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COMPUTER AIDED DESIGN AND DYNAMIC SIMULATION OF  
HYDROSTATIC TRANSMISSION SYSTEMS

by

EROL POLAT

B.S. in M.E., The Technical University of Istanbul, 1981

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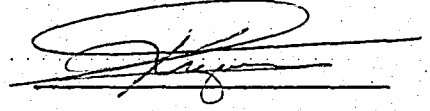
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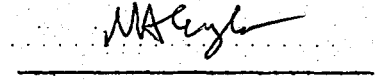
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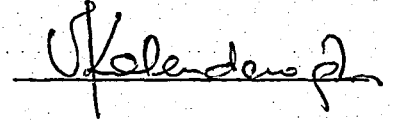
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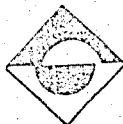


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Erol Polat

COMPUTER AIDED DESIGN AND DYNAMIC SIMULATION OF  
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ABSTRACT

In the present thesis, Computer Aided Design techniques are intended to be used for the design of Hydrostatic Transmission Systems and an user-friendly, interactive computer program package is developed for this purpose. Translational loads and cylinders are also considered so that the final program is an all purpose one.

In addition, a mathematical model is derived for the dynamic behaviour of Hydrostatic Transmission Systems by taking the pressure compensation into consideration as well, and another interactive program is developed for simulation purposes.

Each of the two programs may be used individually or together in an integrated manner. The User's Manuals are also offered to help the user.

The CAD program is used for some example applications and computer simulations are carried out for different cases. The obtained results are presented.

## HİDROSTATİK GÜÇ İLETİMİ SİTEMLERİNİN BİLGİSAYAR DESTEKLİ TASARIMI VE DİNAMİK BENZETİMİ

### KISA ÖZET

Bu çalışmada, Hidrostatik Güç İletimi Sistemlerinin tasarımı için Bilgisayar Destekli Tasarım tekniklerinin kullanılması amaçlanmış ve bunun için bir bilgisayar paket programı hazırlanmıştır. Genel amaçlı bir program olmasını sağlamak üzere, silindirlerin kullanıldığı hidrolik devreler de çalışmaya dahil edilmiştir.

Dinamik davranış gözetilmeden tasarımı yapılan sistemlerin sınırlı çalışabilecekleri göz önünde tutularak, oluşturulan modele uygun bir dinamik davranış benzetim programı, basınç kompansasyonu da dikkate alınarak geliştirilmiştir. Böylece birbirinden bağımsız veya birlikte, birbirini destekleyici olarak kullanılacak iki ayrı program paketi oluşturulmuş ve kullanıcı el kitapçıkları ile beraber kullanıma hazır duruma getirilmiştir. Her iki program da, Boğaziçi Üniversitesi Bilgisayar Merkezinde CDC Cyber 820 için Fortran 77 programlama dili ile kullanıcı etkileşimli olarak yazılmıştır.

Her iki program örnek uygulamalar için çalıştırılmış ve bazı parametrelerin etkisi de gözlemlenerek elde edilen sonuçlar sunulmuştur.

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## LIST OF SYMBOLS

A	System matrix
B	Input vector or Viscous friction coefficient shifted to the motor
$B_l$	Viscous friction coefficient of load
$B_r$	Viscous friction coefficient of reductor
C	Disturbance vector
$D_m$	Motor displacement (geometric delivery)
$D_p$	Pump displacement (geometric delivery)
$E_l$	Energy loss
J	Inertia shifted to the motor axis
$J_l$	Load inertia moment
$J_m$	Motor inertia moment
$J_{r1}$	Reductor inertia on the load side
$J_{r2}$	Reductor inertia on the motor side
$K_p$	Pressure compensation gain
$K_s$	Pump control coefficient
$K_q$	Pump control coefficient
m	Small variation in the shifted load torque
M	Torque shifted to the motor axis
$M_{fl}$	Dry friction torque of load
$M_{fr}$	Dry friction torque of reductor
$M_l$	Load torque

$M_m$	Motor output torque
$n_l$	Load revolution speed
$n_m$	Motor revolution speed
$N$	Gear ratio of reductor
$P_i$	Small variation in the motor inlet pressure
$P_i$	Inlet pressure
$P_o$	Outlet pressure
$\Delta P$	Pressure differentiation
$q_p$	Small variation in the pump flow rate
$Q_m$	Motor flow rate
$Q_p$	Pump flow rate
$Re$	Reynolds number
$t$	Time
$T$	Step size (sampling period)
$u$	Small variation in the pump control current or Input variable
$U$	Pump control current
$V$	Total oil volume in the delivery line
$W$	Mechanical power
$\beta_a$	Bulk modulus of air in oil
$\beta_e$	Equivalent bulk modulus
$\beta_o$	Bulk modulus of oil
$\beta_p$	Bulk modulus of pipe
$\eta_m$	Mechanical efficiency
$\eta_t$	Overall efficiency
$\eta_v$	Volumetric efficiency
$\nu$	Oil viscosity

