RESPONSE CHARACTERISTICS OF ACTUATORS
USED IN AIR BLAST CIRCUIT BREAKERS

by 3735

APPASAHEB TRYAMBAKRAO PATIL

B. E. (Mech.), University of Poona, Poona, India, 1967

A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1970

Approved by:

Ralph O. Turnquist
Major Professor

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NOMENCLATURE

A = Area of the piston, $in.^2$

A_o = Area of control valve opening, $in.^2$

$\bar{A}_o$ = Normalized area of control valve opening.

B_p = Viscous damping coefficient, lbf-sec/in.

C_{cmax} = Ratio of clearance volume to piston area, $\frac{V_{co}}{A}$, in.

C_d = Coefficient of discharge.

C_f = final piston displacement, in.

$\dot{C}_{Amax}$ = Maximum velocity of the piston during a response.

$\dot{C}_{max}$ = Maximum calculated piston velocity, in/sec.

$C(t)$ = Displacement of the piston at time t, in.

$\dot{C}(t)$ = Velocity of the piston at time t, in/sec.

$\dot{C}(\mathbb{P}=max)$ = Velocity of the piston at maximum power, in/sec.

$\bar{\dot{C}}$ = Normalized velocity of the piston.

$\bar{\dot{C}}(\mathbb{P}=max)$ = Normalized velocity of the piston when the power is maximum.

$\ddot{C}(t)$ = Acceleration of the piston at time t, in/sec^2.

F_{cp} = Coulomb friction force, lbf.

F_{max} = Maximum pressure force, lbf.

$F(t)$ = Pressure force acting on the piston at time t, lbf.

$F(\mathbb{P}=max)_{actuator}$ = Pressure force acting on piston at maximum power, lbf.

$F(\mathbb{P}=max)_{load}$ = load force at maximum power, lbf.

$\bar{F}$ = Normalized pressure force.

$\bar{F}(\mathbb{P}=max)$ = Normalized pressure force at maximum power.

g_c = Gravity constant, $\frac{lbm\text{-}in}{lbf\text{-}sec^2}$.

K = Ratio of constant pressure specific heat to constant volume specific heat.

K_a = Ratio of "high pressure" side piston area to spring side piston area.

K_s = Spring constant, lbf/in.

M'_l = Mass of load, $\frac{\text{lbf-sec}^2}{\text{in}}$

M'_p = Mass of piston, $\frac{\text{lbf-sec}^2}{\text{in}}$

M_l = Mass of load, lbm.

M_p = Mass of piston, lbm.

P_{atm} = Atmospheric pressure, psia.

P_{tmax} = Maximum tank pressure, psia.

P_{tf} = Final tank pressure, psia.

$P_c(t)$ = Pressure of air on the "spring side" of the piston at time t, psia.

$P_t(t)$ = Pressure of air in the supply tank at time t, psia.

$\dot{P}_c(t)$ = Rate of pressure change on "spring side" of the piston at time t, lbf/in^2-sec.

$\dot{P}_t(t)$ = Rate of pressure change in the supply tank at time t, $\frac{\text{lbf}}{\text{in}^2\text{-sec}}$.

$\bar{P}_{atm}$ = Normalized atmospheric pressure.

$\bar{P}_c$ = Normalized pressure on "Spring side" of the piston.

$\bar{P}_t$ = Normalized pressure in the supply tank.

T_{sp} = Time specified for piston to first reach final displacement, (critical displacement time), sec.

T'_{sp} = Time computed for piston to first reach final displacement, (computed critical displacement time), sec.

t = time, sec.

t_{vo} = control valve opening time, sec.

V_{co} = Volume on the spring side of the piston including the volume of connection to control valve, when actuator piston is in initial position, in^3.

V_t = Volume of high pressure air tank including the volume on high pressure side of the piston and connection between tank and cylinder when piston is in initial position, in^3.

W'_{max} = Maximum mass flow rate, $\frac{lbf\text{-}sec}{in}$.

W_{max} = Maximum mass flow rate, lbm/sec.

$W'(t)$ = Mass flow rate at time t, $\frac{lbf\text{-}sec}{in}$.

$\bar{W}$ = Normalized mass flow rate.

$\rho_{c(t)}$, $\rho_{t(t)}$ = Densities of working fluid at time t, lbm/in^3.

$\rho'_{c(t)}$, $\rho'_{t(t)}$ = Densities of working fluid at time t, $\frac{lbf\text{-}sec^2}{in^4}$.

$\sigma \underline{\Delta} \zeta .\omega n$, sec^{-1}.

ω_d = Damped frequency, rad/sec.

ω_n = Natural frequency, rad/sec.

$\psi \underline{\Delta} \sin^{-1} \zeta$, rad.

ζ = Damping ratio.

$\mathbb{P}$ = Power, $\frac{lbf\text{-}in}{sec}$.

CHAPTER I

INTRODUCTION

The ever increasing growth of EHV (Extra High Voltage) and UHV (Ultra High Voltage) systems in the world during the last five years shows the large power demands from the electric utility industry. High voltage transmission lines of 345 and 500 KV are already in service; more recently 765 KV lines have been constructed and new levels of 1000 KV are anticipated in next few years for the transmission of large amounts of power (1)*.

The necessary equipment, including switch gear, has been simultaneously developed. The capacity of circuit breakers has increased and interrupting time has been reduced. Looking back to late 30's the typical early circuit breaker had an interrupting time of 22 to 30 cycles. Since then it has been reduced to 8 then to 5 and later to 3 cycles with resultant advantage to power system operation and maintenance (2).

For some time high mass mechanisms of conventional dead tank circuit breakers and basic design problems associated with other types limited interrupting time to a 3-cycle minimum. But the air blast circuit breakers with their remarkable efficiency and high speed light weight design have eliminated the economic and technical barriers on E.H.V. equipment with 2-cycle interrupting time.

Circuit breakers with ultra high speed interrupting capability provide improved margin for system stability and permit the following:

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*References are listed in parenthesis.

1. Higher circuit loadings.

2. Decreased conductor damage on flashover.

3. Decreased damage in transformer from internal faults.

4. Shorter reclosing time on line flashover.

5. Decreased severity of disturbances from voltage dips, etc.

In short, at EHV levels the "two cycle" circuit breaker provides a new "Engineering tool" which offers significant technical and economic advantages in power systems (2).

In air blast circuit breakers the design of the trip mechanism and breaker contact actuator mechanism plays an important part in reducing the interrupting time. In this thesis an attempt is made to analyze and study the dynamic response characteristics of pneumatically actuated air blast circuit breaker contacts similar to those found in the Allis Chalmers Type ABM air blast circuit breaker (3). The following objectives were specifically considered in this thesis:

1. Establishing a design procedure for selecting control valve size and cylinder size for pneumatic actuating systems in general.

2. Calculation of control valve and cylinder size for "2 cycle" response time for pneumatic actuator systems in air blast circuit breaker.

3. Evaluation of the effect of finite control valve opening time on actuating cylinder response time.

4. Evaluation of the effect of decreasing supply pressure on the response time of actuating cylinder during different circuit breaker duty cycles.

In a Master's Report Jung (4) investigated the effect of changes in parameter values on the response of a pneumatic actuating system. Jung focused his attention on a "meter-in flow" type system where the control valve controls the flow of working fluid into the actuator cylinder. Hence the pressure on one side of piston is increasing while the pressure on the other side of the piston is constant. The model studied in this thesis is "meter-out flow" type, where flow away from the actuating cylinder is controlled by the control valve. The pressure on one side of the piston is constant, while on the other side pressure decreases.

Thus the mathematical model of the actuating system studied by Jung was slightly different than the one studied in this thesis.

However the effect of the bhange in some parameters studied by Jung is essentially the same for the response of any valve actuator system in general. Therefore the work of Jung was used as a starting point for the work of this thesis.

CHAPTER II

DESCRIPTION AND ANALYSIS OF A CIRCUIT-BREAKER CONTACT ACTUATING SYSTEM

A. GENERAL DESCRIPTION AND SCHEMATIC DIAGRAM OF THE SYSTEM.

To study an actual contact actuating system for a high capacity circuit breaker in exact detail in this thesis is unnecessary. However a general picture of the working of the mechanism can be obtained by studying the schematic diagram of an Allis Chalmers Type ABM Air Blast circuit breaker contact actuator system. A schematic diagram of the contact actuator system assumed for study in this thesis is shown in Fig. 1. It consists of:

1. A supply tank directly connected to one end of a cylinder.
2. A piston type, spring loaded, single ended double acting cylinder.
3. An on-off type, 3-way, control valve connected to the "spring end" of the cylinder.

It is assumed that the piston is directly connected to the breaker contacts, similar to that found in the Allis-Chalmers Type ABM Air blast circuit breaker (3). This eliminates need for linkages, etc, and hence delay in contact motion.

When the valve is in the closed position the spring side of the cylinder is connected to the supply tank and high pressure air acts on both sides of the piston. When the valve is in the open position, the spring side of the cylinder is opened to the atmosphere and the difference in pressure forces on the piston drives it to the right. In the actual system, used in an Allis Chalmers Type ABM Air Blast circuit breaker, provisions are made for slowing down and stopping the contacts smoothly after a certain critical travel has occurred. This is done by introducing

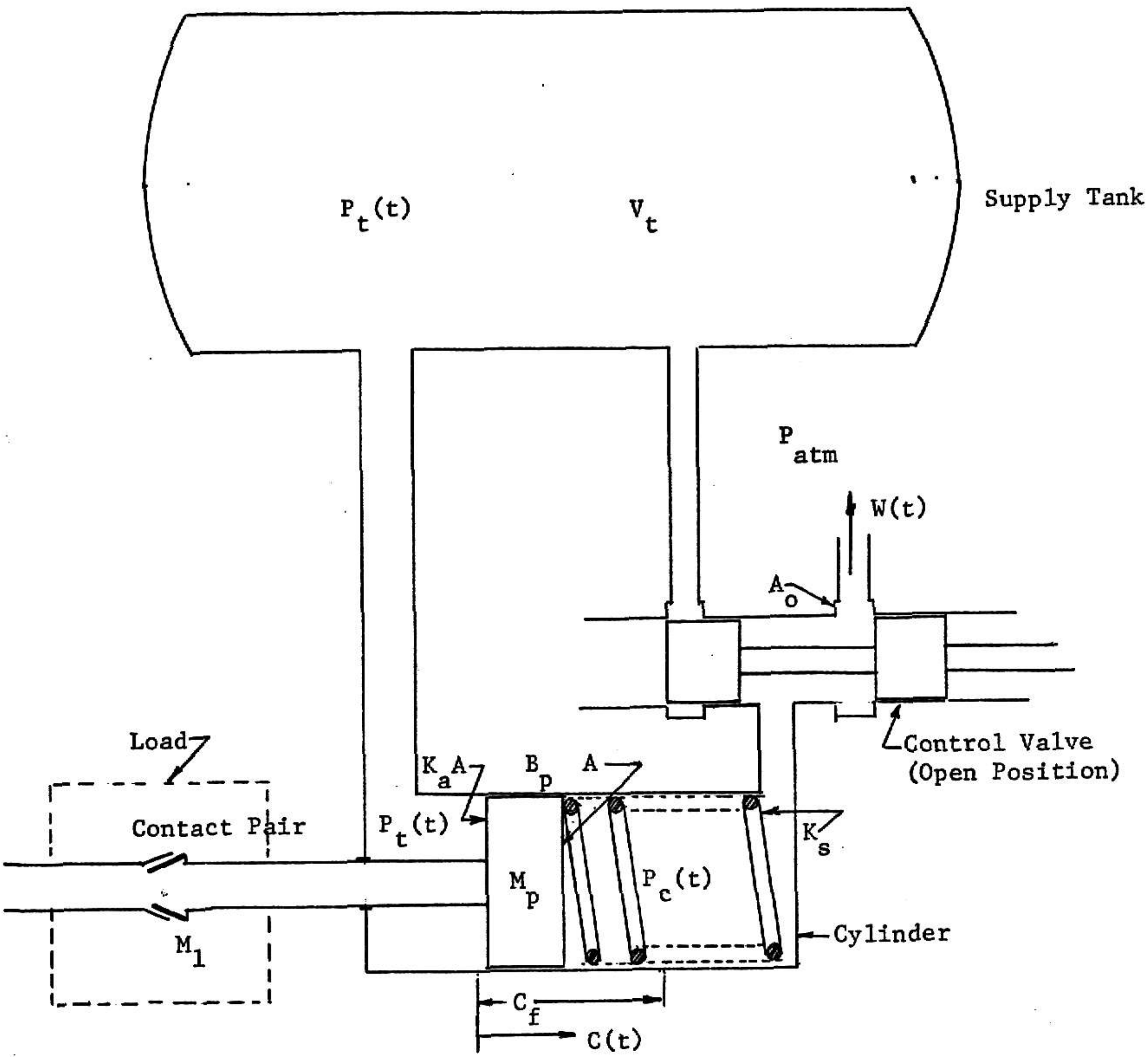

Fig. 1. Schematic Diagram of the Contact Actuator System.

additional spring(s) to the system. But in this thesis attention is focused only on the displacement up to the critical displacement and hence the "slowing down" mechanism is not considered in the schematic diagram shown in Fig. 1. During "breaker open" operation the air in the supply tank is used to actuate many pairs of breaker contacts and also for extinguishing the arc at each pair of contacts. This consumes a large volume of air which considerably decreases the air pressure in the supply tank. This drop in the supply tank air pressure can be simulated for the system of Fig. 1 if appropriate tank volumes are assumed.

During a "breaker close" operation the spring side of the cylinder is opened to the supply tank and the difference in pressure forces on the piston drives it to the left.

The control valve in any real system will take a finite time to fully open. In this thesis the effect of finite valve opening time on system response is studied. The opening motion of the valve is assumed to be linear, or equivalent to an ideal ramp input.

The total number of contact pairs in any circuit breaker unit depends on the capacity of the circuit breaker. A typical Allis Chalmers type ABM air blast circuit breaker, 550 KV, 38,000 MVA, has 8 contact pairs per phase.

The effective area of piston as shown in Figure 1 is not the same on both sides. The difference is in allowing for piston rod area. In Allis Chalmers Type ABM air blast circuit breaker the piston itself acts as a contact and in this case the area on both sides of the piston is the same (3).

B. MATHEMATICAL ANALYSIS OF THE SYSTEM

The following assumptions were used in the analysis of the system of Figure 1:

1. The working fluid, air, behaves like a perfect gas for the operating conditions considered. Therefore,

   $K = 1.4$

   $R = 2.47 \times 10^5 \ in^2/sec^2 - {}^\circ R$

2. The working fluid has constant specific heats.
3. The kinetic energy of the working fluid is negligible.
4. Flow velocity is high enough to consider the flow process adiabatic.
5. Forces due to coulomb type friction are small compared to other forces acting on the piston and can be neglected.
6. The supply tank, cylinder, control valve, and interconnecting lines are rigid.
7. The discharge coefficient of the control valve is constant.
8. Pressure wave effects in working fluid are negligible.
9. Leakage of working fluid across the piston is negligible.
10. Fluid friction in connections and control valve is negligible.
11. Heat transfer between working fluid and its environment is negligible.

1. FLOW OF AIR INTO CYLINDER

As the piston starts moving towards the right (Fig. 1) the air in the supply tank and the left hand side of the cylinder expands. This process is rapid enough so that the expansion can be assumed to be an adiabatic process.

Considering the volume occupied by the air in the supply tank, the left side of the cylinder and the connections as a control volume, the equation for the conservation of the mass in the control volume is:

$$\frac{\partial}{\partial t}\left[\rho'_t(t)(V_t + K_a AC(t))\right] = 0$$

which becomes

$$\dot{\rho}'_t(t)[V_t + K_a AC(t)] + \rho'_t(t)K_a A\dot{C}(t) = 0$$

substituting

$$\dot{\rho}'_t(t) = \frac{\dot{P}_t(t)\rho'_t(t)}{\beta} \qquad \text{(where } \beta = K\,P_t(t)\text{)}$$

gives

$$\frac{\dot{P}_t(t)\rho'_t(t)}{KP_t(t)}(V_t + K_a AC(t)) + \rho'_t(t)K_a A\dot{C}(t) = 0$$

Rearranging and substituting K = 1.4,

$$\dot{P}_t(t) = \frac{-1.4\ P_t(t)K_a A\dot{C}(t)}{(V_t+K_a AC(t))} \tag{1}$$

2. FLOW OF AIR OUT OF CYLINDER

The air in the "spring side" of the cylinder is discharged through the control valve. A control volume can be defined which includes the volume of the cylinder on the "spring side" and the volume of the connection to the control valve.

Applying the principle of the conservation of mass to this control volume gives:

$$W'(t) + \frac{\partial}{\partial t}(\rho_c'(t)\ V_c(t)) = 0$$

Substituting $W(t) = W'(t)g_c$ and $\rho_c(t) = \rho_c'(t)g_c$ gives

$$(W(t) = -\dot{\rho}_c(t)(V_{co}-AC(t)) + \rho_c(t)A\dot{C}(t)$$

Rearranging and substituting $K = 1.4$, $\dot{\rho}_c(t) = \dfrac{\dot{P}_c(t)\ \rho_c(t)}{KP_c(t)}$

$$\dot{P}_c(t) = \frac{1.4\ P_c(t)(\rho_c(t)A\dot{C}(t)-W(t))}{\rho_c(t)\ (V_{co}-AC(t))} \tag{2}$$

3. FLOW OF AIR THROUGH THE CONTROL VALVE

At time $t = 0-$ the valve is closed and at time $t = 0+$ it starts opening. In this study it will be assumed that the valve opening is proportional to time till the valve is wide open. This gives:

$$A_o(t) = \frac{A_{omax}}{t_{vo}}\ t \qquad \text{for } 0 < t < t_{vo}$$

and (3)

$$A_o(t) = A_{omax} \qquad \text{for } t \leq t_{vo}$$

During the flow process through the valve the flow conditions may be critical or subcritical depending only upon the absolute pressure of the air in the cylinder.

For a pressure ratio, $\frac{P_{atm}}{P_c}$, greater than 0.528 the flow is subcritical (5,6). If the initial cylinder pressure is greater than the critical pressure, i.e. $P_c > \frac{P_{atm}}{0.528}$, the flow will be critical until the cylinder pressure drops below the critical pressure. The process then becomes subcritical.

