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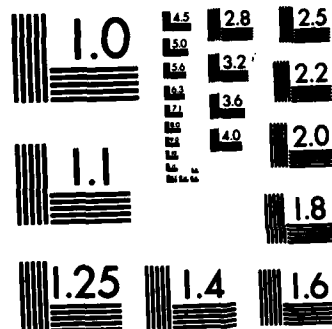
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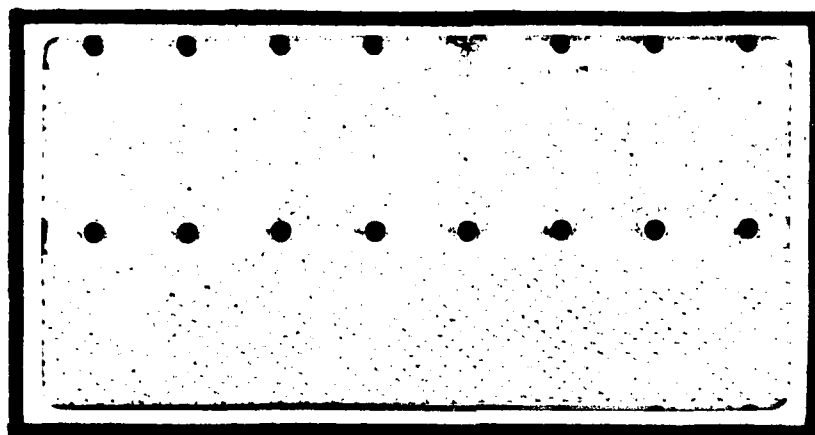
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DYNAMIC CHARACTERISTICS OF
AERIAL REFUELING SYSTEMS

THESIS

AFIT/GAE/AA/82D-4

Thomas J. Carter III
Capt USAF

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DYNAMIC CHARACTERISTICS OF
AERIAL REFUELING SYSTEMS

THESIS

Presented to the Faculty of the School of Engineering
of the Air Force Institute of Technology
Air University
In Partial Fulfillment of the
Requirements for the Degree of
Master of Science

by
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Captain USAF
Graduate Aeronautical Engineering
December 1982

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Thomas J. Carter III

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List of Symbols

A	Area	in^2
A_v	Effective Valve Area	in^2
A_{vo}	Full Flow Valve Area	in^2
c	Sonic Velocity	in/sec
C_d	Discharge Coefficient	
K	Valve Position	% Open
L	Length	in
L()	Line Number ()	
$\dot{m}$	Mass Rate of Flow	in^3/sec
P	Pressure	lb_f/in^2
Q	Flowrate	in^3/sec
T	Time	sec
t	Time	sec
T_c	Valve Closure Time	sec
V	Velocity	in/sec
ρ	Density	$(\text{lb}_f\text{-sec}^2)/\text{in}^4$

Abstract

Two aircraft refueling system configurations were simulated using HYTRAN, an existing transient flow analysis computer program. Transient pressure response subsequent to downstream valve closure was investigated for the KC-135 and a laboratory test rig and compared to available experimental data and previous research. Parametric studies were performed of system variables: valve closure time, valve area versus time (closure curves), and variations in surge attenuation components (surge boot, accumulators). A typical closure curve for quick disconnects was approximated and verified. The simulation results compared favorably with experimental data and previous work. Transient pressure was found to be sensitive to accumulator or surge boot precharge pressure while changes in accumulator volume had little effect. Variations in accumulator entry line diameter or length affected transient pressure and settling time. Increased valve snubbing reduced maximum transient pressures.

DYNAMIC CHARACTERISTICS OF AERIAL REFUELING SYSTEMS

I. Introduction

Problem Background

An aerial refueling system is an auxiliary pumping system installed in a tanker aircraft which is used to transfer fuel to a receiver aircraft. During inflight refueling operations fuel is pumped from the tanker to the receiver through either an extensible, rigid refueling boom or a flexible hose. Only the rigid boom method will be considered in this study.

Fuel spillage during planned or inadvertant separation of the tanker and receiver is prevented by use of a spring loaded shut-off valve at the boom delivery nozzle. A spring loaded valve is also provided at the receiver fuel receptacle.

The rapid closing of the nozzle shut-off valve during separation can cause transient disturbances to propagate through the tanker refueling system. Depending on the time interval of the transient, pressure waves traveling at speeds close to that of sound waves, about 3000 ft/sec, can easily generate energy enough to damage the system. This is the so-called "waterhammer effect". Refueling system performance enhancements such as higher fuel flow rates and faster closing shut-off valves for less fuel spillage, can increase the potential for damage due to transient pressure surges. Therefore, pressure surge effects are of primary interest in the design of new systems and the upgrading of existing configurations. For the

system structure to be maintained within aircraft weight standards and still withstand "worst case" pressure transients, methods must be employed to attenuate excessive pressure surges.

When designing surge attenuation components, reliable design data is needed to make a reasonable estimate of pressure surge magnitude. This design data is commonly obtained from either analytical studies, experiment, or both. General fluid transient phenomena have been researched and documented rather extensively. Wylie and Streeter (Ref 1) provide a comprehensive study of fluid transients and Goodson and Leonard (Ref 2) summarize the development of models for fluid line applications. Streeter and Lai (Ref 3) used numerical methods and the digital computer to solve waterhammer problems; their solutions were verified by experiment with good results. The McDonnell Douglas Corporation (Ref 4,5,6) used computer techniques to simulate complex aircraft and spacecraft hydraulic systems. Numerous experiments were performed to verify various test cases.

However, for aircraft refueling systems, there is a limited amount of well documented transient response experimental data. Some of the available experimental results have been used in extending the applications of existing hydraulic analysis programs to refueling systems. Parks and Franke (Ref 7) investigated modeling of the KC-135 tanker system. Their study focused on the transient conditions during and after a 0.05 sec closure of the nozzle shut-off valve. Kinzig et al. (Ref 8) studied the compatibility of the KC-135 and KC-10 tankers with a simulated receiver system. Shut-off valve closing times of approximately 0.22 and 1.0 sec were used to simulate

the receiver shut-off valve closures during refueling. The objective of the study was to predict transient pressures in the receiver aircraft.

Both of the refueling system simulations discussed above agreed favorably with experimental results for the conditions simulated. However, there is an element of uncertainty critical to all refueling system simulations that must be resolved to achieve accurate solutions. This element is the shut-off valve closure characteristics, i.e., valve area versus time. Kinzig (Ref 8:10) noted that small changes in closure curve shape could result in large differences in pressure surge magnitude. A McDonnell Douglas Corporation (Ref 9:6-7) laboratory test indicated nozzle valve closure times area function of separation rate. Thus, for varying separation rates of tanker and receiver the valve closure characteristics are likely to change. As a consequence, changes in separation rate will influence pressure surge magnitude. Therefore an accurate approximation of valve closure characteristics is a critical input for refueling system simulations.

Scope and Objectives

The purpose of this thesis is to investigate, by computer simulation, the effects of variations in valve closure characteristics and surge attenuation components on the dynamic response of an aircraft refueling system. HYTRAN, a McDonnell Douglas Corporation (Ref 4) developed fluid transient analysis computer program will be used. Two distinct system configurations, the KC-135 and a preliminary KC-10 laboratory test set up, hereafter referred to as the Laboratory Test, will be modeled and simulated with the program.

The specific objectives of the study are to:

- a) Establish an approximation of the actual valve closure characteristics for a typical refueling nozzle.
- b) Verify the general utility of HYTRAN by comparison with results of Streeter and Lai.
- c) Compare the simulated results of this study with experimental results of the Laboratory Test.
- d) Determine, for the Laboratory Test model, the influence of variations in closure curve shape, i.e., valve area versus time, and modifications to surge attenuation components.
- e) Determine, for the KC-135 model, the influence of both closure curve shape variation and closure time.
- f) Examine the effect of varying the KC-135 surge boot precharge pressure.

II. System Description

General

Tanker aircraft are equipped to transfer fuel to receiver aircraft by means of an extensible, rigid boom or a flexible hose arrangement known as the probe and drogue system. This report considers only the rigid boom method. The refueling boom is basically a telescoping tube attached to the underside of the fuselage that is maneuvered into contact with a receiver aircraft. A nozzle at the end of the boom fits into a refueling receptacle on the receiver aircraft to provide a continuous flow path between the two aircraft. A spring-loaded poppet valve forms the seal in the end of the nozzle when the nozzle is not in contact with the receiver receptacle. The poppet valve is automatically opened during insertion of the nozzle into the receptacle by an actuator in the receptacle.

The primary surge attenuation component is the surge boot. A surge boot is an expandable, gas charged envelope that surrounds a perforated section of the boom, normally the aft portion. The basic parts of the surge boot are shown in Fig 1; the inner surface of the envelope seals the perforations in the boom until the fluid pressure in the boom exceeds the precharge on the gas. As the precharge is exceeded, fuel under pressure forces the envelope inner surface away from the outer surface of the boom. Therefore, fuel outflow through the perforations and into the cavity compresses the gas in the surge boot and provides energy dissipation during pressure surges.

During a normal disconnect of tanker and receiver a signal will be automatically sent through the system that shuts off the refueling

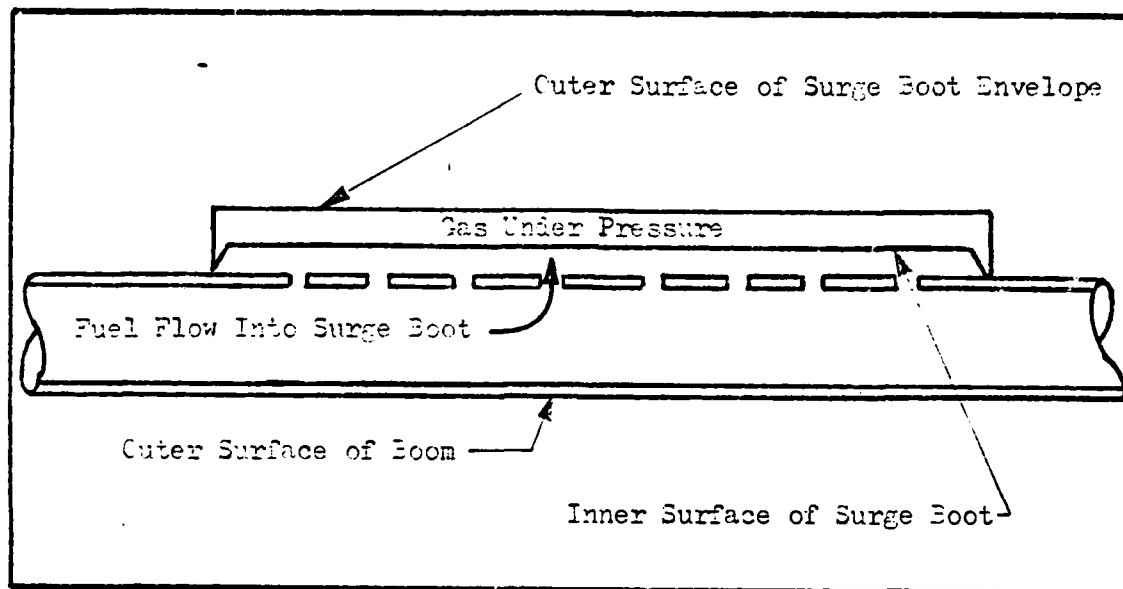


Fig 1. Surge Boot Details

pumps, opens a bypass valve, and initiates boom retraction. The opening of the bypass valve allows fuel to flow back to the tanker fuel tank so there is no sudden flow stoppage during disconnect and retraction. Any surge produced under these conditions will be mild and easily absorbed by the surge boot. Normal disconnects pose little or no threat to the system.

The abnormal disconnect conditions that cause concern are the breakaway situations due to turbulence, emergency actions, or a failure of the automatic process. In these cases it is likely that there will be a flowing disconnect whereupon the pumps continue to run against the closed shut-off valve in the nozzle with no retraction of the boom or bypass of the flow.

Failure of the pumps to shut-off increases the pressure in the line to deadhead pump pressure (in some cases 90 psia or more) in addition to any surge pressure induced by the nozzle shut-off valve

closure (Ref 11:6,16). Since no bypass of fuel occurs there is a stoppage of the full flow through the boom at the time of disconnect. High separation rates of the tanker and receiver, up to the normal minimum of 6.5 ft/sec (Ref 9:7), are usually associated with abnormal disconnects.

KC-135 System (Ref 11,12)

A simplified diagram of the KC-135 refueling system is shown in Fig 2. The KC-135 refueling system is a four pump (constant rpm, each rated at 300 gpm @ 80 psig) system with two refueling pumps in the forward body tank and two in the aft body tank. Reverse flow is prevented to each pump by individual check valves. Fuel flows from all four pumps are united in a main trunk line; fuel then flows through a shut-off valve, pressure regulator, and venturi. Finally the fuel flows through the boom which contains a surge boot with a nominal precharge of 50 psig. The pressure regulator is set to maintain the steady state pressure at the nozzle to 50 ± 5 psig. A secondary method of pressure surge protection is provided by two level control valves located in the lines leading from the refueling manifold to the forward and aft body tanks. The valves' cracking pressure of 85-115 psi allows fuel to flow to the body tanks during pressure surges (Ref 10:3).

Laboratory Test

The McDonnell Douglas Corporation performed an experimental investigation (Ref 9) to study surge pressures developed by the closure of shut-off valves during tanker/receiver disconnects. These

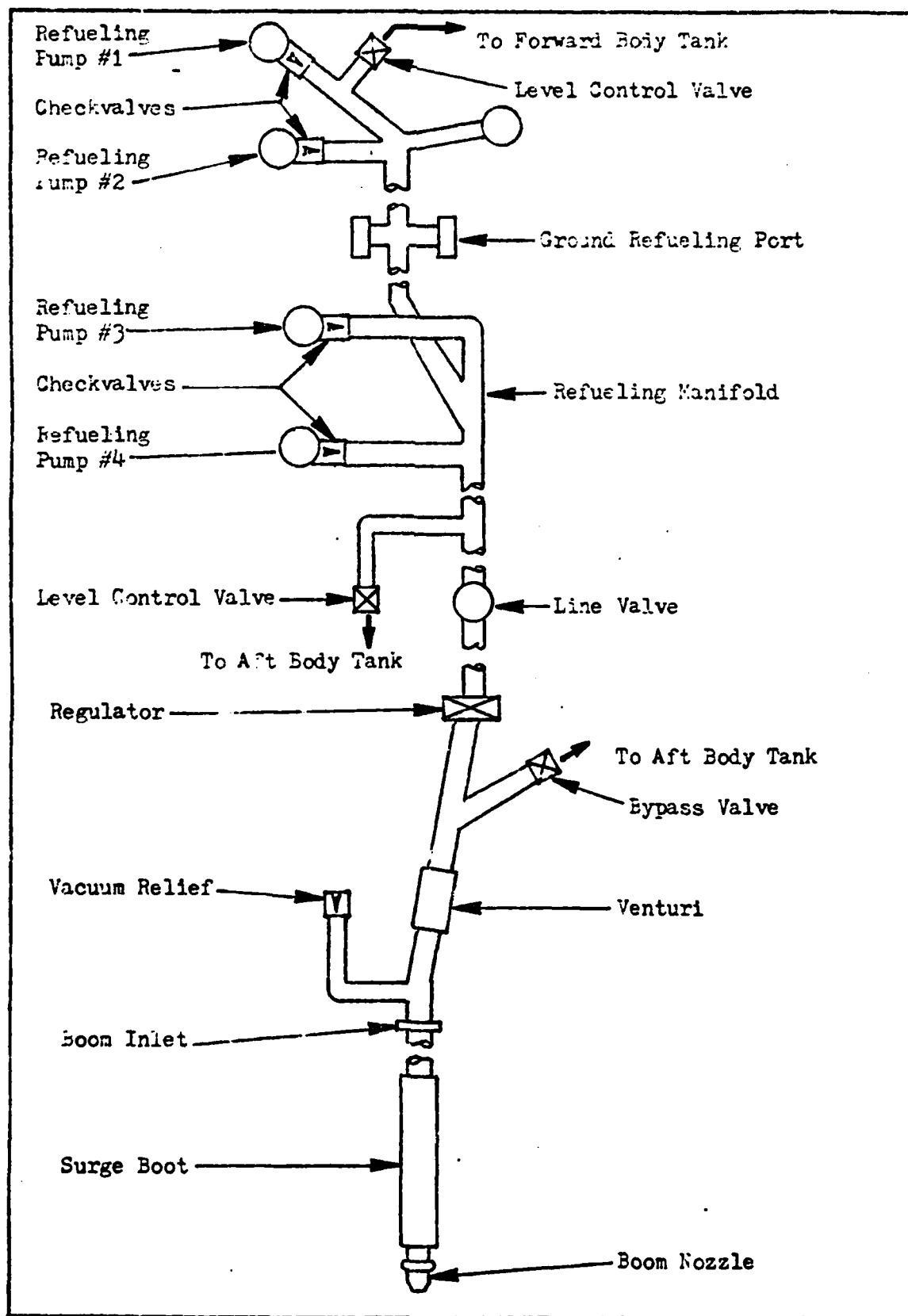


Fig 2. Simplified Diagram of KC-135 Refueling System

tests were performed as part of the Advanced Aerial Refueling Boom/Advanced Aerial Refueling Nozzle development program preceding the KC-10 development.

The experimental setup shown in Fig 3 consisted of an aluminum tube assembly simulating the actual refueling boom, four discrete accumulators to simulate the surge boot, and a laboratory pumping system that supplied fuel to the rig. A refueling nozzle was mounted at the end of the aluminum tube and a refueling receptacle was attached to a carriage that was free to slide between two tracks. The receptacle separation was driven by means of a hydraulic and bungee system.

The test procedure was to establish a predetermined flow rate, set the load on the receptacle for the anticipated separation rate, and activate the disconnect system. Data recorded included flow rate, boom inlet and nozzle pressure, separation rate, and nozzle valve poppet travel. The test conditions are discussed in Chapter IV with the computer simulation model description.

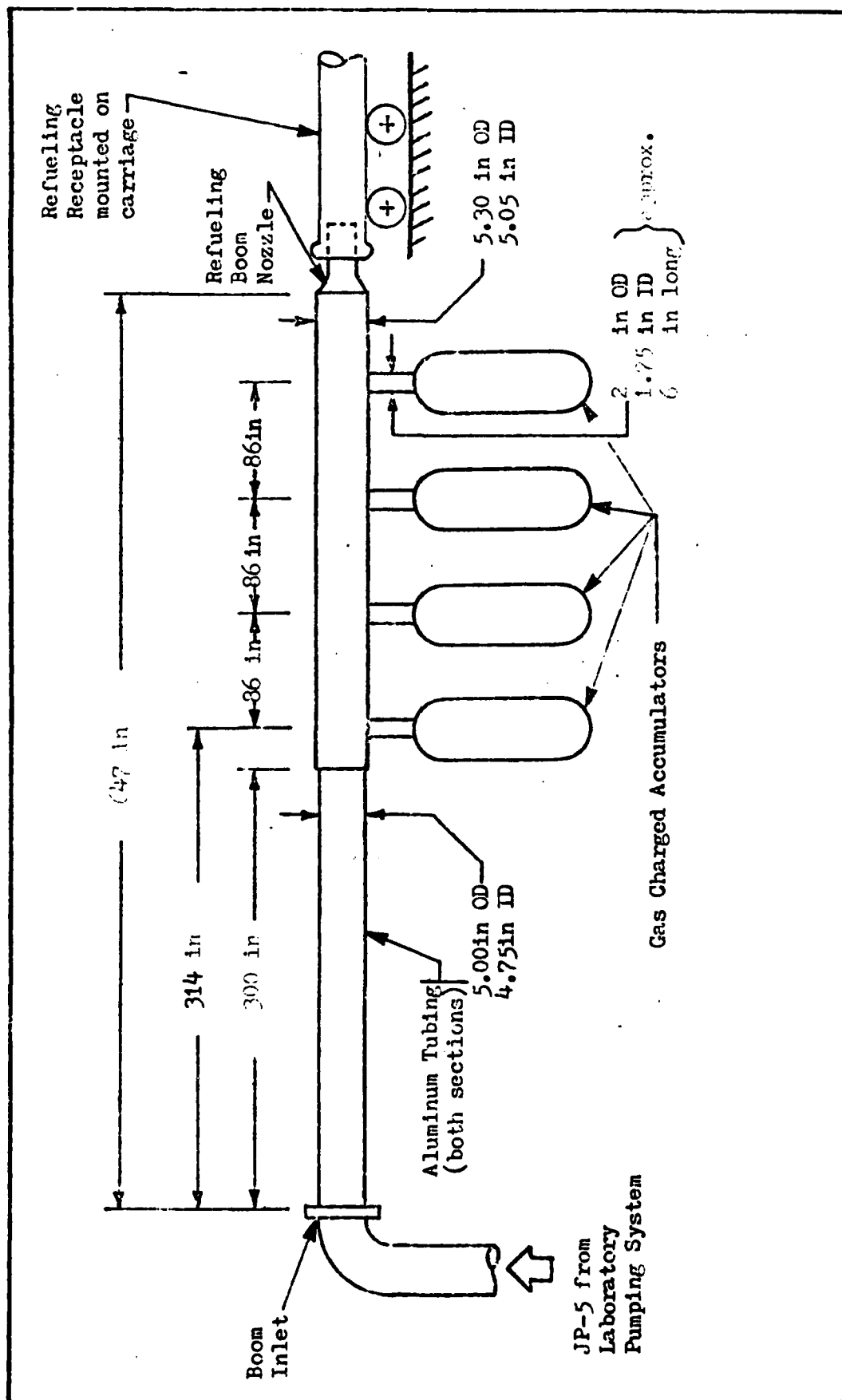


Fig 3. Laboratory Test Configuration

III. Computer Program Description

General

HYTRAN, the hydraulic transient analysis computer program used in this study, was developed by the McDonnell Douglas Corporation under an Aero Propulsion Laboratory contract (Ref 4). The contract included not only program development but also extensive validation and documentation. The program uses a building block approach to facilitate simulation of complete systems. After the configuration is established according to HYTRAN format, the program calculates the values of flows, pressures, and component variables throughout the system.