$\phi$	Small variation in the swivel angle (swash plate angle)
$\Phi$	Swivel angle (swash plate angle)
$\rho$	Oil density
$\tau_p$	Time constant of pump control servomechanism
$\omega_m$	Small variation in the motor angular velocity
$\Omega_l$	Angular velocity of load
$\Omega_m$	Angular velocity of motor

## I. INTRODUCTION

Fluid Power is traditionally associated with the transmission of power by means of pressurized fluids in which the energy predominantly stored as potential rather than kinetic energy. Depending on the type of the fluid used, fluid power applications can be divided into two major groups; hydraulics (pressurized liquid), and pneumatics (compressed gas). In this study, primary attention is given to hydraulics.

Although Hydraulic Power Transmission has been known for a long time it did not gather much interest until recent decades. Nowadays, some unique advantages of Hydraulics Power Transmission together with important developments in manufacturing and control techniques have reawakened an enthusiastic interest in the subject all over the World. As a developing country, Turkey could not escape from this fact. At present in the industrial market of Turkey it is possible to see almost every kind of hydraulic equipment, imported or made in Turkey. Domestic production covers small percentage at the moment, but it is rapidly increasing. At this point it is clear that Turkey vitally needs vast amount of research and qualified men in every category of this field.

There are many application areas of hydraulics, including aircraft controls, flight simulators, numerically controlled machine tools, automation systems, robots, material handling and lifting, porting equipment, construction and agricultural machines, marine applications, etc. There are a lot of points in favour of hydraulic systems:

- high forces (torques) with compact size, i.e. high power density,
- automatic force adaptation,
- movement from standstill possible under full load,
- simple overload protection,



- stepless change (control or regulation) of speed, torque, stroke, force, etc. ,
- suitable for controlling fast movement process and for extremely slow precision movements
- relatively simple accumulation of energy.

In a special case of hydraulic systems in which the actuator is a hydraulic motor rather than a cylinder, the hydraulic pump and motor are specifically matched to work together; the power is transmitted through fluid lines and the system is called "Hydrostatic Transmission System".

Hydrostatic Transmission is preferred to Mechanical Transmission especially in systems where large power is required at varying levels, e.g. heavy duty machines, lifting and porting machines, marine engines and off-the-road vehicles. There is also an increasing trend toward the use of Hydrostatic Transmissions for even high performance over-the-road vehicles. The overall efficiency of a vehicle propulsion system can be greater with a Hydrostatic Transmission (HST) than with a straight mechanical transmission under certain duty cycles like start-stop situations under widely varying loads.