For critical flow conditions.

i.e. for $P_c \geq \frac{P_{atm}}{0.528}$

$$W'(t) = C_d \, A_o(t) \sqrt{K\left(\frac{2}{K+1}\right)^{\frac{K+1}{K-1}} \rho_c'(t) \, P_c(t)}$$

Substituting $W(t) = W'(t)g_c$, $\rho_c(t) = \rho_c'(t)g_c$

gives

$$W(t) = C_d A_o(t) \sqrt{g_c K\left(\frac{2}{K+1}\right)^{\frac{K+1}{K-1}} \rho_c(t) P_c(t)}$$

Substituting K = 1.4 gives:

$$W(t) = 13.45 \, C_d A_o(t) \sqrt{\rho_c(t) P_c(t)} \tag{4}$$

And for subcritical flow conditions

i.e. for $P_c < \frac{P_{atm}}{0.528}$

$$W'(t) = C_d A_o(t) \sqrt{\left(\frac{2K}{K-1}\right) \rho'_c(t)\, P_c(t) \left[1 - \left(\frac{P_{atm}}{P_c(t)}\right)^{\frac{K-1}{K}}\right] \left(\frac{P_{atm}}{P_c(t)}\right)^{2/K}}$$

Changing dimensions for $W'(t)$ and $\rho'_c(t)$ and substituting $K = 1.4$ gives:

$$W(t) = 51.98\, C_d A_o(t) \sqrt{\rho_c(t)\, P_c(t) \left[1 - \left(\frac{P_{atm}}{P_c(t)}\right)^{0.286}\right] \left(\frac{P_{atm}}{P_c(t)}\right)^{1.428}} \qquad (5)$$

4. FORCES ACTING ON PISTON

Considering the dynamic equilibrium of the piston in the system shown in Fig. 1. and considering the inertia load offered by the breaker contact a force balance equation can be written for the piston and the breaker contact together as:

$$P_t(t) K_a A + P_{atm}(1-K_a)A - P_c(t)A - (M'_p+M'_1)\, \ddot{C}(t)$$

$$- B_p \dot{C}(t) - F_{cp} \frac{\dot{C}(t)}{|C(t)|} - K_s C(t)^{*} = 0$$

Rearranging and substituting $M_p = M'_p g_c$

$M_1 = M'_1 g_c$

gives:

$$\ddot{C}(t) = \frac{386.0}{(M_p+M_1)} [(P_t(t)-P_{atm})\, K_a A - (P_c(t)-P_{atm})\, A - B_p \dot{C}(t)$$

$$- F_{cp} \frac{\dot{C}(t)}{|\dot{C}(t)|} - K_s C(t)] \qquad (6)$$

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*A spring preload force F_o, if present, can also be considered in this equation, by substituting the term $K_s C(t)$, representing spring force, by the term $K_s C(t)+F_o$.

CHAPTER III

DESIGN OF CYLINDER AND VALVE FOR DESIRED TRANSIENT RESPONSE CHARACTERISTICS

The design of an actuator and valve involves finding a configuration that will be economical and one which can do the job. There is no well defined or standard procedure for finding out the sizes of the cylinder and valve combination needed, in case of pneumatic actuating systems, to drive a given load with a desired dynamic response. Experience and common sense are ordinarily used, but this may lead to needlessly expensive and oversize components.

With the help of an analog computer calculations of component sizes is possible using trial and error methods.

Once the analog circuit for a typical system is set up, various requirements and their effects can be studied and within a few hours optimum design values could be obtained. This saves time, money and effort compared to a method using experimental models (5).

The approach taken in this thesis is based on determining the maximum power required to drive a given load with a desired dynamic response and is similar to the procedure sometimes used for determining the actuator and control valve size for hydraulic actuating systems (5).

The required output motion of the load gives the designer information to estimate the maximum power that a valve and actuator combination must be able to provide, and also to construct a velocity-force characteristic for the load. If the velocity-force curve, for a valve-actuator combination, completely encloses the load velocity-force curve then that combination can be used for the specified load. But the region which is enclosed by a velocity-force curve of the valve-actuator, but not by the load velocity-force curve represents uneconomical overdesign. (Fig. 2). Therefore the designers task is to minimize the area of this region in the most practical fashion. If the velocity-force curve of a given valve-actuator exactly matches the load velocity-force curve then it would be the best valve-actuator combination for the given load. However this is impossible to do in actual practice. Instead the best design comprise usually results if the point of maximum power output of the valve actuator matches the point of maximum power required by the load (5).

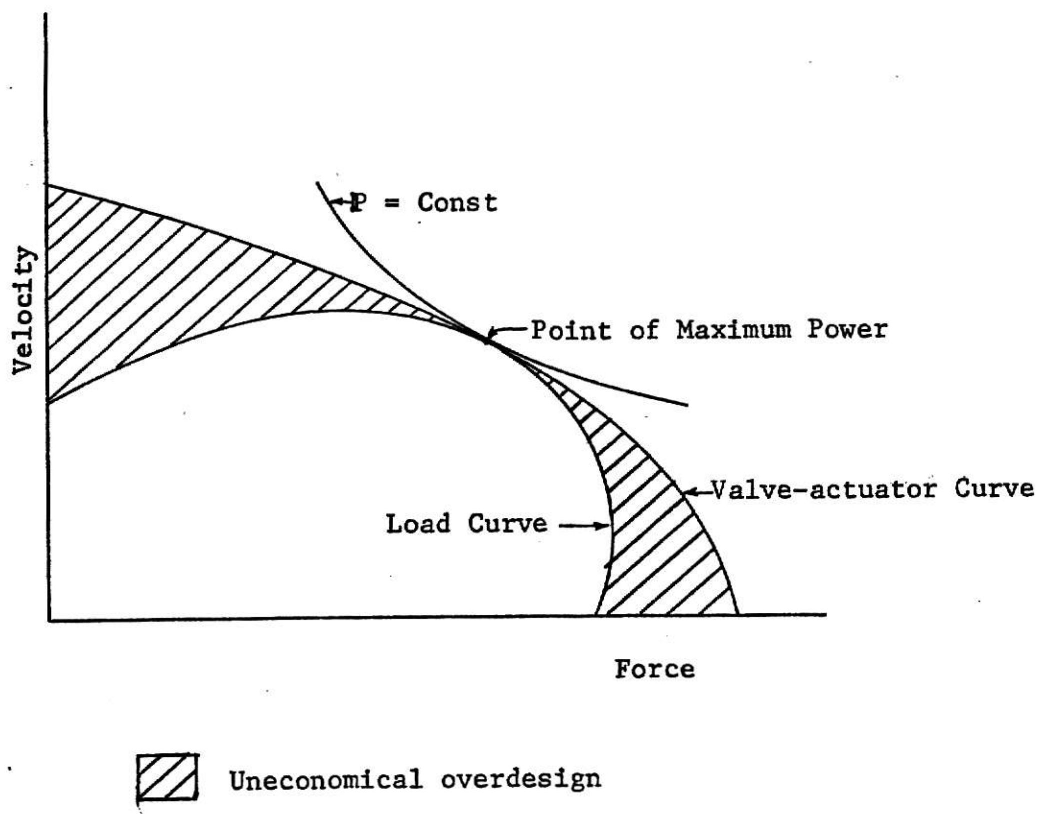

Fig. 2. Velocity-Force Curves of Load and Actuator for a Design Example.

A. DETERMINING VELOCITY-FORCE CHARACTERISTIC FOR THE LOAD TO BE DRIVEN

The actual mechanical load encountered in practice may have widely varying characteristics, and hence is not always easy to define. But specifying the required load force in terms of position, velocity and acceleration of load, gives a clear description of the load, and with this a velocity-force curve for the load can be easily plotted.

In the case of a circuit breaker the breaker contacts form a purely inertia load, except for a possible momentary stiction force which is usually small compared to the inertia of the contacts. Additional load is offered by viscous damping in cylinder, inertia of the piston and by the spring. If the coulomb friction is neglected, the piston, the damping and the spring, together with the contacts, form a damped second order system as the load for the circuit breaker actuating system assumed in Fig. 1.

If the opening time of the control valve is sufficiently small it can be assumed to be a step input. Therefore the piston-load response may be expected to be similar to the step response of damped second order system as shown in Fig. 3. This is the type of the response desired for the contacts in an air blast circuit breaker. The load velocity-force curve for such a response can be fully defined if the appropriate parameter values are specified.

The time, hereafter to be known as the critical displacement time, T_{sp}, for the contacts to reach a desired critical displacement, C_f, is of primary importance in a circuit breaker, and depends upon breaker type and capacity. This time may be different for different circuit breakers. In an Allis-Chalmers type ABM air blast circuit breaker $T_{sp} \simeq 0.008$ sec, to

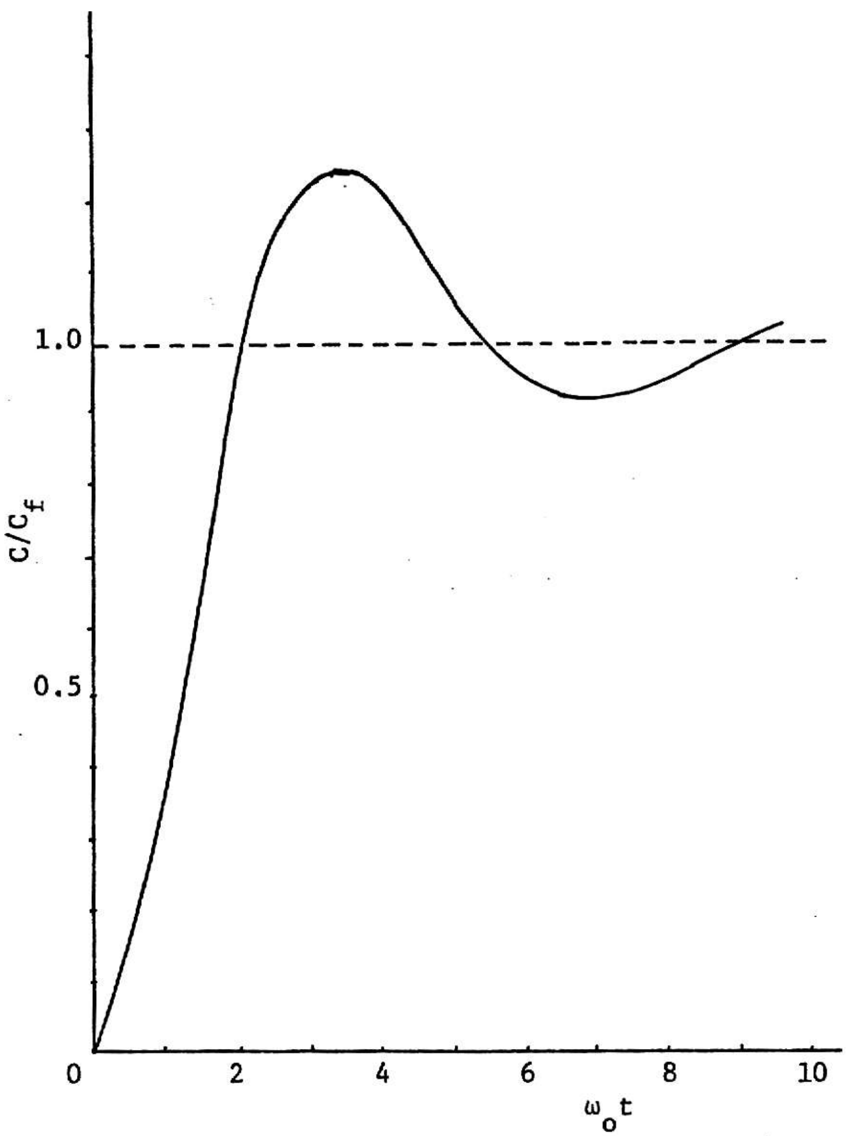

Fig. 3. Response of Damped Second Order System to Unit Step Input For $\zeta = 0.4$.

reach a displacement of $C_f = 2.436$ in.

The other system parameters are related to each other and can be defined by specifying the damping ratio ζ and the total mass of piston and contacts.

For a given natural frequency, too low values of $\zeta(<0.4)$ give undesirable overshoot and too high values of $\zeta(>1.0)$ give considerably slower response.

For this thesis a damping ratio of $\zeta = 0.4$ was chosen since it provides fast response without excessive overshoot or oscillation.

At this point in determining the velocity-force characteristics of the load no information is available regarding the actuator size and hence the piston mass and actuator damping. Therefore an estimation of the combined contact and piston mass must be made. It should be noticed here that since the value of ζ is specified earlier, actuator damping is fixed if the mass is fixed. This means that both mass and damping should be checked after the actuator size is fixed to see if both are physically possible. If the assumed combination is not valid, necessary changes in assumed mass and damping should be made and the design problem reworked.

1. Load velocity-force curve calculations.

A step response of an underdamped second order system can be given in terms of step input, spring constant, natural frequency and damping ratio (7). Using $\sigma \triangleq \zeta\omega_n$ and $\Psi \triangleq \sin^{-1}\zeta$ it can be written as:

$$C(t) = (F/K_s)[1 - \frac{1}{\cos\Psi} e^{-\sigma t} \cos(\omega_d t - \Psi)] \qquad (7)$$

The response curve assumed is the step input response of a damped second order system with damping ratio $\zeta = 0.4$. (Fig. 3). If $C(\infty)$ is made equal to C_f, and the time when response first reaches C_f is made equal to "T_{sp}" then this response curve very nearly represents the desired transient response curve for the contacts in an air blast circuit breaker, when a "breaker open" signal is generated.

Substituting $C(t) = C_f$ at time $t = T_{sp}$ in equation 7 gives:

$$\cos(\omega_d T_{sp} - \Psi) = 0$$

$$\omega_d = \frac{\Psi + \pi/2}{T_{sp}} \tag{8}$$

The relations between natural frequency, dapmed frequency, mass, damping and spring constant are

$$\omega_n = \frac{\omega_d}{\sqrt{1 - \zeta^2}} \qquad \text{if } \zeta < 1.0$$

$$2\zeta\omega_n = \frac{B_p}{(M'_p + M'_1)} \tag{9}$$

$$\omega_n^2 = \frac{K_s}{(M'_p + M'_1)}$$

Value of "T_{sp}" depends on the rating of the circuit breaker to be designed. Therefore once "T_{sp}" is given, natural frequency and damped frequency are fixed, for a fixed value of ζ. The natural frequency and damped frequency are given by equation 8 and 9.

As previously mentioned, value of piston and contact masses, together, is now estimated, and using equations 9 damping and spring constant are determined.

By differentiating equation 7, equations for velocity and acceleration can be obtained:

$$\dot{C}(t) = \frac{C_f e^{-\sigma t}}{\cos\psi} \left[\sigma\cos(\omega_d t-\psi) + \omega_d \sin(\omega_d t-\psi)\right] \tag{10}$$

$$\ddot{C}(t) = \frac{C_f e^{-\sigma t}}{\cos\psi} \left[(\omega_d^2-\sigma^2)\cos(\omega_d t-\psi) - 2\sigma\omega_d \sin(\omega_d t-\psi)\right] \tag{11}$$

Now the load force at any time t can be calculated as:

$$F(t) = (M_p'+M_1')\ddot{C}(t) + B_p\dot{C}(t) + K_s C(t) \tag{12}$$

Now the velocity-force plot can be made using equations 7, 10, 11 and 12. The point of maximum power can be determined by drawing constant power curves $\dot{C}F$ = Constant. (Fig. 4.)

However in this particular case, where the step input response of damped second order system is assumed, the force is constant and is determined by considering equation 12 at any point in time where displacement, velocity and acceleration is defined. For example, at $t = \infty$ where $C(\infty)$, $\dot{C}(\infty)$ and $\ddot{C}(\infty)$ are known. Since force is constant throughout the response the maximum power point is given by the point where velocity is maximum and can be determined if maximum velocity is determined.

For the velocity to be a maximum the acceleration must be zero. Using this condition and equations 10 and 11, maximum value of the velocity can be determined. Thus velocity and force at maximum power can be determined. The nature of velocity force diagram for this case is shown in Fig. 4.

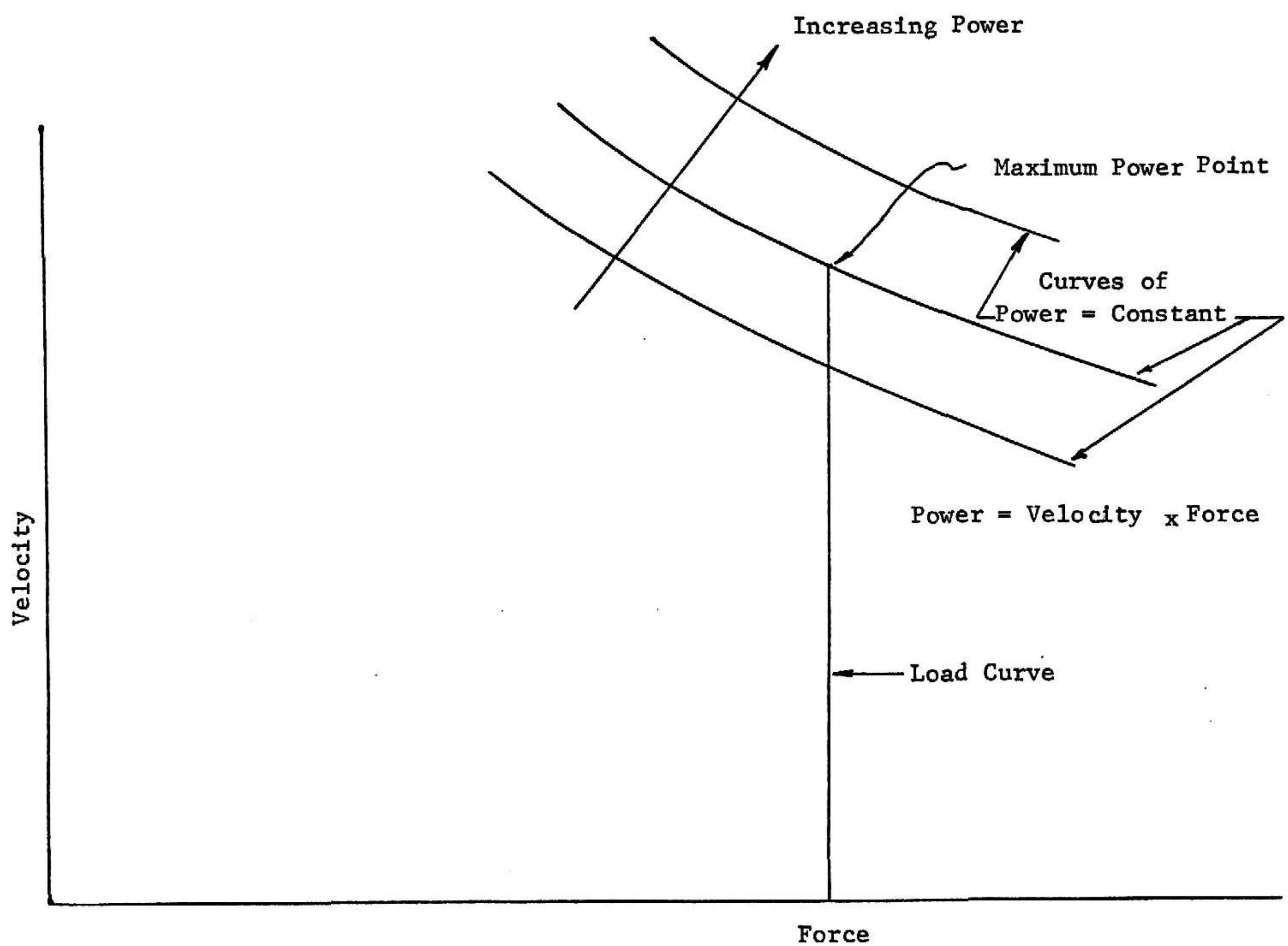

Fig. 4. Velocity-Force Characteristic of the Load

B. ACTUATOR VELOCITY-FORCE CHARACTERISTICS

The velocity-force characteristics for the valve and cylinder of Fig. 1 can be obtained by transformation of the valve flow-pressure characteristics. Therefore the first step in determining the velocity-force characteristic is to determine the valve flow-pressure characteristic.