The basic theory of HYTRAN is that a change in flow (momentum) results in a change in pressure. Flow is brought to rest at the expense of an increase in pressure. By coupling the continuity and momentum equations with component equations, pressure and flow as a function of time and line distance can be determined. The mathematical solution technique used by the program is the method of characteristics. The major assumptions upon which the program is based are:

- a) Fluid temperature is held constant during the entire run.
- b) Flow is one dimensional, that is, the fluid properties are constant across any transverse cross section of pipe.
- c) Pipes have circular cross sections and friction factors based on smooth, drawn tubing.
- d) Stresses in pipes are always below the elastic limit.

e) Pipe geometry is such that the "thin wall" case is valid.

f) Pipe and liquid are perfectly elastic (all energy dissipation is due to shearing stresses at the walls).

Each component subroutine has its own set of assumptions since each model is broken down into its most basic equations of motion and flow.

The controlling input to the system is normally a valve motion, which causes a disturbance to propagate through the system at or near the speed of sound. The components of the system each have a response to the pressure and flow changes.

HYTRAN is composed of four basic parts: input, steady state calculations, transient calculations, and output. A simplified flowchart is shown in Fig 4.

Included in the input data are line and component parameters, the system configuration, i.e., how the lines and components are connected, initial conditions, fluid properties, and various directive parameters such as time steps, run time, plotting intervals, etc.

The steady state section of the program balances the pressures and flows in the system and calculates the initial values for all the system variables. Once the initial values are established at zero time, the program starts by calculating, for a small change in time (ΔT), new flows, pressures, and values for the component variables such as valve position, accumulator fluid volumes, etc.

The program continues forward in ΔT time intervals, first calculating the line and then the component variables until the desired simulation time period is covered.

The output is time histories of selected system variables which

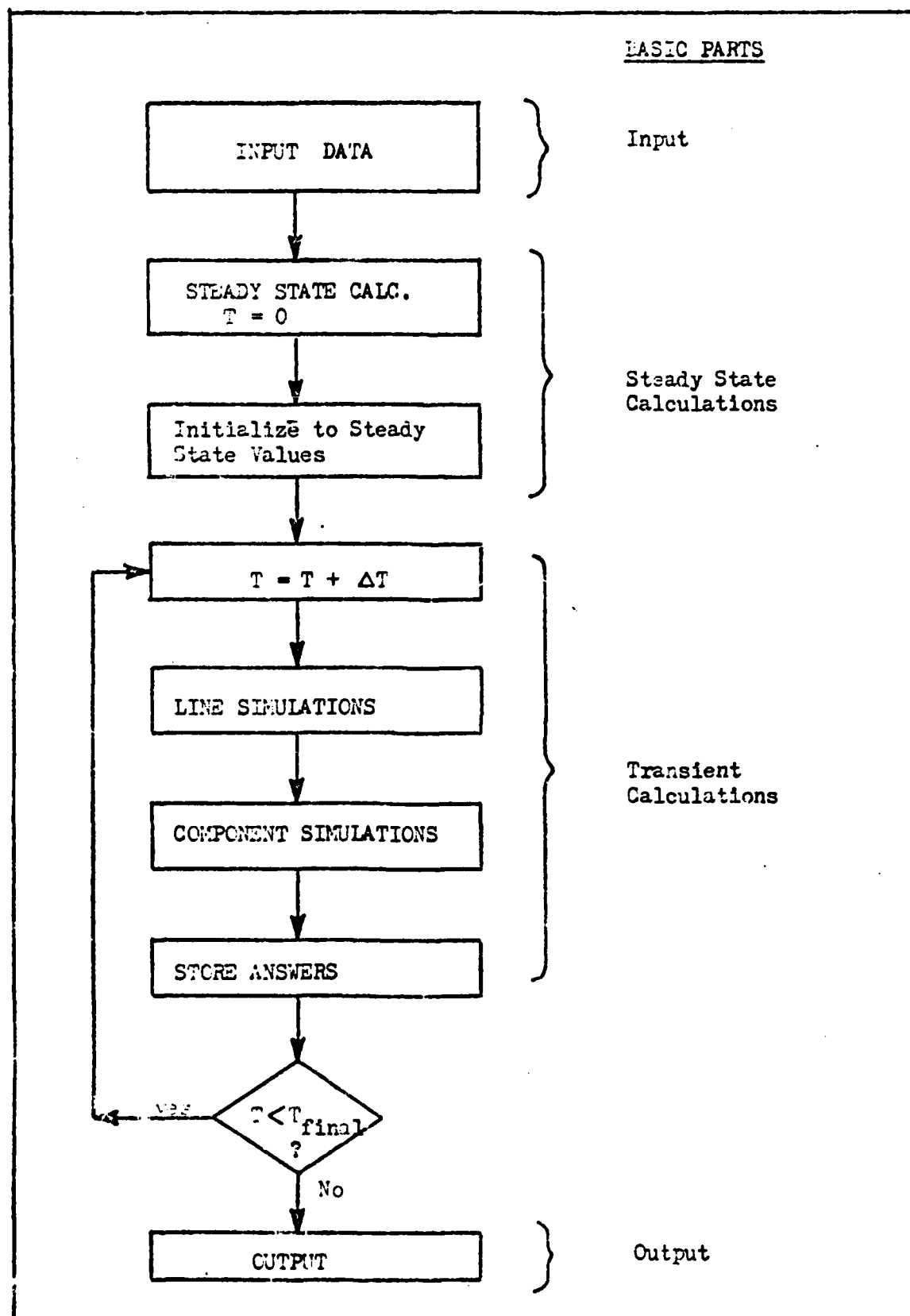


Fig 4. Simplified HYTRAN Flowchart

have been disturbed by the controlling input. A typical output plot would be a pressure versus time curve for before, during, and after a valve closure.

A detailed description of program structure, methods of solution and component models is provided in Ref 4. The program requires approximately 170,000 core memory and a run time between 80 and 300 sec.

Fuel System Usage

Although HYTRAN was developed for use with hydraulic systems, the basic assumptions and solution techniques are common to most fluid systems. The main alterations required for use with fuel systems was to change the fluid properties from hydraulic fluid to fuel.

System Modeling

The first step in developing a HYTRAN model is to construct the system schematic in HYTRAN format. This involves the assignment of line, component, node, and leg numbers throughout the system as well as determining which component models will be selected to simulate each individual component. References 13 and 14 explain in detail how to translate a system into HYTRAN format.

The specific HYTRAN component models used for the simulation models discussed in Chapter IV and V are discussed briefly in the following paragraphs. Detailed descriptions of each component model can be found in Ref 4 and 15.

Fuel tanks and flow sources were simulated with HYTRAN TYPE 61 constant pressure reservoirs. The reservoir is assumed to have an infinitely large gas volume so that pressure remains unchanged.

Lines were joined at branches and junctions with HYTRAN TYPE 11 branches. The branches do not incorporate any losses associated with changes in diameter and flow direction or divisions, i.e., they are frictionless branches.

Fixed orifices were modeled using HYTRAN TYPE 41 restrictors. The model assumes flow in either direction. The discharge coefficient is the same for flow in each direction and constant over the entire flow regime.

The HYTRAN TYPE 31 check valve model was used to simulate check and relief valves. The check valve is modeled as an orifice of variable area. The area at any given time is determined by the position of the force-balanced poppet. Reverse flow can take place until the valve closes.

Control valves were simulated using HYTRAN TYPE 21 two-way valves. The valve is modeled as an orifice of variable area and a constant discharge coefficient over the entire flow range. A time history of valve closure is input in the form of an effective valve area versus time table. The effective valve is defined as the valve area at a particular time multiplied by the discharge coefficient. In this study, valve area versus time plots are presented in the nondimensional form of valve position versus percent of closure time. The valve position, given in percent open, is the valve area with respect to the full flow valve area. Likewise, the percent of closure time values are the time values for each valve position with respect to the total closure time. Therefore, the effective area with respect to time for program input is obtained by multiplying the full flow area by valve position and discharge

coefficient is shown below

$$A_v(t) = K(t)C_d A_{vo} \quad (1)$$

where

$A_v(t)$ = effective valve area with respect to time

$K(t)$ = valve position with respect to time

C_d = discharge coefficient

A_{vo} = full flow valve area

Accumulators were simulated with HYTRAN TYPE 71 accumulator models. Each unit is modeled as a simple gas charged piston type accumulator with the frictional and inertial effects neglected. The model assumes a nonvarying gas constant and no heat transfer between the gas and the accumulator walls.

IV. Laboratory Test Computer Model

The Laboratory Test configuration in Fig 3 was modeled using the HYTRAN components discussed in Chapter III. A schematic of the test configuration in HYTRAN input format is shown in Fig 5.

Input data for all lines except those entering the accumulators were obtained from the Laboratory Test report (Ref 9). The outside diameter of the accumulator entry lines was not available and was estimated from measurements of test rig photographs. An assumed inside diameter was based on the thickness of other tubing used in the experiment.

Fluid properties were altered from hydraulic fluid to those of JP-5 at 60 F. The properties used were:

Viscosity	$= 0.0362 \times 10^{-2} \text{ in}^2/\text{sec}$
Density	$= 0.720 \times 10^{-4} \text{ lb}_f\text{-sec}^2/\text{in}^4$
Bulk Modulus	$= 0.139 \times 10^6 \text{ lb}_f/\text{in}^2$
Vapor Pressure	$= 0.5 \text{ psia}$

Time steps (ΔT) of 0.0005, 0.0004, and 0.000333 sec were used over a simulation interval of 0.4 sec. This interval was chosen to match the time period over which the experimental data were recorded. The various time steps were used, when required, to insure a whole number of time steps over all segments of each of the effective valve area versus time input tables.

The nozzle was modeled using a TYPE 21 two-way valve. The valve closure was input as an effective valve area versus time as discussed

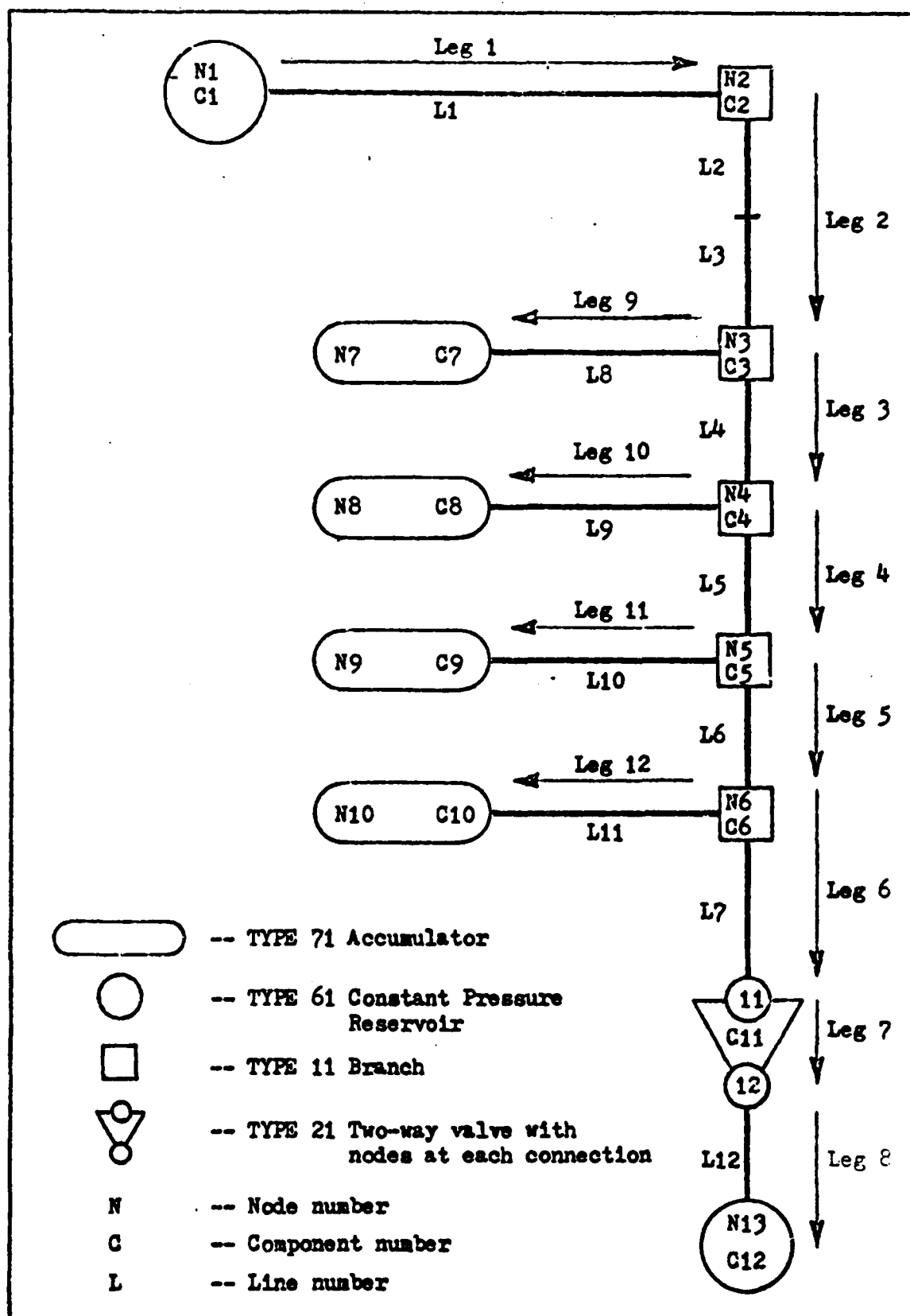


Fig 5. HYTRAN Diagram of Laboratory Test

in Chapter III. This method simplified modifications and the usage of various closure curves.

The full flow area times discharge coefficient was calculated from a flow versus pressure difference across the nozzle curve for the actual nozzle (Ref 16:44). A flow rate of 2018 in³/sec was used in the Laboratory Test. This flowrate corresponds to $\Delta P = 5.1$ psi from the curve for the nozzle. Using the orifice equation,

$$A_{vo} C_d = \frac{Q}{\sqrt{\frac{2 \Delta P}{\rho}}} \quad (2)$$

a steady state value of $A_{vo} C_d = 5.35$ in² was computed.

The steady state flow was exhausted into a TYPE 61 constant pressure reservoir to simulate the fuel return system used in the test. The constant pressure was set at 47.5 psia. This pressure was computed by subtracting the ΔP of the nozzle from the measured steady state flowing pressure of 52.6 psia just upstream of the nozzle (Node 11). This back pressure was consistent with the conditions used to generate the flow curve for the nozzle.

TYPE 71 gas charged accumulators were used to simulate the test accumulators. Volume and precharge pressure was the only definitive data available for the accumulators; therefore, some assumptions were required to complete the models. Entry lines were scaled from photographs to be 6 in long with an outside diameter of 2 in. A wall thickness of 0.125 in was assumed. The volume of each accumulator was given as 5 gal (1155 in³); a usable volume of 1000 in³ was used in the simulation to account for the unknown

internal configuration. Assuming that the gas would be compressed to a maximum of ten percent of its original volume resulted in a maximum fuel volume for each accumulator of 900 in^3 . The total fuel volume for the four accumulators totaled 3600 in^3 , consistent with the actual volume of 3565 in^3 (Ref 17) for the KC-10.

No data other than photographs was available for the Laboratory pumping system. Since the system provided a constant steady state flow and pressure, a TYPE 61 constant pressure reservoir was used to model the pump as a constant pressure source. The pressure of the reservoir was adjusted to provide the test condition steady state flow.

A complete listing of input parameters for the Laboratory Test model is given in Appendix A.

V. KC-135 Computer Model

The KC-135 system was modeled using the technique of Parks (Ref 10) and the HYTRAN component models discussed in Chapter III. The use of a similar model and test conditions provided a cross-check and extension of the previous results. Although Parks used an earlier version of HYTRAN which required recomputation of some input parameters, the basic model configuration was unchanged. A schematic of the system in HYTRAN format is shown in Figures 6 and 7.

Fluid properties were altered from hydraulic fluid to JP-4 at 60 F. The properties used were:

Viscosity	=	$0.136 \times 10^{-2} \text{ in}^2/\text{sec}$
Density	=	$0.77 \times 10^{-4} \text{ lb}_f\text{-sec}^2/\text{in}^4$
Bulk Modulus	=	$0.16 \times 10^{-6} \text{ lb}_f/\text{in}^2$
Vapor Pressure	=	2.0 psia

Time steps (ΔT) of 0.0005, 0.0004, and 0.000333 sec were used over a simulation interval of 1.0 sec. This interval was selected to match Parks' simulation time interval (Ref 10:23).

The various time steps were used, when required, to insure a whole number of time steps over all segments of the valve closure input tables.

Parks simulated the KC-135 system with three pumps on and one pump off (Ref 10:23); this study used the same configuration to facilitate comparisons with Parks. Each pump was modeled as a TYPE 61 reservoir at a constant pressure of 30 psia. Pump check valves

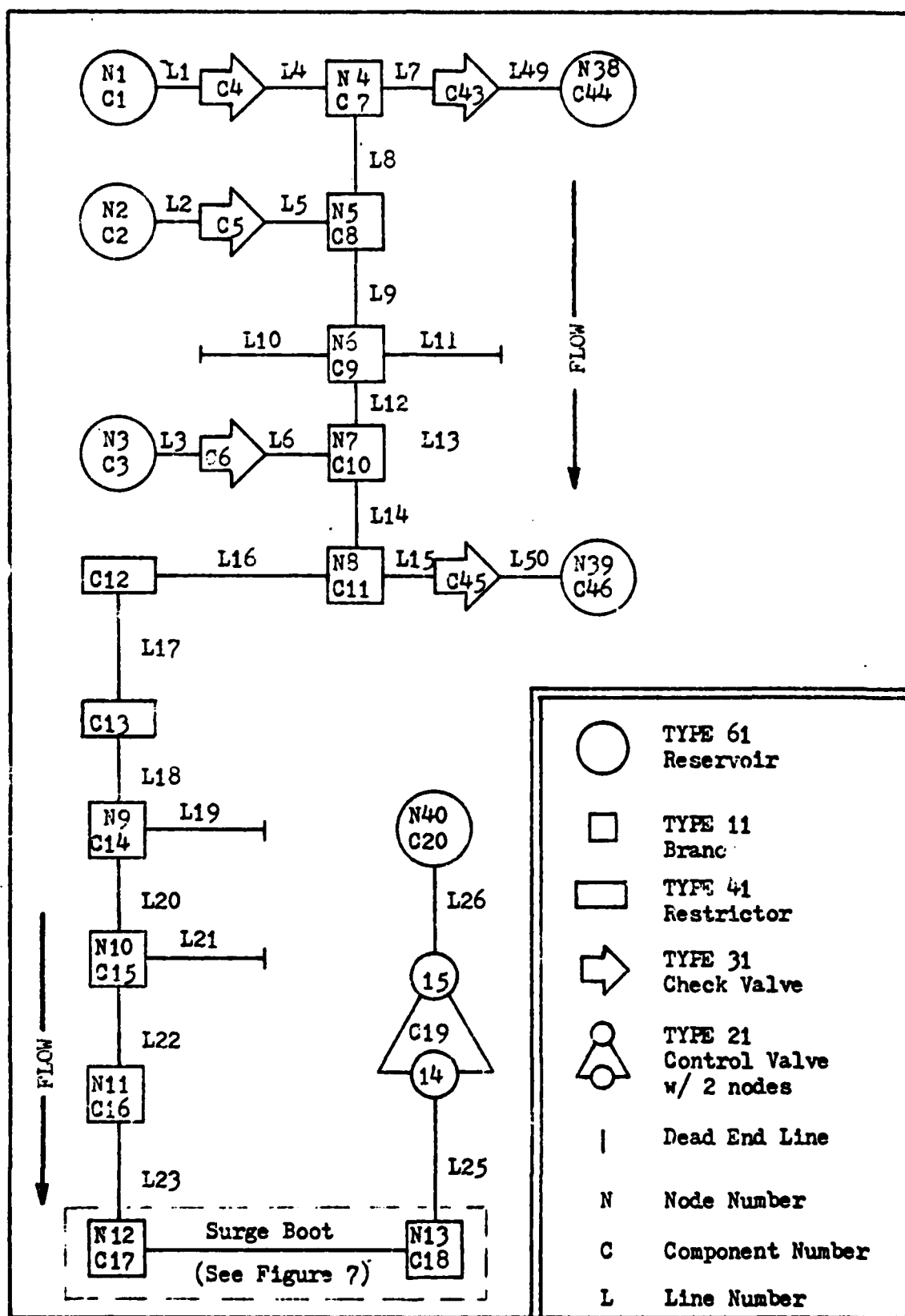


Fig 6. HYTRAN Schematic of KC-135 Refueling System



were simulated as TYPE 31 checkvalves.

