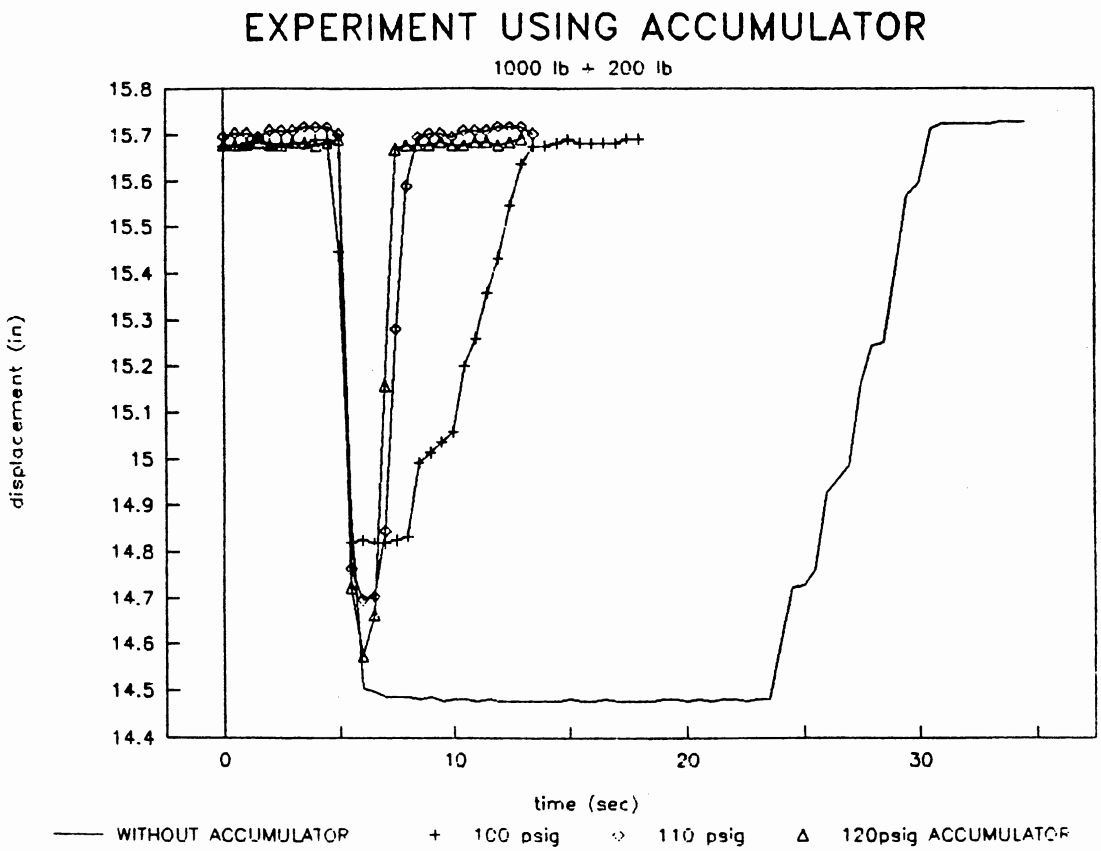


Figure 37. Experiment Using Accumulator on the 1000 lb Load System

the system. One thing should be noticed, however, it requires more time to charge this accumulator. Using this accumulator in the situation when there is only a short time to recharge it (the settling time is short) will give about the same result with the simple tube type accumulator.

Experiment Compared to Simulation

Figures 38, 39, 40, 41, 42, and 43 juxtapose the simulation result with the experimental output. The first three figures (figures 38, 39, 40) depict the nonaccumulator system with 900 lb initial load. The results are very close. Considering the displacement/time curve of figure 38, the experiment starts about 0.03 inch higher than the simulation. When the load is added, the simulation shows platform dropping about 0.001 inch more than the experimental results -- after adjusting with initial height. When the load has been overcome, the simulation shows the return 0.1 inch higher than the experiment. The simulation of the load removal gives maximum displacement 0.06 inch higher and 2.0 second later than the experiment. The discharging speed is faster in the simulation than in the experiment. This quicker discharging speed causes the platform in the simulation to drop lower. This difference is obvious in the pressure graph of figure 39.

The next three figures (figures 41, 42, 43) exhibit the comparison of the system with a 1000 lb base load. Simulation result also shows more drop when the additional

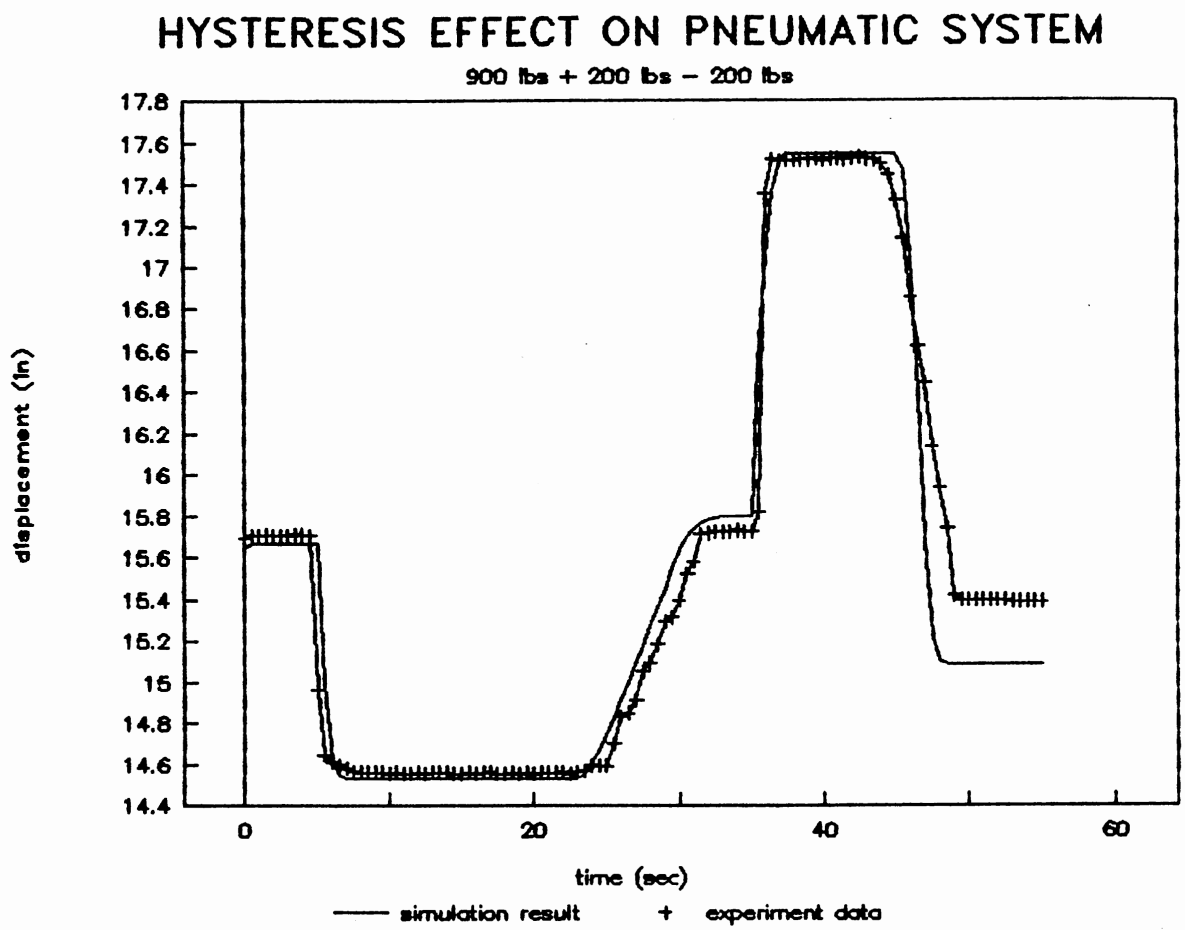


Figure 38. Displacement Curve Simulation-experiment Comparison (900 lbs)

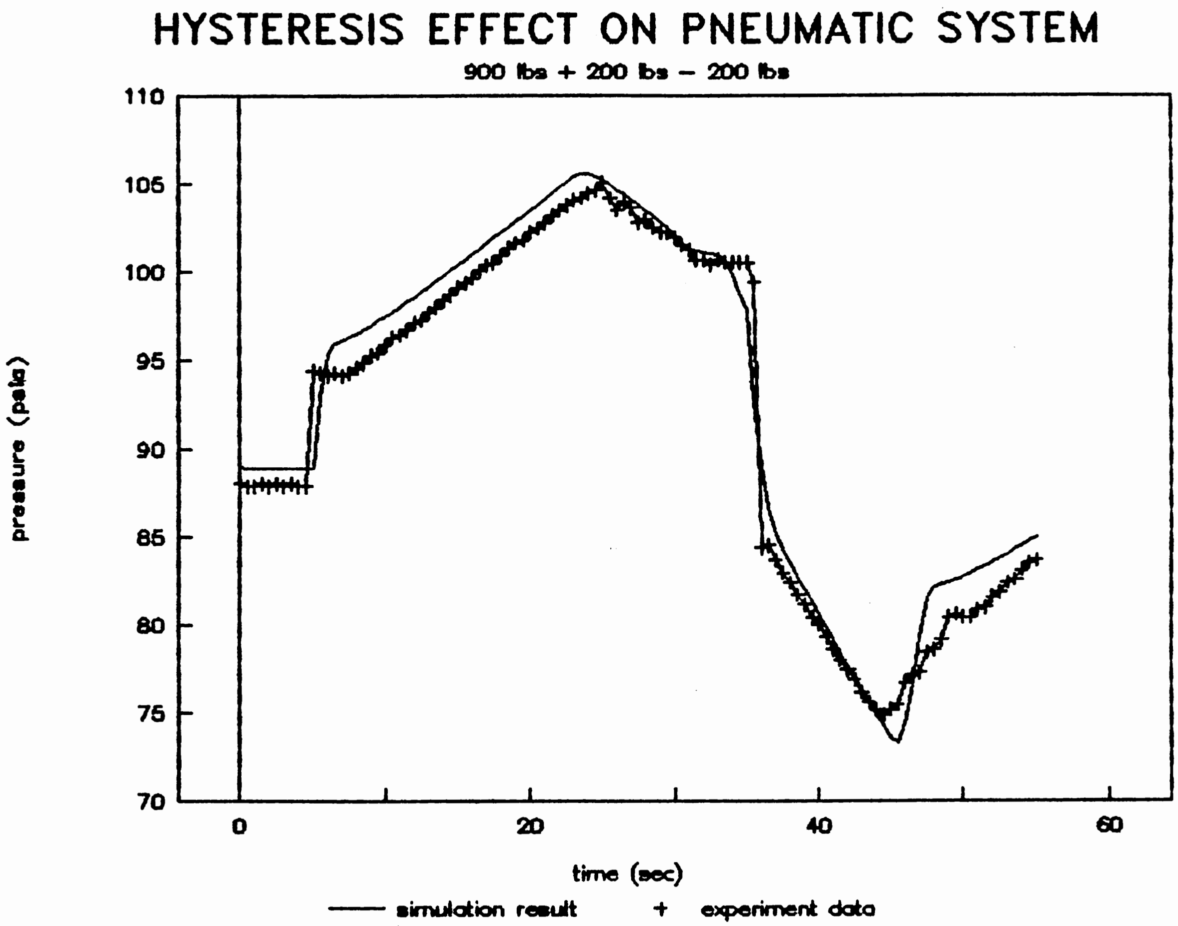


Figure 39. Pressure Curve Simulation-experiment Comparison (900 lbs)