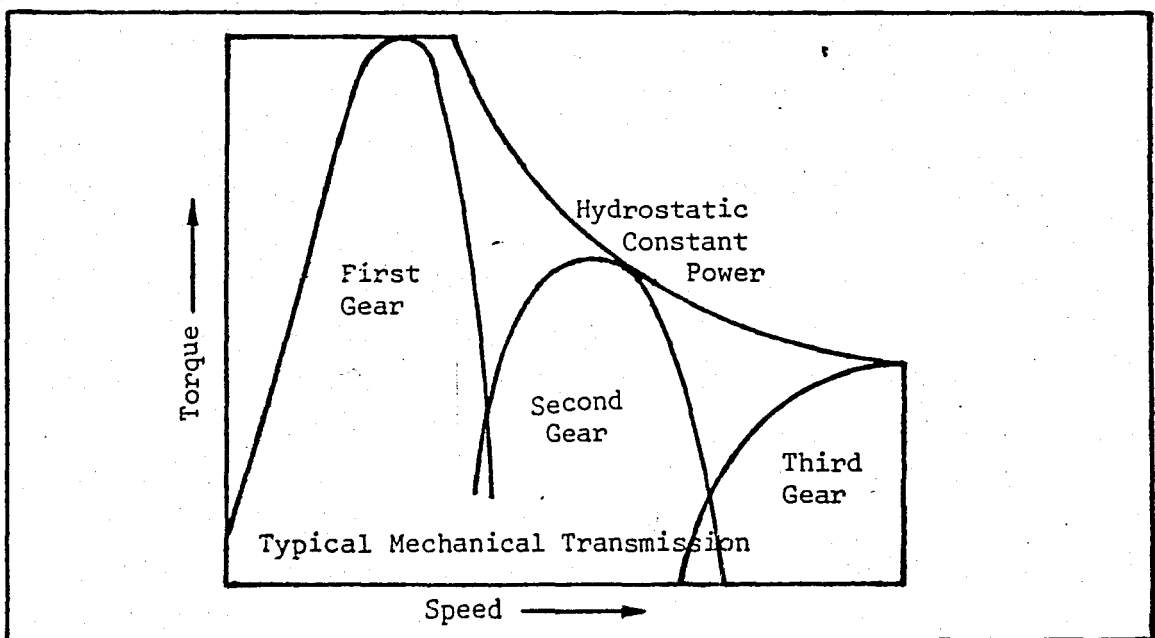


FIGURE 1.1 - Comparison of torque-speed characteristics of hydrostatic transmission and mechanical transmission

Fig.1.1 shows the difference in operation of the hydrostatic transmission compared to a three speed gear transmission. The smooth curve represents the uniform matching of torque and speed requirements by the hydrostatic transmission. The gear transmission has only three points of peak power while the hydrostatic transmission offers a continuous curve without peaks and valleys.

HST systems have longer life, necessitate less maintenance, are easily controllable, but their investment cost is higher. That is why it should be provided to work under very suitable conditions by choosing the best suitable circuit components and also by considering the dynamic behaviour of the system during the design step. Consequently, the best selection of components to fulfill the load requirements is critical from both performance and economic point of view. The dynamic behaviour should also be considered and observed during the design.

At the present time the design and analysis of hydraulic systems is usually conducted by a trial and modification procedure that involves sketching the circuit and calculation of the various component and system parameters. Industrial designers usually rely on past experience and after much testing and redesign usually achieve acceptable hydraulic system performance. This conventional design procedure may sometimes be quite drastic, time consuming, and expensive. At this point, Computer-Aided Design techniques have been considered as a tool which could be used to streamline the design procedure. For this purpose a computer program has been constructed. This program makes possible the design of a hydraulic circuit - of course it should be in the design limits of the program - by the selection of appropriate elements from existing files, which is the first step of the task.

As the second step the dynamic model of Hydrostatic Transmission Systems is used to perform a digital simulation of the dynamic behaviour of the designed system. So it is possible to consider the dynamic behaviour, i.e transient response, up to a point depending on precision of the system parameter values and capability of the model in predicting the real system. So the final computer program package is thought to be an all purpose tool for designing of hydraulic systems, specifically hydrostatic transmission systems.

## II. HYDROSTATIC TRANSMISSION SYSTEMS

### 2.1 INTRODUCTION

In a typical hydraulic system, a pump drives the system's fluid. The intake of the pump is connected to a fluid source ( a reservoir ). The fluid discharged by the pump is used to control the position or speed of resisting loads. Motion is usually provided by a hydraulic actuator - either a linear hydraulic cylinder or a rotary hydraulic motor. The flow and direction of the pressurized liquid supplied by the pump are controlled by valves farther downstream.

Hydrostatic transmission ( HST ) pumps and motors are designed and matched to optimize energy transmission [1] . They offer many important operating features. They will

- remain stalled and undamaged under full load at low power loss,
- hold a preset speed accurately against driving or braking loads,
- may operate in reverse direction at controlled speed within design limits, unaffected by output loads,
- provide faster response than any other transmission, and
- provide dynamic breaking.

In the following sections the versions of HST systems are investigated. The variable pump-fixed motor version is particularly studied, which is the basic and most widely used form of hydrostatic power transmission.

In the latter sections system elements which form a HST system are investigated. In order to extend the design limits of the computer aided design program some elements ,e.g.cylinders, are also considered. So it may be possible to develop a general purpose program for designing hydraulic systems.

## 2.2 CLASSIFICATION OF HST SYSTEMS

Hydraulic technology has evolved from open circuit constant flow systems to open circuit demand flow systems, and is now moving toward closed loop, electrohydraulic servo systems [1]. Because HST systems have evolved as a definitive branch of fluid power technology, they first reached the market as closed circuit systems, followed by open circuit systems. Fig.2.1 summarizes the four basic HST configurations, and conforms with international control technology usage. The entire spectrum of HST shown in Fig.2.1 is now available commercially.