The line valve and regulator were modeled as TYPE 41 restrictors; the level control valves as TYPE 31 checkvalves and the fuel tanks as TYPE 61 reservoirs, respectively. The surge boot was modeled as twelve discrete accumulators; the total volume of the twelve units was 1300 in³. Each was modeled as an individual TYPE 71 accumulator with a minimum gas volume equal to ten percent of the initial gas volume.

A TYPE 21 control valve was used to simulate the boom nozzle. The valve closure was input as an effective valve area versus time as discussed in Chapter III.

All dimensions, where applicable, were obtained from the Parks model (Ref 10:70-77).

The steady state flow was exhausted into a TYPE 61 reservoir to simulate the fuel return system. The reservoir pressure was set to 43 psig (Ref 10:36).

A complete listing of input parameters can be found in Appendix B.

VI. Discussion of Valve Closure

General (Ref 18:69)

As discussed in Chapter I, a nozzle valve closure causes a pressure surge to propagate through a refueling system. The direct relationship of pressure surge magnitude to valve closure can be illustrated by considering the deceleration of an incompressible fluid flowing frictionlessly in a rigid pipe of uniform area A with a velocity V . The pipe has a length L , an inlet pressure P_1 , and a pressure P_2 at L . Assume, at length L , there is a valve which can reduce the velocity at L to $V - \Delta V$. The mass rate of flow for a pressure wave traveling at sonic velocity c , is $\dot{m} = \rho A c$. From the impulse-momentum equation, for this application

$$(\rho A c) (V - \Delta V - V) = P_2 A - P_1 A$$

or the increase in pressure is given by

$$\Delta P = -\rho c \Delta V \quad (3)$$

thus a change in velocity is directly proportional to a change in pressure. The time for a pressure wave to travel the length of pipe L and return is $t = 2L/c$. If the time of closure T_c is less than or equal to t , the approximate pressure rise is given by

$$\Delta P \approx \frac{-2\rho V L}{T_c} \quad (4)$$

thus a decrease in closure time will cause an increase in pressure rise.

Since the reduction in flow velocity is determined by the valve closure, it is apparent that the closure curve is a critical input to the simulation of aircraft refueling system response subsequent to nozzle valve closures.

Valve Closures from Previous Research

Parks (Ref 10:19-20) used a valve closure relationship, hereafter referred to as curve "A", described by Eq (1) where

$$K(t) = \left[1 - \frac{t}{T_c} \right]^3 \quad (5)$$

The closure curve corresponding to this relationship is shown in Fig 8 and is characterized by a rapid reduction of the valve opening initially, followed by an almost asymptotic reduction to full closure. This slower area reduction during the final stage of the closure curve is referred to as "valve snubbing". Parks justified his usage of the curve on the basis of a comparison to experimental data.

Typical curves used by Kinzig (Ref 8:25) in his study of receiver pressure surges are shown in Fig 9.

For use in this study, the curves were reduced to an analytical expression of the following form:

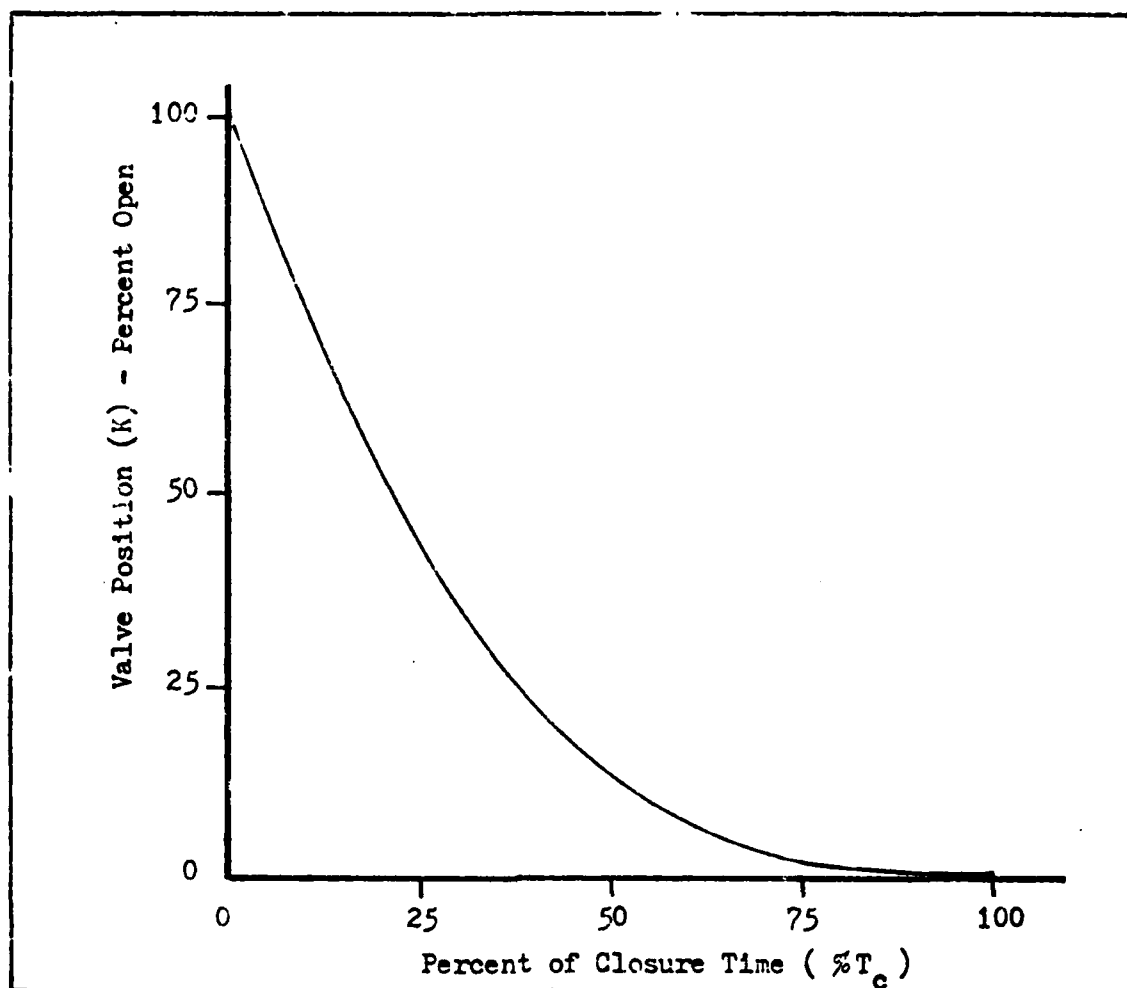


Fig 8. Closure Curve 'A'

$$K(t) = \left[1 - (t / T_c) \right] \left[a - b(1 - (t / T_c))^n \right] \quad (6)$$

where for

curve "B"

curve "C"

$$a = 1.45$$

$$a = 2.60$$

$$b = 1.00$$

$$b = 2.00$$

$$n = 1.30$$

$$n = 1.25$$

Both curves represent a relatively gentle closure characteristic of receiver shut-off valves.

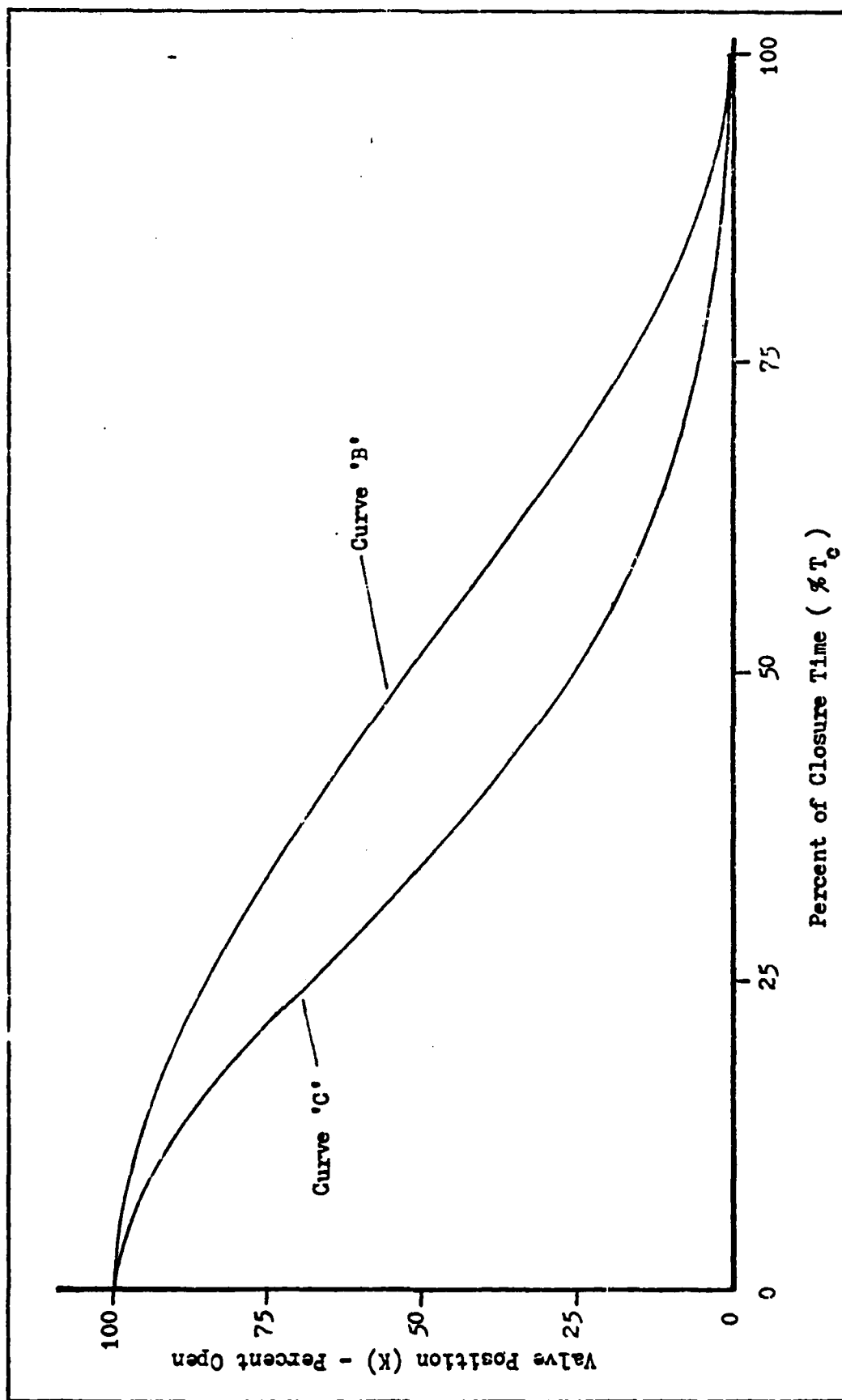


Fig 9. Closure Curves 'B' and 'C'

Nozzle Valve Closure

Valve poppet position versus time from the Laboratory Test (Ref 9:24) and drawings of the valve mechanism were used to plot an actual closure curve for a refueling nozzle. The curve, in nondimensional form, is shown in Fig 10. From fully open, the valve closes at an increasing rate as it accelerates to a constant closure rate. The closure rate decreases during the final stage of area reduction, approximately the last ten percent. This snubbed closure is expected since the valve poppet incorporates a damper that is designed to slow the closure during the last stage of poppet travel.

Nozzle / Receptacle Combination

While the previous curves, Figs 8 and 9, may be typical for normal disconnects, abnormal disconnects at high separation rates are likely to result in more severe closures. Before establishing the effect of high separation rates on the valve closure curve, the interaction of the combined nozzle and receptacle configuration was examined.

Shown in Fig 11 are (a) the receptacle sleeve valve, (b) the nozzle valve, (c) the nozzle and receptacle in connected position. The spring-loaded nozzle poppet valve is opened mechanically on contact with the fixed pedestal in the receiver aircraft receptacle in conjunction with the opening of the sliding sleeve valve in the receptacle by the nozzle tip.

During quick disconnects, the damper in the poppet can slow the poppet closure such that the poppet and pedestal do not remain in contact during separation of the nozzle and receptacle. Therefore, nozzle valve closure is independent of separation rate. Conversely,

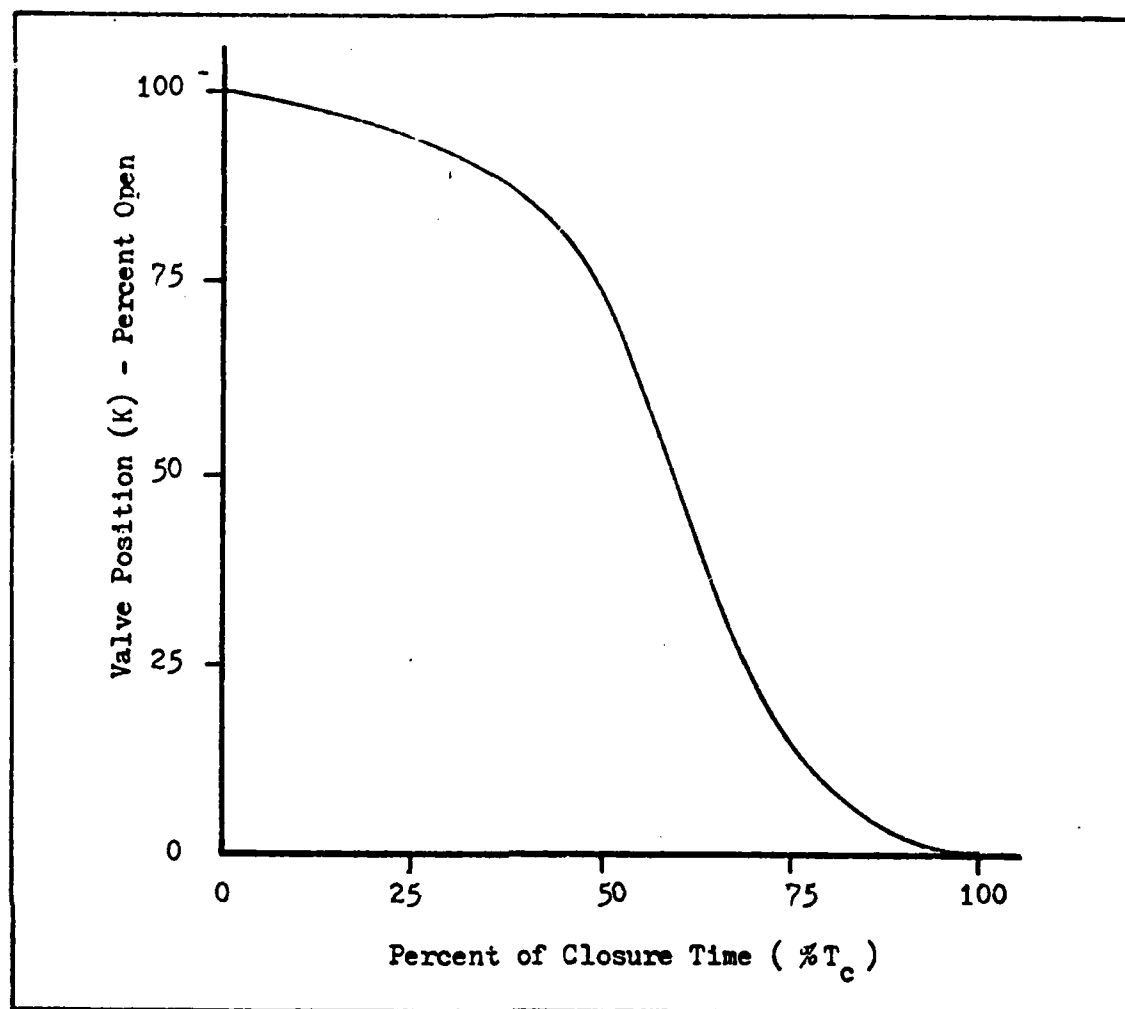


Fig 10. Nozzle Closure Curve Computed from the Laboratory Test Measurements

since the receptacle sleeve valve is spring-loaded and undamped, its closure rate is determined by the rate of separation. A typical closure curve for the receptacle sleeve valve, hereafter referred to as curve "D", is shown in Fig 12. The closure accelerates to a constant rate with no appreciable snubbing.

To reach the fully closed position, the required travel for the receptacle sleeve valve is 1.0 in (Ref 9:7) and for the nozzle valve poppet a minimum of 1.15 in is required to reach the closed position (Ref 16:50). Therefore, when the nozzle and receptacle have separated