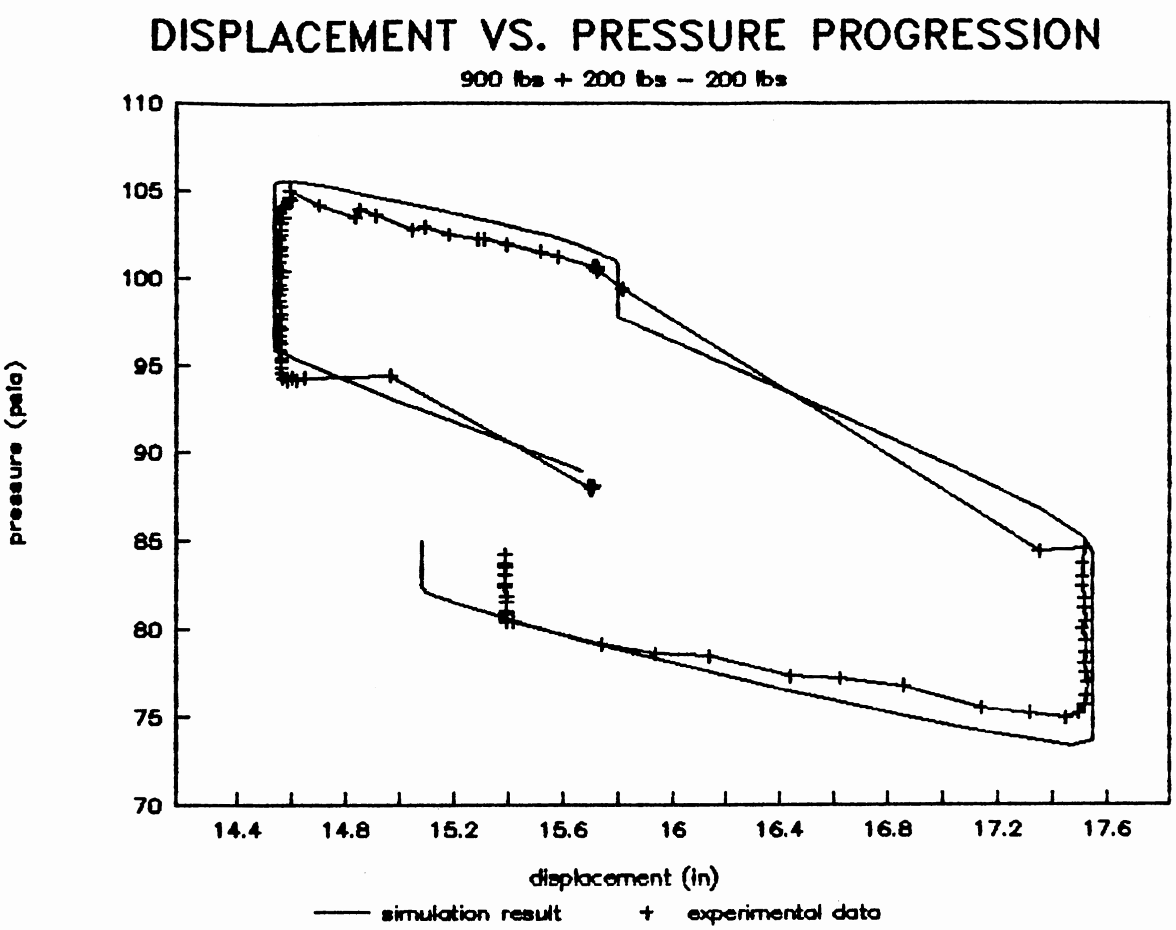


Figure 40. Displacement-Pressure Curve Simulation-experiment Comparison (900 lbs)

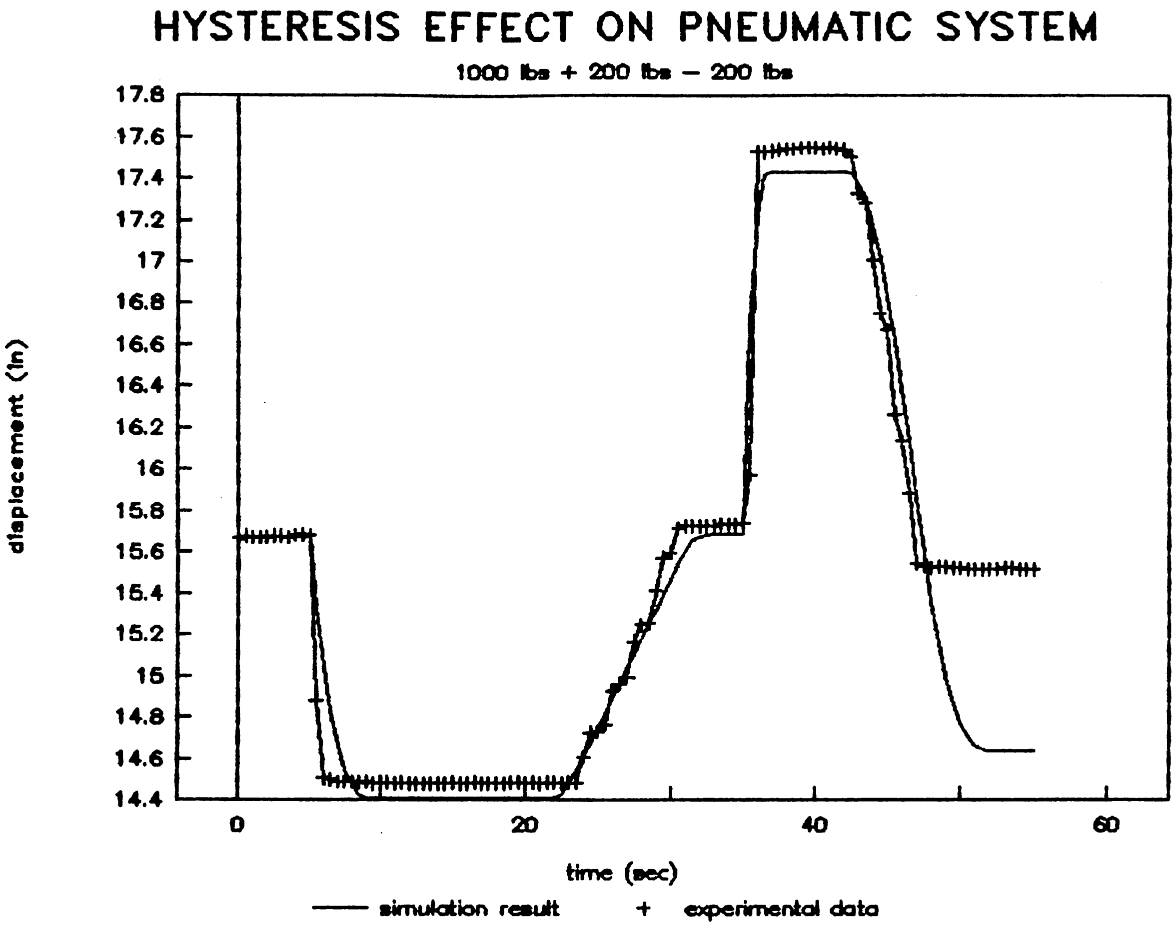


Figure 41. Displacement Curve Simulation-experiment Comparison (1000 lbs)

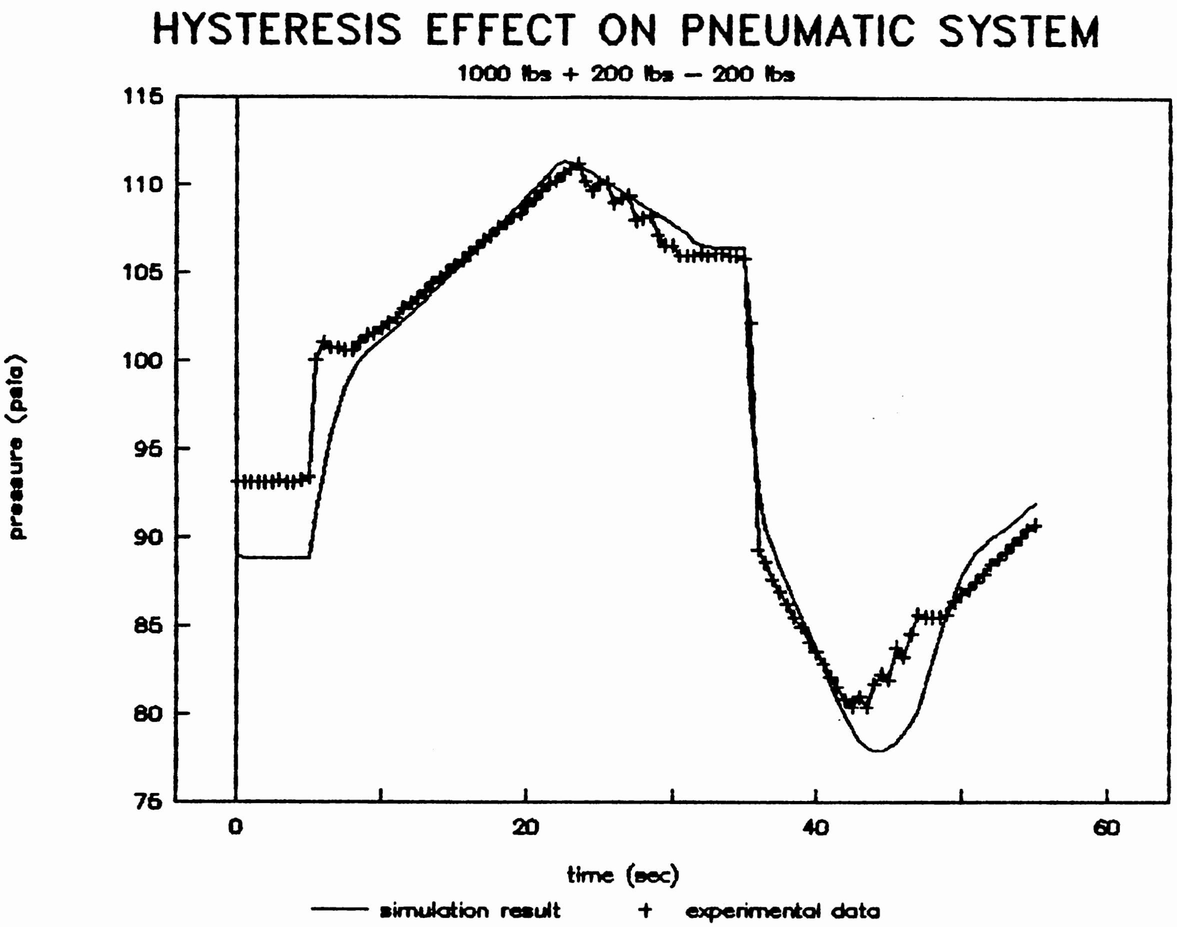


Figure 42. Pressure Curve Simulation-experiment Comparison (1000 lbs)