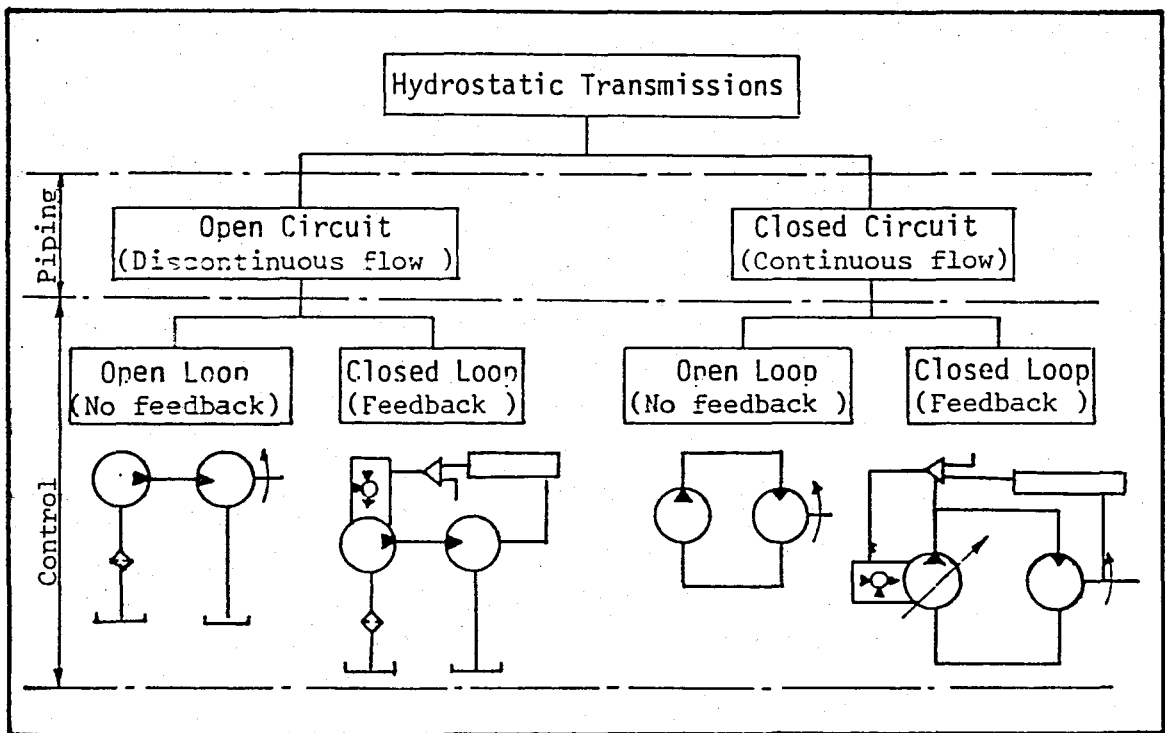


FIGURE 2.1-Basic HST configurations

Another classification of HST systems is based on the types of pumps and motors used, i.e. fixed or variable displacement, with their corresponding output characteristics. According to this classification there are four types which are investigated in the following sections.

The speed range of the system is an important parameter to assess the system effectiveness. It is defined as the range within the ratio between output speed and prime mover speed varies [2].

### 2.2.1 Fixed-Displacement Pump and Fixed-Displacement Motor ( PFMF )

This is the simplest form of hydrostatic transmission. Although this transmission is inexpensive, its applications are limited [2]. Because pump displacement is fixed it must be sized to drive the motor at a fixed speed under full load. However, in order to obtain variable output speed, a controllable bypass valve should be provided, which returns fluid directly from the delivery line into the tank [3].

The PFMF transmission is not recommended for applications requiring frequent starts and stops, because of the low efficiency under these conditions [1]. However if the load has constant speed and a constant speed prime mover is available, this type of transmission may be used.

### 2.2.2 Fixed-Displacement Pump and Variable-Displacement Motor ( PFMV )

If the pump has a fixed displacement and the motor has a variable displacement, the transmission can deliver constant power. If flow to motor is constant, and motor displacement is varied to maintain the product of speed and torque constant, then the delivered power may be kept constant.

The PFMV transmission can not accelerate a load from rest without a bypass or a zero speed prime mover, so its use is limited. Also its efficient speed range is only about 4:1, [2].

### 2.2.3 Variable-Displacement Pump and Variable-Displacement Motor ( PVMV )

Varying the displacements of pump and motor provides infinite ratios of torque and power. When motor displacement is at maximum, varying pump output directly varies speed and horsepower output, while torque remains constant. Decreasing motor displacement at full pump displacement increases motor speed to its maximum, while torque varies inversely with the speed, horsepower remains constant.

A PVMV type transmission is the most flexible, covering speed ranges of up to 20:1 in some cases [2]; but this only true as long as the speed limit of the motor increases with reduced displacement, and has to be verified in every instant [3].

### 2.2.4 Variable-Displacement Pump and Fixed-Displacement Motor ( PVMF )

The hydrostatic transmission incorporating a variable displacement pump and a fixed displacement motor is the basic and most widely used form of hydrostatic power transmission [3]. For most applications this configuration unifies all advantages of performance and still has reasonable simplicity.