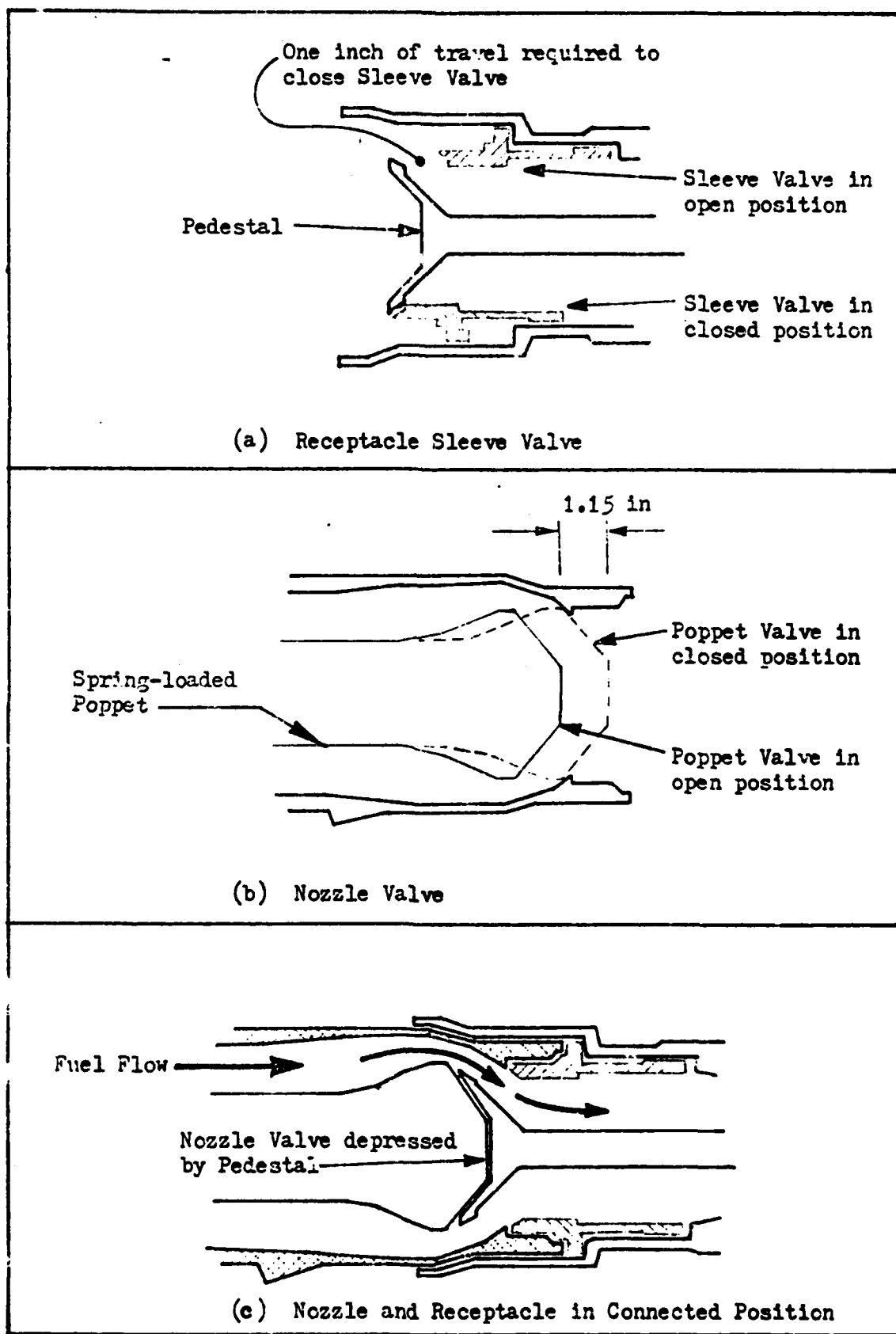


Fig 11. Nozzle / Receptacle Details

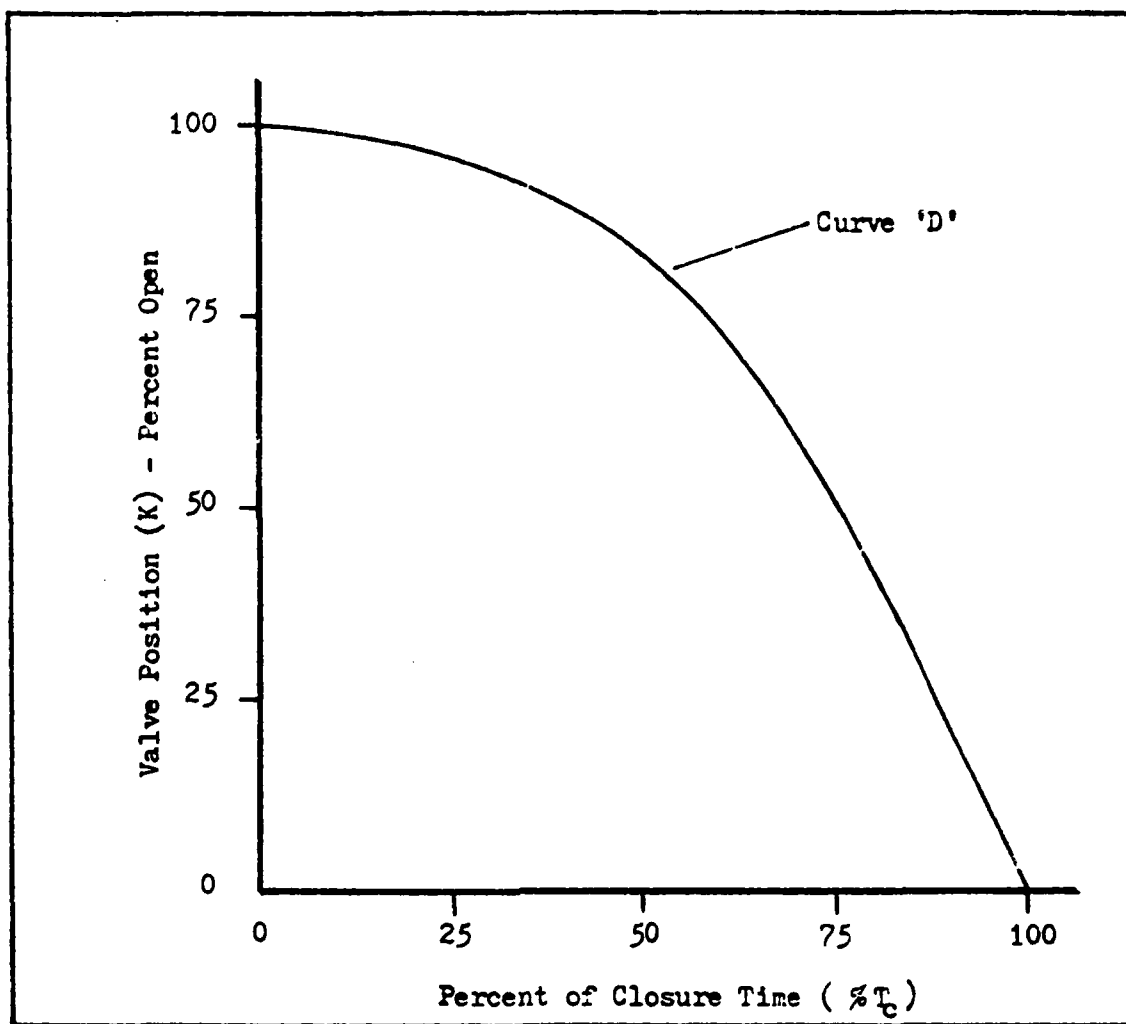


Fig 12. Receptacle Closure Curve

1.0 in, Fig 13, the receptacle sleeve valve is closed. At the 1.0 in separation point the nozzle valve has reached a maximum of approximately 90% of full closure if the poppet and pedestal have remained in contact, less if the poppet travel lags the relative separation of the two components. The penetration of the nozzle into the receptacle is sufficient for the two components to be essentially sealed together at this position of separation. The actual closure for quick disconnects is determined by the receptacle sleeve valve and can initiate a surge that propagates into the tanker

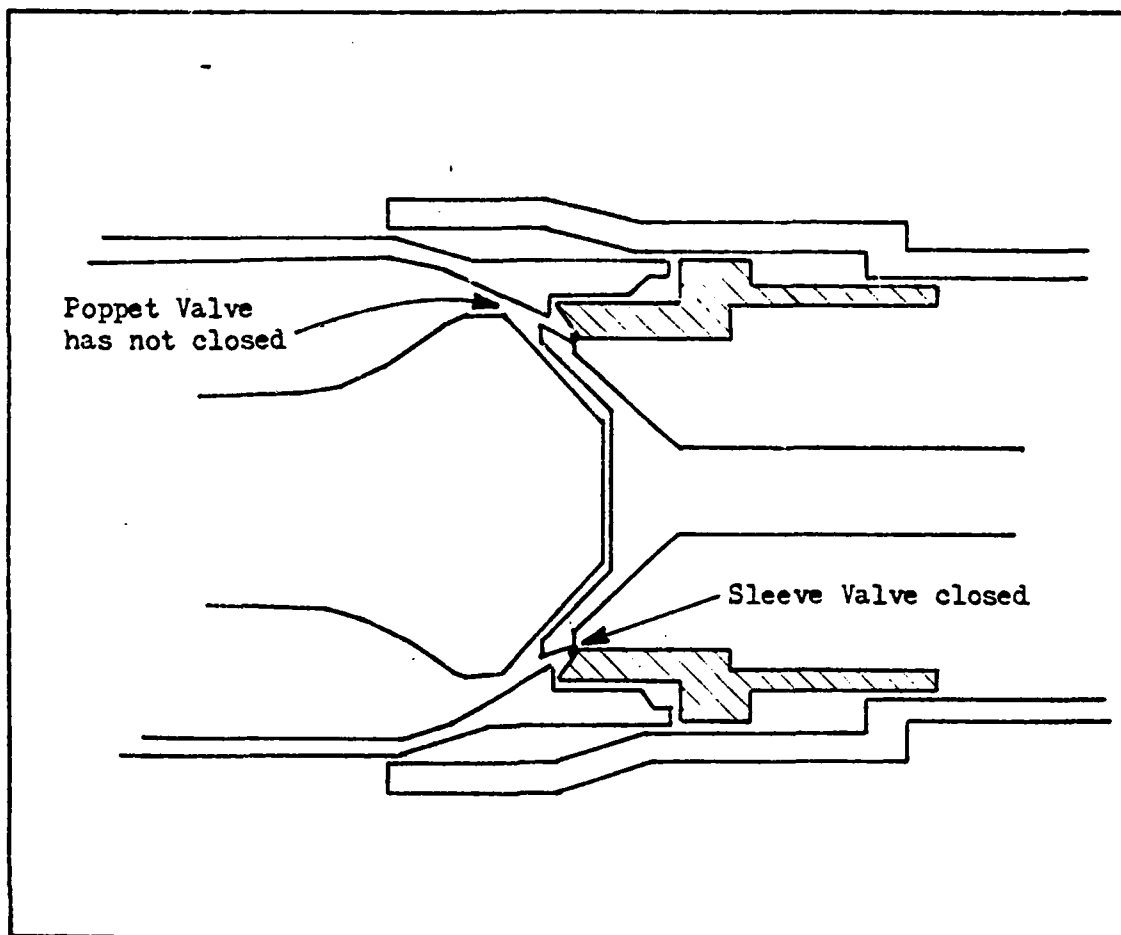


Fig 13. Nozzle and Receptacle at the One Inch Separation Point

system even though the nozzle valve is still open. This supposition is supported by pressure measurements obtained during the Laboratory Test (Ref 9:10) which indicate peak pressure occurs coincident with the 1.0 in separation point for separation rates of 5-10 ft/sec.

Actual Valve Closure Approximation

With a relationship between separation rate and valve closure established, a reasonable approximation to the minimum effective closure time can be established. The maximum disconnect speed is 6.5 ft/sec (Ref 9:7). Assuming instantaneous acceleration to this

rate, the time required for the nozzle and the receptacle to separate one inch is 0.013 sec. The separation rates measured during the Laboratory Test (Ref 9:10) indicate a minimum time for one inch separation of approximately 0.02 sec at a separation rate of 6 ft/sec.

Based on the previous discussion in this chapter the closure curve for quick disconnects is theorized to be basically the receptacle closure curve as shown in Fig 12 (curve D). Typically the closure rate would increase to a constant rate with no snubbing. The overall effect of the combined nozzle/receptacle closure is to eliminate the snubbing portion of the closure action of the receptacle.

VII. Results and Discussion

HYTRAN Verification

A basic waterhammer problem with known results was solved using HYTRAN. Streeter and Lai (Ref 3) obtained an analytical solution for the configuration shown in Fig 14 and verified it by experiment. After establishing a steady state flowrate in the line, the valve was closed according to Eq (1) where

$$K(t) = \left[1 - \frac{t}{T_c} \right]^2 \quad (7)$$

The HYTRAN format configuration is shown in Fig 15. A closure time of 0.09 sec and a steady state flowrate of 5 in³/sec was used. The working fluid was water at 60 F.

Comparison of HYTRAN and Streeter results (Fig 16), indicate excellent agreement in phasing, curve shape, and magnitude of the first peak. Subsequent peaks of the HYTRAN solution exhibit more attenuation than the Streeter solution due to the inclusion of dynamic friction effects in HYTRAN that Streeter neglected.

Laboratory Test Simulation

Comparison with Experimental Results. The steady state conditions prior to the separation of nozzle and receptacle, i.e., valve closure, were as described in Chapter IV. The accumulator precharge was set at 90 psia (Ref 9:10, 24). The valve was closed

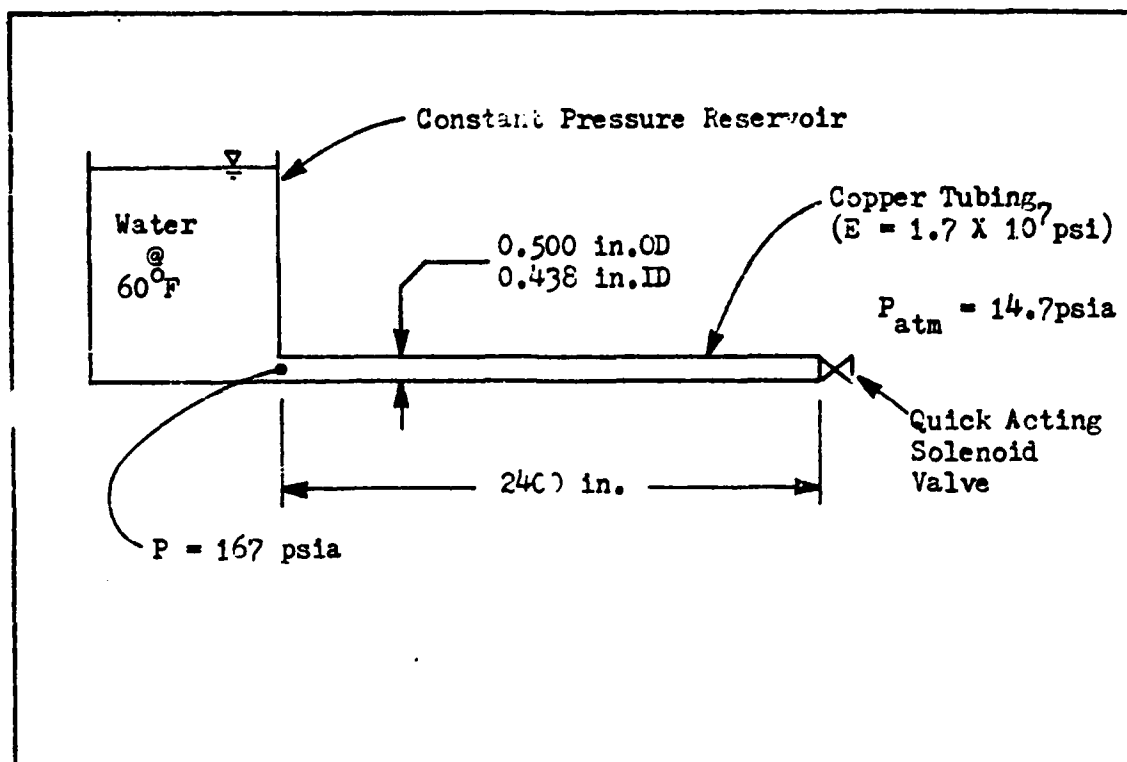


Fig 14. Streeter's Experimental Configuration

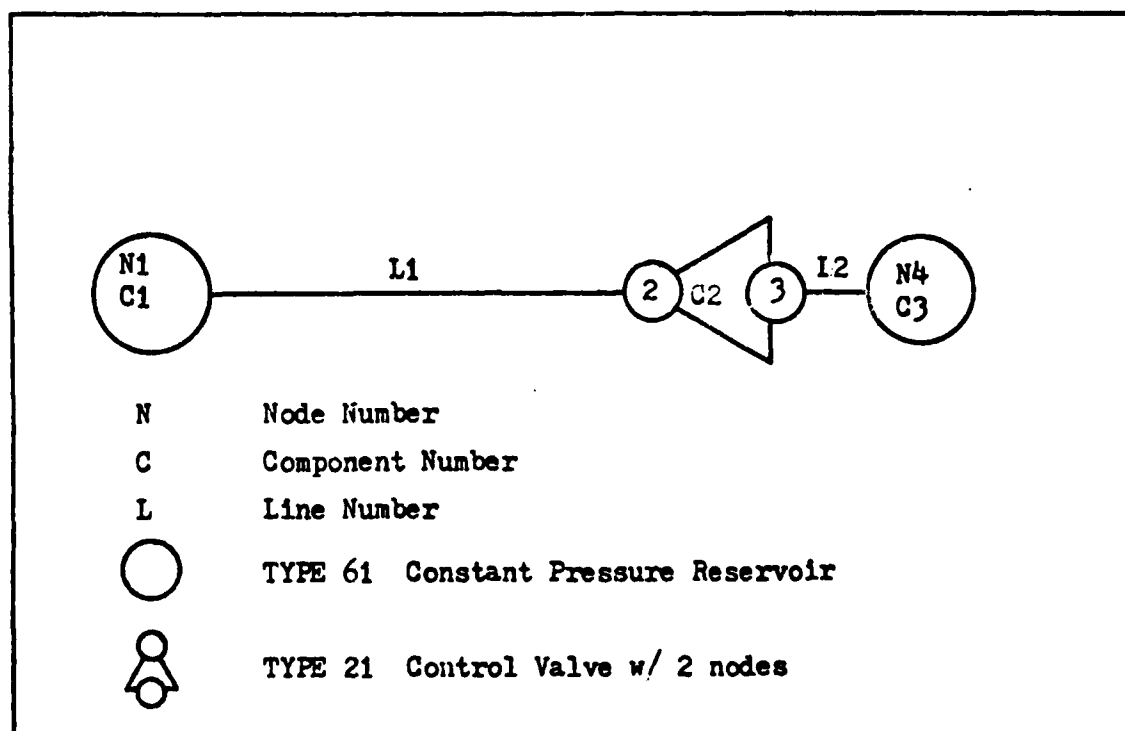


Fig 15. HYTRAN Diagram for Streeter's Experiment

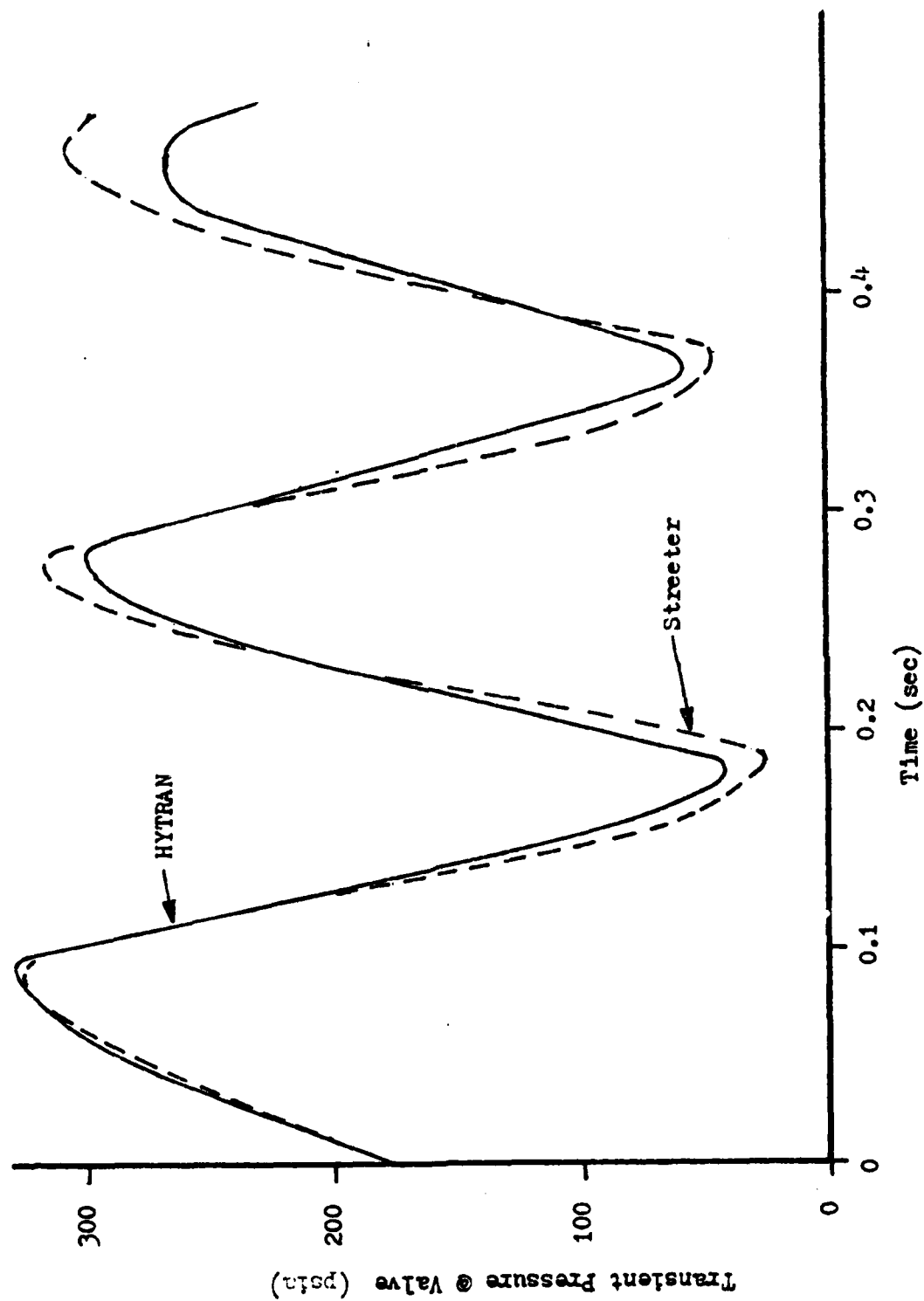


Fig 16. Comparison of Transient Response Due to Valve Closure:
HYTRAN versus Streeter

according to Eq (1) with $K(t)$ from curve D of Chapter VI, Fig 12.

Closure time was 0.03 sec. An oscillograph trace of the experimental response for the conditions of the simulation was obtained from the test report (Ref 9:24).

Preliminary results exhibited reflected peaks not shown in the experimental data that indicated the laboratory pumping system did not reflect pressure waves precisely as a constant pressure reservoir. As a result, the laboratory pumping system was modeled as a constant pressure reservoir connected to a long line, the length of which prevented the return of the reflected wave during the time interval of interest (0.0 to 0.4 sec). The line from source to test section (L1) was lengthened to accomplish this modification of the model. The pressure of the constant pressure source was adjusted to provide the desired steady state flow conditions.

Comparison of experimental and simulated transient response to the valve closure is shown in Fig 17. The first peak in both cases corresponds to the complete closure of the valve. Excellent agreement is seen in comparison of the five percent settling times. The settling time is the time required for the response to decrease to a specified percentage of its final value. The initial and final values are virtually the same for both.

Comparison of peak magnitudes and phasing are less favorable. The computer simulation underpredicts the magnitude of the first peak and overpredicts the subsequent peaks. There are twice as many peaks in the experimental results as in the simulation; a result, perhaps, of the accumulator model limitations.