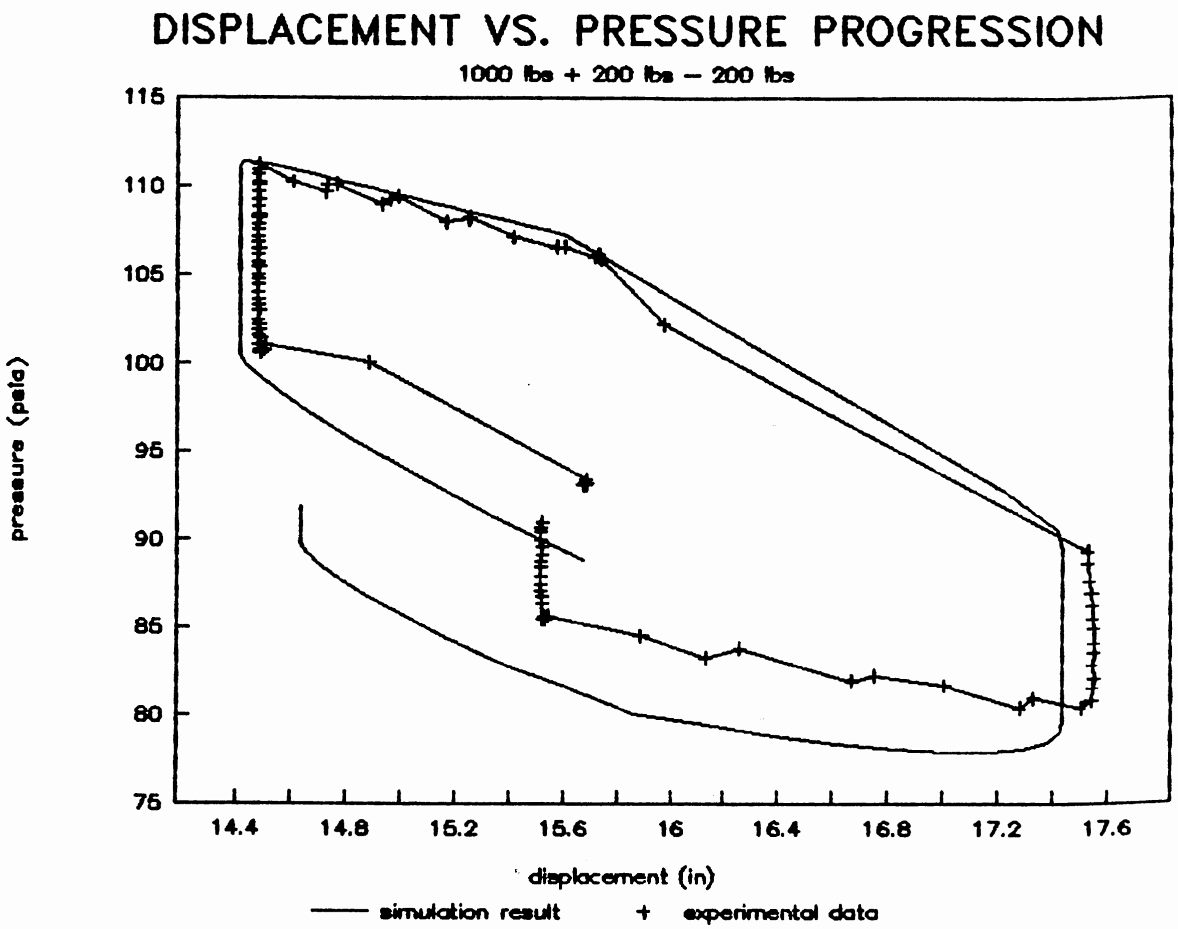


Figure 43. Displacement-Pressure Curve
Simulation-experiment Comparison (1000 lbs)

200 lb load is added. Also the simulated platform moves lower than the real experiment. The discharging procedure makes the simulation result about 0.11 inch lower than the experiment. The difference between computer simulation and laboratory experiment in the displacement-pressure curve is more distinct with the 1000 lb base load than with the 900 lbs. Starting pressure is lower and the final pressure is lower. The cause of this difference in load removal is that in the simulation, a single fixed polytropic process ($n = 1.1$) was assumed to be valid to represent the whole processes experienced by the system. In the experiment, actually there occured three different gaseous processes: adiabatic, isothermal, and polytropic. Adiabatic process is the process experienced during sudden expansion or compression since there is not enough time for the system to exchange heat. Isothermal process occurs when the spring stays unmoved. Polytropic processes are experienced when the spring is returning to the desired height. The assumption of constant and uniform air density also contributes to the deviation along with the linearizing of the valve, gas transmission, air spring area, etc.

The simulation result of the system with a 900 lbs base load using an accumulator is very close to the experimental results (figure 44). The experiment exceeded the simulation in producing better results with the accumulator. More lift was provided by experimental accumulator than the simulation one. Also the time needed to return upward is shortened.

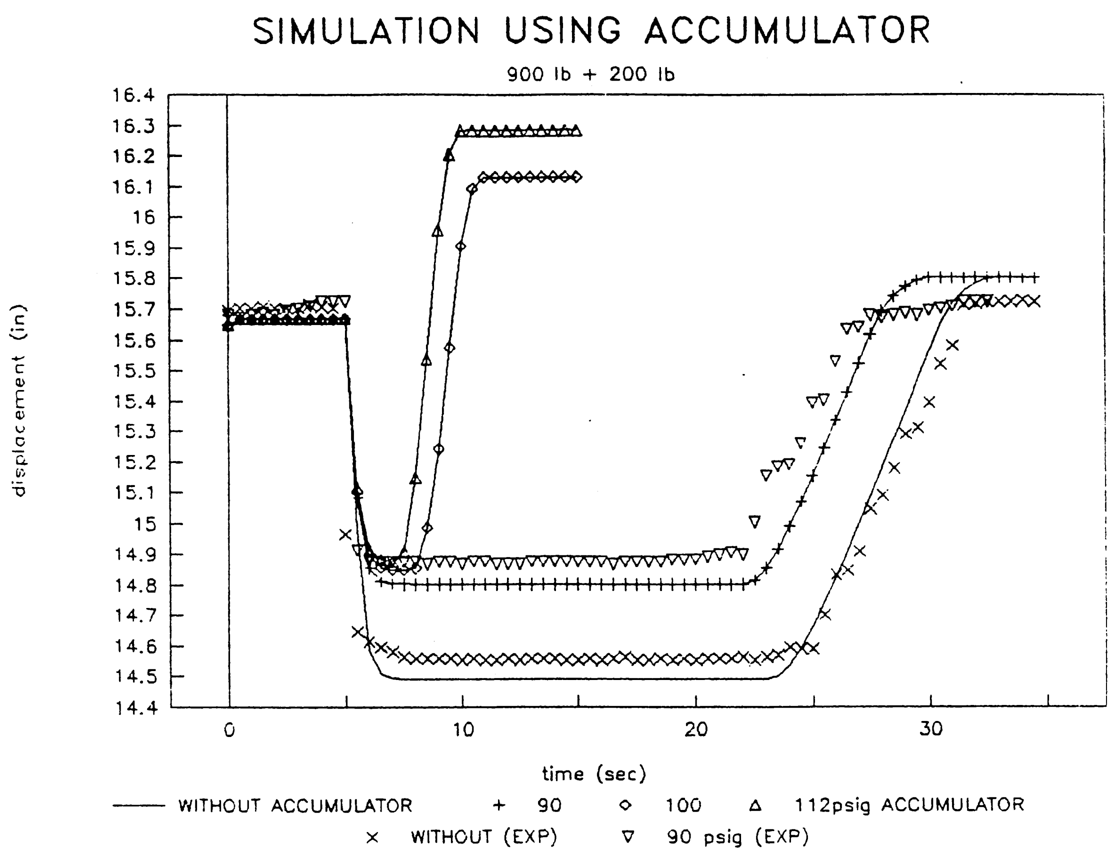


Figure 44. Comparison Between Simulation and Experiment on Accumulator System (900 lbs)