This combination is also called a constant torque transmission. Torque output is constant at any speed because torque depends only on fluid pressure and motor displacement. Power output is varied by adjusting pump displacement with changes motor speed. The efficient speed range is limited to about 6:1, [2].

There are two possible circuit configuration; open and closed circuit. In the open circuit transmission, with constant input speed, the flow is determined by the displacement setting of the pump which must be controlled according to the load requirements. Although the flow varies with the displacement setting it is almost unaffected by the pressure. Therefore this is a constant-flow system with variable pressure as long as displacement setting is held constant.

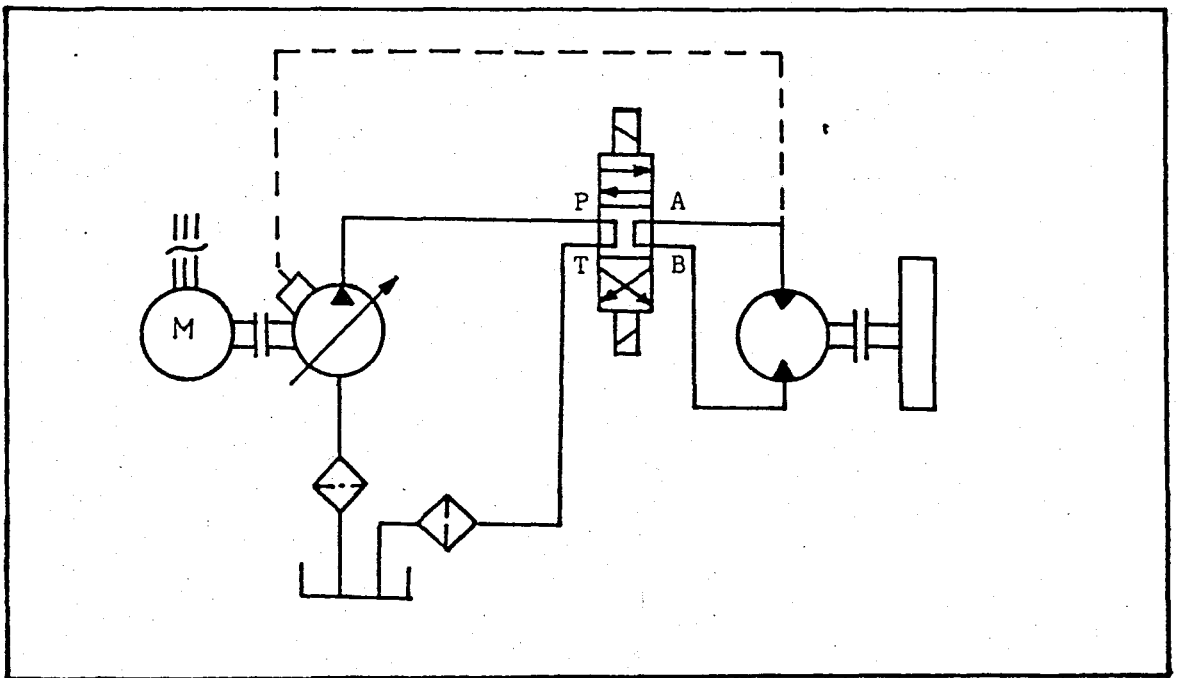


FIGURE 2.2- Open circuit HST system

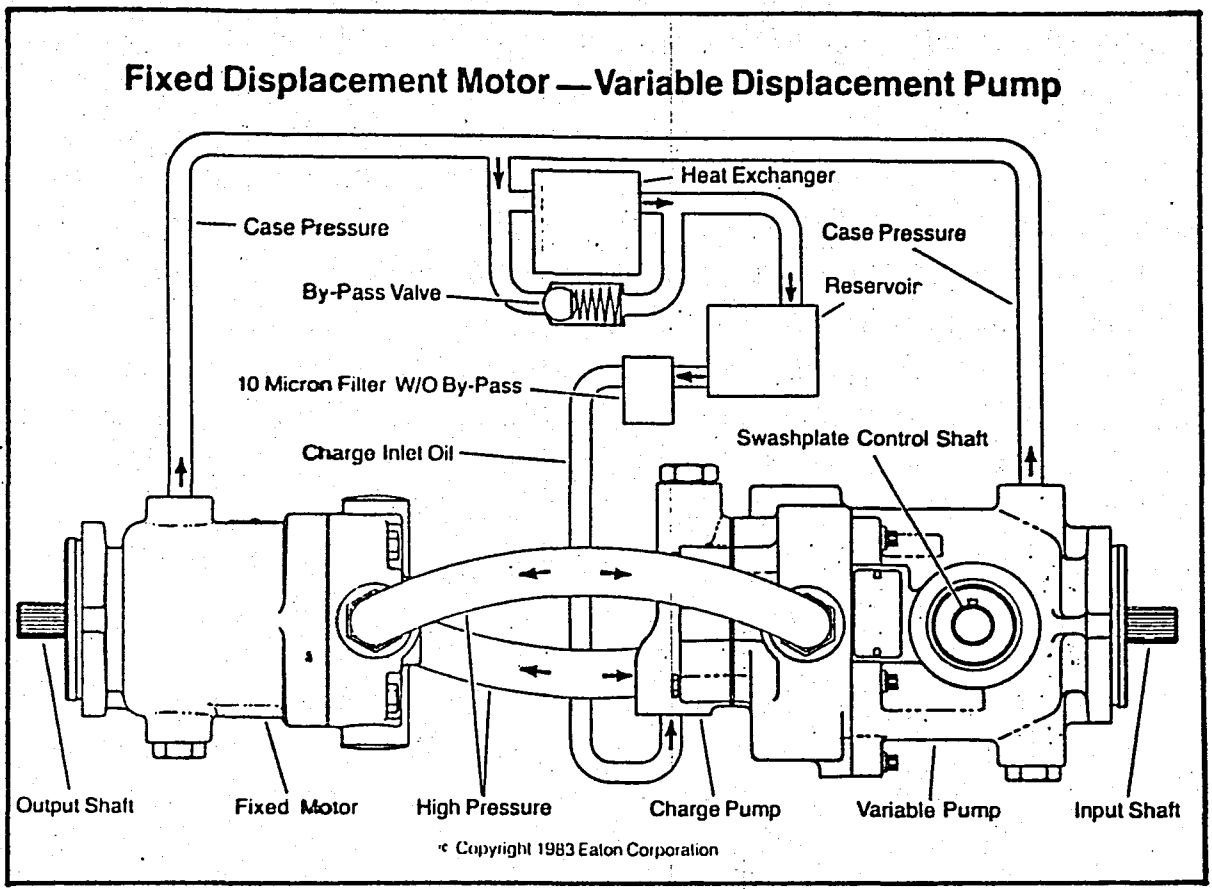


FIGURE 2.3 - A closed circuit PVMF type hydrostatic transmission (Eaton Co.,USA)

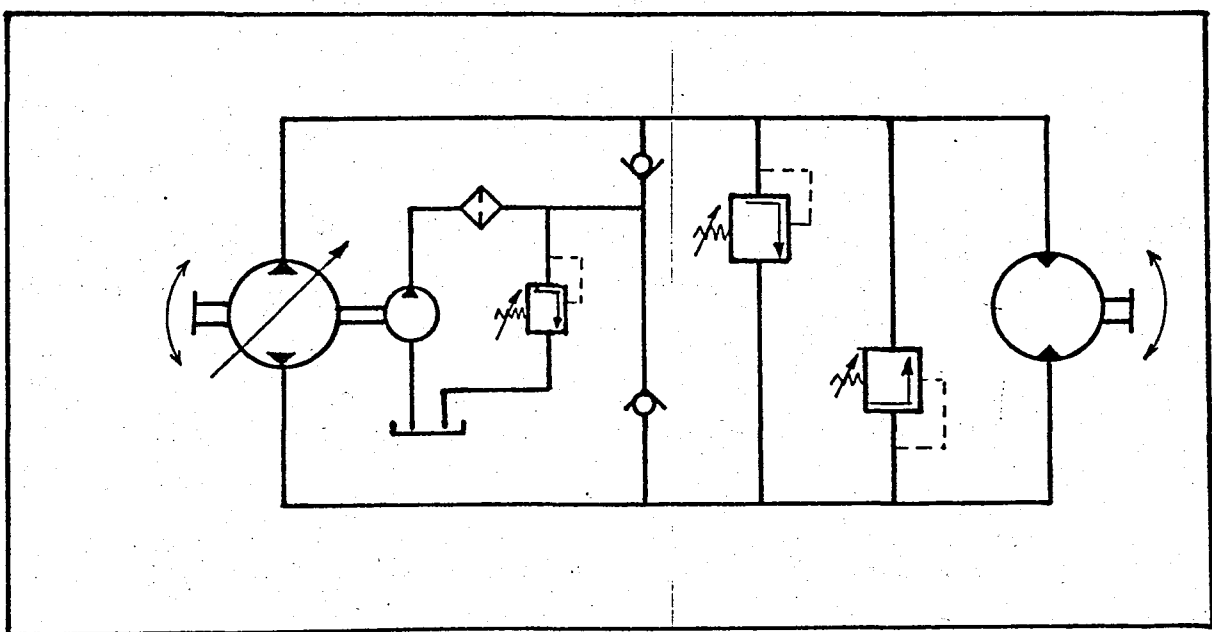


FIGURE 2.4 - Closed circuit PVMF type HST in symbolic representation

The closed circuit is more complicated and has more components, but it is frequently used because motor shaft can be reversed and/or braked. In all applications where these requirements are vital, as in vehicle drives, the closed circuit is an absolute necessity.