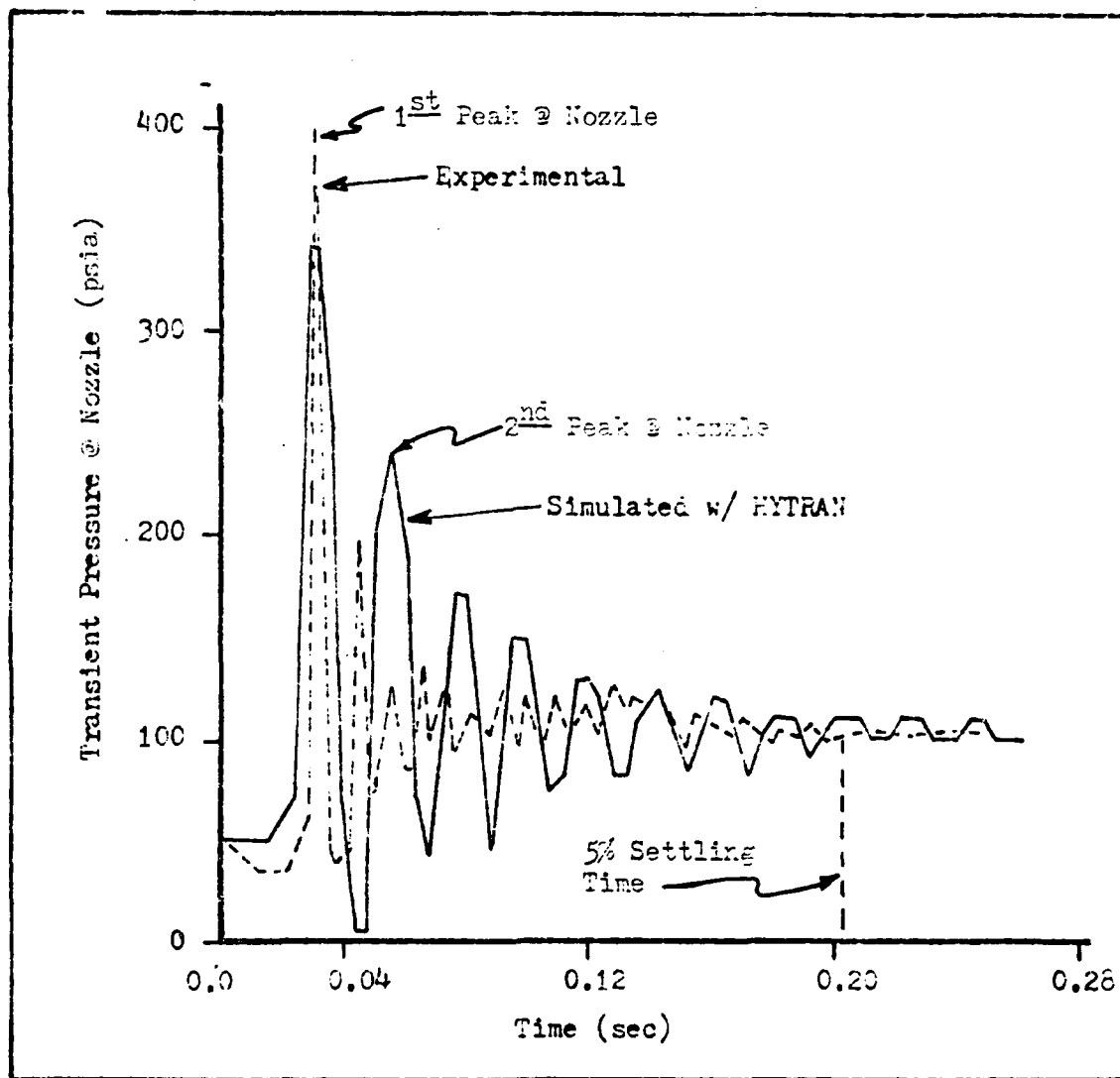


Fig 17. Comparison of Experimental and Simulated Transient Response at Nozzle to Valve Closure for the Laboratory Test Model

Accumulator Volume and Precharge. Four accumulator volume configurations with four different precharge settings per configuration were investigated. All simulation conditions with the exception of accumulator parameters were outlined in the previous section. Shown in Figs 18-21 are the maximum transient pressures versus precharge pressures for each volume configuration. Refer to Fig 3 for the relative locations of the boom inlet and nozzle.

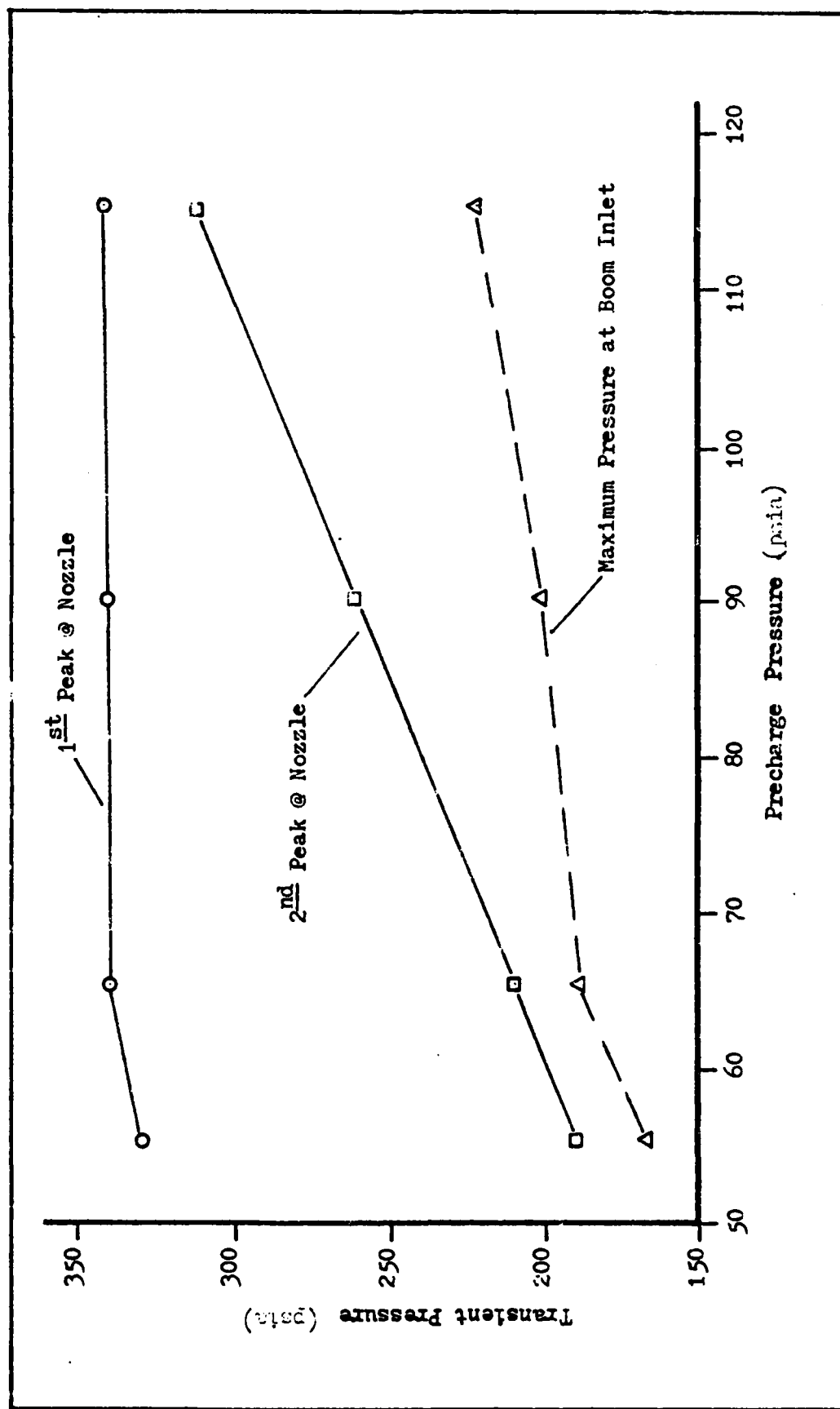


Fig 18. Effect of Accumulator Precharge on Maximum Transient Pressure for Accumulator Volume of 1000 cubic inches for the Laboratory Test Model

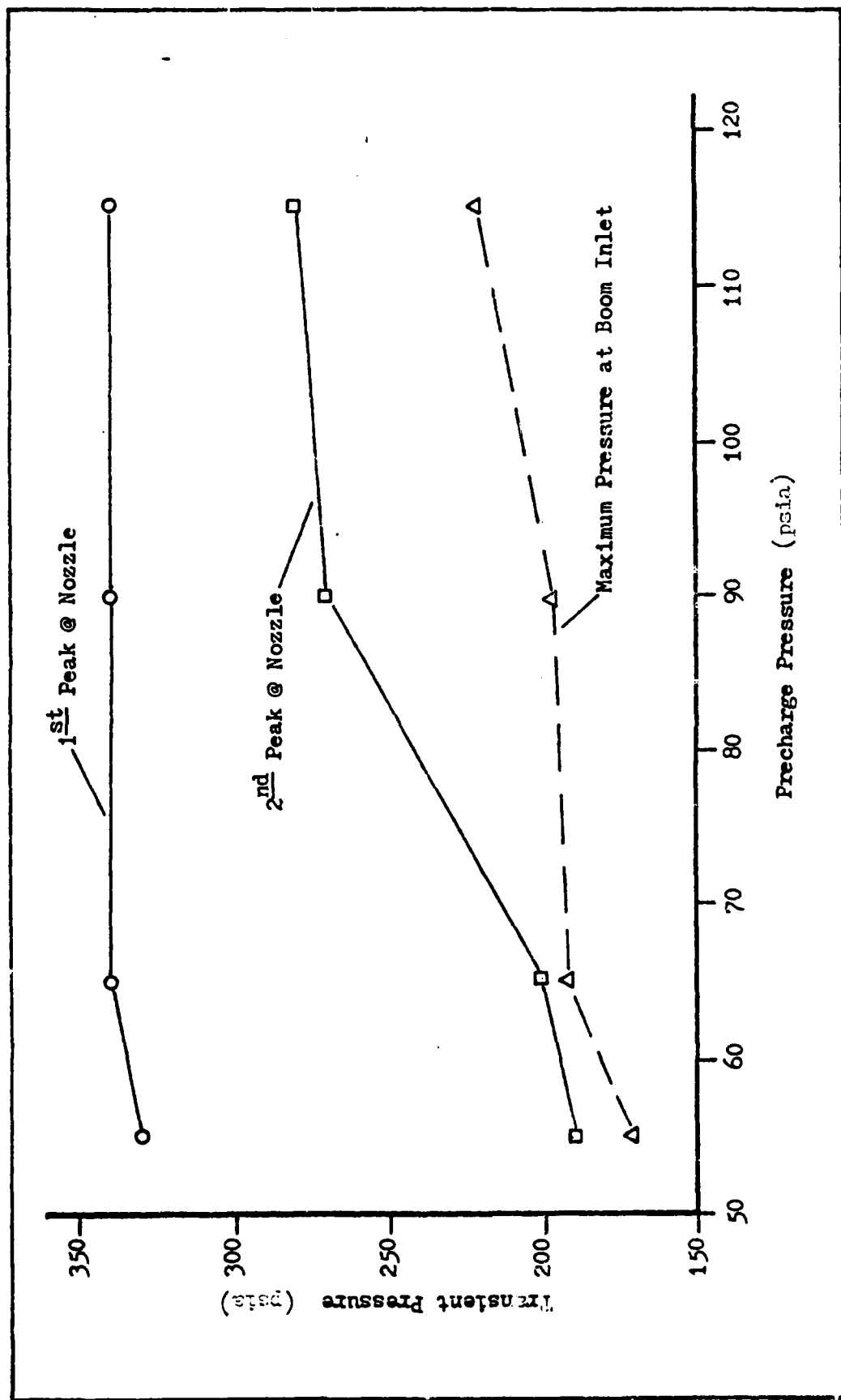


Fig 19. Effect of Accumulator Precharge on Maximum Transient Pressure for Accumulator Volume of 660 cubic inches for the Laboratory Test Model

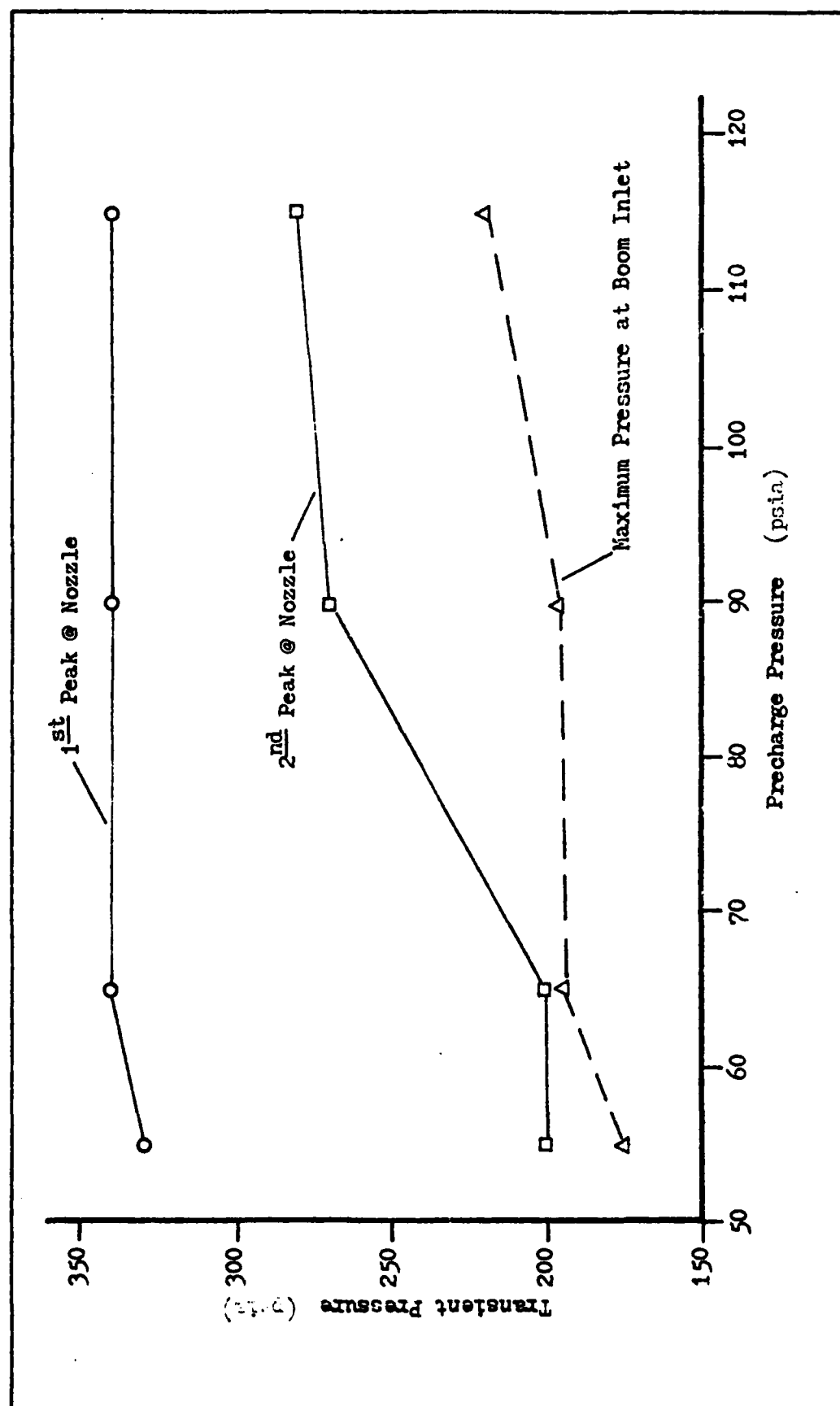


Fig 20. Effect of Accumulator Precharge on Maximum Transient Pressure for Accumulator Volume of 500 cubic inches for the Laboratory Test Model

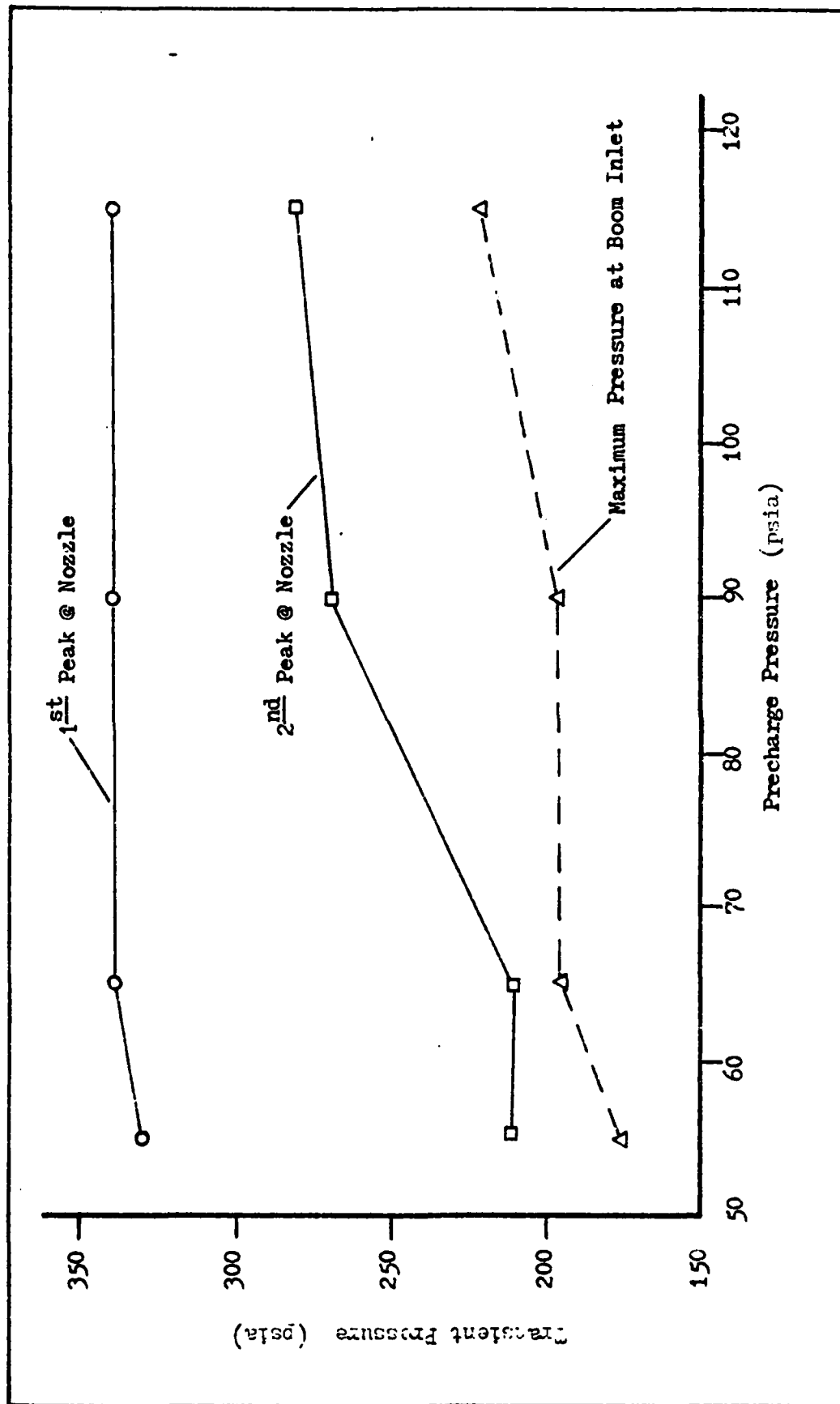


Fig 21. Effect of Accumulator Precharge on Maximum Transient Pressure for Accumulator Volume of 250 cubic inches for the Laboratory Test Model

Volume changes had a negligible effect on the maximum transient pressure at either the nozzle or boom inlet. A small change in the second peak pressures at low precharge as the volumes are decreased is due to the accumulators reaching their fluid capacity limits. The dominant influence was precharge pressure, increasing precharge pressure resulted in higher peak pressures at both nozzle and boom inlet. Note that the first peak is unaffected by changes in either volume or precharge. This is due to the accumulator time response lag. The initial impulse is not attenuated as readily as surges that build up over a longer period of time since a finite period of time is required for flow into the accumulators to begin.

Accumulator Entry Line Diameter. Simulations with accumulator entry line diameters from 1.25 to 2.75 in were run to observe the general effects of diameter variation. Also, since the actual accumulator entry line diameters for the Laboratory Test were not known, a verification of the choice of 1.75 in was desired. All simulation conditions except for entry line diameter were as outlined in the previous section. The basis for comparison was the maximum transient pressure, five percent settling time, and the number of peaks to settling time.