From equations 4 and 5 the flow through the control valve is:

For $P_c < \dfrac{P_{atm}}{0.528}$

$$W(t) = 51.98\ C_d\ A_o(t) \sqrt{\rho_c(t) P_c(t)\ [1-(\frac{P_{atm}}{P_c(t)})^{0.286}](\frac{P_{atm}}{P_c(t)})^{1.428}} \quad \text{(5 repeated)}$$

and for $P_c \geq \dfrac{P_{atm}}{0.528}$

$$W(t) = 13.45\ C_d\ A_o(t) \sqrt{\rho_c(t)\ P_c(t)} \quad \text{(4 repeated)}$$

Assuming that valve opening time is negligible and making the equations dimensionless to generalize them, by defining:

$$W_{max} = 13.45\ C_d\ A_{omax} \sqrt{P_{tmax} \rho_{tmax}}$$

$$\bar{W} = \frac{W}{W_{max}}; \quad \bar{P}_c = \frac{P_c}{P_{tmax}}, \quad \bar{P}_{atm} = \frac{P_{atm}}{P_{tmax}}$$

Equations 4 and 5 can be written as:

For $\bar{P}_c < \frac{\bar{P}_{atm}}{0.528}$

$$\bar{W} = 3.86\sqrt{(\bar{P}_c^{\;0.286} - \bar{P}_{atm}^{\;0.286})\bar{P}_{atm}^{\;1.428}} \tag{13}$$

and for $\bar{P}_c \geq \bar{P}_{atm}/0.528$

$$\bar{W} = \bar{P}_c^{\;0.8575} \tag{14}$$

using equations 13 and 14 the flow pressure characteristic of the valve can be determined for P_{tmax} = constant and is shown in Fig. 5.

At this stage the transformation of the flow-pressure characteristic of Fig. 5 into the velocity-force characteristic for the valve cylinder combination of Fig. 1. is not possible unless some simplifying assumptions regarding the parameter variations in the system are made. It should also be made clear that the effects of these assumptions can only be observed after studying actual system response.

Consider equation 2 which defines the unsteady flow from the spring side of the cylinder. If substitutions are made for W' and ρ_c' the equation becomes

$$W = -\frac{\rho_c \dot{P}_c}{KP_c}(V_{co} - AC) + \rho_c A\dot{C}$$

Using $\dot{C}_{max} = W_{max}/\rho_{tmax} A$

$$\bar{\dot{C}} = \frac{\dot{C}}{\dot{C}_{max}}, \quad \bar{\dot{P}}_c = \frac{\dot{P}_c}{P_{tmax}}$$

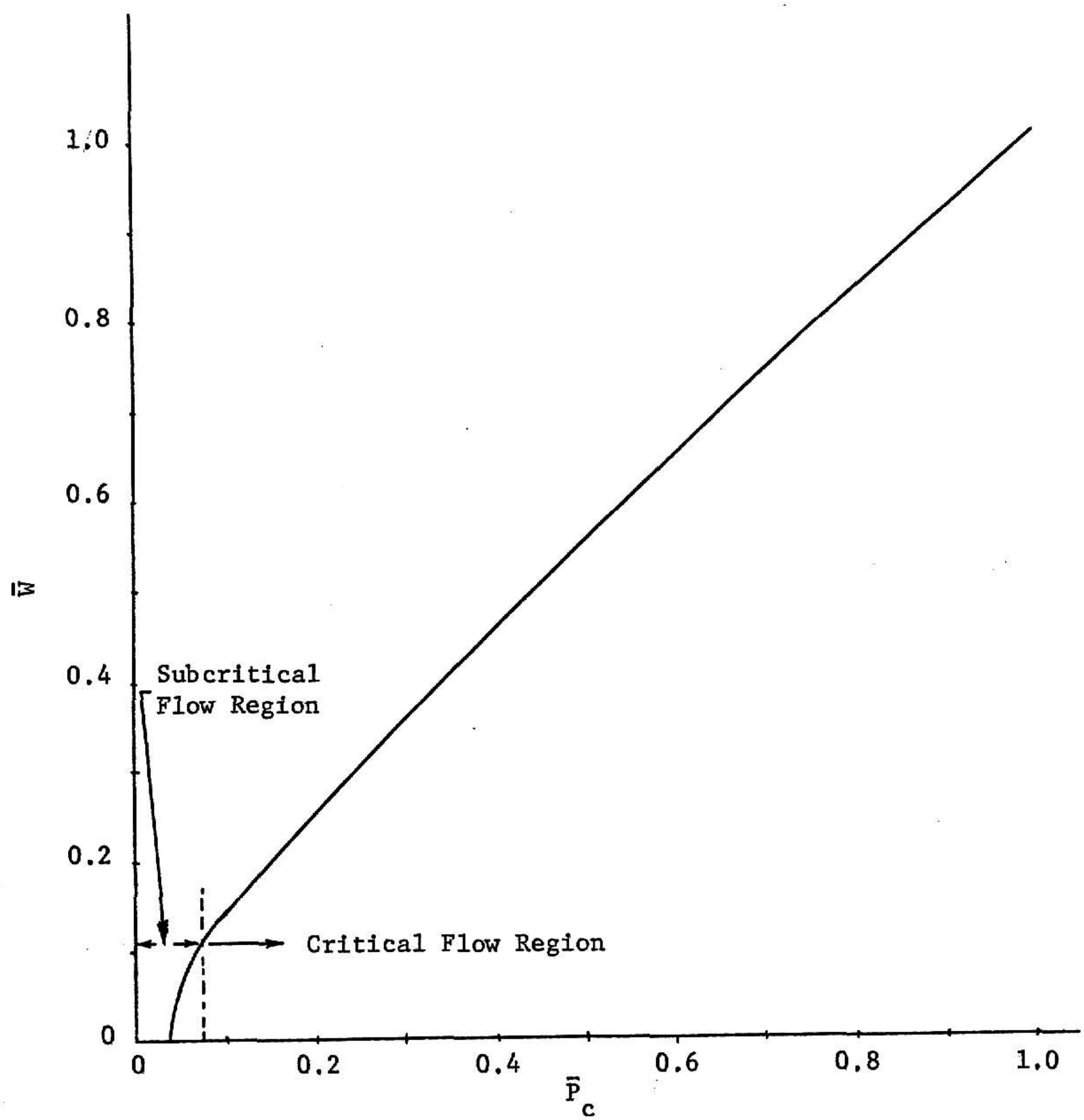

Fig. 5. Valve Flow-Pressure Characteristic. P_{tmax} = 384.7 psia.

The following is obtained:

$$\bar{W} = \frac{-\bar{P}_C^{\,1/K}\,\dot{\bar{P}}_C(V_{co}-AC)}{K\,\bar{P}_C\,\dot{C}_{max}\,A} + \bar{P}_C^{\,1/K}\,\dot{\bar{C}}$$

Rearranging and defining $C_{cmax} = \frac{V_{co}}{A}$

$$\dot{\bar{C}} = \bar{W}\,\bar{P}_C^{\,-1/K} + \frac{\dot{\bar{P}}_C(C_{cmax}-C)}{K\,\bar{P}_c\,\dot{C}_{max}} \tag{15}$$

The second term on the right hand side in the above equation takes into consideration the compressibility effect of the working fluid. If the working fluid is practically incompressible (i.e. mineral oils, or liquids) then the value of this term will be negligible in comparison with other terms. But for gases this term is not always negligible.

This term presents the difficulty in transforming the flow-pressure characteristics of the valve into the velocity-force characteristic for the valve-cylinder system of figure 1. For example, no information is available about the pressure variation in the cylinder, no relation is available between the displacement and pressure and no prediction about the value of $\dot{C}_{max}$ can be made. Neither is any information available at this point concerning the piston area.

One approach would be to assume the second term to be constant. But even then its value cannot be defined.

A feasible way to handle this situation is to neglect the term at this stage, then come back later and define it when information is available so that at least an approximate value of the term is possible.

Therefore, making this assumption

$$\bar{\dot{C}} \simeq \bar{W}\,\bar{P}_c^{\,-1/K} \tag{16}$$

Now consider the forces acting on the piston, let

$$F = (M_p' + M_l')\,\ddot{C} + B_p C + F_{cp}\frac{\dot{C}}{|\dot{C}|} + K_s C$$

Then the summation of the pressure forces on the piston can be written as:

$$K_a\, A\, P_t + (1-K_a)\, A\, P_{atm} - A\, P_c = F$$

At this stage no direct relation between P_t and P_c is available. Therefore it will be assumed that P_t is constant throughout the actuator piston displacement, therefore let $P_t = P_{tmax}$.

and

$$F = K_a\, A\, P_{tmax} + (1-K_a) A\, P_{atm} - A\, P_c \tag{17}$$

Defining

$$F_{max} = P_{tmax}\, A, \text{ and } \bar{F} = \frac{F}{F_{max}}$$

equation 17 becomes

$$\bar{F} = \bar{P}_{atm}\,(1-K_a) + K_a - \bar{P}_c \tag{18}$$

Using equations 16 and 18 a velocity-force characteristic for the system shown in Fig. 1 can be plotted (Fig. 6). But at the same time it should be remembered that some error will exist because of neglecting the second term in the expression for $\bar{\dot{C}}$ as given by equation 15.

The maximum power output point on the velocity-force characteristic can be determined mathematically using equations 16 and 18 if it lies in the region of critical flow for the valve. Otherwise it has to be determined by computing $\bar{\mathbb{P}} = \bar{F}\,\bar{\dot{C}}$ for the values of $\bar{F}$ near the maximum power output point obtained from Fig. 6, by drawing "$\mathbb{P}$ = constant" curves.

C. CALCULATION OF VALVE AND CYLINDER SIZE

As previously stated, after the individual load and actuator velocity-force characteristics are obtained the designer's work is to match the two curves at their maximum power point. From the load velocity-force characteristic the velocity and force at the maximum power point can be obtained as previously described. From the actuator velocity-force characteristic the velocity and force at the maximum power point can be obtained as previously described. From the actuator velocity-force characteristic the velocity and force at the maximum power are defined in terms of valve and piston parameters.

If maximum power points are matched

$$F(\mathbb{P} = \max)_{load} = F(\mathbb{P} = \max)_{actuator}$$

$$F(\mathbb{P} = \max)_{load} = \frac{F(\mathbb{P} = \max)_{actuator}}{F_{max}} \cdot F_{max}$$

$$= \bar{F}(\mathbb{P} = \max)_{\dot{A}ctuator}\, A\, P_{tmax}$$

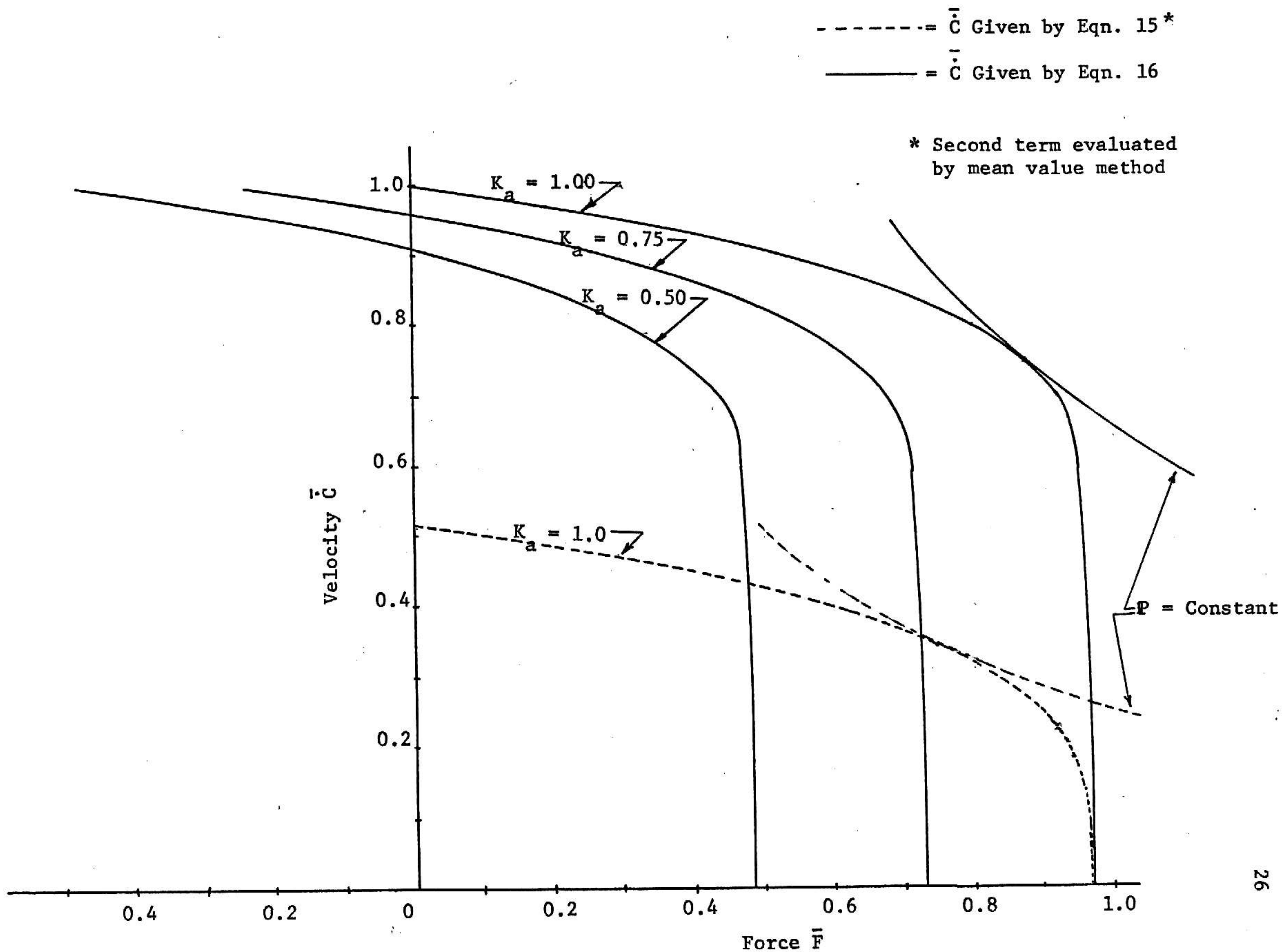

Fig. 6. Actuator Velocity-Force Characteristics

which gives

$$A = \frac{F(\mathbb{P} = \max)_{load}}{\bar{F}(\mathbb{P} = \max)_{actuator}\ P_{tmax}} \tag{19}$$

also

$$\dot{C}(\mathbb{P} = \max)_{load} = \dot{C}(\mathbb{P} = \max)_{actuator}$$

or

$$\dot{C}(\mathbb{P} = \max)_{load} = \bar{\dot{C}}(\mathbb{P} = \max)_{actuator}\ \dot{C}_{max}$$

substituting $\dot{C}_{max} = W_{max}/\rho_{tmax}\ A$

and $\qquad W_{max} = 13.45\ C_d\ A_o \sqrt{\rho_{tmax}\ P_{tmax}}$

$$A_o = \frac{A\ \dot{C}(\mathbb{P} = \max)_{load}\ \ \rho_{tmax}}{13.45\ \bar{\dot{C}}(\mathbb{P} = \max)_{actuator}\ C_d\ P_{tmax}} \tag{20}$$

Equations 19 and 20, after substituting value of $\bar{\dot{C}}(\mathbb{P} = \max)$ and $\bar{F}(\mathbb{P} = \max)$ obtained from the valve and piston velocity-force characteristic, and substituting values of C_d, ρ_{tmax} and P_{tmax}, yield the equations of A and A_o in terms of $\dot{C}(\mathbb{P} = \max)_{load}$ and $F(\mathbb{P} = \max)_{load}$. And using the values of velocity and force at the maximum power point on the load velocity-force curve, the valve and piston size can be determined.

D. "IMPROVED" VALUES FOR VALVE AND CYLINDER SIZE

Now it is possible to "improve" the values of A and A_o resulting from use of equations 19 and 20 by estimating the second term, in the equation 15, which was previously neglected. The second term of the equation 15 is

$$\frac{\dot{\bar{P}}_c(C_{cmax}-C)}{K\,\bar{P}_c\,\dot{C}_{max}}$$

Value of this term will be changing during a response depending upon the values of $\bar{P}_c$, $\dot{\bar{P}}_c$, and C. However over a considerable portion of the response a drastic variation in the value of the second term seems unlikely. This may be explained as follows.

Consider $\bar{P}_c$ and $\dot{\bar{P}}_c$. When $\bar{P}_c$ is high the flow rate through the control valve will also be high, therefore $\bar{P}_c$ will drop rapidly and $\dot{\bar{P}}_c$ will be high. When $\bar{P}_c$ is low the flow rate will also be low, therefore $\bar{P}_c$ will drop slowly and $\dot{\bar{P}}_c$ will be low. Consider $\bar{P}_c$ and $(C_{cmax}-C)$. C will increase slowly when $\bar{P}_c$ is high, then increase more rapidly as $\bar{P}_c$ decreases. Therefore $(C_{cmax}-C)$ decreases slowly when $\bar{P}_c$ is high, then decreases more rapidly as $\bar{P}_c$ decreases. Therefore the value of this term may be expected to vary slowly during an actuator response.

Hence approximating the second term of equation 15 by a constant, in order to "improve" the values of A and A_o, appears to be a feasible approach to take.

The value of the second term in equation 15 can be estimated by using two different methods.

1. MEAN VALUE METHOD:

The value of the second term in equation 15 can be estimated assuming that $\bar{P}_c$ varies linearly with time. This gives good approximation for $\dot{\bar{P}}_c$ and a mean value of $\bar{P}_c$. Also the mean value of C is used in calculating $(C_{cmax}-C)$.

$\dot{C}_{max}$ is a reference term and its value depends on A_o and A. Increasing $\frac{A_o}{A}$ will increase $\dot{C}_{max}$. This is also true for the maximum velocity of the piston $\dot{C}_{Amax}$, during an actual response. For the same value of piston area A, if control valve area A_o is increased, the flow through the control valve will increase and the response will be faster. Therefore $\dot{C}_{Amax}$ will increase.