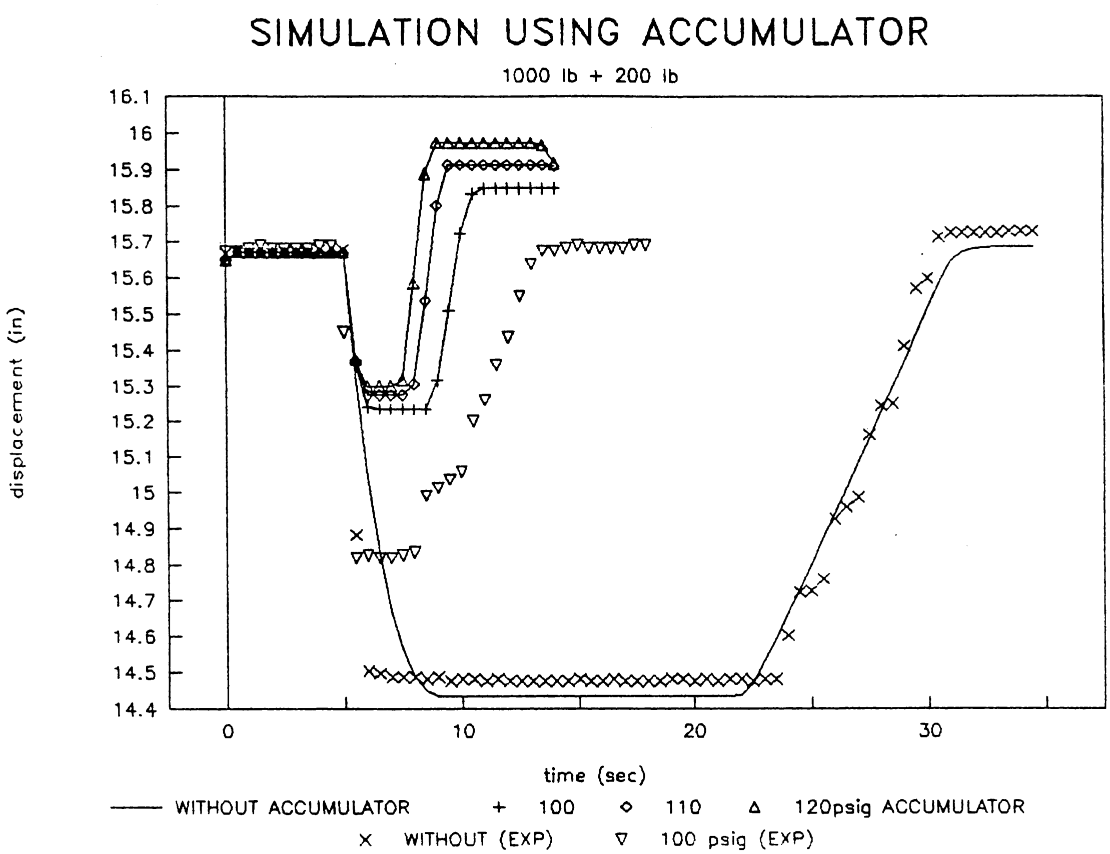


Figure 45. Comparison Between Simulation and Experiment on Accumulator System (1000 lbs)

In the 1000 lb system, the experimental results were also better than the simulation data in faster return to the reference height, see figure 45. However, notice that in the experiment the platform drops further than the simulation indicates. The suggested explanation is the inaccuracy of linear assumption for the effective air spring area. Another striking thing in the simulation is that the platform goes too high while returning from the minimum position (this is also true for the 900 lb system.) The reason is also the linearizing of the air spring area along with the other assumptions.

CHAPTER V

CONCLUSION

Conclusion

Pneumatic spring systems provides characteristics that cannot be obtained by other spring systems. Its low stiffness during small displacements and high stiffness during large displacements are essential in protecting the system from vibration. In leveling systems, however, even a small displacement could be troublesome. This problem can best be handled by returning the displaced system to its initial position as soon as possible. The use of rubber in the pneumatic spring produces the complication of hysteresis. The difference between the force needed to expand and to retract the rubber produces a hysteresis effect. This hysteresis causes problems in the pneumatic spring leveling system during loading and unloading. A load placed suddenly on the air spring displaces it downward and for some time the spring cannot move due to hysteresis. This also occurs during load removal, the spring jumps to a new height and quite a considerable time elapses before the spring can return to its desired height.

In this thesis a simulation of a pneumatic actuator system has been presented. A mathematical model of an on-off controlled pneumatic system consisting of a compressor, transmission pipe, a solenoid valve, a vent valve, and pneumatic spring has been developed. Hysteresis characteristic found in the pneumatic spring due to the use of the rubber was derived from direct measurement and was integrated into the model. This entire load supported by the pneumatic spring was the input and the load displacement was the output. The control objective was to control the compressor and all the valves to maintain the displacement unchanged when the load was increased or decreased. The mathematical model was written in state equation format and was simulated by the Runge-Kutta method of iteration. The simulation was tested by adding and removing load to the system. The results show a considerable response delay when the system is subjected to varying loads due to hysteresis.

A solution to the delay caused by hysteresis is presented by adding an accumulator into the system. This accumulator alone cannot improve the system performance and may even produce a greater problem since the system volume would be larger. This extra volume absorbs the system energy and can make the system responds even slower. Assuming that the accumulator would function well to accelerate the system if it is in a fully charged condition and decelerate the system if it is uncharged, a simple algorithm was proposed to make the best use of the accumulator.

Hence, an accumulator and valve model were added to the initial simulation model. Using the algorithm developed for the system with an accumulator to execute the final model subjected to the same input, a second simulation result is presented. The new results justify the use of accumulator. That is the response delays are shortened.

Two sets of experiment were conducted and the results are presented to verify the computer simulation. The first experiment is of the system without an accumulator. The second is the same experiment of the system with accumulator.

Comparison between simulation and experiment results is presented and shows good correlation.

Suggestion for Future Research

The performance of the accumulator system used in this thesis could still be improved. Future research should be directed toward the determination of which accumulator is the best for this pneumatic spring system or perhaps the design of the new accumulator may be needed. The control algorithm is another interesting point. During a major load change, the valves instead of immediately closing or opening could be delayed. It is also practical to do research on the system accuracy to determine how much steady state error can be eliminated.

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APPENDIXES

APPENDIX A

PARAMETER SENSITIVITY

PARAMETER SENSITIVITY

In this thesis, some of the simulation parameters were hard to define. It was necessary to assume some reasonable numbers to be used in the simulation. One of those parameters was polytropic constant, n.

Polytropic constant is used in this simulation for modelling the state of the pipe, accumulator, and air spring. This polytropic constant, n, determines the compressibility of the air. It is important in calculating the rate of change of pressure with respect to time.

All the simulation results presented in this thesis use n = 1.1 as the parameter. For comparison, in this appendix, the simulation results of applying this value is showed together with the results of if n = 1.0 or n = 1.4 is used instead. Consult figure 46 through 53.

Using n = 1.0 allows greater displacement while isentropic n = 1.4 limits the displacement. This is already predicted by equation (10), greater n greater rate and vice versa. In the figure, it can also be seen that the pressure inside the air spring does not change very much with the parameter switching. It is also obvious that the pressure graph of n = 1.0 lags behind the pressure of n = 1.1 and these both pressure lag behind that of n = 1.4.

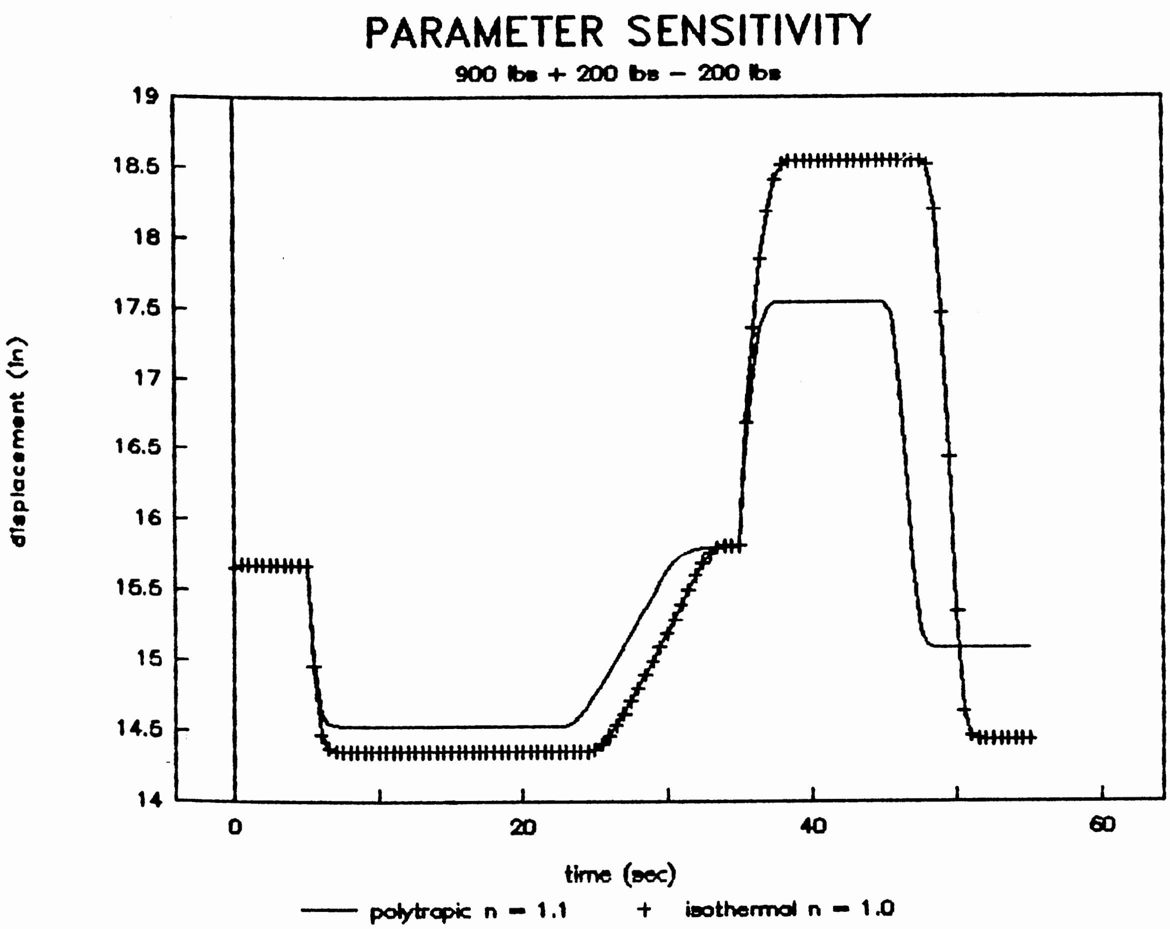

Figure 46. Displacement Graph Comparison Between Using n = 1.1 and n = 1.0 (900)