Changes in diameter did not affect the maximum pressure at the boom nozzle. As in the previous section, this is attributed to the accumulator response time. The maximum pressure at the boom inlet is affected because during the time required for the pressure wave to reach the boom inlet, flow into the accumulators has begun. The increase in pressure magnitude at the boom inlet for a decrease in the diameter is shown in Fig 22. This increase is expected due to the

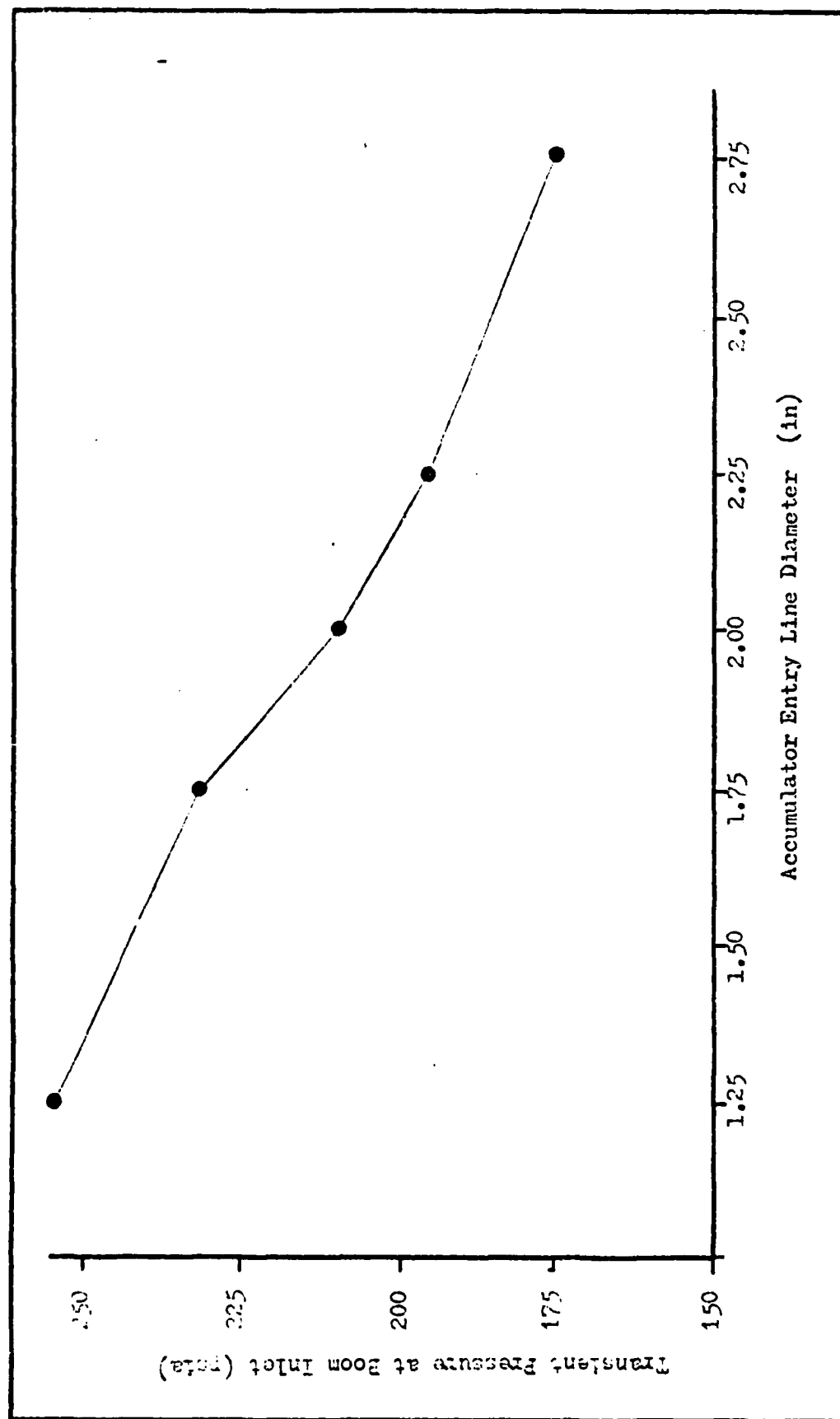


Fig. 22. Effect of Variation of Accumulator Entry Line Diameter on Maximum Transient Pressure at Boom Inlet for the Laboratory Test Model

additional resistance to flow in smaller pipes, thus negating some of the capability of the accumulator to function as an energy absorption component.

Settling time and number of peaks to settling time were favorably affected by decreasing the line diameter, Fig 23 and 24. Settling time was reduced by 20 to 50 percent for the range of diameters. The number of peaks from initiation to settling time was reduced 50 to 70 percent. In this case the increase in flow restriction to the accumulator resulted in increased damping and less pulsation added to the system due to the release of energy stored in the accumulators.

Two runs were made with longer than standard (14 in longer) accumulator entry lines. The results for the longer lines exhibited the same response trends as decreasing the diameter, see Fig 23.

Examination of settling times at the nozzle supports the choice of the 1.75 in entry line for the Laboratory Test simulation. The settling time of approximately 0.02 sec agrees with the measured response (Fig 17).

Closure Curve Study

Laboratory Test Model. Simulations were run using each of six closure curves, shown in Fig 25, to examine the effects of curve shape. Steady state conditions were as described in Chapter IV with an accumulator precharge of 90 psia. Valve closure was according to Eq (1) with $K(t)$ given by the closure curves. Time of closure was 0.03 sec.

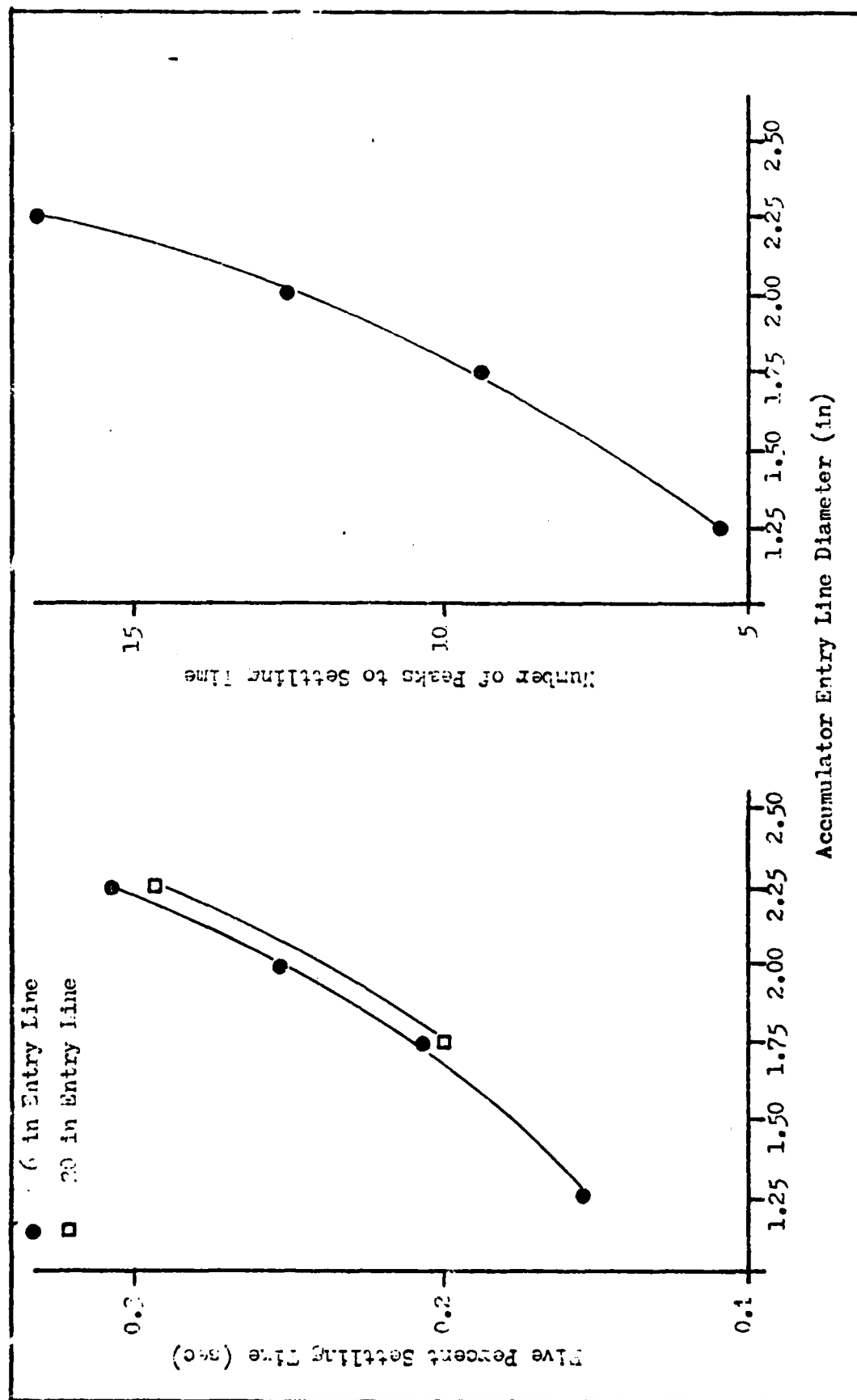


Fig 23. Effect of Variation of Accumulator Entry Line Diameter on Transient Response at Nominal for the Laboratory Test Model

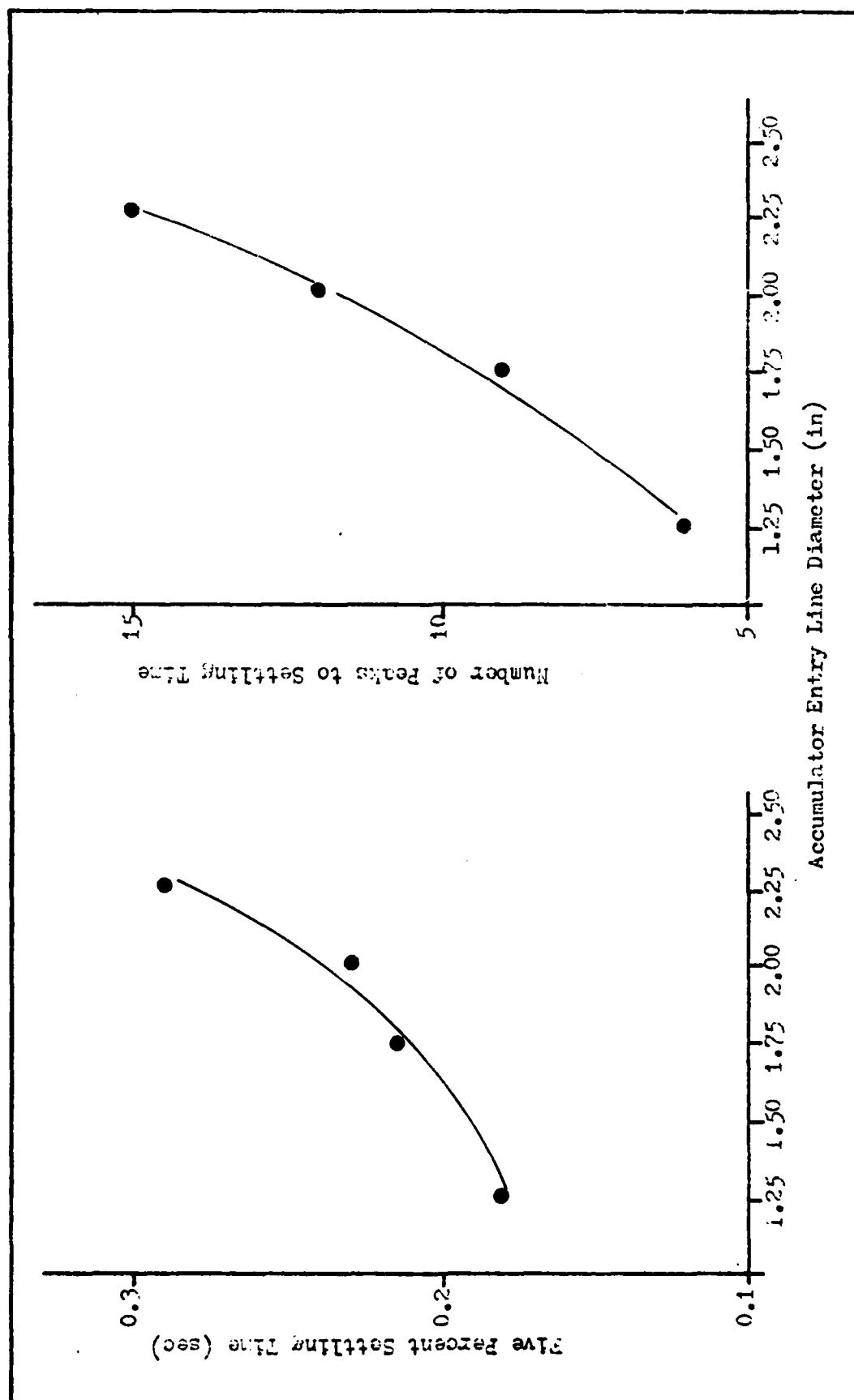


Fig 24. Effect of Variation of Accumulator Entry Line Diameter on Transient Response at Boom Inlet for the Laboratory Test Model

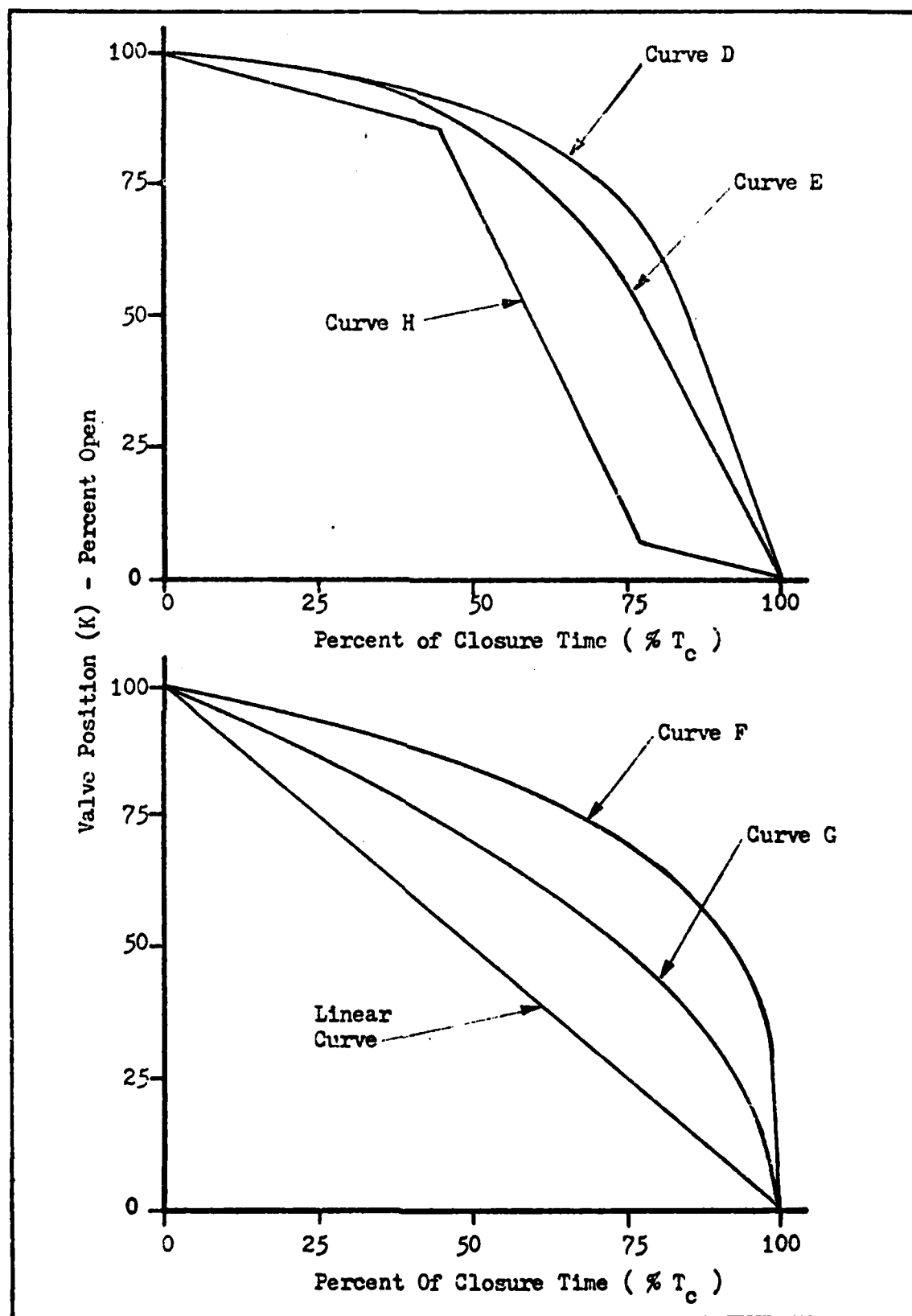


Fig 25. Closure Curves for Laboratory Test Model Closure Curve Study

TABLE I

Effect of Curve Type on Transient Pressure and
Settling Time for the Laboratory Test Model

<u>Maximum Transient Pressure (psia)</u>			
<u>Curve</u>	<u>Nozzle</u>	<u>Boom Inlet</u>	<u>Settling Time (sec)</u>
Linear	310	197	.243
D	340	210	.246
E	340	190	.232
F	340	206	.247
G	340	220	.247
H	260	122	.242

Results for the six curves are shown in Table I. Curves D, E, F, and G, produced similar results, not unexpected since the general curve shape is the same for all four (Refer to Fig 25). Maximum transient pressure at the nozzle and boom inlet for those four curves are the same except for curve E. Curve E has a less severe closure which results in a slightly lower pressure at the boom inlet; settling time is also shorter, again a reflection of the less severe closure.

The linear closure resulted in a reduction of maximum transient pressure at both the nozzle and boom inlet; settling time was somewhat less than curves D, F, and G but greater than curve E. The reduction in pressure was due to the constant rate of closure without changes in slope.

Curve H yielded the lowest maximum transient pressure of the group. Settling time was of the same order as the linear case. The

reduction in pressure peaks was due to the "snubbing" effect during the final stage of closure. This effect is examined further in the next section.

In Chapter VI curve D was selected as the representative curve shape for a quick disconnect; the results of this section indicate a similar response to curves D, E, F, and G; therefore, a general analytical relationship applicable to all and hence, a quick disconnect can be given as

$$K(t) = \left[1 - \frac{t}{T_c} \right]^m \quad (8)$$

where $m < 1$

For the quick disconnect case, the linear and snubbed curves are not applicable because of the lower maximum pressures generated.