Using the values of A and A_o estimated, when the second term in the equation 15 was neglected, the corresponding values of $\dot{C}_{max}$ and $\dot{C}_{Amax}$ can be obtained and denoted as $\dot{C}_{max1}$ and $\dot{C}_{Amax1}$. Now assuming that any change in $\dot{C}_{max}$ gives a corresponding change in $\dot{C}_{Amax}$, the value of $\dot{C}_{max}$, to be used in the estimation of the second term, can be obtained as follows:

$$\dot{C}_{max} = \frac{\dot{C}_{max1}}{\dot{C}_{Amax1}} \cdot (\dot{C}_{Amax})$$

where $(\dot{C}_{Amax})$ is expected value of maximum piston velocity during the system response. Since the system is expected to have the assumed response, discussed in section A of Chapter III, $(\dot{C}_{Amax})$ is substituted by the maximum velocity calculated in the assumed response.

2. MAXIMUM POWER METHOD.

Equation 15 is used to determine the velocity-force characteristic of the actuator and for determining velocity at maximum power, which is of main importance as far as the calculations of the piston and valve sizes are concerned. Therefore if value of the second term could be estimated at the maximum power point, it would seem logical that better design values for piston and valve could be obtained.

A possible way of determining the value of the second term is to consider the response obtained using the values of A and A_o determined when the second term was neglected. If the values of $\bar{P}_c$, $\bar{\dot{P}}_c$, C and $\dot{C}_{max}$ obtained at the maximum power point in this response are used, the value of the second term in equation 15 can be approximately estimated.

Equation 15 can now be used instead of equation 16 with a constant substituted for the second term on the right hand side of the equation. And the velocity-force curve can now be modified accordingly. For a given value of $\bar{P}_c$, $\bar{F}$ is same but the value of $\bar{\dot{C}}$ is changed by a constant. Therefore in new velocity-force plot the curve is shifted but the nature of the curve remains the same. The new velocity-force curve based on mean value method is also plotted in Fig. 6. for example problem in section F.

E. COMPUTER SOLUTION OF THE SYSTEM RESPONSE

The design procedure to determine size of valve and piston configuration, developed hereto, is based on theoretical analysis of the system using some simplifying assumptions. The validity of this design procedure can be determined by studying the response of a system having valve and piston sizes based on results obtained from the design procedure developed. Also any alterations and/or improvements in the procedure can be specified by comparative study between an assumed theoretical response and the actual response obtained. However, to build an actual system in practice and then to study its response would be an expensive, laborous and time consuming method, more so, when other reliable methods of determining a system response are more easily available.

Analog and digital computers using a simulation technique, or numerical technique,have been used satisfactorily, in many cases. Considering the nonlinearities involved, a digital computer can be very efficiently used to compute the response of a system.

1. Computer solution for the response of a system.

Response of the system studied in this thesis to a change in valve area A_o can be obtained by solving the simultaneous differential equations 1, 2 & 6. These differential equations can be solved by using any one of the different numerical techniques developed based on finite difference method (8, 9). These techniques differ in their accuracy and in the amount of computation work involved. As a rule of thumb a smaller step size of independent variable in the solution of differential equation will generally give a better accuracy. But it is not true in all cases and in some cases it may lead to a serious error being introduced in the solution, if a considerably small step size of independent variable is used. This phenomenon is known as numerical instability (8). For a numerically stable method, e.g. the Runge-Kutta method, a smaller step size will give a better accuracy. However a smaller step size means more computations for a given interval of independent variable, and this may lead to inaccuracy. Beyond a certain step size, the optimum value of step size, it is not possible to obtain more accuracy by making step size smaller. This is because, in case of digital computer, round off errors and computer inherent errors may be present at every step and this may result in serious errors being introduced in the solution. Therefore an optimum step size of the independent variable should always be used to get maximum accuracy. However an optimum step size cannot be predicted beforehand, therefore comparison of results using different step sizes seems to be a possible way of determining the optimum step size. If no appreciable change in results (with respect to desired accuracy) is observed, then the step size used need not be changed. The computer program, in Fortran IV, for computing the response of the system studied in this thesis is given in Appendix A.

F. SOLUTION OF AN EXAMPLE PROBLEM

A general design procedure for determining the valve and piston sizes for an air blast circuit breaker actuating system has been set up in this chapter. This design procedure will now be verified and studied further by considering and working out an example problem.

The data required for the example problem can be divided into two types. The first type of data is related to the rating and capacity of the circuit breaker, and cannot be altered. The other data may be given or assumed and can be changed according to the requirements. The remaining parameters can be computed, using the design data, as previously discussed in this chapter.

For the example problem the desired data and corresponding values used are given below:

Fixed Data:

1. Critical displacement time, T_{sp} = 0.0167 sec.
2. Critical displacement of the contacts, C_f = 2.44 in.
3. Supply pressure, P_{tmax} = 384.7 psia

Assumed Data:

1. Mass of piston and contact, $M_P + M_1 = 12$ lbm*
2. Damping ratio, $\zeta = 0.4$
3. Ratio of piston area on high pressure side to the piston area on "spring" side, $K_a = 1.0$
4. Ratio of clearance volume to piston area, $C_{cmax} = 3.66$ in.

For these assumed values of the parameters given above the other parameter values obtained are:

Natural frequency, $\omega_n = 129.51$ rad/sec

Damped frequency, $\omega_d = 118.7$ rad/sec

Damping coefficient, $B_P = 3.2211 \ \frac{\text{lbf-sec*}}{\text{in}}$

Spring constant, $K_s = 521.46$ lbf/in

Using the method discussed in section B of chapter II the maximum power required during the desired ideal step response is given by:

Force at maximum power, $F(P = \max)_{load} = 1275.4$ lbf

and Velocity at maximum power $\dot{C}(P = \max)_{load} = 190.6$ in/sec

Using these parameter values and the design procedure discussed in section C of chapter II, the valve and piston sizes necessary were calculated to be:

Area of actuator piston, $A = 3.77$ sq in.

Area of the control valve, $A_o = 0.1298$ sq in.

These values were corrected using the mean value method discussed in section D of chapter II. The corrected values of valve and piston sizes

---

*It should be noted that the value of mass and damping coefficient depends on actuator actually used. However it will be assumed that the assumed values can be realized in actual practice. If not, then the values may be changed when such information is available.

are:

Area of actuator piston $A = 4.446$ sq in.

Area of control valve $A_o = 0.3345$ sq in.

The design procedure will now be evaluated by comparing the step response of a system having the calculated piston and valve areas, and the assumed mass and damping with the desired step response characteristic, i.e. that having $T_{sp} = 0.0167$ sec. This will be done in the next chapter of this thesis.

CHAPTER IV

EVALUATION OF DESIGN PROCEDURE

In chapter III the size of valve and piston for the system considered in this thesis was determined for an example problem using the design procedure developed. The response of this system, employing a numerical technique to solve its differential equations, was computed on an IBM 360/50 digital computer. The computed response is shown in fig. 7. The assumed response for the system is also shown in the same figure. With the help of the two responses the design procedure developed can be evaluated.

Before studying the system response the following point should be noticed. In any circuit breaker there are three critical parameter values of importance, as far as the mechanical operation is considered.

1. A minimum contact displacement is required during "open operation" previously referred to as "critical displacement" in this thesis and denoted by C_f.
2. Time taken by the contacts to first react the critical displacement from its "close" position, previously referred to as "critical displacement time" and denoted by T_{sp}.
3. The velocity of the contacts measured between two specified points.

These three parameter values play an important role in determining the capacity and rating of a circuit breaker.

Considering the above mentioned points it can be seen that it is less important to study the latter part of the response and therefore attention is only focused on the first part of the response, i.e. from initial

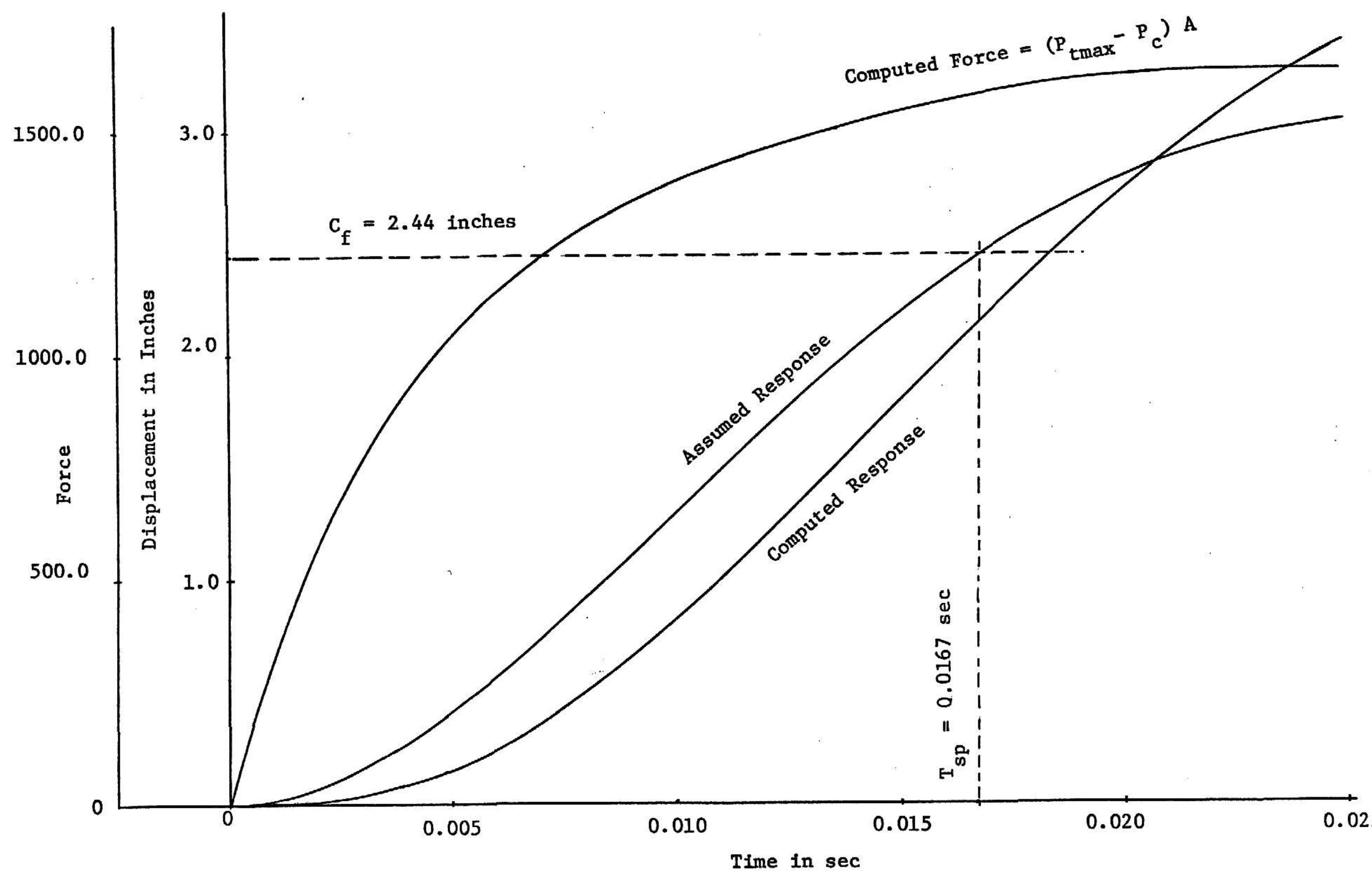

Fig. 7. Assumed Response, Computed Response and Computed Force Curves of the System

position of no displacement to the point when piston (and contact) displacement first equals "critical displacement". In the example problem the critical displacement was assumed to be 2.44 inches.

In the assumed system response the piston reaches this displacement at the specified critical time, T_{sp}. But in computed responses studied hereafter, the time taken by the piston to first reach the critical displacement can be different and it will be denoted as T'_{sp}.

Taking the assumed response as a reference, a delay (lag) or advance (lead) in time for a computed response can be determined and plotted for all values of displacement. In this plot the assumed response will lie on the displacement axis, and therefore comparative study between the assumed response and computed response can be easily made. This plot can further be simplified by normalizing difference in time, lag or lead, by the critical displacement time, T_{sp}, and by normalizing displacement by the critical displacement C_f. This plot of error in response against displacement for the response in Fig. 7 is shown in Fig. 8. Delay in response time, with respect to the assumed response, is shown negative as "lag" and advance in response time, with respect to the assumed response, is shown positive as "lead".

From Figure 7 it can be seen that time required for the piston to first reach critical displacement, T'_{sp} is 0.0183 sec., more than the desired critical displacement time T_{sp} = 0.0167 sec. Fig. 7 and 8 show that there is a time difference between the assumed response and the computed response, and the computed response is lagging. This can be explained by following reasons:

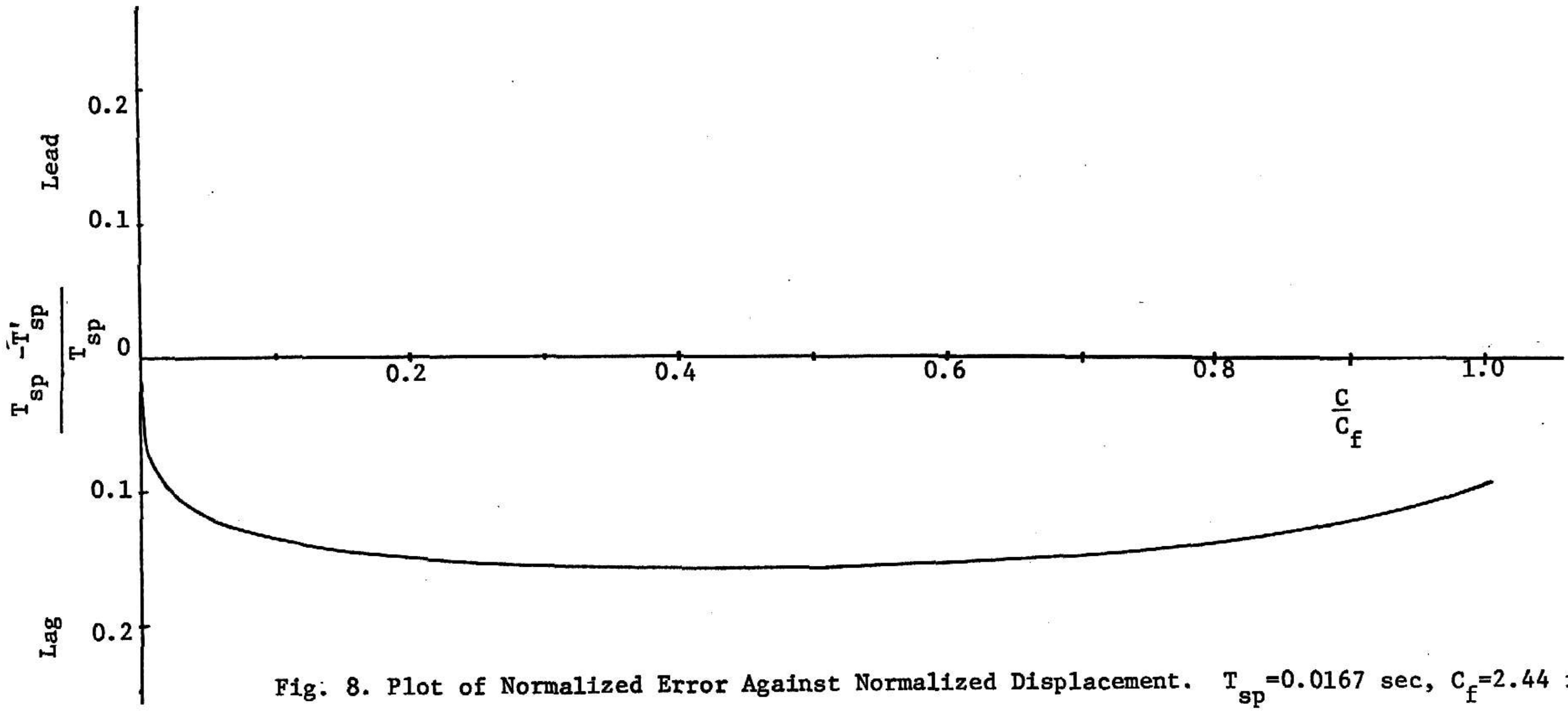

Fig. 8. Plot of Normalized Error Against Normalized Displacement. T_{sp}=0.0167 sec, C_f=2.44 inch.

1. The force curve in Fig. 7 shows that at time $t = 0+$ the force is zero and it builds up as time increases, thus requires a finite time to build up to its maximum value. This shows that the step input in force assumed for the assumed response is not obtained and hence an error is introduced in the input so that the computed response lags behind the assumed response.
2. Estimation of A and A_o in the design procedure developed has some error in it since the estimation of second term in the equation 14, using mean value method, is not exact.
3. There may be some error introduced by computer in computing system response. However the error due to the computer is not considerable when compared with other errors.

From Fig. 7 it can be anticipated that a faster rise time for the force exerted on the piston would probably give a better approximation towards a step input and therefore would probably give a response which is faster and closer to the assumed response. Since the pressure force acting on the piston is the result of the pressure difference $P_t - P_c$ across the piston, a faster rise time for the pressure force can be obtained by a faster drop in cylinder pressure P_c. This can be obtained by increasing the ratio $\frac{A_o}{A} = \bar{A}_o$.

Jung studied the effect of change in ratio of valve area to piston area on the response of a system where the valve was between the supply tank and the cylinder (4). He pointed out that increase in $\bar{A}_o$ gives a faster response but there is a practical limit for the value of the ratio $\bar{A}_o$. If $\bar{A}_o$ is increased beyond this practical limit the reduction in response time is less and less significant. Fig. 9 shows response curves for different values of $\bar{A}_o$ for the system studied by Jung.

For the example problem studied here the values for A_o and A determined by the design procedure used have an $\bar{A}_o$ ratio of 0.075. Different responses, keeping area of the piston the same as obtained from the design procedure, but for different values of A_o are shown in Fig. 10 It is observed that changing $\bar{A}_o$ from 0.075 to 0.1, response time T'_{sp} reduces by 8.34%. Increasing $\bar{A}_o$ further from 0.1 to 0.2 and then from 0.2 to 0.3 correspondingly reduces the response time by 11.8% and 3.34%.

If normalized error in system response time at critical displacement, i.e. $\frac{T_{sp} - T'_{sp}}{T_{sp}}$, is plotted against ratio of piston area to valve area, $\frac{A}{A_o}$ (i.e., $\bar{A}_o^{-1}$), it gives the plot shown in Fig. 11. It is observed that the curve is almost a straight line and for practical purpose it can be assumed to be a straight line between $\bar{A}_o^{-1}$ values of 3 to 15. (i.e. $\bar{A}_o$ values of 0.33 to 0.066). This covers a practical working range of piston area to valve area ratios. This graph can be used to correct $\bar{A}_o$ for desired responses, keeping in mind that curve plotted for any particular set of working conditions will be parallel to the curve shown in Fig. 11.