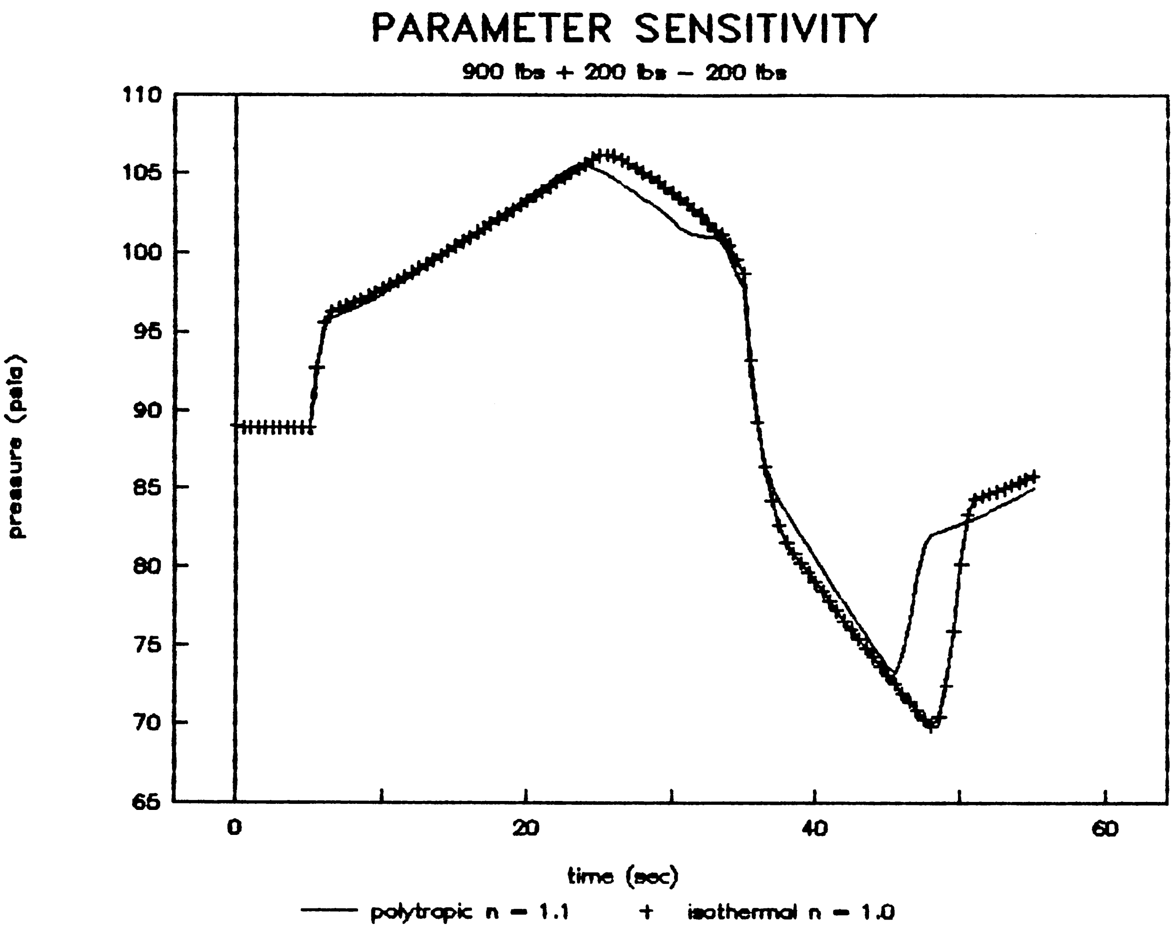

Figure 47. Pressure Graph Comparison Between Using n = 1.1 and n = 1.0 (900)

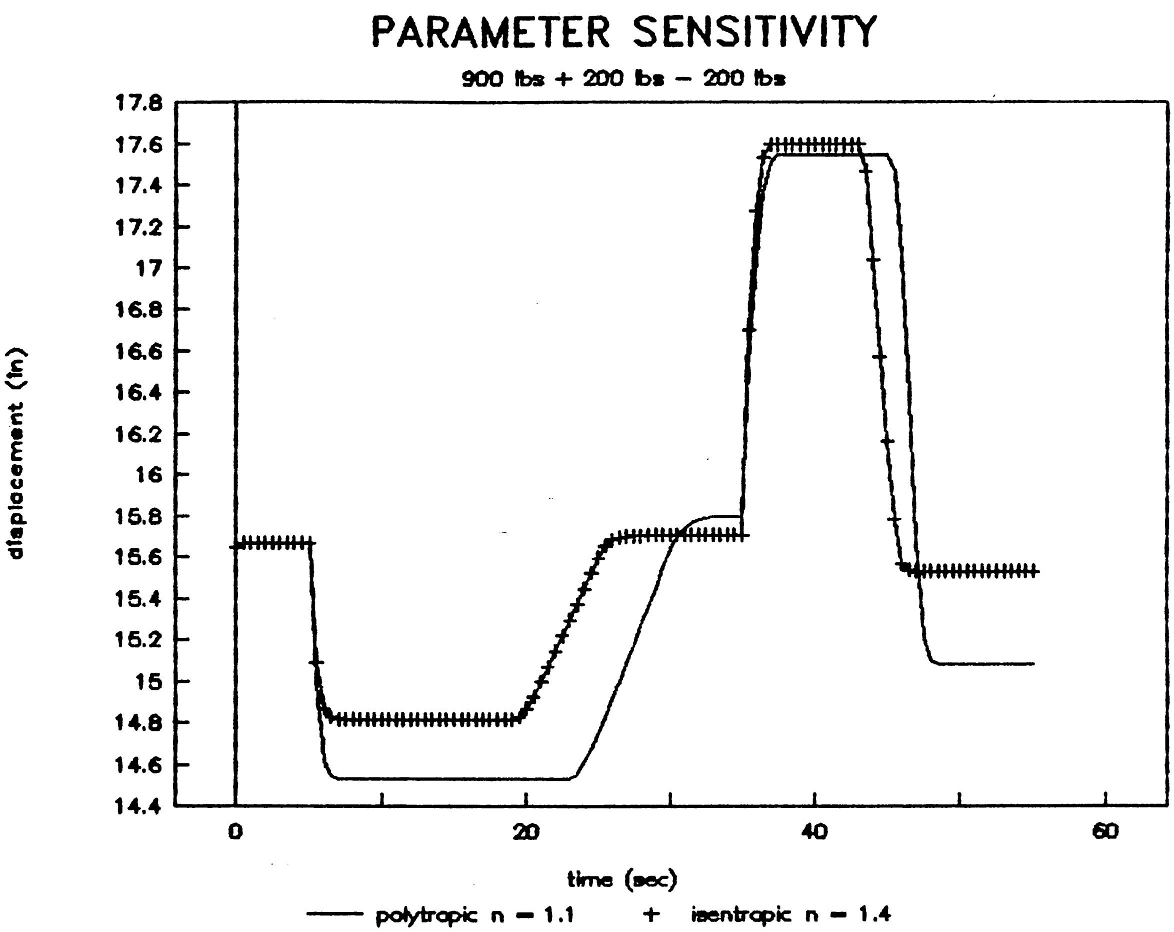

Figure 48. Displacement Graph Comparison
Between Using n = 1.1 and n = 1.4 (900)

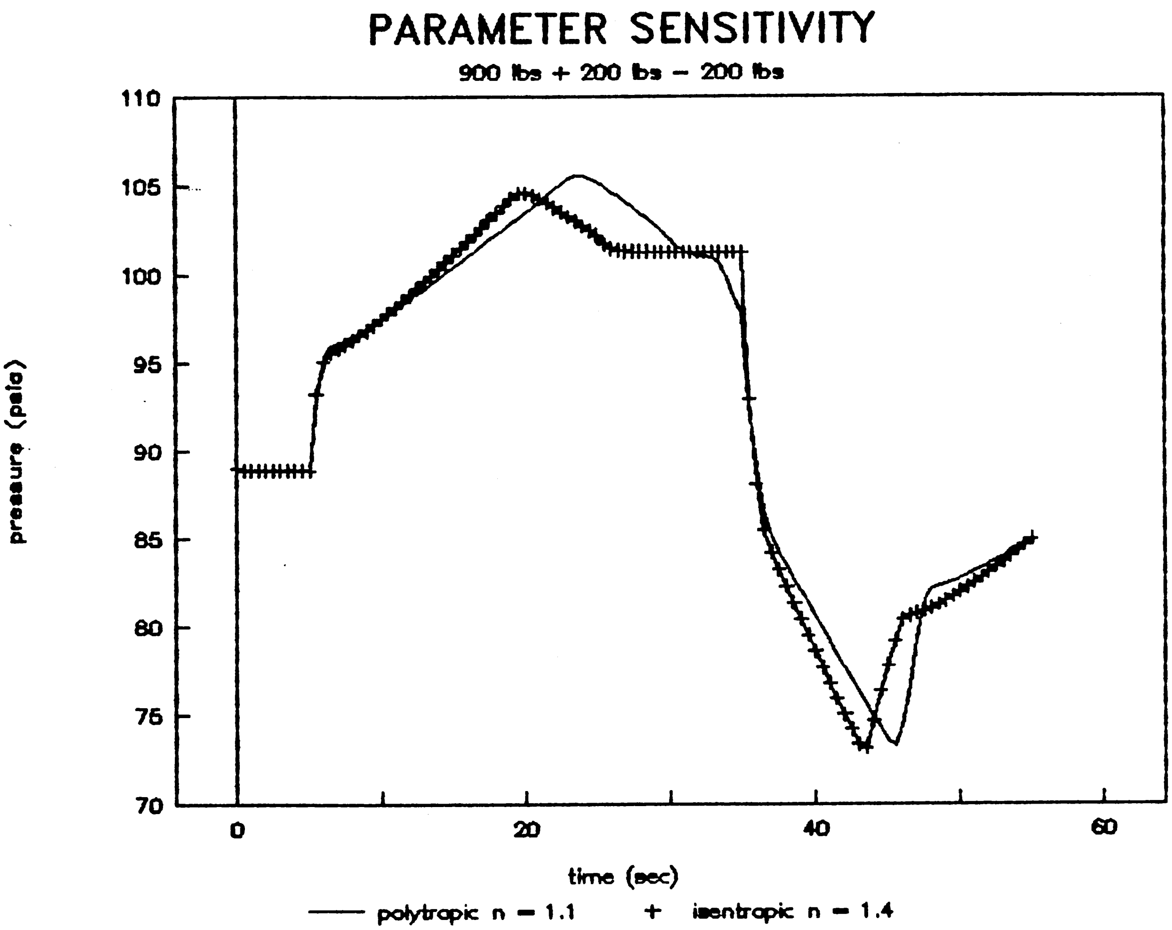

Figure 49. Pressure Graph Comparison
Between Using n = 1.1 and n = 1.4 (900)