Valve Snubbing Effects. Valve snubbing was discussed in Chapter VI and is defined for this study as the reduction in the rate of valve area decrease during the final ten percent of valve area reduction. The effect of valve snubbing was investigated by running the Laboratory Test simulation with snubbed and unsnubbed curves. Time of closure and steady state conditions were the same as the previous runs. The valve closure was given by Eq (1) and $K(t)$ from the modified closure curve H as shown in Fig 26a. Four closures were evaluated; an unsnubbed curve, and three with snubbing of 3.7%, 7.5%, and 11%, respectively. The 7.5% snubbed curve is the basic curve H. The effect of the increased percentages of snubbing is shown in Fig 26b. Increasing the snubbing percentage decreases

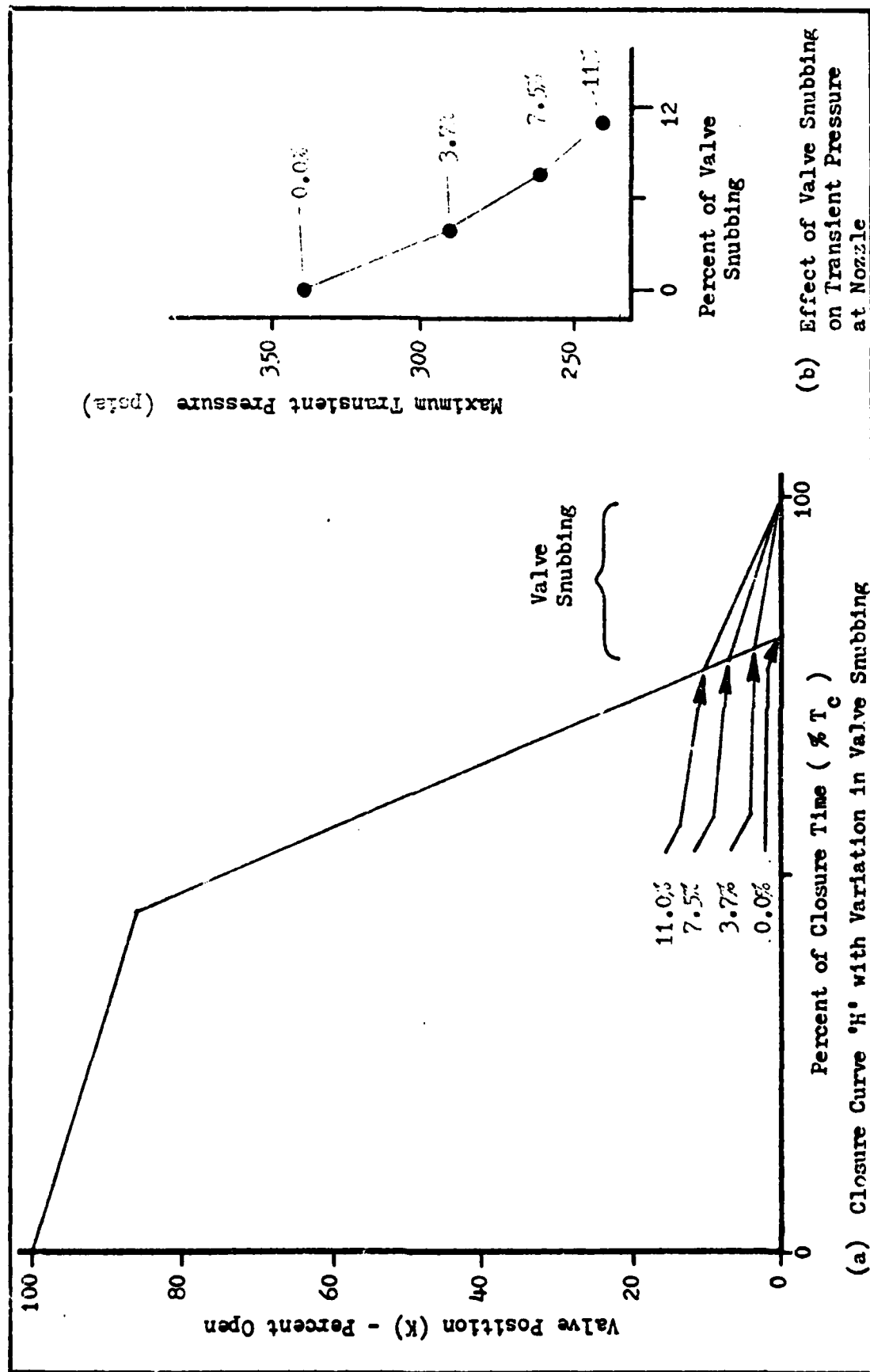


Fig 26. Effect of Variation in Valve Snubbing on Transient Pressure

the maximum transient pressure.

KC-135 Model. The effect of closure curve shape and time of closure on the KC-135 system described in Chapter V was examined by varying the time of closure for several curves. The valve closure was given by Eq (1) with $K(t)$ from closure curves A, B, C, D, E, F, G, H, and the linear curve. Time of closure was varied from 0.01 and 0.05 sec to encompass the minimum closure time discussed in Chapter VI and the minimum used by Parks (Ref 10:53). The surge boot precharge was 50 psig and all other conditions were as described in Chapter V.

The magnitude of the first peak at the nozzle was chosen as the basis for comparison of the results; the peak numbering is illustrated in Fig 27. The response of the system subsequent to the first peak was basically unaffected by changes in closure time curve shape. Normally the most important peak for design purposes is the first peak after valve closure, unless subsequent peaks are of greater magnitude. The latter case is possible in complex systems like the KC-135 where different concentrations of reflected pressure waves coalesce to form peaks greater than previous peaks. The transient response in Fig 27; closure curve A, $T_c = 0.05$ sec; is an example of such a case.

Shown in Fig 28 is the pressure of the first peak at the nozzle versus closure time for each curve.

For all curves there is an increasing pressure with decreasing closure time relationship. Curves D, F, and G are least affected by changes in closure time. These curves have closure rates that are high during the latter stages of closure thus most of the closure takes place over a short percentage of any closure time. The slope of the curve does not reverse, i.e., go from positive to negative or vice versa,

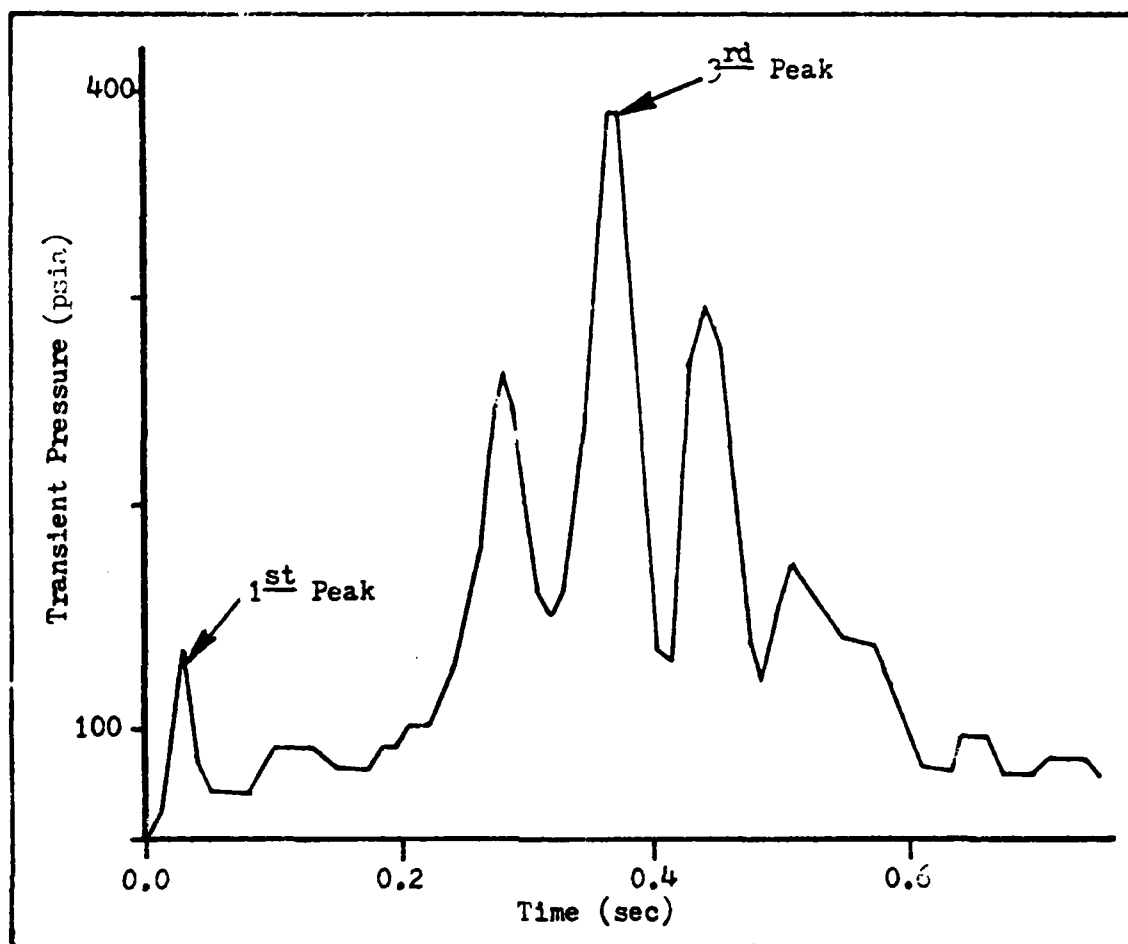


Fig 27. Numbering of Transient Pressure Peaks for a Typical Response to Valve Closure for the KC-135 Model using curve A and $T_c = 0.05$ sec.

over the closure interval.

Curves B, C, and H have similar responses to decreasing closure time. All of these curves appear to be very dependent on changes to closure time, especially below 0.03 seconds. The extreme effects of closure time variation is shown by the approximately 200% increase in pressure for a reduction in closure time from 0.03 to 0.01 sec.

The linear curve exhibits a relationship of pressure increase versus closure time that is similar to curves H, B, and C except that the increase is not quite so dramatic. This is a consequence of the constant slope with no snubbing.

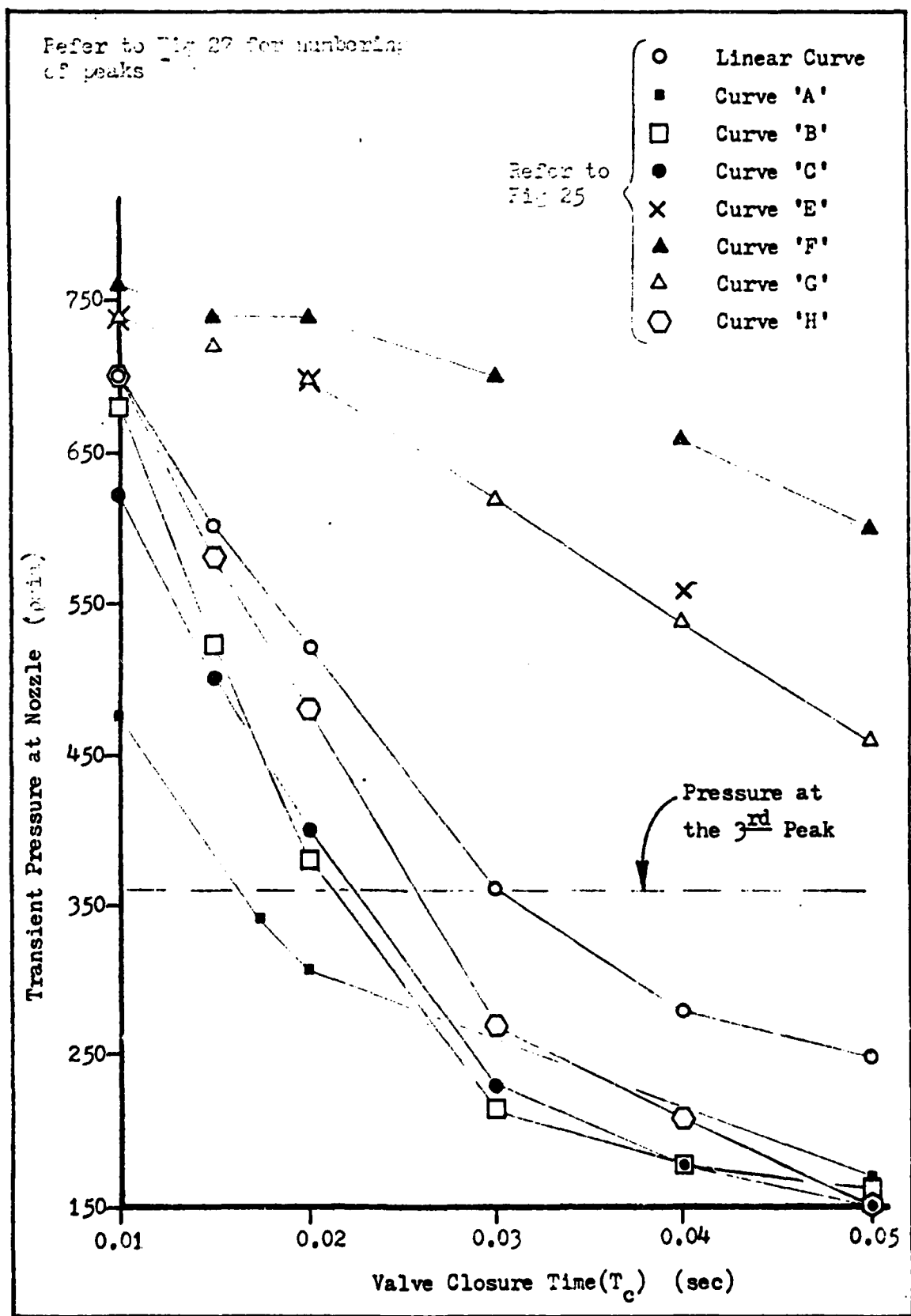


Fig 23. Effect of Closure Time Variation on 1st Peak Transient Pressure for the KC-135 Model

Using curve A results in a modest change over the range of closure times. This is to be expected since the curve is heavily snubbed and has no reversal of slope. Note that neither curve A nor curves E, F, and G have slope reversals; the effect of changes in closure time is similar for all. The lower values of curve A are due to snubbing effects.

Overall, for the KC-135, the effect of reducing closure time is to shift the dominant pressure peak from the third peak to the first peak for all curves except E, F, and G. First peak pressure for all cases increases with reduction in closure time.

The consistency of the third peak value can be attributed to the system configuration, since for complex systems the compression wave initiated by the valve closure does not bring the flow to rest as it might in simple systems; therefore the time required for flow stoppage is characteristic of the system. As shown by Eq (3), a pressure peak will occur at the point of complete flow stoppage and dependent on the system, it is likely to maintain a consistent magnitude without regard to closure time. For the KC-135 model, complete flow stoppage occurs coincident with the timing of the third peak. This is illustrated in Fig 29 for a typical response upstream of the nozzle.

Parks (Ref 10:53-55) investigated pressure versus closure time for the interval 0.05 and 0.15 sec and found little change. For the closure used by Parks; curve A, $T_c = 0.05$ sec; the results previously discussed indicate that the third peak would be predominant and consistently the same magnitude. Therefore, the results Parks obtained are consistent with the results of this study.

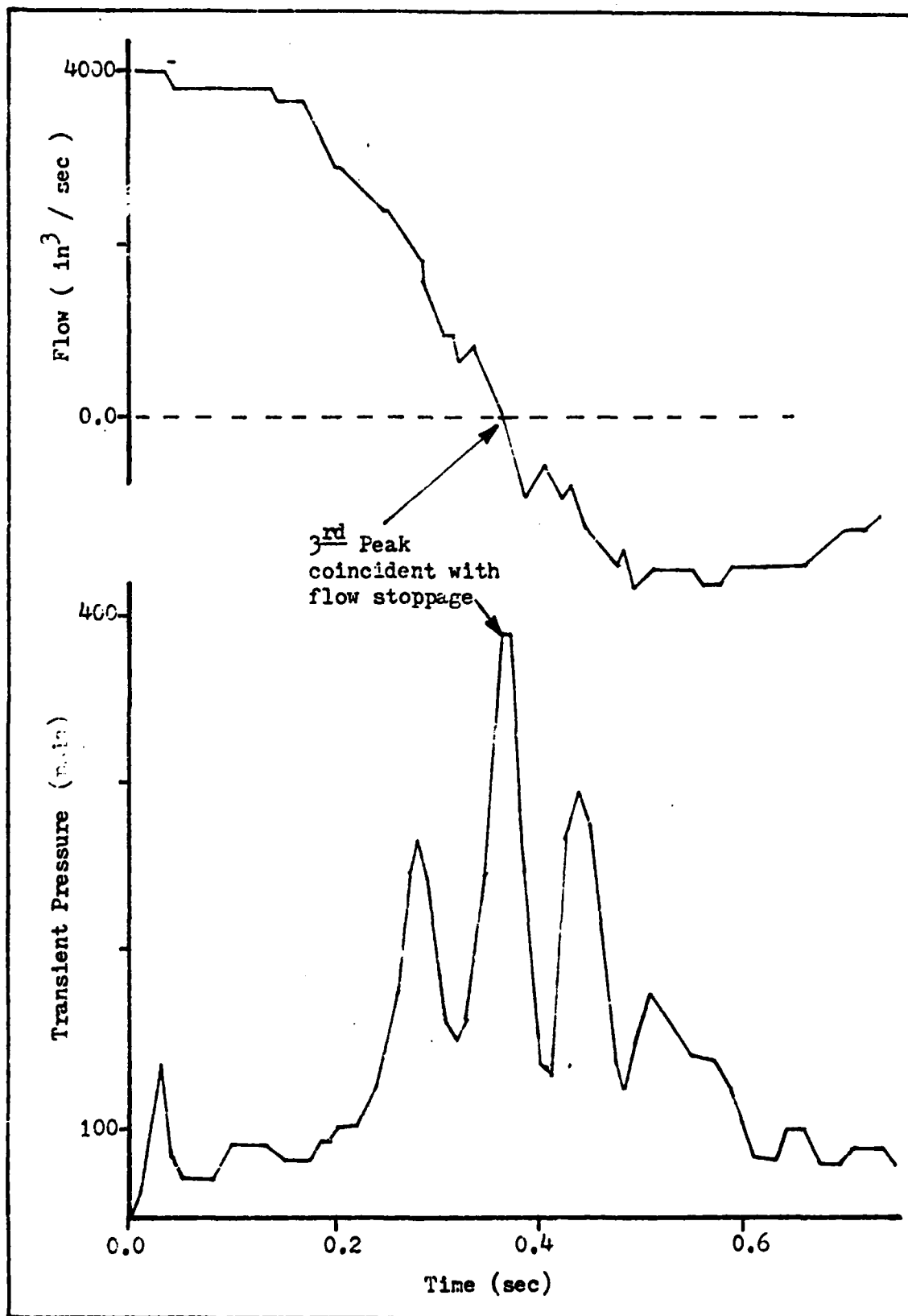


Fig 29. Typical Effect of Flow Stoppage on Transient Response for the KC-135 Model using curve A and $T_c = 0.05$ sec

Variation of KC-135 Surge Boot Precharge Pressure

The effect of surge boot precharge was investigated by varying the precharge from 50 to 100 psig. A linear closure curve was used with closure times of 0.03 and 0.05 sec. All other conditions were as described in Section V.

Results, shown in Fig 30, indicate an increase in pressure for the first peak and a reduction in pressure for the third peak over the range of increasing precharge pressure. Nozzle pressure versus precharge pressure curve slopes for the first peak are similar for both closure times with a higher transient pressure for the 0.03 sec closure. The third peak has the same response for both closure times.

Parks' study (Ref 10:53) indicated a decreasing maximum transient pressure with increasing precharge pressure relationship. As discussed earlier, due to the slow closure ($T_c = 0.05$ sec) and extreme snubbing of the closure curve used by Parks, the third peak was predominant in his study. Therefore, the results obtained by Parks and the third peak results of this study are in agreement.

The differing effects on the first and third peaks result from the inability of the accumulator to respond quickly to short duration impulses. The third peak pressure builds at a slower rate as the flow decreases to zero. For the first peak, increasing the precharge pressure acts to harden the system, providing progressively less energy absorption for the impulse. Conversely, the slower building pressure of the third peak is attenuated with the increased precharge since more energy is required to compress the gas in the accumulators.

The effect of precharge for the KC-135 model is consistent with the results obtained for the Laboratory Test model. The Laboratory

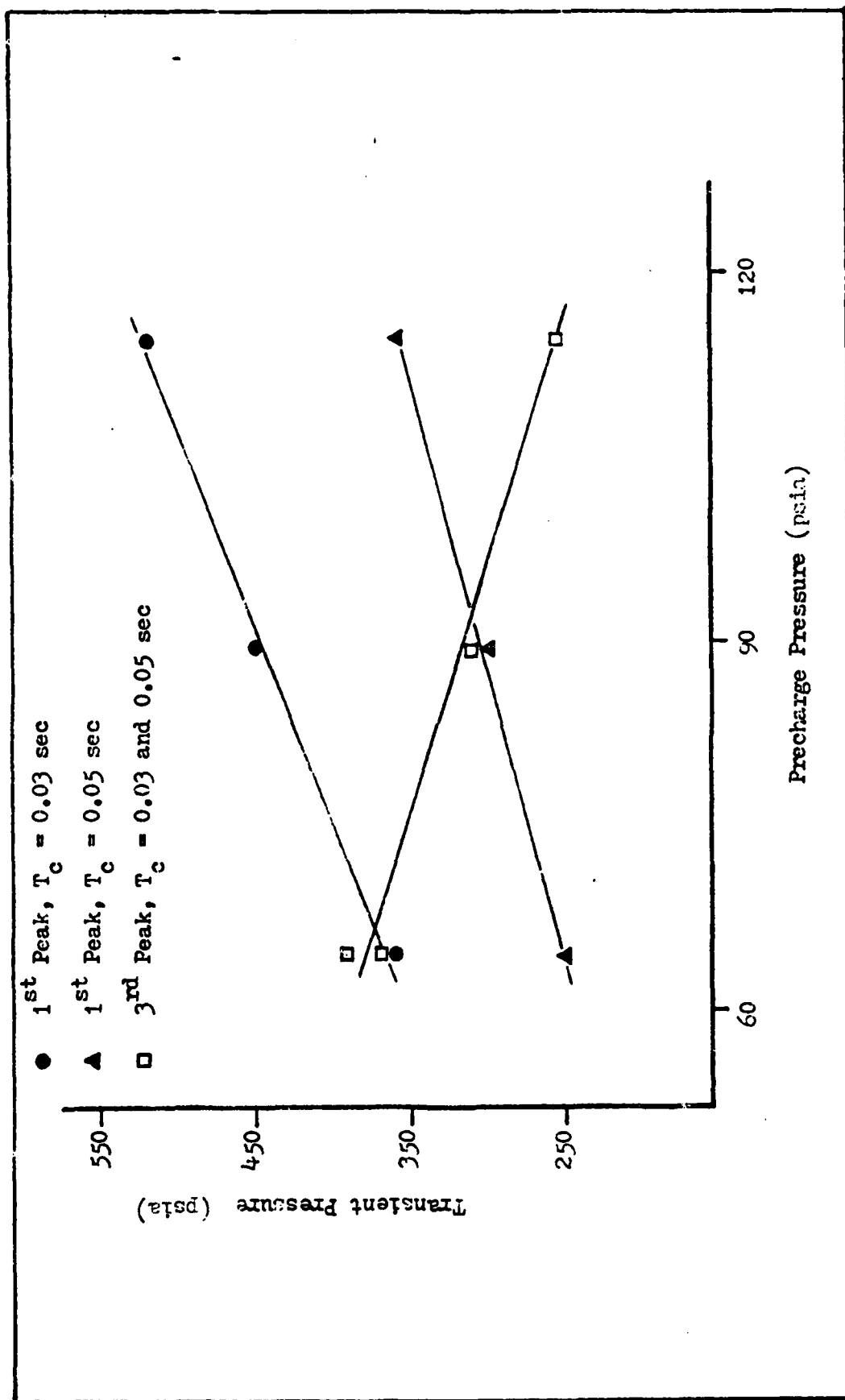


Fig 10. Effect of Variation in KC-135 Surge Boot Precharge on Transient Pressure at Nozzle

Test is a relatively simple system with the first peak predominant;
the maximum pressure increased with precharge pressure increases.