The curve plotted in Fig. 11 gives normalized error for different $\bar{A}_o^{-1}$ values for the example problem. Using the value of $\bar{A}_o^{-1}$, when normalized error is zero, gives the value of $\bar{A}_o \approx 0.105$. Therefore if the same piston area, as obtained from the design procedure, is used with $\bar{A}_o \approx 0.105$, it gives a response with a critical displacement time ≈ 0.0167, same as the specified critical displacement time T_{sp}.

This result can be used to improve design procedure so that the response obtained will be within acceptable limits. However at this stage

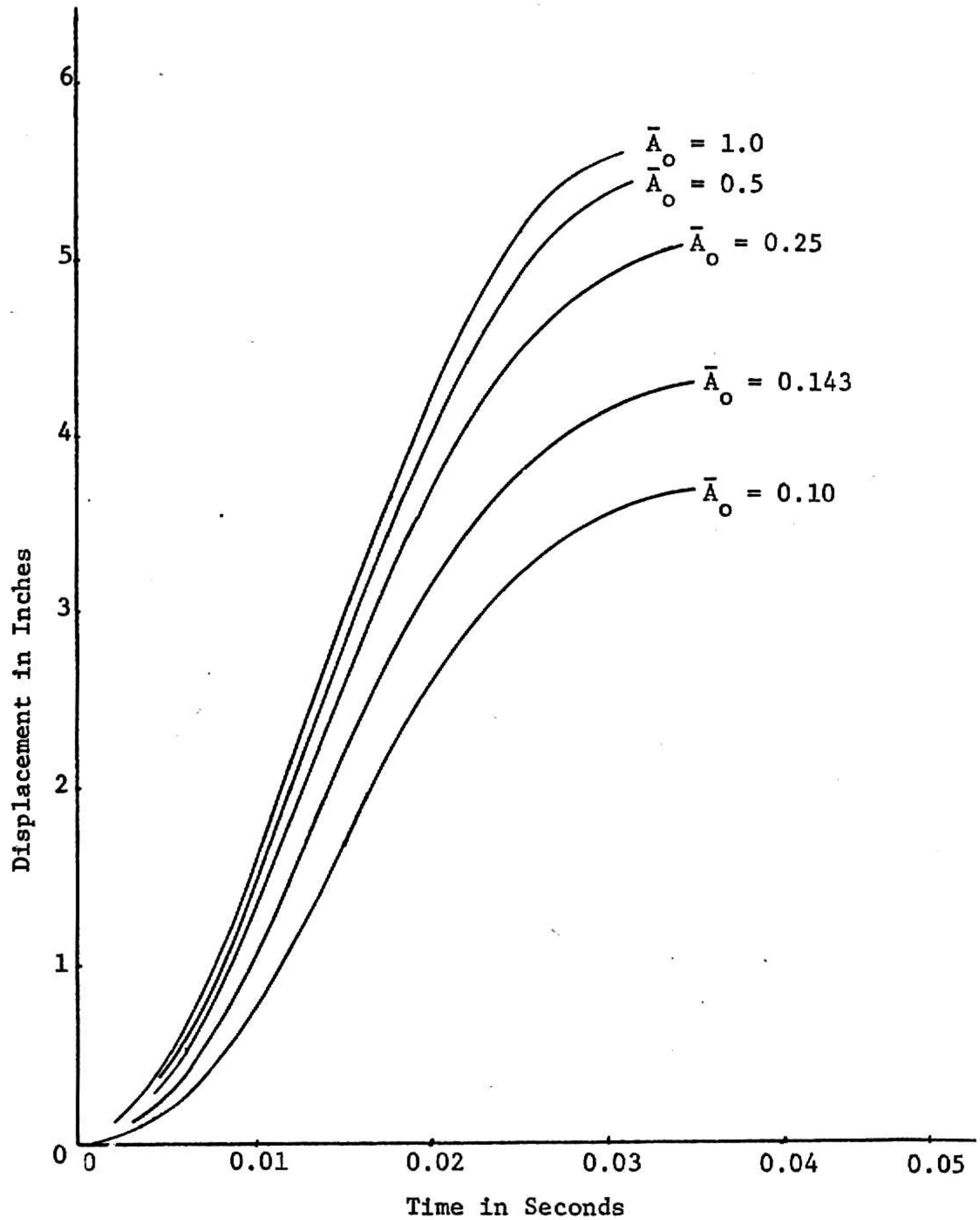

Fig. 9. Actuator-Displacement Characteristics.

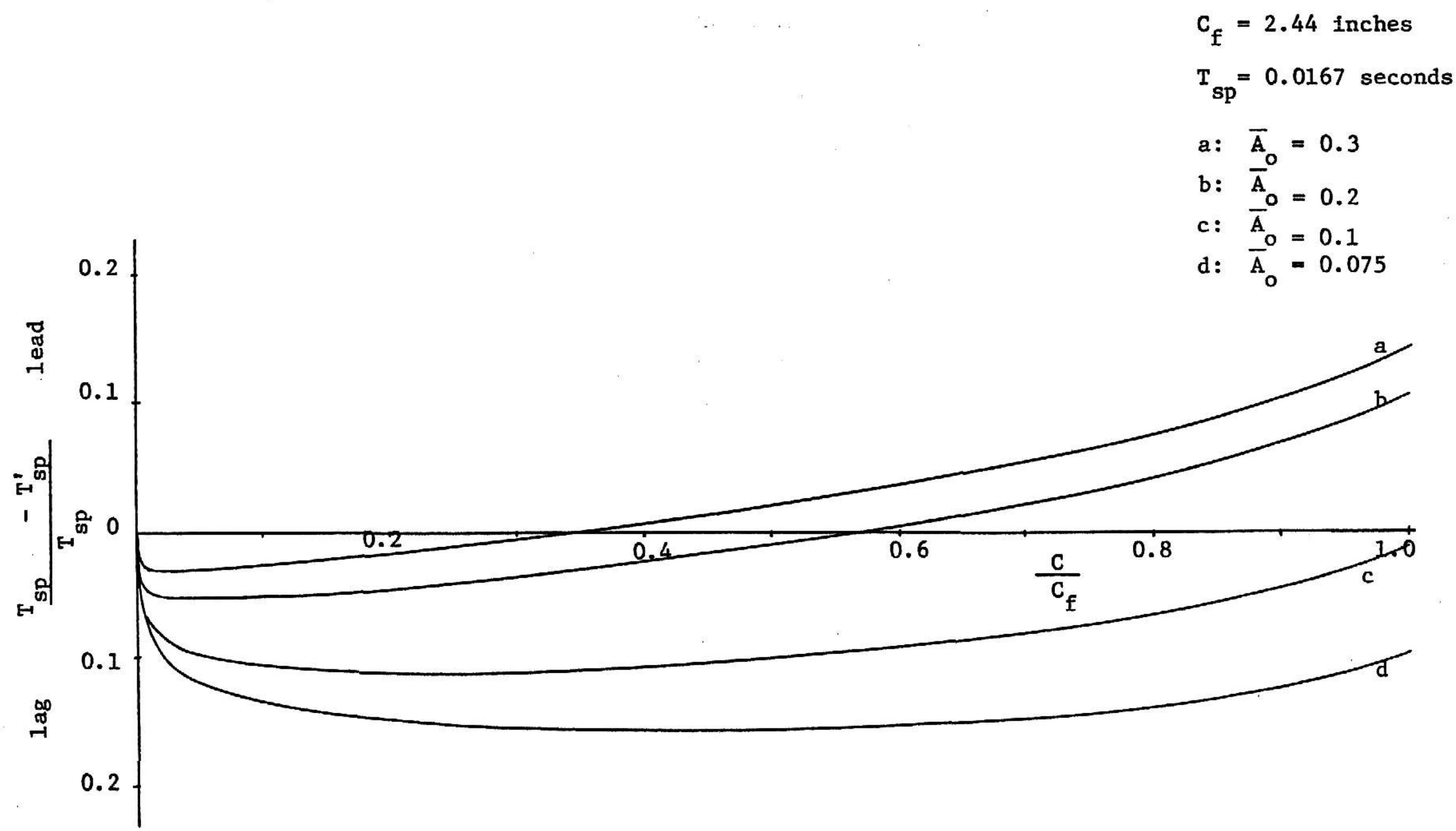

Fig. 10. Plot of Normalized Error Against Normalized Displacement for Different Values of $\bar{A}_o$.

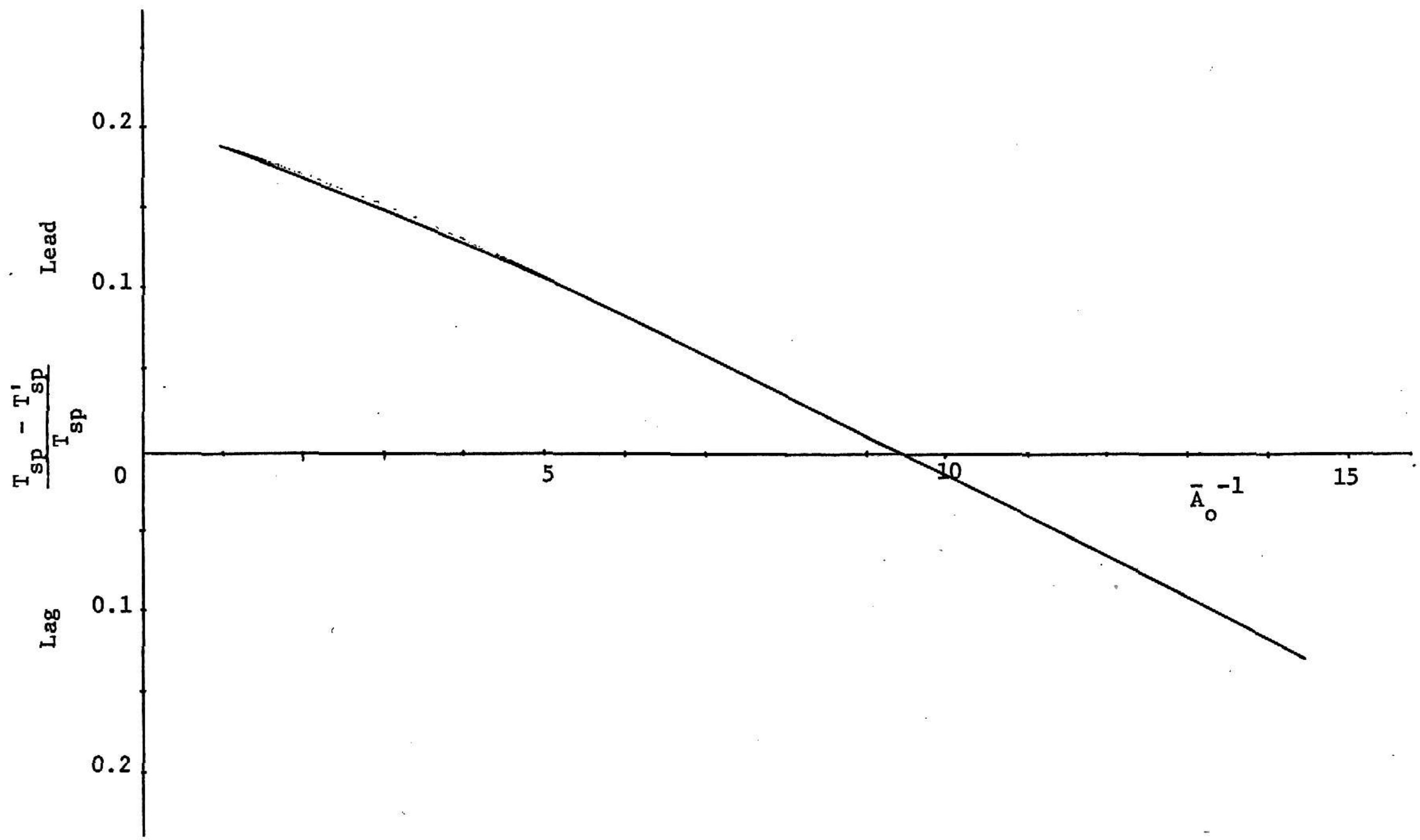

Fig. 11. Plot of $(T_{sp} - T'_{sp})/T_{sp}$ Against A/A_o For $A = 4.445\ in^2$ and $T_{sp} = 0.0167$ sec.

it is not possible to predict what value of $\bar{A}_o$ will give satisfactory response for all conditions to be considered hereafter. But for uniformity and ease in comparing future results a value of $\bar{A}_o = 0.2$ will be used for all system responses computed.

It can be concluded that the design procedure developed in chapter III does not give a valve and piston combination which has the desired second order response characteristic which was assumed. It gives a combination which has a slower response, but a faster response can be obtained using the same piston area and a larger valve area than given by the design procedure.

The example problem was worked out using a supply pressure, P_{tmax} of 384.7 psia, specified critical displacement time $T_{sp} = 0.0167$ sec, and mass of piston and load $M_p + M_l = 12$ lbm. Therefore the response obtained in this chapter and the results obtained from it apply only to this particular set of parameter values; and hence the results should be checked for other values of the parameters mentioned above. This will be done in the next chapter of this thesis.

CHAPTER V

EFFECT OF CHANGE IN DESIGN PARAMETER VALUES

In the previous chapter the system response of the valve and piston combination designed to satisfy an example problem was studied and it was observed that the design procedure developed, gives a piston and valve combination which, after modification of valve size, can give a satisfactory response. However the design procedure should be checked using different values of design parameters to study its validity over a wide range of parameter values. The main parameters of interest that one would anticipate having a dominant effect on the system behaviour include supply pressure, critical displacement time, mass, and clearance volume. The validity of the design procedure can be examined by applying the design procedure developed for determining piston area A for a higher value (twice the original value) and for a lower value (half the original value), for each parameter value used in the example problem. For uniformity $\bar{A}_o = 0.2$ will be used in all the cases, as previously mentioned.

A. EFFECT OF CHANGE IN DESIGN SUPPLY PRESSURE.

Fig. 12 shows the response of systems designed using different values of supply pressure. In all three cases all the parameter values, except the design supply pressure, are the same as used in example problem worked out in chapter III, section F.

Considering system response for a design supply pressure of 769.4 psia as opposed to 384.7 psia, it can be seen that for the higher design pressure the response is slightly faster. For a 100% increase in design supply pressure there is a 1.27% decrease in critical displacement time. For a decrease in design supply pressure from 384.7 psia to 192.35 psia, a 50% decrease in design supply pressure, the critical displacement time is

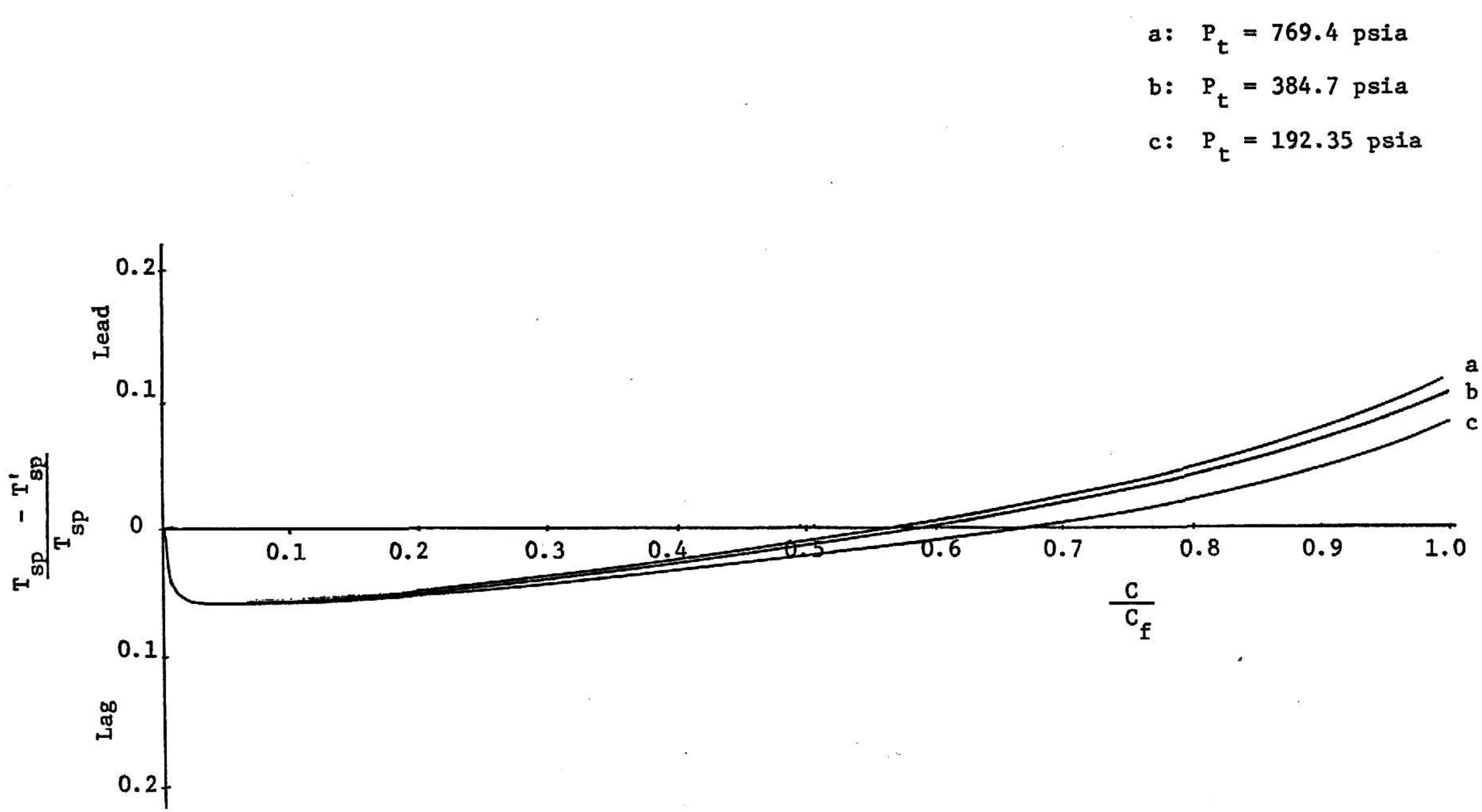

Fig. 12. Plot of Normalized Error Against Normalized Displacement for Different Design Supply Pressures. T_{sp} = 0.0167 sec. C_f = 2.44 in. $\overline{A}_o$=0.2

increased by 3.01%.

It should be noted that for lower values of supply pressure the piston area required increases. For a design supply pressure of 192.35 psia the required piston area is 8.76 sq in, but for a design supply pressure of 769.4 psia the piston area required is only 2.25 sq in.

This is expected since in all three cases the load force on the piston remains the same since the desired step response characteristic is the same for each case.

B. EFFECT OF CHANGE IN SPECIFIED CRITICAL DISPLACEMENT TIME.

In the design procedure developed the natural frequency of the actuating cylinder is determined by its specified critical displacement time, T_{sp}. An increase in specified critical displacement time gives a lower natural frequency; a decrease in critical displacement time gives a higher natural frequency.