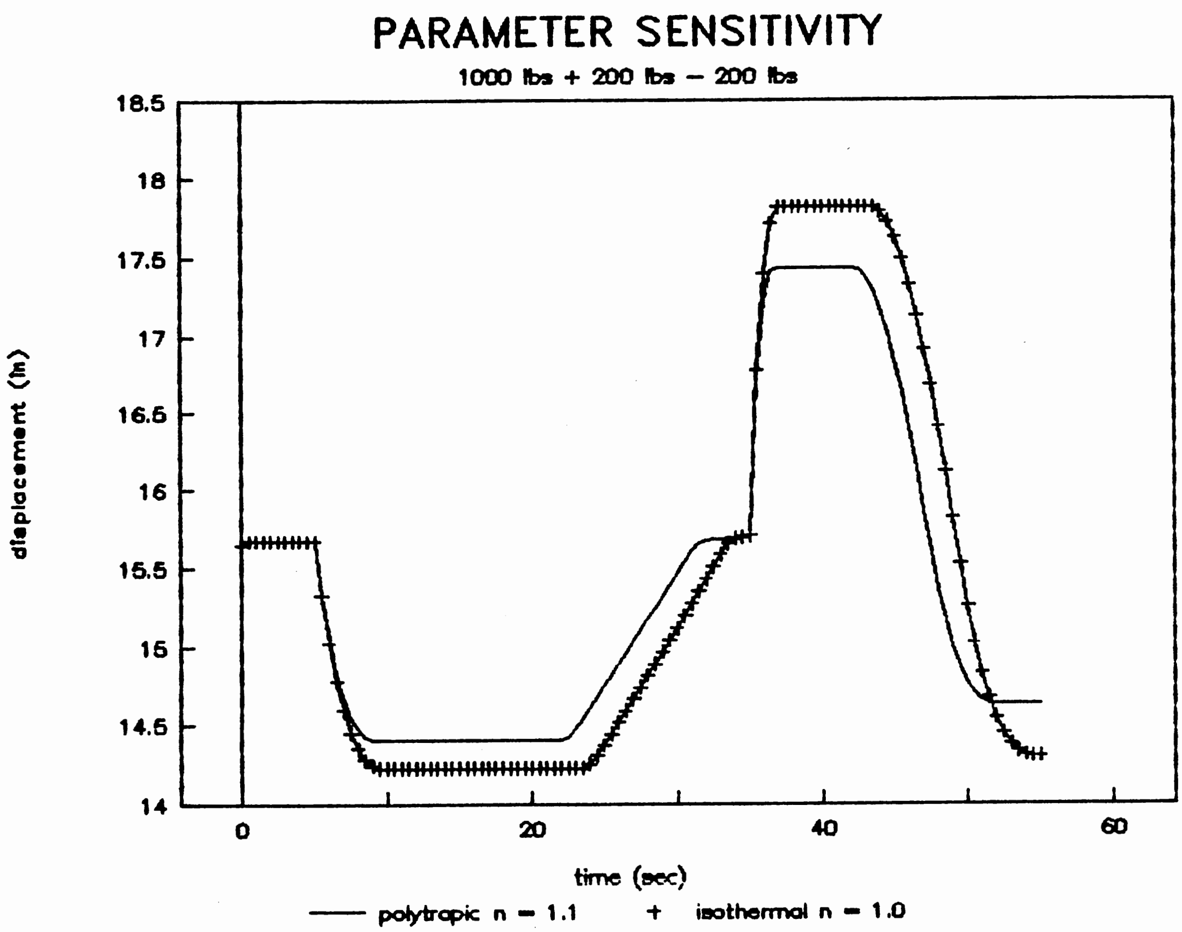

Figure 50. Displacement Graph Comparison Between Using n = 1.1 and n = 1.0 (1000)

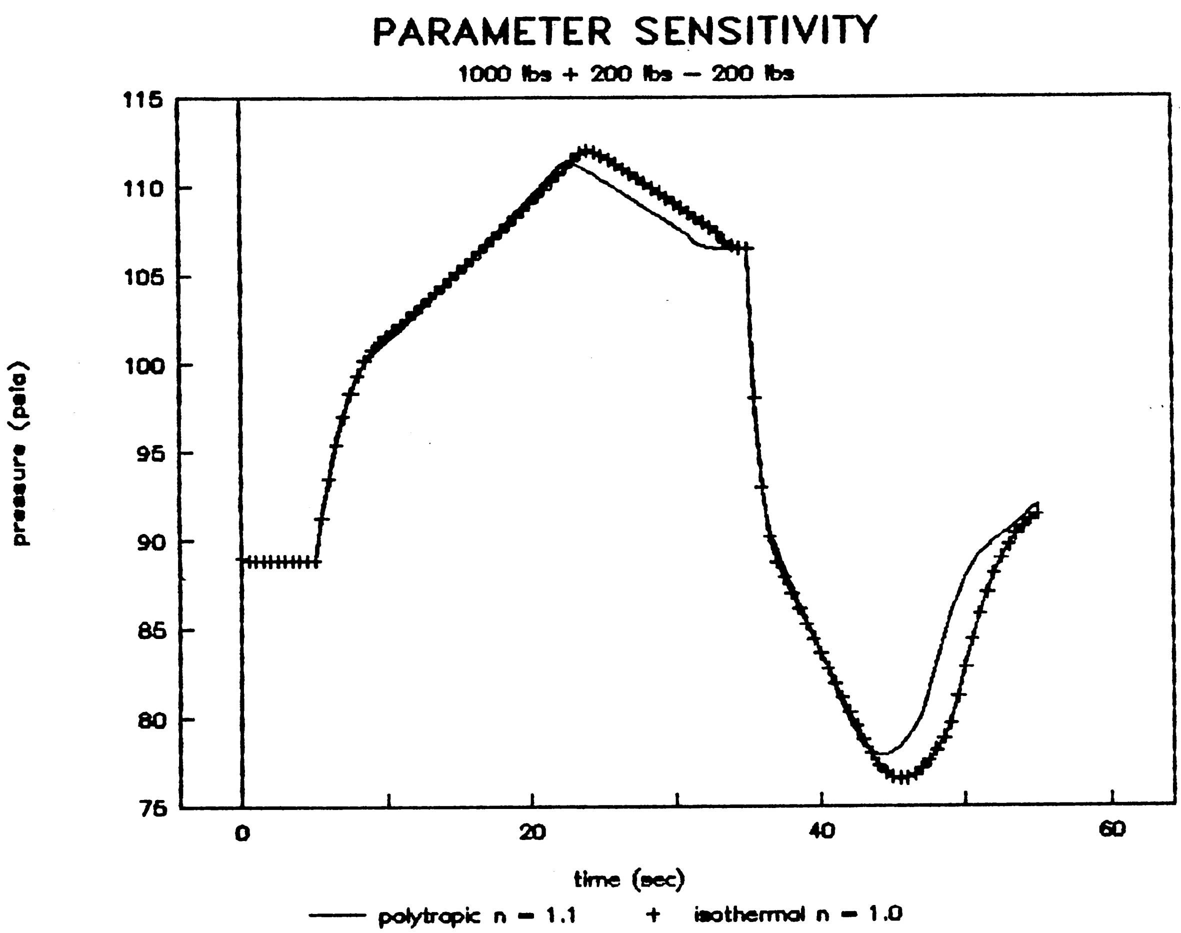

Figure 51. Pressure Graph Comparison
Between Using n = 1.1 and n = 1.0 (1000)

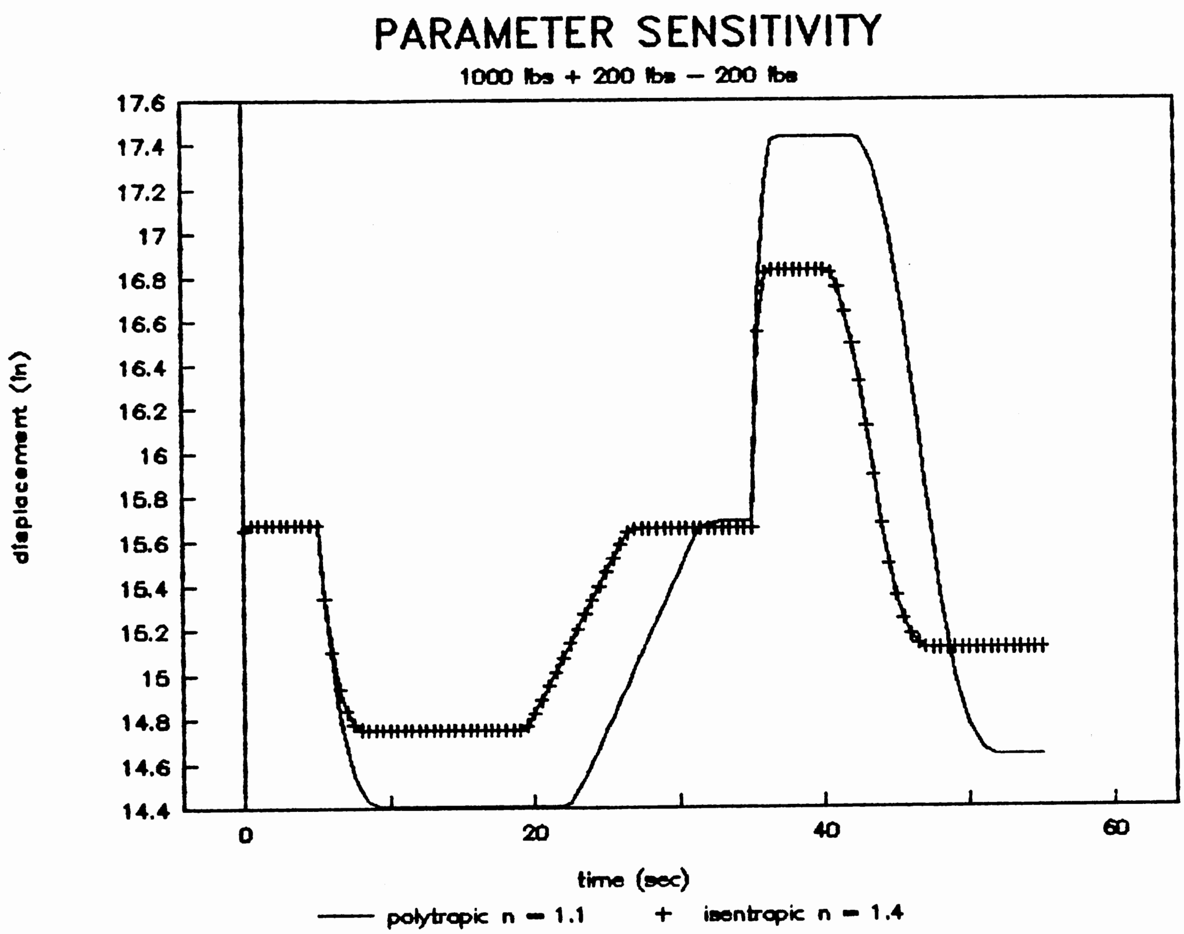

Figure 52. Displacement Graph Comparison Between Using n = 1.1 and n = 1.4 (1000)