VIII. Conclusions

Based on the research and simulations conducted and reported herein, it is concluded that:

1. The HYTRAN computer program can predict the transient response of simple systems with good results.
2. Simulation of the Laboratory Test resulted in a favorable agreement with experimental findings for the general transient response but underpredicted maximum transient pressure by approximately 15%.
3. For the Laboratory Test simulation:
 - a) A 75% decrease in accumulator volume with other variables held constant had a negligible effect on maximum transient pressure.
 - b) Increasing the accumulator precharge pressure with other variables held constant caused an increase in transient pressure.
 - c) Decreasing the diameter or increasing the length of the accumulator entry lines with all other variables held constant caused an increase in transient pressure and a decrease in the time required to settle to a steady pressure after valve closure.
4. The representative closure curve for quick disconnects must be derived from analysis of the combined nozzle/receptacle dynamics. The curve developed in this study gave good results when used in the Laboratory Test simulations and compared to the experimental results.
5. Increased percentages of valve snubbing decreases transient pressures; this is consistent with the findings of Kinzig (Ref 8).

4. For the KC-135:

a) The overall effect of decreasing nozzle valve closure time is to increase transient pressures; the dominant peak in the transient response is shifted from the third peak to the first peak; the third peak is essentially constant for all closure times.

b) Increasing the surge boot precharge pressure increased the first peak pressure and decreased the third peak pressure.

IX. Recommendations for Further Study

Additional verification of computer simulations with experimentally measured data is needed to gain confidence in the computer predictions. A scaled down system similar in configuration to the Laboratory Test could be used in the laboratory for parameter variation experiments.

A fuel systems oriented derivative of HYTRAN is available; this program, FUELTRAN, uses the basic HYTRAN program with additional subroutines added for fuel system component models (Ref 19). FUELTRAN was not used for this study because confidence could not be developed in the results obtained from it. FUELTRAN should be evaluated thoroughly to determine what, if any, usage limitations exist.

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APPENDIX A

Input Data for the Laboratory Test Model

TABLE II

[illegible]

TABLE II, continued

REAL DATA	UPST	DOWNST	LEG NO	UPST NODE NO	DOWNST NODE NO	NO OF ELEMENTS	FLOW GUESS	UPST PRESS	DOWNST PRESS
1	1	2	1	1	2	3	2013.00000	0.00000	0.00000
2	2	3	2	2	3	4	2014.00000	0.00000	0.00000
3	3	4	3	3	4	5	2015.00000	0.00000	0.00000
4	4	5	4	4	5	6	2016.00000	0.00000	0.00000
5	5	6	5	5	6	7	2017.00000	0.00000	0.00000
6	6	7	6	6	7	8	2018.00000	0.00000	0.00000
7	7	8	7	7	8	9	2019.00000	0.00000	0.00000
8	8	9	8	8	9	10	2020.00000	0.00000	0.00000
9	9	10	9	9	10	11	2021.00000	0.00000	0.00000
10	10	11	10	10	11	12	2022.00000	0.00000	0.00000
11	11	12	11	11	12	13	2023.00000	0.00000	0.00000
12	12	13	12	12	13	14	2024.00000	0.00000	0.00000
13	13	14	13	13	14	15	2025.00000	0.00000	0.00000
14	14	15	14	14	15	16	2026.00000	0.00000	0.00000
15	15	16	15	15	16	17	2027.00000	0.00000	0.00000
16	16	17	16	16	17	18	2028.00000	0.00000	0.00000
17	17	18	17	17	18	19	2029.00000	0.00000	0.00000
18	18	19	18	18	19	20	2030.00000	0.00000	0.00000
19	19	20	19	19	20	21	2031.00000	0.00000	0.00000
20	20	21	20	20	21	22	2032.00000	0.00000	0.00000
21	21	22	21	21	22	23	2033.00000	0.00000	0.00000
22	22	23	22	22	23	24	2034.00000	0.00000	0.00000
23	23	24	23	23	24	25	2035.00000	0.00000	0.00000
24	24	25	24	24	25	26	2036.00000	0.00000	0.00000
25	25	26	25	25	26	27	2037.00000	0.00000	0.00000
26	26	27	26	26	27	28	2038.00000	0.00000	0.00000
27	27	28	27	27	28	29	2039.00000	0.00000	0.00000
28	28	29	28	28	29	30	2040.00000	0.00000	0.00000
29	29	30	29	29	30	31	2041.00000	0.00000	0.00000
30	30	31	30	30	31	32	2042.00000	0.00000	0.00000
31	31	32	31	31	32	33	2043.00000	0.00000	0.00000
32	32	33	32	32	33	34	2044.00000	0.00000	0.00000
33	33	34	33	33	34	35	2045.00000	0.00000	0.00000
34	34	35	34	34	35	36	2046.00000	0.00000	0.00000
35	35	36	35	35	36	37	2047.00000	0.00000	0.00000
36	36	37	36	36	37	38	2048.00000	0.00000	0.00000
37	37	38	37	37	38	39	2049.00000	0.00000	0.00000
38	38	39	38	38	39	40	2050.00000	0.00000	0.00000
39	39	40	39	39	40	41	2051.00000	0.00000	0.00000
40	40	41	40	40	41	42	2052.00000	0.00000	0.00000
41	41	42	41	41	42	43	2053.00000	0.00000	0.00000
42	42	43	42	42	43	44	2054.00000	0.00000	0.00000
43	43	44	43	43	44	45	2055.00000	0.00000	0.00000
44	44	45	44	44	45	46	2056.00000	0.00000	0.00000
45	45	46	45	45	46	47	2057.00000	0.00000	0.00000
46	46	47	46	46	47	48	2058.00000	0.00000	0.00000
47	47	48	47	47	48	49	2059.00000	0.00000	0.00000
48	48	49	48	48	49	50	2060.00000	0.00000	0.00000
49	49	50	49	49	50	51	2061.00000	0.00000	0.00000
50	50	51	50	50	51	52	2062.00000	0.00000	0.00000
51	51	52	51	51	52	53	2063.00000	0.00000	0.00000
52	52	53	52	52	53	54	2064.00000	0.00000	0.00000
53	53	54	53	53	54	55	2065.00000	0.00000	0.00000
54	54	55	54	54	55	56	2066.00000	0.00000	0.00000
55	55	56	55	55	56	57	2067.00000	0.00000	0.00000
56	56	57	56	56	57	58	2068.00000	0.00000	0.00000
57	57	58	57	57	58	59	2069.00000	0.00000	0.00000
58	58	59	58	58	59	60	2070.00000	0.00000	0.00000
59	59	60	59	59	60	61	2071.00000	0.00000	0.00000
60	60	61	60	60	61	62	2072.00000	0.00000	0.00000
61	61	62	61	61	62	63	2073.00000	0.00000	0.00000
62	62	63	62	62	63	64	2074.00000	0.00000	0.00000
63	63	64	63	63	64	65	2075.00000	0.00000	0.00000
64	64	65	64	64	65	66	2076.00000	0.00000	0.00000
65	65	66	65	65	66	67	2077.00000	0.00000	0.00000
66	66	67	66	66	67	68	2078.00000	0.00000	0.00000
67	67	68	67	67	68	69	2079.00000	0.00000	0.00000
68	68	69	68	68	69	70	2080.00000	0.00000	0.00000
69	69	70	69	69	70	71	2081.00000	0.00000	0.00000
70	70	71	70	70	71	72	2082.00000	0.00000	0.00000
71	71	72	71	71	72	73	2083.00000	0.00000	0.00000
72	72	73	72	72	73	74	2084.00000	0.00000	0.00000
73	73	74	73	73	74	75	2085.00000	0.00000	0.00000
74	74	75	74	74	75	76	2086.00000	0.00000	0.00000
75	75	76	75	75	76	77	2087.00000	0.00000	0.00000
76	76	77	76	76	77	78	2088.00000	0.00000	0.00000
77	77	78	77	77	78	79	2089.00000	0.00000	0.00000
78	78	79	78	78	79	80	2090.00000	0.00000	0.00000
79	79	80	79	79	80	81	2091.00000	0.00000	0.00000
80	80	81	80	80	81	82	2092.00000	0.00000	0.00000
81	81	82	81	81	82	83	2093.00000	0.00000	0.00000
82	82	83	82	82	83	84	2094.00000	0.00000	0.00000
83	83	84	83	83	84	85	2095.00000	0.00000	0.00000
84	84	85	84	84	85	86	2096.00000	0.00000	0.00000
85	85	86	85	85	86	87	2097.00000	0.00000	0.00000
86	86	87	86	86	87	88	2098.00000	0.00000	0.00000
87	87	88	87	87	88	89	2099.00000	0.00000	0.00000
88	88	89	88	88	89	90	2100.00000	0.00000	0.00000
89	89	90	89	89	90	91	2101.00000	0.00000	0.00000
90	90	91	90	90	91	92	2102.00000	0.00000	0.00000
91	91	92	91	91	92	93	2103.00000	0.00000	0.00000
92	92	93	92	92	93	94	2104.00000	0.00000	0.00000
93	93	94	93	93	94	95	2105.00000	0.00000	0.00000
94	94	95	94	94	95	96	2106.00000	0.00000	0.00000
95	95	96	95	95	96	97	2107.00000	0.00000	0.00000
96	96	97	96	96	97	98	2108.00000	0.00000	0.00000
97	97	98	97	97	98	99	2109.00000	0.00000	0.00000
98	98	99	98	98	99	100	2110.00000	0.00000	0.00000
99	99	100	99	99	100	101	2111.00000	0.00000	0.00000
100	100	101	100	100	101	102	2112.00000	0.00000	0.00000
101	101	102	101	101	102	103	2113.00000	0.00000	0.00000
102	102	103	102	102	103	104	2114.00000	0.00000	0.00000
103	103	104	103	103	104	105	2115.00000	0.00000	0.00000
104	104	105	104	104	105	106	2116.00000	0.00000	0.00000
105	105	106	105	105	106	107	2117.00000	0.00000	0.00000
106	106	107	106	106	107	108	2118.00000	0.00000	0.00000
107	107	108	107	107	108	109	2119.00000	0.00000	0.00000
108	108	109	108	108	109	110	2120.00000	0.00000	0.00000
109	109	110	109	109	110	111	2121.00000	0.00000	0.00000
110	110	111	110	110	111	112	2122.00000	0.00000	0.00000
111	111	112	111	111	112	113	2123.00000	0.00000	0.00000
112	112	113	112	112	113	114	2124.00000	0.00000	0.00000
113	113	114	113	113	114	115	2125.00000	0.00000	0.00000
114	114	115	114	114	115	116	2126.00000	0.00000	0.00000
115	115	116	115	115	116	117	2127.00000	0.00000	0.00000
116	116	117	116	116	117	118	2128.00000	0.00000	0.00000
117	117	118	117	117	118	119	2129.00000	0.00000	0.00000
118	118	119	118	118	119	120	2130.00000	0.00000	0.00000
119	119	120	119	119	120	121	2131.00000	0.00000	0.00000
120	120	121	120	120	121	122	2132.00000	0.00000	0.00000
121	121	122	121	121	122	123	2133.00000	0.00000	0.00000
122	122	123	122	122	123	124	2134.00000	0.00000	0.00000
123	123	124	123	123	124	125	2135.00000	0.00000	0.00000
124	124	125	124	124	125	126	2136.00000	0.00000	0.00000
125	125	126	125	125	126	127	2137.00000	0.00000	0.00000
126	126	127	126	126	127	128	2138.00000	0.00000	0.00000
127	127	128	127	127	128	129	2139.00000	0.00000	0.00000
128	128	129	128	128	129	130	2140.00000	0.00000	0.00000
129	129	130	129	129	13				

APPENDIX B

Input Data for the KC-135 Model

TABLE III

Input Data for the KC-135 Model

LINE NO.	EXTERNAL DIA.	WALL THICKNESS	MODULUS OF ELASTICITY	OFFL	CHARACTERISTIC VELOCITY OF SOUND	IMPEDANCE
1	1.0000	.0720	.100E+00	1.0000	.2200	2000.0000
2	1.0000	.0720	.100E+00	1.0000	.2200	2000.0000
3	1.0000	.0720	.100E+00	1.0000	.2200	2000.0000
4	1.0000	.0720	.100E+00	1.0000	.2200	2000.0000
5	1.0000	.0720	.100E+00	10.0000	.2200	20000.0000
6	1.0000	.0720	.100E+00	10.0000	.2200	33976.4700
7	1.0000	.0300	.100E+00	10.0000	1.6433	33976.4700
8	1.0000	.0720	.100E+00	10.0000	.2200	33976.4700
9	1.0000	.0720	.100E+00	17.1667	.2200	33976.4700
10	1.0000	.0720	.100E+00	10.0000	.2200	33976.4700
11	1.0000	.0720	.100E+00	20.0000	.2200	33976.4700
12	1.0000	.0720	.100E+00	17.3333	.2200	33976.4700
13	1.0000	.0720	.100E+00	21.0000	.2200	33976.4700
14	1.0000	.0720	.100E+00	21.0000	.2200	33976.4700
15	1.0000	.0720	.100E+00	10.0000	1.6433	33976.4700
16	1.0000	.0720	.100E+00	17.0000	.2200	33976.4700
17	1.0000	.0720	.100E+00	17.3333	.2200	33976.4700
18	1.0000	.0720	.100E+00	22.0000	.2200	33976.4700
19	1.0000	.0500	.100E+00	17.5000	.4053	34500.4763
20	1.0000	.0720	.100E+00	10.6667	.2200	33976.4700
21	1.0000	.0300	.100E+00	20.0000	4.1575	37470.0067
22	1.0000	.1250	.100E+00	19.1519	.2200	33976.4700
23	1.0000	.1000	.100E+00	20.0000	.1774	33976.4700
24	1.0000	.1000	.100E+00	17.0000	.1774	33976.4700
25	1.0000	.1000	.100E+00	20.0000	.1774	33976.4700

APPENDIX III. continued

[illegible]

ATT-5 III, continued

1	12	37	4500-0000	0-0000	0-0000
13	14	15	4500-0000	0-0000	0-0000
16	17	18	4500-0000	0-0000	0-0000
19	20	21	4500-0000	0-0000	0-0000
22	23	24	4500-0000	0-0000	0-0000
25	26	27	4500-0000	0-0000	0-0000
28	29	30	4500-0000	0-0000	0-0000
31	32	33	4500-0000	0-0000	0-0000
34	35	36	4500-0000	0-0000	0-0000
37	38	39	4500-0000	0-0000	0-0000
40	41	42	4500-0000	0-0000	0-0000
43	44	45	4500-0000	0-0000	0-0000
46	47	48	4500-0000	0-0000	0-0000
49	50	51	4500-0000	0-0000	0-0000
52	53	54	4500-0000	0-0000	0-0000
55	56	57	4500-0000	0-0000	0-0000
58	59	60	4500-0000	0-0000	0-0000
61	62	63	4500-0000	0-0000	0-0000
64	65	66	4500-0000	0-0000	0-0000
67	68	69	4500-0000	0-0000	0-0000
70	71	72	4500-0000	0-0000	0-0000
73	74	75	4500-0000	0-0000	0-0000
76	77	78	4500-0000	0-0000	0-0000
79	80	81	4500-0000	0-0000	0-0000
82	83	84	4500-0000	0-0000	0-0000
85	86	87	4500-0000	0-0000	0-0000
88	89	90	4500-0000	0-0000	0-0000
91	92	93	4500-0000	0-0000	0-0000
94	95	96	4500-0000	0-0000	0-0000
97	98	99	4500-0000	0-0000	0-0000
100	101	102	4500-0000	0-0000	0-0000
103	104	105	4500-0000	0-0000	0-0000
106	107	108	4500-0000	0-0000	0-0000
109	110	111	4500-0000	0-0000	0-0000
112	113	114	4500-0000	0-0000	0-0000
115	116	117	4500-0000	0-0000	0-0000
118	119	120	4500-0000	0-0000	0-0000
121	122	123	4500-0000	0-0000	0-0000
124	125	126	4500-0000	0-0000	0-0000
127	128	129	4500-0000	0-0000	0-0000
130	131	132	4500-0000	0-0000	0-0000
133	134	135	4500-0000	0-0000	0-0000
136	137	138	4500-0000	0-0000	0-0000
139	140	141	4500-0000	0-0000	0-0000
142	143	144	4500-0000	0-0000	0-0000
145	146	147	4500-0000	0-0000	0-0000
148	149	150	4500-0000	0-0000	0-0000
151	152	153	4500-0000	0-0000	0-0000
154	155	156	4500-0000	0-0000	0-0000
157	158	159	4500-0000	0-0000	0-0000
160	161	162	4500-0000	0-0000	0-0000
163	164	165	4500-0000	0-0000	0-0000
166	167	168	4500-0000	0-0000	0-0000
169	170	171	4500-0000	0-0000	0-0000
172	173	174	4500-0000	0-0000	0-0000
175	176	177	4500-0000	0-0000	0-0000
176	177	178	4500-0000	0-0000	0-0000
177	178	179	4500-0000	0-0000	0-0000
178	179	180	4500-0000	0-0000	0-0000
179	180	181	4500-0000	0-0000	0-0000
180	181	182	4500-0000	0-0000	0-0000
181	182	183	4500-0000	0-0000	0-0000
182	183	184	4500-0000	0-0000	0-0000
183	184	185	4500-0000	0-0000	0-0000
184	185	1	4500-0000	0-0000	0-0000

[illegible]

TABLE III, continued

[illegible]

Vita

Thomas J. Carter III was born 3 August 1947 in Louisville, Kentucky. He graduated from high school in Jackson, Tennessee in 1965 and entered the USAF in April 1966. He was assigned to the 3550th Field Maintenance Squadron at Moody AFB, Georgia until June 1967; from June 1967 until March 1969 he served with the 20th Special Operations Squadron in Southeast Asia. During the period March 1969 to September 1974 his assignments were to the 6510th Field Maintenance and Civil Engineering Squadrons, Edwards AFB, California; the 3550th Air Base Group (Civil Engineering), Moody AFB, Georgia; and the 5010th Civil Engineering Squadron, Eielson AFB, Alaska. He entered the Airman Education and Commissioning Program in October 1974 at Auburn University from which he received the degree of Bachelor of Mechanical Engineering in August 1977. He received his commission through CTS in December 1977. He then served as a flight test engineer, 6520th Test Group, Edwards AFB, California, until entering the School of Engineering, Air Force Institute of Technology, in June 1981.

Permanent Address: 506 N. Royal Street

Jackson, Tennessee

33301

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		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Thomas J. Carter III Captain		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Air Force Institute of Technology (AFIT-EN) Wright-Patterson AFB, Ohio 45433		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS 4950 th Test Wing/FFDF Capt. Scott E. Parks Wright-Patterson AFB, Ohio 45433		12. REPORT DATE December, 1982
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19. KEY WORDS (Continue on reverse side if necessary and identify by block number) Refueling Systems Transient Flow Analysis KC-135, Refueling System Computer Model		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number) Two aircraft refueling system configurations were simulated using HYTRAN, an existing transient flow analysis computer program. Transient pressure response subsequent to downstream valve closure was investigated for the KC-135 and a laboratory test rig and compared to available experimental data and previous research. Parametric studies were performed of system variables: valve closure time, valve area versus time (closure curves), and variations in surge attenuation components		

(surge boot, accumulators). A typical closure curve for quick disconnects was approximated and verified. The simulation results compared favorably with experimental data and previous work. Transient pressure was found to be sensitive to accumulator or surge boot precharge pressure while changes in accumulator volume had little effect. Variations in accumulator entry line diameter or length affected transient pressure and settling time. Increased valve snubbing reduced maximum transient pressures.

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