In the open circuit configuration (Fig.2.2), the fluid to the pump is supplied from a reservoir and delivered to the control block and motor inlet. Return and leakage flows come back to the reservoir. The direction of movement is determined by a directional control valve. The motor speed can be varied by either using a variable displacement pump or adjusting motor inlet flow by a servovalve in the control block.

In the closed circuit configuration (Fig.2.3 and 2.4) motor outlet is connected to the pump inlet. Either the direction of movement or the speed is varied by the variable displacement, double direction pump. A boost pump (charge pump) is used to compensate leakages and provide the oil to circulate even in stops. Another function of the boost pump is to fix and regulate the rated pressure in the return line at a level above the cavitation limit.

The capacity of the boost pump is normally selected as 10 or 15 per cent of the maximum system flow and its energy loss is negligible in practice 3. Another advantage is that the reservoir size is greatly reduced, according to the capacity of the boost pump and not with respect to the maximum flow rate of the main pump.

## 2.3 SYSTEM ELEMENTS

In this section the elements which form a HST system as a whole unit are investigated. In addition, translational loads and cylinders are also considered so that the final computer program will be an all purpose one.

### 2.3.1 Load

#### 2.3.1.1 Translational Load

Although a proper HST is not intended to drive translational loads, they are also taken into the design limits of the objective computer program. A detailed study, however, is not done since primary attention