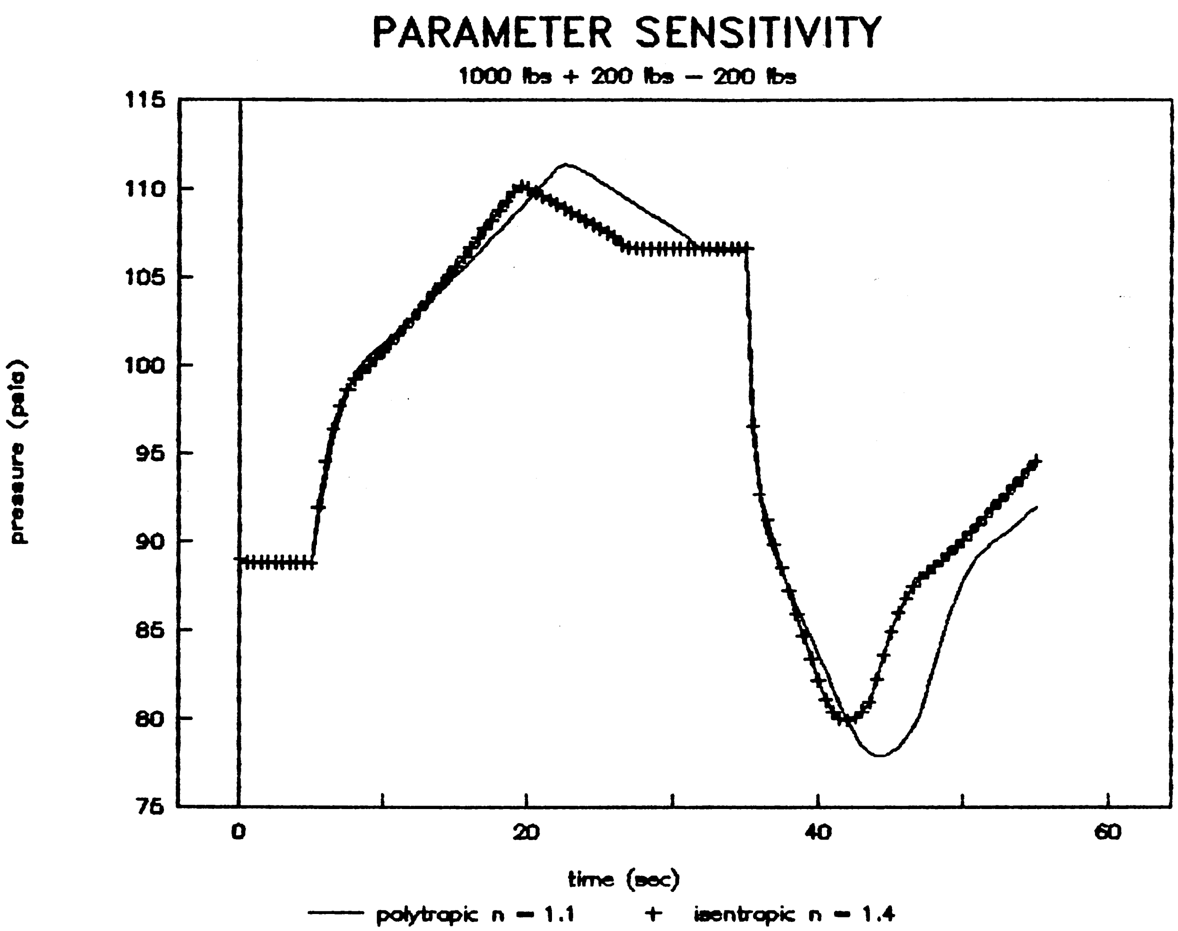

Figure 53. Pressure Graph Comparison
Between Using n = 1.1 and n = 1.4 (1000)