Fig. 13 shows responses for systems with different specified critical displacement times. It can be seen that for higher values of specified critical displacement time the response is faster and vice versa.

Changing the specified critical displacement time from 0.0167 sec to 0.0334 sec, i.e. for 100% increase in specified critical displacement time, response time decreases by 5.06%. And by decreasing specified critical displacement time from 0.0167 sec to 0.00835 sec, 50% decrease in specified critical displacement time, response time increases by 13.15%.

It should be pointed out here, that increase in the specified critical displacement time decreases the piston area required and vice versa. Piston area, obtained from the design procedure developed, for the specified critical displacement time of 0.0167 sec is 4.45 sq in., whereas for the

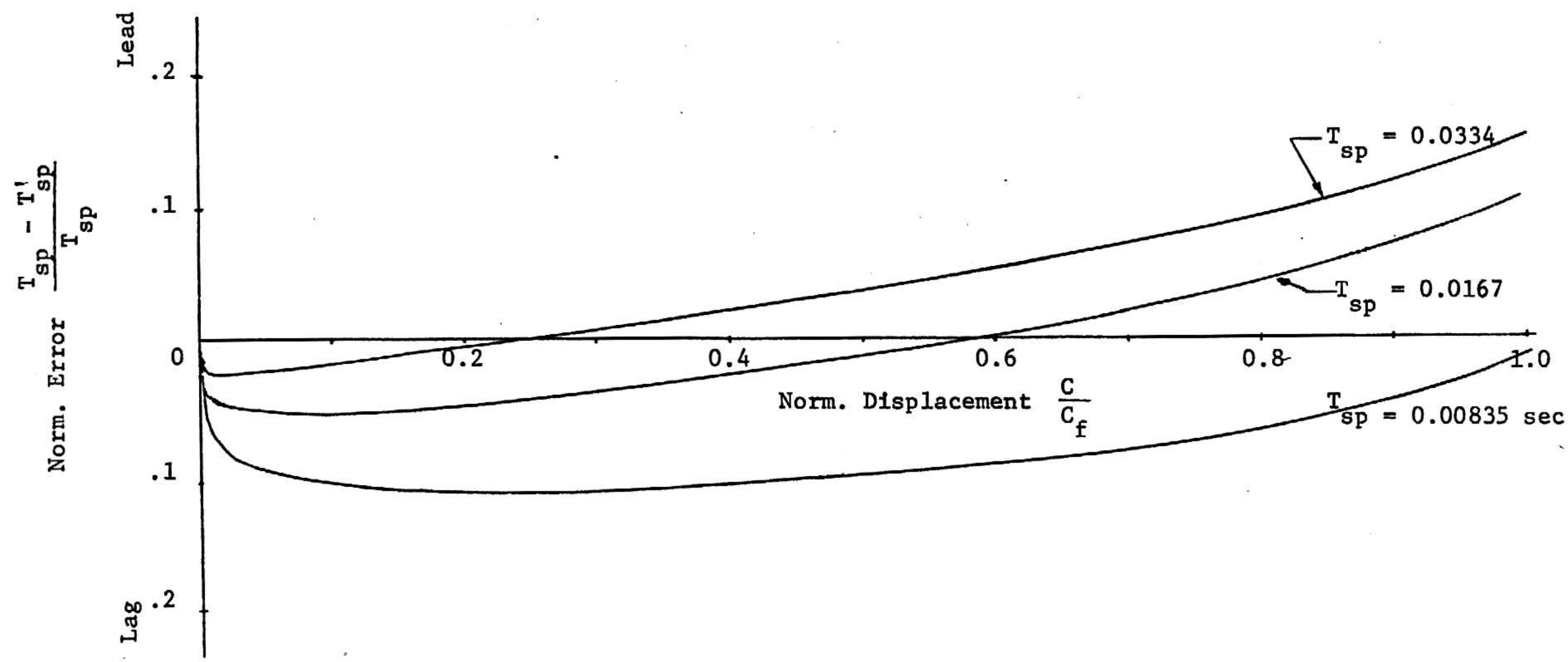

Fig. 13. Plot of Normalized Error Against Normalized Displacement for Different Critical Displacement Time. C_f = 2.44 in. $\overline{A}_o$ =0.2

specified critical displacement time of 0.0334 sec, the piston area obtained is 1.11 sq in. and for the specified critical displacement time of 0.00835 sec the piston area obtained is 17.78 sq in. This shows that the required piston area varies inversely as the square of the specified displacement time. This can be easily explained as follows:

Considering equation 12, at time $t = \infty$, shows that the force varies as the spring constant K_s, if the $C(\infty)$ is fixed; and equation 9 gives that spring constant, K_s, varies as the square of the natural frequency if the mass of load and piston remains constant. Hence the force F varies as the square of the natural frequency, or, if ζ is constant, F varies inversely as the square of the specified critical displacement time, T_{sp}. (i.e., $F \propto 1/T_{sp}^2$). For the same design supply pressure and actuator characteristics, equation 19 gives that piston area A varies with the force F which concludes that the piston area A varies inversely as the square of the specified critical displacement time, T_{sp}. (i.e., $A \propto 1/T_{sp}^2$).

C. EFFECT OF CHANGE IN RATIO OF CLEARANCE VOLUME TO SWEPT VOLUME

Fig. 14 shows that as the clearance volume to swept volume ratio, AC_{cmax}/AC_f, increases the response becomes faster. Increasing clearance volume to swept volume ratio from 1.5 to 2.0 gives 5.3% faster response. And decreasing the ratio from 1.5 to 1.25 gives a 1.62% slower response.

Before considering these results any further the effect of increasing clearance volume, keeping piston area the same, should be studied.

If piston area is kept the same and clearance volume is increased the cylinder pressure will drop more slowly (after the valve is opened), since for a given pressure drop more air has to be discharged through the

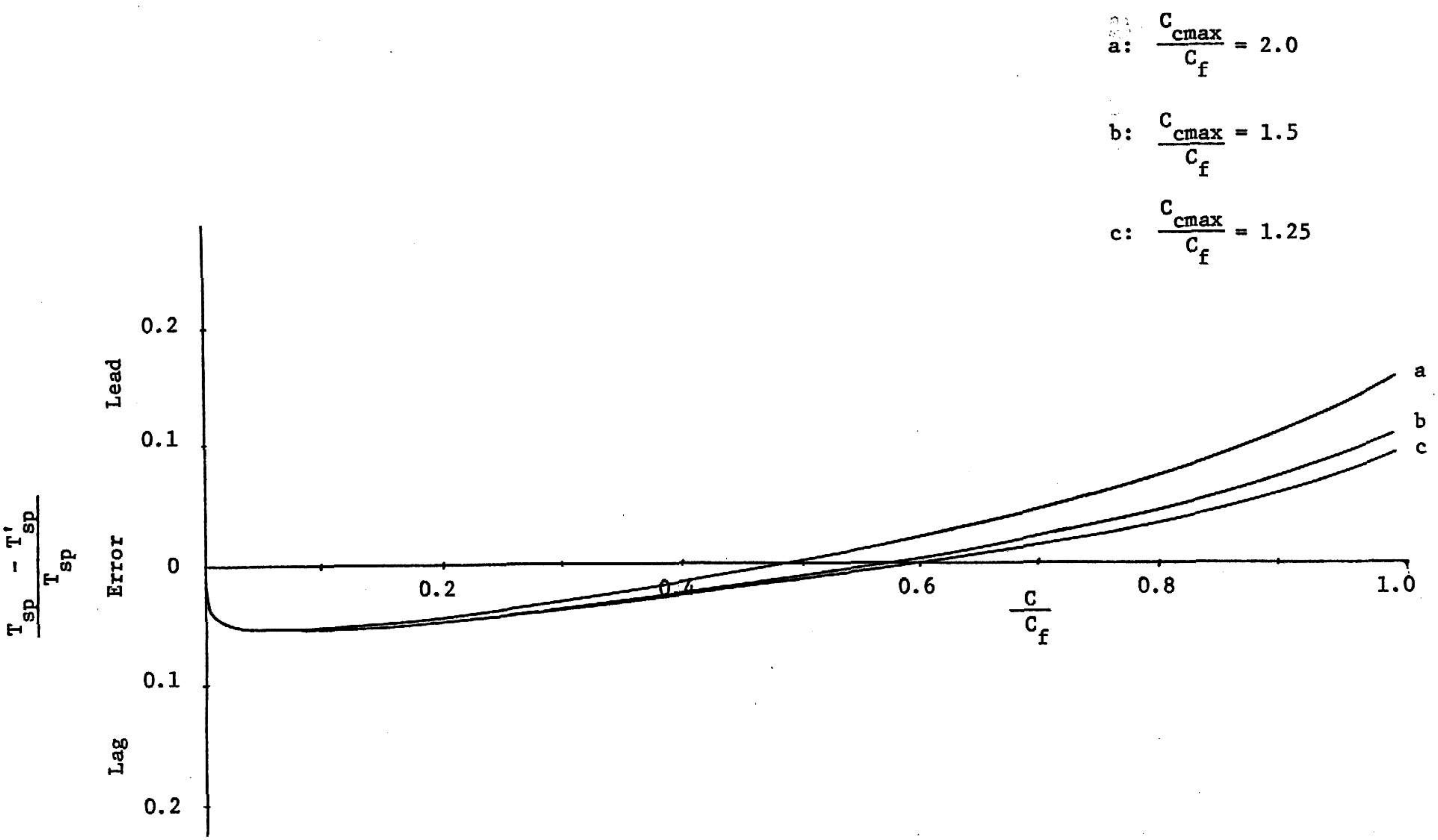

Fig. 14. Plot of Normalized Error Against Normalized Displacement for Different "$\frac{\text{Clearance Volume}}{\text{Swept Volume}}$" Designs. $T_{sp} = 0.0167$ sec. $C_f = 2.44$ in. $\overline{A}_o = 0.2$

valve, which requires more time. Therefore the effective pressure force on the piston will increase slowly resulting in a slower response. However the effect observed in Fig. 14 is contrary to this. This is because of the way that the second term in equation 15 is evaluated. The value of this term depends on the clearance volume to swept volume ratio and it increases with the ratio of clearance volume to swept volume. And the actuator characteristic changes with the value of the constant used to substitute the second term in the equation 15. This results in higher values of piston area and valve area if the ratio of clearance volume to swept volume is increased and hence a faster response for the same specified critical displacement time.

If specified critical displacement time, T_{sp}, and damping ratio, ζ, are kept the same, a change in load mass changes the inertia load force, spring constant and actuator damping, proportionally, therefore the piston area required also changes proportionally, but the response obtained remains the same.

It can be concluded, now, that changes in design values of all parameters changes the response of the system designed, (using the design procedure developed) and this effect should be considered in the design procedure. This will be discussed further in Chapter 7.

For all the responses studied in this chapter the supply pressure is assumed to be constant, also the valve is assumed to open instantaneously. However both these assumptions may not hold true in practice and hence the effect of pressure drop during a response, and effect of finite valve opening time on the system response will be studied in the next chapter.

CHAPTER VI

EFFECT OF CHANGE IN SUPPLY PRESSURE - AND VALVE OPENING TIME

A. SUPPLY PRESSURE

The supply pressure was assumed to remain constant to simplify the mathematical analysis in the design procedure developed for the valve and piston combination studied in this thesis. However in actual practice a drop in supply pressure may occur during a system response. This may occur due to expansion of supply air into the actuator cylinder, if the cylinder volume is considerable compared with supply tank volume; or it may occur due to some other reasons, as in an Allis Chalmers type ABM air blast circuit breaker where supply air is used to extinguish the arcs formed during the breaker "open" operation (1). This consumption of air produces a considerable drop in supply pressure and affects contact actuating system response. In some cases the supply pressure is not recharged to its initial value before another system operation takes place. In this case there will be an initial supply pressure drop in addition to the supply pressure drop during a system response.

The effects of these supply pressure drops are studied in this chapter. In the following cases studied the supply pressure drops assumed are taken from the actual pressure drops observed is an "Allis Chalmers type ABM air blast circuit breaker (3). Table 1 shows initial and final supply pressure of the final open operation during different duty cycles of the circuit breaker.

DUTY CYCLE	INITIAL AND FINAL SUPPLY PRESSURE FOR FINAL OPEN OPERATION
1. Open	384.7 psia - 349.7 psia
2. Open-Close-Open	347.7 psia - 324.7 psia
3. Close-Open-Close-Open	324.7 psia - 302.7 psia

Table 1: Initial and Final Supply Pressures For Different Circuit Breaker Duty Cycles.

The exact nature of the decrease in supply pressure during a system response is not known. However since it occurs in a matter of milliseconds it may be assumed to occur adiabatically. The desired adiabatic pressure drop can be realized by using the appropriate supply tank volume. For a sufficiently small supply tank volume the expansion of the higher pressure air from the supply tank into the cylinder will create a pressure drop in the supply tank. If the process is assumed to be adiabatic, an equation for final supply tank pressure can be obtained in terms of initial supply tank pressure, supply tank volume and swept volume of the piston, as:

$$P_{tf} = P_{tmax}\left(\frac{V_t}{V_t + AC_{max}}\right)^K$$

rearranging, substituting K = 1.4, and solving for V_t gives:

$$V_t = \frac{A\ C_{max}}{\left[\left(\frac{P_{tmax}}{P_{tf}}\right)^{1/1.4} - 1.0\right]} \qquad (21)$$

Equation 21 was used to determine a supply tank volume which will give an adiabatic pressure drop in supply tank pressure from a desired initial supply tank pressure to a desired final supply tank pressure.

Fig. 15 shows the responses for the three cases given in Table 1. along with a system response for a constant supply pressure of 384.7 psia. For all cases studied the system was actually designed assuming a constant supply pressure of 384.7 psia, and $\bar{A}_o$ was same, i.e. 0.2. Other system specifications are the same as for the example problem in Chapter III.

Studying Fig. 15 shows that a supply pressure drop during response slows the response. This is expected, since a drop in supply pressure reduces pressure force across piston which means reduction in input, compared with a response without supply pressure drop, and hence a slower response. For a supply pressure drop from 384.7 psia to 349.7 psia, i.e. 9.1% drop in supply pressure, there is a 2.34% increase in critical displacement time.

For initial drop in supply pressure the response will be slower since the pressure force across the piston will never reach the maximum value attained during a response without any supply pressure drop. There will be more delay in response if supply pressure is allowed to drop further during system response. This is confirmed by the system response shown in Fig. 15 for an initial and final supply pressure values of 347.7 psia and 324.7 psia respectively. This gives a response which has a 10.38% increase in critical displacement time, when compared with the critical displacement time for a constant supply pressure of 384.7 psia. Also the slowest response which occurs during a duty cycle, for the lowest pressures mentioned in Table 1, is also shown in Fig. 15. In this case delay in critical displacement time is 17.6% when compared to the critical displacement time for a constant supply pressure of 384.7 psia.

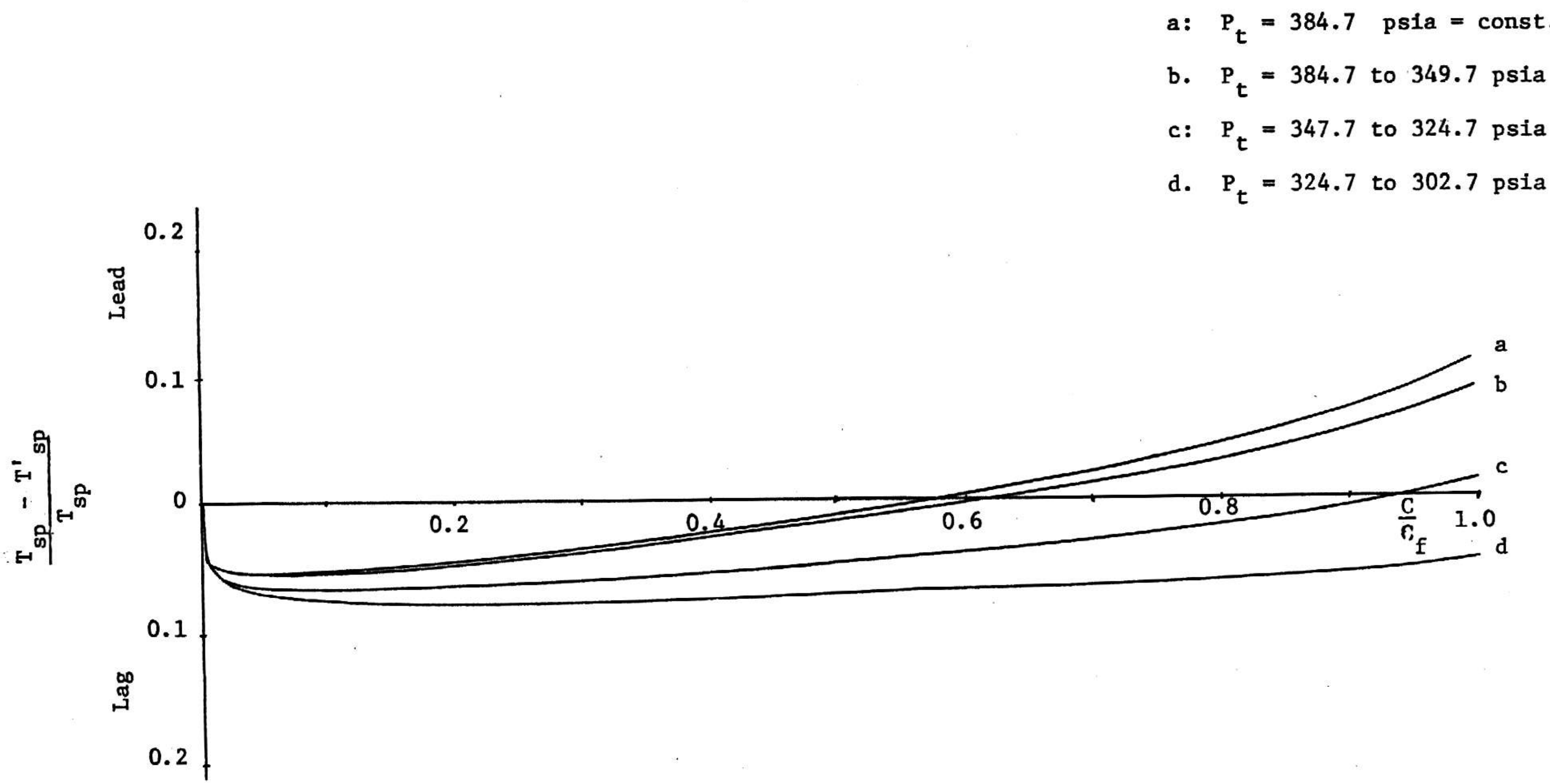

Fig. 15. Plot of Normalized Error Against Normalized Displacement for Different Values of Supply Pressures. T_{sp} = 0.0167. sec. $\overline{A}_o$ = 0.2

B. VALVE OPENING TIME

Any control valve whether hydraulically, pneumatically or electrically operated requires a finite time for its movement from a closed position to a fully open position. The design of a valve actuating mechanism, like solenoid or pneumatic actuator, is not dealt with in this thesis. The valve actuating mechanism in an air blast circuit breaker is an independent unit by itself and improvement in its response time has been accomplished by some manufacturers. Reduction in response time from 11.7 to 5 milliseconds has been achieved (10). An opening time as short as possible is desirable but, of course, zero opening time is impossible.