APPENDIX B

COMPUTER PROGRAM FOR

SIMULATION

COMPUTER PROGRAM FOR SIMULATION

In the following page is printed the Pascal program that simulate the pneumatic system. This program is using the diaphragm accumulator with characteristic given in figure 22. To build the program that calculates the simple accumulator, we just need to make the accumulator volume fixed and so the rate of change of this volume with respect with time is zero. To simulate the non-accumulator system, simply adjust the procedure on_off. In this mode, the boolean variable accumulator is always false.

```
program Pneumatic_Spring;

const
  pi               = 3.14159;
  Pa               = 14.7;     { psia }
  Vpipe            = 12.5;     { cuin }
  Charged_pressure = 126.7;    { psi }
  Cdsu             = 0.8;
  Cdsd             = 0.4;
  Cdv              = 0.12;
  Cda              = 0.3;
  Dsolenoid        = 2.0;      { mm }
  Dvent            = 2.2;      { mm }
  Dacc             = 2.2;
  yref             = 15.7;     { in }
  Upper_band       = 15.8;     { in }
  Lower_band       = 15.6;     { in }
  Kc               = 0.926;    { lb in / V }
  ec               = 13.5;     { volts }
  J                = 0.25;     { lb in sec^2 }
  Bc               = 0.1;      { lb in sec }
  Dc               = 0.0161;   { in^3 / rad }
  Cc               = 0.0001;   { in^3 psi / sec }
  Bup              = 62.0;     { lb sec / in }
  Bdown            = 10.0;     { lb sec / in }
  Z                = 0.95;
  n                = 1.1;      { polytropic constant }
  k                = 1.4;      { specific heat ratio }
  T                = 535;      { Rankine }
  R                = 640.08;   { in lbf / lbm R }
  g                = 386.4;    { in / sec^2 }
  gc               = 386.4;    { (lbm in / sec^2) / lbf }
  time1            = 5.0;
  time2            = 35.0;
  time3            = 55.0;
  dt               = 0.01;     { sec }
  NN               = 50;

var
  i                                                : integer;
  phi, phi1, Pcrit, C1, C2, ratio, As, Av, Aa      : real;
  Load, fLoad, prld, DL, CC1, CC2, Ae, Ae0         : real;
  Kcec, KcecJ, BcJ, DcJ                            : real;
  cgv, constant, Dcconstant, Ccconstant            : real;
  m, UPlimit, DPlimit, B                           : real;
  Ws, Wa                                           : real;
  upsol, upsol1, upsol2, dnsol, dnsol1, dnsol2     : real;
  upvnt, upvnt1, upvnt2                            : real;
  dnvnt, dnvnt1, dnvnt2, dnvnt1Pa, dnvnt2Pa        : real;
  upacc, upacc1, upacc2, dnacc, dnacc1, dnacc2     : real;
  time, dt2, t1, t2, t3                            : real;
  prP1, P1, d1P1, d2P1, d3P1, d4P1                 : real;
  prP2, P2, d1P2, d2P2, d3P2, d4P2                 : real;
```

```
prP3, P3, d1P3, d2P3, d3P3, d4P3                         : real;
promega,omega,d1omega, d2omega, d3omega, d4omega: real;
pry, y, dy, d1y, d2y, d3y, d4y, y0                       : real;
prvel, vel, dvel, d1vel, d2vel, d3vel, d4vel             : real;
prV, fV, V, d1V, d2V, d3V, d4V, V0                       : real;
Vacc, dVadt                                              : real;
motor, solenoid, vent, accumulator                       : boolean;


procedure define;              { DEFINING CONSTANT }
begin
dt2 := dt / 2;
t1 := time1 - dt2;
t2 := time2 - dt2;
t3 := time3 - dt2;
constant := gc / g * R * T;
phi := (k - 1.0) / k;
phi1 := 1 - phi;
Pcrit := exp( Ln( 2.0 / (k + 1) ) / phi );
C1 := g * sqrt(2.0 / R / g / phi);
C2 := g * sqrt(k / R / g / exp( (1+2/phi/k)*Ln(k/2+0.5)
));

As := pi * sqr(Dsolenoid/25.4) / 4.0;
Av := pi * sqr(Dvent/25.4) / 4.0;
Aa := pi * sqr(Dacc/25.4) / 4.0;

upsol := Cdsu * As / sqrt(T);
upsol1 := upsol * C1;
upsol2 := upsol * C2;
dnsol := -Cdsd * As / sqrt(T);
dnsol1 := dnsol * C1;
dnsol2 := dnsol * C2;
upvnt := Cdv * Av / sqrt(T);
upvnt1 := upvnt * C1;
upvnt2 := upvnt * C2;
dnvnt := -Cdv * Av /sqrt(T);
dnvnt1 := dnvnt * C1;
dnvnt1Pa := dnvnt1 * Pa;
dnvnt2 := dnvnt * C2;
dnvnt2Pa := dnvnt2 * Pa;
upacc := Cda * Aa / sqrt(T);
upacc1 := upacc * C1;
upacc2 := upacc * C2;
dnacc := -Cda * Aa / sqrt(T);
dnacc1 := dnacc * C1;
dnacc2 := dnacc * C2;

Kcec := Kc * ec;
KcecJ := Kcec / J;
BcJ := Bc / J;
DcJ := 2.0 * Dc / J;
Dcconstant := Dc / constant;
```

```
Ccconstant := Cc / constant;
cgv  := Z * constant * n / Vpipe;
end;



function fCC1 : real;        { INTERPOLATION OF EFFECTIVE
                                  AREA OF THE SPRING }
var
  A1, A2, A3, A4      : real;
begin
DL := Load / 100;
A1 := 6.711826 * DL;
A2 := -0.644472 * sqr(DL);
A3 := 0.026514 * DL * DL * DL;
A4 := -0.000393 * sqr(DL) * sqr(DL);
fCC1 := -12.176065 + A1 + A2 + A3 + A4;
end;



function fCC2 : real;
begin
fCC2 := 0.1271 + 0.0121 * DL;
end;



procedure on_off;            { ON-OFF CONTROLLER }
begin
if y < Lower_band then
  begin
  motor := true;
  solenoid := true;
  vent := false;
  if P3 > P1 then accumulator := true
  else accumulator := false;
  end

else if y > Upper_band then
  begin
  motor := false;
  if P3 < Charged_pressure then
    begin
    if P3 < P1 then accumulator := true
    else accumulator := false;
    end
  else
    accumulator := false;
  solenoid := true;
  vent := true;
```

```
  end

else
  begin
  solenoid := false;
  vent := false;
  if P3 >= Charged_pressure then
    begin
    motor := false;
    accumulator := false;
    end
  else
    begin
    motor := true;
    if P3 < P1 then accumulator := true
    else accumulator := false;
    end;
  end;

end;




function alp : real;          { COMPRESSOR ANGULAR
                                    ACCELERATION}
begin
if motor then
  begin
  if omega > 1e-6 then
    alp :=  KcecJ - BcJ * omega - DcJ * (P1 - Pa)
  else if omega < -1e-6 then
    alp :=  KcecJ - BcJ * omega
  else
    alp := KcecJ;
  end

else
  begin
  if omega > 1e-6 then
    alp :=   - BcJ * omega - DcJ * (P1 - Pa)
  else if omega < -1e-6 then
    alp :=   - BcJ * omega
  else
    alp := 0.0;
  end
end;
```

```
function comprflow : real;   { COMPRESSOR FLOW RATE }
begin
if motor then
  begin
  if omega > 1e-4 then
    begin
    if P1 - Pa > 1e-4 then
      comprflow := P1 * ( Dcconstant * omega - Ccconstant
* (P1 - Pa) )
    else
      comprflow := P1 * ( Dcconstant * omega );
    end
  else
    comprflow := 0.0;
  end

else
  comprflow := 0.0;
end;

function accflow : real;     { ACCUMULATOR FLOW RATE }
begin
if accumulator then
  begin
  if P1 - P3 > 1e-6 then
    begin
    ratio := P3 / P1;
    if ratio <= Pcrit then
      accflow := upacc2 * P1
    else
      accflow := upacc1 * P1 * exp( phi1*Ln(ratio) )
                 * sqrt( 1 - exp(phi*Ln(ratio)) );
    end
  else if P1 - P3 < -1e-6 then
    begin
    ratio := P1 / P3;
    if ratio <= Pcrit then
      accflow := dnacc2 * P3
    else
      accflow := dnacc1 * P3 * exp( phi1*Ln(ratio) )
                 * sqrt( 1 - exp(phi*Ln(ratio)) );
    end
  else
    accflow := 0.0;
  end

else
  accflow := 0.0;
end;
```

```
function solflow : real;     { FLOW RATE THROUGH SOLENOID
                                          VALVE }
begin
if solenoid then
  begin
  if P1 - P2 > 1e-6 then
    begin
    ratio := P2 / P1;
    if ratio <= Pcrit then
      solflow := upsol2 * P1
    else
      solflow := upsol1 * P1 * exp(phi1*Ln(ratio))
              * sqrt( 1 - exp( phi * Ln(ratio) ) );
    end

  else if P1 - P2 < -1e-6 then
    begin
    ratio := P1 / P2;
    if ratio <= Pcrit then
      solflow := dnsol2 * P2
    else
      solflow := dnsol1 * P2 * exp(phi1*Ln(ratio))
              * sqrt( 1 - exp( phi * Ln(ratio) ) );
    end
  else
    solflow := 0.0;
  end

else
  solflow := 0.0;
end;



function ventflow :real;     { DISCHARGED FLOW RATE }
begin
if vent then
begin
  if P1 - Pa > 1e-6 then
    begin
    ratio := Pa / P1;
    if ratio <= Pcrit then
      ventflow := upvnt2 * P1
    else
      ventflow := upvnt1 * P1 * exp(phi1*Ln(ratio))
              * sqrt( 1.0001 - exp( phi * Ln(ratio) ) );
    end
  else if P1 - Pa < -1e-6 then
    begin
    ratio := P1 / Pa;
    if ratio <= Pcrit then
      ventflow := dnvnt2Pa
```

```
    else
      ventflow := dnvnt1Pa * exp(phi1*Ln(ratio))
              * sqrt( 1 - exp( phi * Ln(ratio) ) );
    end
  else
    ventflow := 0.0;
  end

else
  ventflow := 0.0;
end;



function apc : real;         { RATE OF CHANGE OF PRESSURE
                                 IN THE ACCUMULATOR }
begin
apc := constant * Wa / (Vacc/Z/n + P3*dVadt);
end;



function ppc : real;         { RATE OF CHANGE OF PRESSURE
                                 IN THE PIPE }
begin
ppc := cgv * ( comprflow - Wa - Ws - ventflow );
end;


function spc : real;         { RATE OF CHANGE OF PRESSURE
                                 IN THE PNEUMATIC SPRING }
begin
spc := Z * n / V * (constant * Ws - P2 * fV);
end;



procedure Plimit;            { HYSTERESIS CHARACTERISTIC }
begin

if Load < 950.0 then {if abs(Load - 900.0) < 1e-3 then}
  begin
  UPlimit := -3.43 * y + 130.0;
  DPlimit := UPlimit - 11.0;
  end

else if Load < 1050.0 then {else if abs(Load - 1000.0) <
1e-3 then}
  begin
  UPlimit := -3.67 * y + 139.33;
  DPlimit := UPlimit - 10.5;
  end
```

```
  else if Load < 1150.0 then {else if abs(Load - 1100.0) <
le-3 then}
    begin
    UPlimit := -3.5 * y + 141.5;
    DPlimit := UPlimit - 10.5;
    end

  else {else if abs(Load - 1200.0) < le-3 then}
    begin
    UPlimit := -3.67 * y + 149.33;
    DPlimit := UPlimit - 10.5;
    end;

  end;



  function acc : real;          { ACCELERATION OF THE LOAD }
  begin
  if P2 - Pa > UPlimit then
    acc := gc / m * ( (P2 - Pa - UPlimit) * Ae - B * vel )
  else if P2 - Pa < DPlimit then
    acc := gc / m * ( (P2 - Pa - DPlimit) * Ae - B * vel )
  else
    acc := gc / m * ( - B * vel );
  end;





begin
define;

time := 0.0;                    { INITIAL CONDITION }
Load := 900.0;      { lbf }
m := Load / g * gc + 50.0;
fLoad := -Load;

motor := true;
vent := false;
accumulator := false;
solenoid := true;

CC1 := fCC1;
CC2 := fCC2;
y0 := 15.65;
Ae0 := CC1 + CC2 * y0;
V0 := 215.0;
```

```
omega := 83.2649;
P1 := 89.0;
P2 := 89.0;
P3 := Charged_pressure;
V := V0;
y := y0;
vel := 0.5067;
Plimit;

writeln(time:8:4,'    ',y:10:4,'    ',P2:15:4,'   ',P1:15:4,'
',P3:15:4);

i := 0;
repeat                      { RUNGE-KUTTA ITERATION STARTS }
begin
prld := Load;
promega := omega;
pry := y;
prvel := vel;
prV := V;
prP1 := P1;
prP2 := P2;
prP3 := P3;


if time > t2 then               { LOAD CHANGE }
  Load := 900.0
else if time > t1 then
  Load := 1100.0;

if abs(prld - Load) > 1e-3 then
  begin
  m := Load / g * gc + 50.0;
  fLoad := -Load;
  CC1 := fCC1;
  CC2 := fCC2;
  end;




if P3 < 40.0 then               { CHARACTERISTIC OF DIAPHRAGM
                                          ACCUMULATOR }
  begin
  Vacc := 20.0;
  dVadt := 0.0;
  end
else if P3 < 106.7 then
  begin
  Vacc := 0.1546 * P3 + 13.8176;
  dVadt := 0.1546;
  end
```

```
else if P3 < 112.7 then
  begin
  Vacc := 75.531097 * P3 - 8028.85463;
  dVadt := 75.531097;
  end
else
  begin
  Vacc := 0.75 * P3 + 398.975;
  dVadt := 0.75;
  end;

if vel > 0.0001 then B := Bup { UPWARD DAMPER }
else if vel < -0.0001 then B := Bdown; { DOWNWARD DAMPER }



Wa := accflow;
Ws := solflow;
Ae := CC1 + CC2 * y;
fV := Ae * vel;
Plimit;
d1omega := alp * dt;
d1P1 := ppc * dt;
d1P2 := spc * dt;
d1P3 := apc * dt;
d1V := fV * dt;
d1y := vel * dt;
d1vel := acc * dt;

omega := promega + d1omega/2;
P1 := prP1 + d1P1/2;
P2 := prP2 + d1P2/2;
P3 := prP3 + d1P3/2;
V := prV + d1V/2;
y := pry + d1y/2;
vel := prvel + d1vel/2;
Wa := accflow;
Ws := solflow;
Ae := CC1 + CC2 * y;
fV := Ae * vel;
Plimit;
d2omega := alp * dt;
d2P1 := ppc * dt;
d2P2 := spc * dt;
d2P3 := apc * dt;
d2V := fV * dt;
d2y := vel * dt;
d2vel := acc * dt;

omega := promega + d2omega/2;
P1 := prP1 + d2P1/2;
P2 := prP2 + d2P2/2;
P3 := prP3 + d2P3/2;
```

```
V := prV + d2V/2;
y := pry + d2y/2;
vel := prvel + d2vel/2;
Wa := accflow;
Ws := solflow;
Ae := CC1 + CC2 * y;
fV := Ae * vel;
Plimit;
d3omega := alp * dt;
d3P1 := ppc * dt;
d3P2 := spc * dt;
d3P3 := apc * dt;
d3V := fV * dt;
d3y := vel * dt;
d3vel := acc * dt;

omega := promega + d3omega;
P1 := prP1 + d3P1;
P2 := prP2 + d3P2;
P3 := prP3 + d3P3;
V := prV + d3V;
y := pry + d3y;
vel := prvel + d3vel;
Wa := accflow;
Ws := solflow;
Ae := CC1 + CC2 * y;
fV := Ae * vel;
Plimit;
d4omega := alp * dt;
d4P1 := ppc * dt;
d4P2 := spc * dt;
d4P3 := apc * dt;
d4V := fV * dt;
d4y := vel * dt;
d4vel := acc * dt;

time := time + dt;
omega := promega + (d1omega + 2*d2omega + 2*d3omega +
d4omega) / 6;
P1 := prP1 + (d1P1 + 2*d2P1 + 2*d3P1 + d4P1) / 6;
P2 := prP2 + (d1P2 + 2*d2P2 + 2*d3P2 + d4P2) / 6;
P3 := prP3 + (d1P3 + 2*d2P3 + 2*d3P3 + d4P3) / 6;
V :=  prV + (d1V + 2*d2V + 2*d3V + d4V) / 6;
y :=  pry + (d1y + 2*d2y + 2*d3y + d4y) / 6;
dvel := (d1vel + 2*d2vel + 2*d3vel + d4vel) / 6;
vel := prvel + dvel;

on_off;


i := (i + 1) mod NN;
if i = 0 then
  begin
```

```
  writeln(time:8:4,'      ',y:10:4,'      ',P2:15:4,'
',P1:15:4,'   ',P3:15:4);

  end;

end
until time > t3;


end.
```

VITA

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Master of Science

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