To start with, a finite valve opening must be allowed and taken into consideration in the design of valve and piston configuration. In this thesis, however, the effect of valve opening has been neglected thus far (it has been assumed that the valve opens instantaneously) while the effects of other parameters on system response have been considered. Now the effect of valve opening time will be considered and studied.

For this study it will be assumed that valve displacement with time is linear such that at time $t = 0+$, the effective valve area is zero and at time $t = t_{vo}$ the valve is fully open, as described by the following equations:

$$A_o(t) = A_{omax} \cdot \frac{t}{t_{vo}} \qquad \text{for } 0 < t < t_{vo}$$

and (3 repeated)

$$A_o(t) = A_{omax} \qquad \text{for } t \geq t_{vo}$$

Fig. 16 shows the plot of error in displacement time against normalized displacement for four different values of t_{vo}, namely $t_{vo} = 0.0\ T_{sp}$, $t_{vo} = 0.1\ T_{sp}$, $t_{vo} = 0.2\ T_{sp}$ and $t_{vo} = 0.4\ T_{sp}$, and shows the effect of valve opening time on system response. It shows that, in the latter part of the system response (for all valve opening times), the curves are separated from each other by a constant delay in response time. This delay in response time depends on the valve opening time t_{vo}. For valve opening times of 10%, 20% and 40% of T_{sp}, the corresponding increased delay in critical displacement time is 4.61%, 8.71% and 16.0% of T_{sp} when compared to a critical displacement time for zero valve opening time. If a response curve with zero valve opening time is taken as basis, a plot of normalized delay in response time against normalized valve opening time can be constructed. This plot is shown in Fig. 17 and may be used to estimate the effect of valve opening time on critical displacement time.

Fig. 18 shows a response where a drop in supply pressure during a response, and a finite valve opening time occurred at the same time. This response shows that for a pressure drop from 384.7 psia to 349.7 psia and for a valve opening time of 0.2 T_{sp}, the increased delay in critical displacement time is 10.78% of T_{sp} when compared to critical displacement time for a constant supply pressure of 384.7 psia and zero valve opening time. This equals the delay caused by drop in supply pressure, i.e. 2.07% T_{sp}, plus the delay caused by finite valve opening time, i.e. 8.71% T_{sp}, when each was considered separately.

Therefore it can be concluded that the delay in critical displacement time due to drop in supply pressure and finite valve opening time considered at the same time, can be obtained by adding the delays caused by these two effects when considered separately.

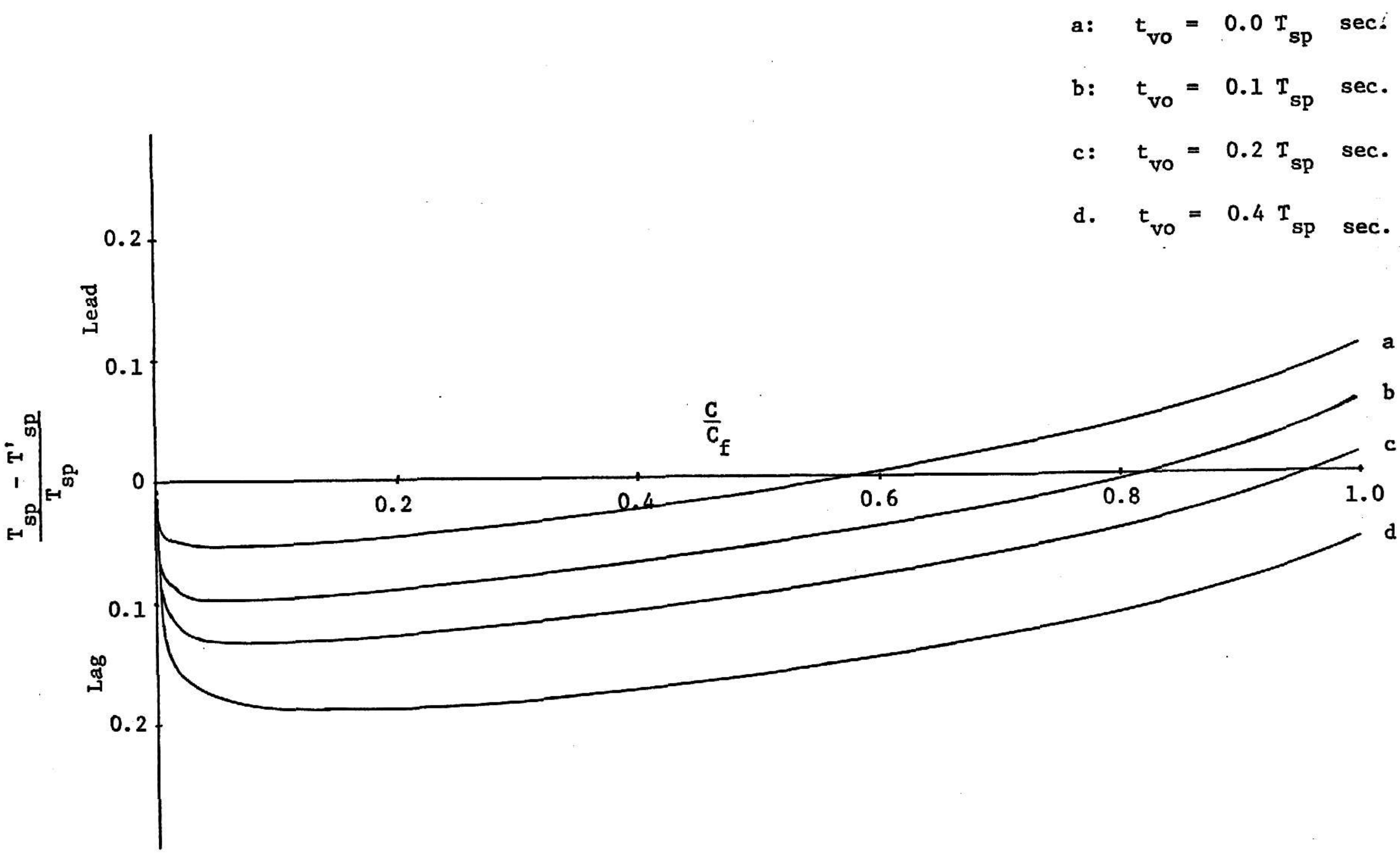

Fig. 16. Plot of Normalized Error Against Normalized Displacement for Different Valve Opening Times.
T_{sp} = 0.0167, C_f = 2.44 in. $\overline{A}_o$ = 0.2

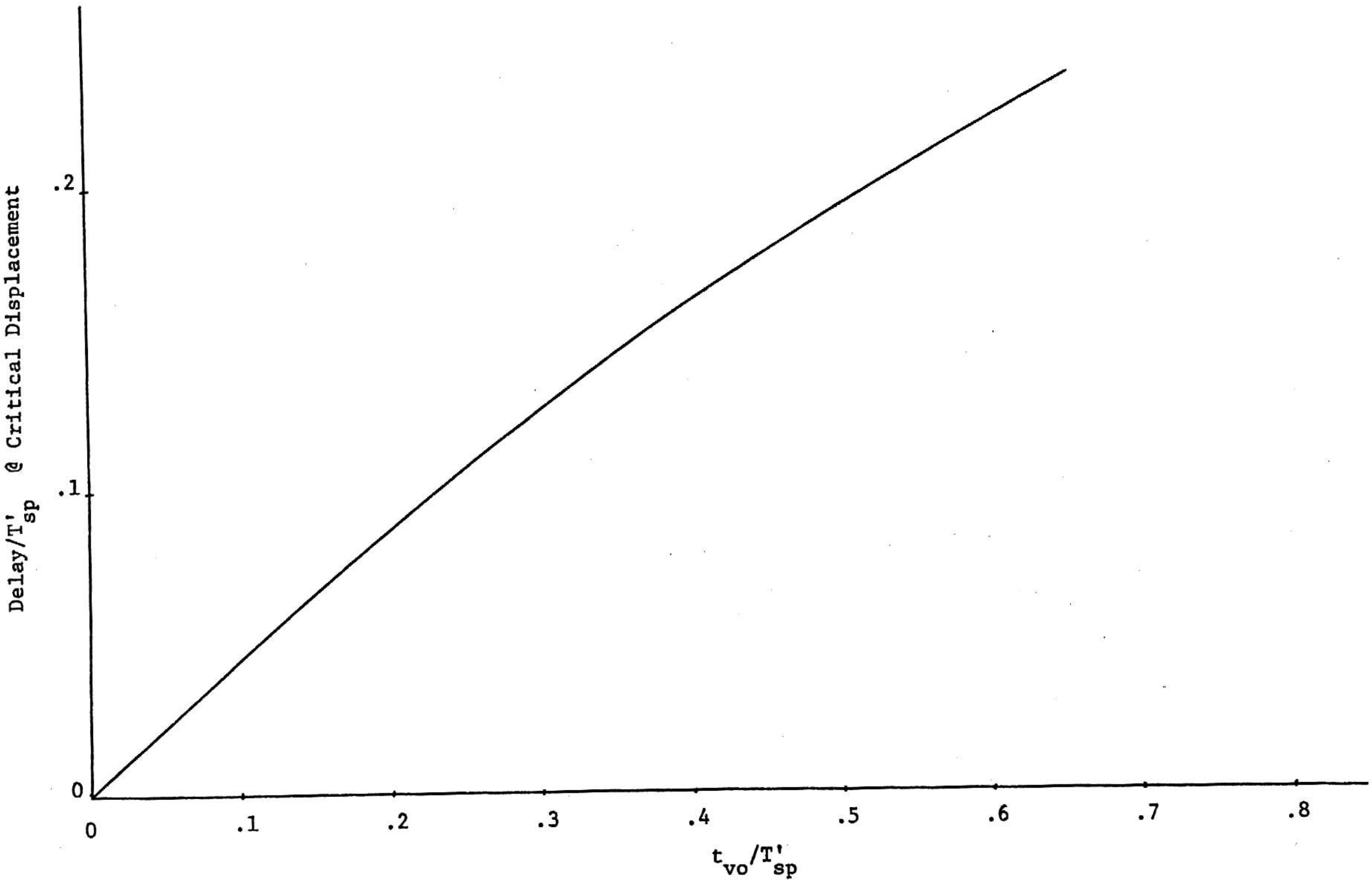

Fig. 17. Plot of Normalized Delay Against Normalized Valve Opening Time

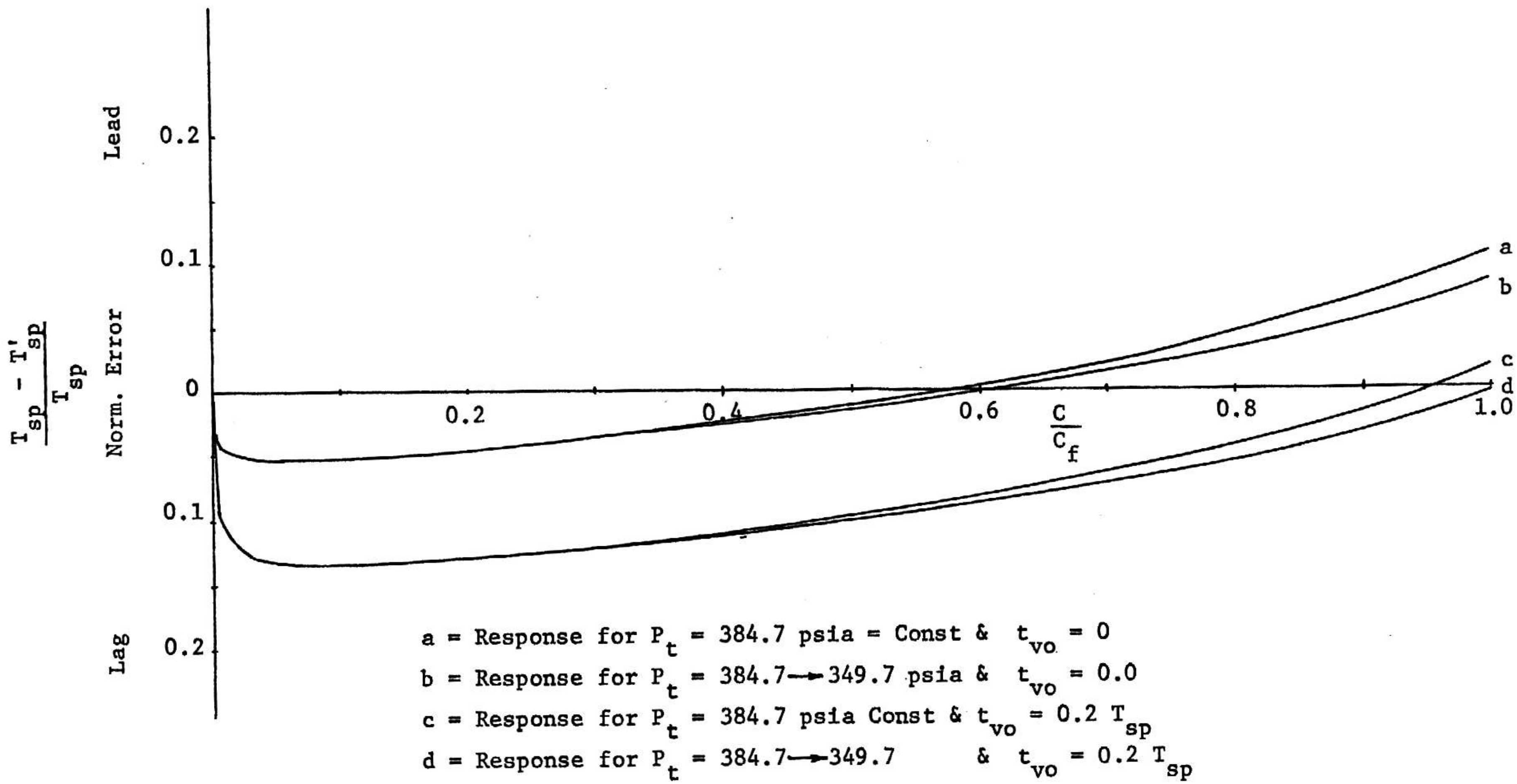

Fig. 18. Plot of Normalized Error Against Normalized Displacement for Different Conditions.

$\overline{A}_o = 0.2$

The delay due to drop in supply pressure both before and during system response can be compensated in the design procedure by two different ways; first, to base the design of the system on the lowest value of supply pressure attained, and second, to base the design on a corrected desired critical displacement time. If the design is based on the lowest supply pressure reached, then the response time of the system for this supply pressure value will be within close limits of the specified critical displacement time and all higher supply pressure responses will be faster thus giving critical displacement times less than the specified critical displacement time. However at higher supply pressures the pressure forces on the piston will be more and a considerable increase in maximum displacement reached is unavoidable. But if attention is to be focused only on the critical displacement of the piston and system response till this point, the design procedure based on the lowest supply pressure is acceptable.

In case of a "corrected" critical displacement time design procedure, the original specified critical displacement time can be reduced by a certain amount of time to allow for delay which will occur due to reduction in supply pressure. Therefore the response obtained from a system designed using a "corrected" critical displacement time will give a response that will be slower than the assumed "corrected" response, but will be within acceptable limits as far as the original specified critical displacement time is concerned.

The delay caused by finite valve opening time can also be compensated for by using the "corrected" critical displacement time technique described above. Using Fig. 17, the approximate delay time at critical displacement due to finite valve opening time t_{vo} can be obtained. Subtracting this amount from the original design value for critical displacement time a new design value for critical displacement time can be obtained. The system designed using this "corrected" critical displacement time will give a response within acceptable limits. The delay in critical displacement time due to both, drop in supply pressure and finite valve opening time, can be considered simultaneously using this procedure to realize the desired critical displacement time.

It should be obvious by now that the response of the valve and piston combination, considered in this thesis, is never the ideal step response of a second order system;primarily because a finite time is required for cylinder pressure P_c to drop down to near atmospheric and a finite time required for valve opening. This introduces error since the force input acting on the system is never a true step input. Thus assuming an ideal step input response as a desired response in the design procedure will always give some error.

If a design procedure is based on the assumption that the piston actually responds to a force input which is either ramp or parabolic in nature (as shown in Figure 19), to allow for the effect of slow drop in cylinder pressure and finite valve opening time, then better results may be possible. The primary difficulty in such an approach is in being able to specify the time at which the ramp or parabolic portion becomes a constant input. This time is approximatley the same as the time required for cylinder pressure to drop down to near atmospheric.

The assumed response would be obtained by solving equation (12) where F(t) is either a ramp or parabolic forcing function. Using this assumed response in the design procedure would result in new values of force and velocity at maximum power. The final result would be new values of valve and piston area. This approach should be studied to further improve the design procedure developed in this thesis.

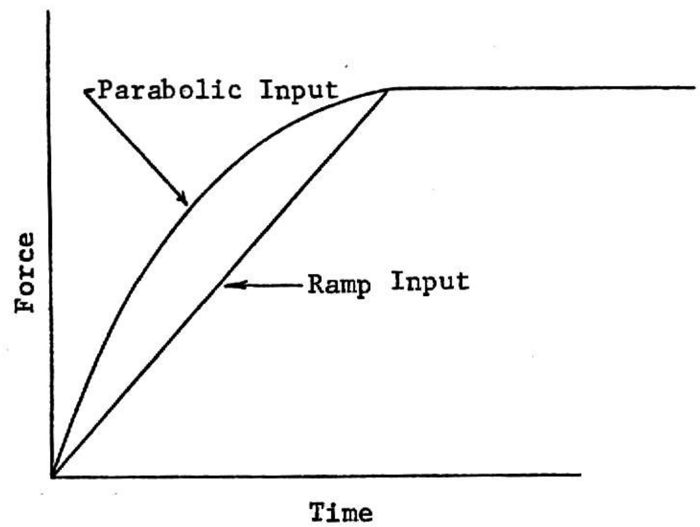

Fig. 19. Suggested Types of Input to the System, Shown in Fig. 1, for Further Study.

CHAPTER VII

SUMMARY OF DESIGN PROCEDURE

After studying the effect of valve opening time and drop in supply pressure on the system response, it was shown that the design procedure can be improved to compensate for these effects. Also any change in the response due to change in other design parameters can be anticipated, and valve size could be changed for compensating these effects. In the design of a new system, therefore, the use of the results obtained in this thesis is necessary and important.

As mentioned previously, the design data, which must be specified in order to solve a given problem, includes critical displacement of the piston or contacts, critical displacement time, supply pressure, valve opening time and supply pressure drop.

The other parameter values that need to be estimated or used in the design procedure are left to the designer's choice. Previous experience or knowledge will be useful in estimating values of these parameters which include clearance volume to swept volume ratio, $\frac{C_{cmax}}{C_f}$, damping ratio, mass of the load and piston and damping of the cylinder.

The design procedure will now be summarized:

1. With the specified value of valve opening time estimate the time delay which will result using Fig. 17 and subtract it from the specified critical displacement time. (The value of specified critical displacement time should be used for T'_{sp} in Fig. 17.)
2. Estimate the time delay due to drop in supply pressure using Fig. 15 for the value of critical displacement time resulting from step 1.
3. Subtract the time delay, obtained from step 2, from the critical displacement time, resulting from step 1, to get a "corrected"

value of critical displacement time to be used in the design procedure.

4. Use the design procedure described in Chapter III to determine the required actuator parameters.

5. Estimate the effect on response, when $C = C_f$, of changes in initial design value of supply pressure, critical displacement time and clearance volume to swept volume ratio, from their reference values as used in this thesis i.e., $P_{tmax} = 384.7$, $T_{sp} = 0.0167$ sec, and $\frac{C_{cmax}}{C_f} = 1.5$, (using Fig. 12, 13 and 14 respectively), to get a total time lag or lead as a fraction of critical displacement time.

6. Add the result of step 5 above to every point on the curve in Fig. 11 to get a new curve which gives a normalized error in response against $\bar{A}_o^{-1}$ for the system using the actuator designed in step 4.

   The value of $\bar{A}_o^{-1}$, where the new curve cuts the abscissa, gives the correct value of $\bar{A}_o^{-1}$ to be used.

7. Since A is already known (step 4) use result of step 6 above to calculate value of A_o.

The actuator and valve combination thus obtained should give a response close to or faster than the desired response at critical displacement. However it should be noted that it is still possible to make the response faster, if necessary, by increasing the valve size.

CHAPTER VIII

CONCLUSIONS

In this thesis the design of a valve-actuator system to give a second order transient response was studied. The actuating system used in an Allis-Chalmers air blast circuit breaker was considered as a typical system and was studied for parameter values of the same order of magnitude as found in such a circuit breaker.

It was observed that an exact second order transient response is not possible for this case and therefore the design procedure developed for calculating valve and actuator dimensions giving the desired response was not possible. However a system can be designed, if all other conditions are specified, to give a specified displacement in a specified time. It was observed that this system cannot give the specified displacement in the same specified time if any of the specified conditions are changed. However it is possible to design a system giving the specified displacement in less or the same time as the specified time if the range of parameter values which change during the system response can be estimated in advance.

In the course of developing a design procedure the following conclusions were reached. These conclusions are valid for the system shown in Fig. 1.

1. A drop in supply pressure, during the response, gives a slower response, but it is not considerable. For 384.7 psia supply pressure a drop of 35 psia during the response gives only 2.31% increase in critical response time.
2. Lower values of supply pressure give a considerably slower response if the system was originally designed for a higher supply pressure.

3. For the same piston area, faster response is obtained if the valve area is increased. For the same decrease in A/A_o, (or $1/\bar{A}_o$) decrease in response time is approximately the same.
4. Change in load inertia force (in this case the mass of piston and contacts) changes the required piston area, spring constant and damping coefficient proportionally (if the critical displacement, C_f, critical displacement time, T_{sp}, and supply pressure P_{tmax} are the same) and the response remains the same.
5. The other parameter values remaining the same, the required piston area and spring constant vary inversely with the square of the critical displacement time, T_{sp}, while damping varies inversely as critical displacement time.
6. The effect of valve opening time and drop in supply pressure on system response, considered simultaneously, can be given by the total effect of valve opening time and drop in supply pressure considered separately (If the valve opening curve is similar to a ramp input).
7. The effect of finite valve opening time (ramp input) is a constant time lag in later part of the response compared with a response where valve opening time is assumed to be zero.

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APPENDIX A

FORTRAN IV PROGRAM

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                RESPONSE CHARACTERISTICS OF ACTUATORS USED IN
                        AIR BLAST CIRCUIT BREAKERS
••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••••
         THIS PROGRAM USES FOURTH ORDER RUNGE KUTTA METHOD TO SOLVE THI
      INITIAL VALUE PROBLEM  WHICH GIVES THE RESPONSE OF THE ACTUATOR

 200 FORMAT(5F16.8)
 201 FORMAT(I5)
 501 FORMAT(1H-,5X,'A=AREA OF THE ACTUATOR PISTON=',F10.5)
 502 FORMAT(1H-,5X,'AO=AREA OF THE VALVE=',F10.5)
 503 FORMAT(1H-,5X,'VCO=CLEARENCE VOLUME=',F10.5)
 504 FORMAT(1H-,5X,'VT=HIGH PRESSURE AIR TANK VOLUME =',F10.5)
 505 FORMAT(1H-,5X,'PTMAX=MAXIMUM TANK PRESSURE=',F10.5)
 506 FORMAT(1H1,5X,'**** AK=RATIO OF H.P.SIDE UPON SPRING SIDE AREA',
    1F10.5,'****')
 508 FORMAT(1H-,5X,'AOB=RATIO OF VALVE AREA UPON PISTON AREA=',F8.5)
 511 FORMAT(1H-,5X,'SK=SPRING CONSTANT=',F10.4)
 512 FORMAT(1H-,5X,'BP=DAMPING COEFF=',F10.4)
 513 FORMAT(1H-,5X,'WD=DAMPED FREQUENCY=',F10.4)
 514 FORMAT(1H-,5X,'WN=NATURAL FREQUENCY=',F10.5)
 515 FORMAT(1H-,5X,'VOT=VALVE OPENING TIME=',F8.5)
 516 FORMAT(1H-,5X,'PTF=FINAL TANK PRESSURE=',F8.3)
 517 FORMAT(1H-,5X,'DDT=2ND INCREMENT IN TIME=',F8.5)
  95 FORMAT('1'5X,1HT,12X,2HPT,14X,2HPC,12X,4HCDOT,11X,5HCDDOT,15X,1HC
    112X,4HCCAL,8X,5HFORCE)
  80 FORMAT(1H0,F7.5,7E16.8)
     PI=3.14159265
     XCONS=386.0*1.4*((125.0*125.0)/(216.0*216.0))
     YCON=SQRT(386.0*7.0)
     READ(1,201) J
     IREAD=1.0
 666 READ(1,200)DT,DK,PM,CM,C,CDOT,BP,SK,A,AO,PT,PC,AK,PTF,VOT
     READ(1,200) WD,ZETA,SI,CMAX,DDT
     READ(1,201) K
     RATM=0.433319/10000.0
     PATM=14.7
     KKOUNT=0.0
     KOUNT=28
     T=0.0
     PTMAX=PT
     CDDOT=0.0
     SAVET=0.0
     CCAL=0.0
     VCO=1.5*A*CMAX
     VT=AK*A*CMAX/(((PTMAX/PTF)**(1.0/1.4))-1.0)
     RT=RATM*(PT/PATM)
     RC=RATM*(PC/PATM)
```

```
      AOB=AO/A
      CCMAX=1.4*CMAX
      AOMAX=AO
      PCCRT=((PTMAX-PATM)*AK+PATM)
      CKOSSI=COS(SI)
      WN=WD/SQRT(1-ZETA**2)
      SIGMA=ZETA*WN
      DEFT=PI/WD
      WRITE(3,506)AK
      WRITE(3,501)A
      WRITE(3,502)AO
      WRITE(3,508) AOB
      WRITE(3,503)VCO
      WRITE(3,504)VT
      WRITE(3,505)PT
      WRITE(3,511) SK
      WRITE(3,512) BP
      WRITE(3,513) WD
      WRITE(3,514)WN
      WRITE(3,515) VOT
      WRITE(3,516) PTF
      WRITE(3,517) DDT
      WRITE(3,95)
      WRITE(3,80)T,PT,PC,CDOT,CDDOT,C,CCAL
      T=DT
      DO 10 N=1,K
      IF(C.GE.CMAX)  GO TO 998
      IF(SAVET.GT.DEFT) GO TO 998
      IF(T.GE.VOT) GO TO 19
      AO=AOMAX*T/VOT
      GO TO 18
   19 AO=AOMAX
   18 IF(N.EQ.KOUNT) GO TO 20
      GO TO 21
   20 WRITE(3,95)
      KOUNT=KOUNT+29
   21 RCOLD=RC
      PTOLD=PT
      PCOLD=PC
      CR=PATM/PC
      IF(CR-0.528)30,30,32
   30 W=DK*AO*SQRT(XCONS*RC*PC)
      GO TO 60
   32 S=1.0-(PATM/PC)**0.286
      IF(S)40,33,33
   33 W=SQRT(S*((PATM/PC)**1.428)*RC*PC)
      W=YCON*DK*AO*W
      GO TO 60
   40 S=1.0-(PC/PATM)**0.286
      W=-SQRT(S*((PC/PATM)**1.428)*PATM*RATM)
      W=YCON*DK*AO*W
   60 A1=-1.4*AK*A*CDOT*PT/(VT+AK*A*C)*DT
```

```
      B1=1.4*PC*(RC*A*CDOT-W)/(RC*(VCO-A*C))*DT
      C1=386.0/(PM+CM)*((PT-PATM)*AK*A-(PC-PATM)*A-BP*CDOT-SK*C)*DT
      D1=CDOT*DT
      IF(PC-PCCRT)121,70,70
   70 IF(C1.LT.0.0) C1=0.0
      IF(D1.LT.0.0)  D1=0.0
  121 RC=RCOLD*((PC+B1/2.0)/PCOLD)**0.714
      IF(CR-0.528) 130,130,132
  130 W=DK*AO*SQRT(XCCNS*RC*(PC+B1/2.0))
      GO TO 160
  132 S=1.0-(PATM/PC)**0.286
      IF(S)140,133,133
  133 W=SQRT(S*((PATM/(PC+B1/2.0))**1.428)*RC*(PC+B1/2.0))
      W=YCON*DK*AO*W
      GO TO 160
  140 S=1.0-(PC/PATM)**0.286
      W=-SQRT(S*(((PC+B1/2.0)/PATM)**1.428)*PATM*RATM)
      W=YCON*DK*AO*W
  160 A2=-1.4*AK*A*(CDOT+C1/2.0)*(PT+A1/2.0)/(VT+AK*A*(C+D1/2.0))*DT
      B2=1.4*(PC+B1/2.0)*(RC*A*(CDOT+C1/2.0)-W)/(RC*(VCO-A*(C+D1/2.0)))*
     1DT
      C2=386.0/(PM+CM)*(((PT+A1/2.0)-PATM)*AK*A-((PC+B1/2.0)-PATM)*A-BP*
     1(CDOT+C1/2.0)-SK*(C+D1/2.0))*DT
      D2=(CDOT+C1/2.0)*DT
      IF(PC-PCCRT)221,170,170
  170 IF(C2.LT.0.0)  C2=0.0
      IF(D2.LT.0.0)  D2=0.0
  221 RC=RCOLD*((PC+B2/2.0)/PCOLD)**0.714
      IF(CR-0.528) 230,230,232
  230 W=DK*AO*SQRT(XCCNS*RC*(PC+B2/2.0))
      GO TO 260
  232 S=1.0-(PATM/PC)**0.286
      IF(S)240,233,233
  233 W=SQRT(S*((PATM/(PC+B2/2.0))**1.428)*RC*(PC+B2/2.0))
      W=YCON*DK*AO*W
      GO TO 260
  240 S=1.0-(PC/PATM)**0.286
      W=-SQRT(S*(((PC+B2/2.0)/PATM)**1.428)*PATM*RATM)
      W=YCON*DK*AO*W
  260 A3=-1.4*AK*A*(CDOT+C2/2.0)*(PT+A2/2.0)/(VT+AK*A*(C+D2/2.0))*DT
      B3=1.4*(PC+B2/2.0)*(RC*A*(CDOT+C2/2.0)-W)/(RC*(VCO-A*(C+D2/2.0)))*
     1DT
      C3=386.0/(PM+CM)*(((PT+A2/2.0)-PATM)*AK*A-((PC+B2/2.0)-PATM)*A-BP*
     1(CDOT+C2/2.0)-SK*(C+D2/2.0))*DT
      D3=(CDOT+C2/2.0)*DT
      IF(PC-PCCRT)321,270,270
  270 IF(C3.LT.0.0)  C3=0.0
      IF(D3.LT.0.0)  D3=0.0
  321 RC=RCOLD*((PC+B3)/PCOLD)**0.714
      IF(CR-0.528) 330,330,332
  330 W=DK*AO*SQRT(XCCNS*RC*(PC+B3))
      GO TO 360
```

```
332 S=1.0-(PATM/PC)**0.286
    IF(S)340,333,333
333 W=SQRT(S*((PATM/(PC+B3))**1.428)*RC*(PC+B3))
    W=YCON*DK*AO*W
    GO TO 360
340 S=1.0-(PC/PATM)**0.286
    W=-SQRT(S*(((PC+B3)/PATM)**1.428)*PATM*RATM)
    W=YCON*DK*AO*W
360 A4=-1.4*AK*A*(CDOT+C3)*(PT+A3)/(VT+AK*A*(C+D3))*DT
    B4=1.4*(PC+B3)*(RC*A*(CDOT+C3)-W)/(RC*(VCO-A*(C+D3)))*DT
    C4=386.0/(PM+CM)*(((PT+A3)-PATM)*AK*A-((PC+B3)-PATM)*A-BP*(CDOT+
   1C3)-SK*(C+D3))*DT
    D4=(CDOT+C3)*DT
    IF(PC-PCCRT)400,370,370
370 IF(C4.LT.0.0)  C4=0.0
    IF(D4.LT.0.0)  D4=0.0
400 PT=PT+(A1+2.0*A2+2.0*A3+A4)/6.0
    PC=PC+(B1+2.0*B2+2.0*B3+B4)/6.0
    C=C+(D1+2.0*D2+2.0*D3+D4)/6.0
    CDOT=CDOT+(C1+2.0*C2+2.0*C3+C4)/6.0
    CDDOT=(C1+2.0*C2+2.0*C3+C4)/(6.0*DT)
    RC=RCOLD*(PC/PCOLD)**0.714
    FORCE=(PT-PATM)*AK*A-(PC-PATM)*A
    IF(PC-PCCRT) 13,13,11
 13 KKOUNT=KKOUNT+1
    IF(KKOUNT.EQ.1.0) GO TO 50
    GO TO 51
 50 T=T-DT
    PT=PT-(A1+2.0*A2+2.0*A3+A4)/6.0
    PC=PC-(B1+2.0*B2+2.0*B3+B4)/6.0
    C=C-(D1+2.0*D2+2.0*D3+D4)/6.0
    CDOT=CDOT-(C1+2.0*C2+2.0*C3+C4)/6.0
    RC=RCOLD
    DT=DDT
    GO TO 10
 51 IF(KKOUNT.EQ.2.0) SAVET=T
    CT=T-SAVET
    IF(CT.EQ.0.0) GO TO 14
    CCAL=CMAX*(1.0-COS(WD*CT-SI)/(EXP(SIGMA*T)*CKOSSI))
    GO TO 12
 14 PC=PCCRT
    C=0.0
    CDOT=0.0
 12 WRITE(3,80)CT,PT,PC,CDOT,CDDOT,C,CCAL,FORCE
    GO TO 10
 11 WRITE(3,80)T,PT,PC,CDOT,CDDOT,C,CCAL,FORCE
 10 T=T+DT
    IF(IREAD.EQ.J) GO TO 1000
    IREAD=IREAD+1.0
    GO TO 666
1000 STOP
    END
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ACKNOWLEDGEMENT

The author wishes to express his heartfelt gratitude to his major adviser, Dr. Ralph O. Turnquist, who provided the original idea and gave constant encouragement and helpful suggestions for this thesis.

Also he wishes to express his sincere thanks to Dr. Preston E. McNall, Head of the Department of Mechanical Engineering; Dr. Kenneth K. Gowdy, Assistand Dean of Engineering; and Dr. S. Thomas Parker, Professor of Mathematics for being a member of his advisory committee.

Finally he wishes to express his sincere thanks to Thomas F. Clark, Chief Engineer Development, Allis Chalmers for his invaluable cooperation and to Allis Chalmers Manufacturing Company for providing necessary funds for this project.

VITA

Appasaheb Tryambakrao Patil

Candidate for the Degree of

Master of Science

Thesis: RESPONSE CHARACTERISTICS OF ACTUATORS USED IN AIR BLAST CIRCUIT BREAKERS

Major Field: Mechanical Engineering

Biographical:

Personal Data: Born in Kolhapur, Maharashtra, India, August 14, 1945, the son of Shri and Smt. Trambakrao Krishnarao Patil.

Education: Attended Sadanand Vidhyalaya; graduated from Irwin High School, Kolhapur in 1961; received the Bachelor of Engineering degree from University of Poona, India with a major in Mechanical Engineering in 1967; completed requirements for Master of Science degree in February 1970.

Professional experience: Worked as an Assistant Lecturer in Mechanical Engineering in Government Polytechnic, Kolhapur, Maharashtra, India from January 1967 to August 1968.

RESPONSE CHARACTERISTICS OF ACTUATORS
USED IN AIR BLAST CIRCUIT BREAKERS

by

APPASAHEB TRYAMBAKRAO PATIL

B. E. (Mech.), University of Poona, Poona, India, 1967

AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1970

ABSTRACT

This thesis presents a comprehensive study of the response characteristics of actuators similar to those used in air blast circuit breakers. The study consists of an analysis of a typical actuator used in the air blast circuit breakers, development of a design procedure for determining the size of the piston and the valve of the actuator to give a desired response*, and evaluation of the design procedure by studying the response characteristics of an actuator having dimensions obtained using the design procedure.

A contact actuating system, similar to the one used in an Allis Chalmers ABM Air Blast Circuit Breaker, is used as a typical system for study. This system consists of a supply tank, a piston type double acting actuator, and an on-off type three-way control valve.

A mathematical analysis of the system is carried out to define the system by a set of simultaneous differential equations. A design procedure, based on considering the maximum power required, to determine piston and valve size for a desired response is established. This design procedure is used to solve an example problem of designing an actuator and valve to obtain a response characteristic which is similar to the step response of an underdamped second order system.

The response curves for different specified design conditions are computed using a numerical method. The design procedure is evaluated using different design values of the parameters having dominant effect on the system response.

The results indicate that the response time is significantly influenced

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*Reaching a desired displacement at a specified time by the control valve area. It is observed that a drop in supply pressure during system response slows the response, but the response is affected more by a lower supply pressure at the beginning of the response. It is observed that the effect of finite valve opening time (assuming the valve opening is a ramp input) is a constant delay in time in the system response. It is also observed that the effect of finite valve opening time and drop in supply pressure on the system response, considered simultaneously, is the sum of the effect of finite valve opening time and drop in supply pressure considered separately.

Finally, it is observed that the design procedure, modified by the results obtained in this thesis, can be used to specify valve and piston size of an actuator to give a desired displacement in specified time.