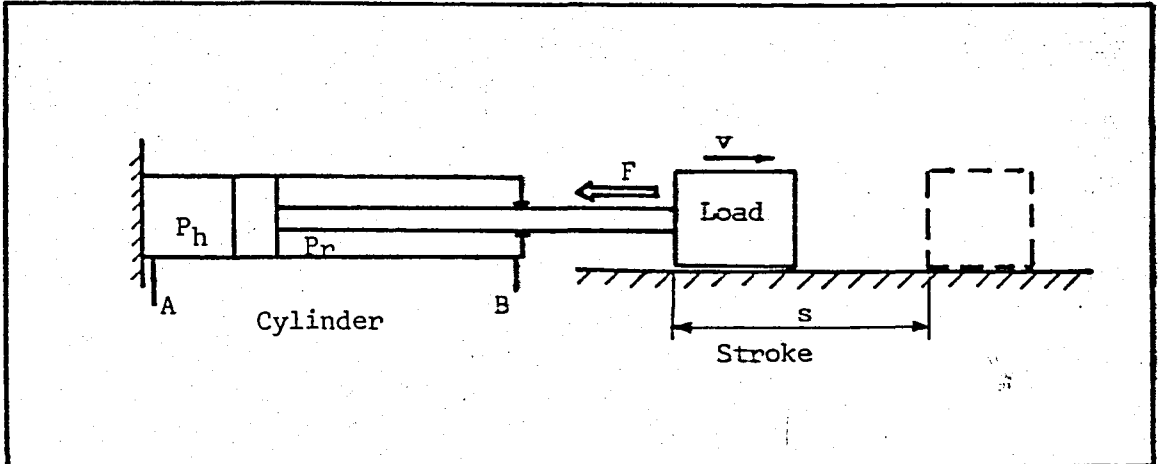


FIGURE 2.5 - Translational load drive system and parameters

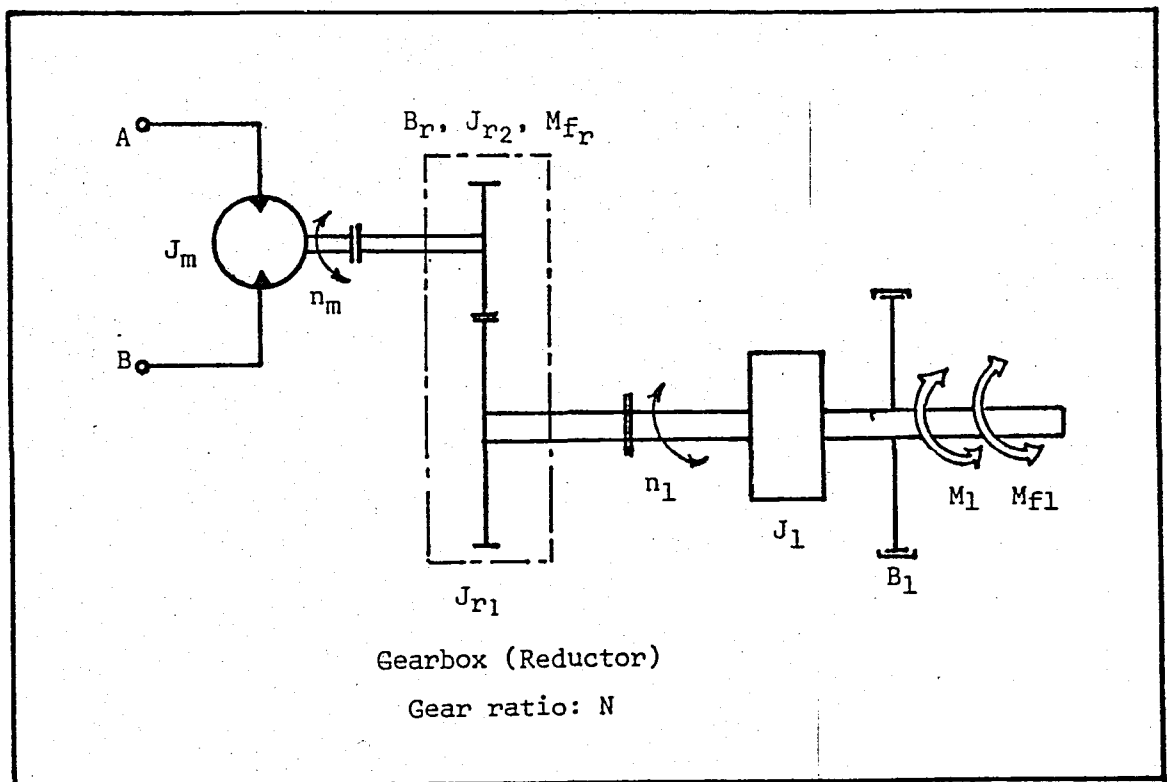


FIGURE 2.6 - Rotational load drive system and parameters

is given to rotational loads.

As shown in Fig.2.5, a translational load is identified by the force applied by the load to the cylinder, drive speed and maximum distance to drive (stroke).

### 2.3.1.2 Rotational Load

A typical rotational load drive system is shown in Fig.2.5 . As shown in the figure the load may be connected to the motor via a reductor with a gear ratio  $N$ . In choosing the gear ratio the following criterion gives a proper reductor-motor pair [4] :

-for inertial, starts-and-stops type loading

$$N^2 = J_1 / J_m \quad (2.1)$$

-for frictional, constant-speed type loading

$$N^2 = B_1 / B_m \quad (2.2)$$

The parameters shifted to the motor axis are given as follows [4] :

$$J = (J_1 + J_{r1}) / N^2 + (J + J_{r2}) \quad [\text{kg-m}] \quad (2.3)$$

where

- $J$  = inertia shifted to the motor axis
- $J_1$  = load inertia moment
- $J_m$  = motor inertia moment
- $J_{r1}$  = reductor inertia moment on the load side
- $J_{r2}$  = reductor inertia moment on the motor side

and,

$$B = B_1 / N^2 + B_r \quad [\text{Nm/rd/s}] \quad (2.4)$$

where

- $B$  = viscous friction coefficient shifted to the motor axis
- $B_1$  = viscous friction coefficient of the load
- $B_r$  = viscous friction coefficient of the reductor

and,

$$M = (M_1 + M_{f1}) / N + M_{fr} \quad [\text{Nm}] \quad (2.5)$$

where

$M$  = torque shifted to the motor axis

$M_l$  = load torque

$M_{fl}$  = dry friction torque of the load

$M_{fr}$  = dry friction torque of the reductor

The speed of motor is also given by

$$n_m = N \cdot n_l \quad [\text{rpm}] \quad (2.6)$$

or for angular speed,

$$\Omega_m = N \cdot \Omega_l \quad [\text{rd/s}] \quad (2.7)$$

If the reductor is not used, by taking  $N$  equal to unity and the parameters of the reductor equal to zero, in the equations above, we can get the parameters which affect the motor axis.

For the steady state operation the torque which should be created by the motor can be found by this equation:

$$M_m = M + B \cdot \Omega_m \quad [\text{Nm}] \quad (2.8)$$

### 2.3.2 Actuator

Actuator type will be chosen according to the type of the load. If one has a translational load it will be a cylinder, if a rotational load then a hydraulic motor should be selected.

#### 2.3.2.1 Cylinder

Hydraulic cylinders serve to carry out translatory (straight) movements and to transfer force by so doing. There are some types of cylinders, but here only differential cylinder (cylinders with rod on one side) are considered (Fig.2.5).

The steady state load equation is

$$P_h A_h - P_r A_r = F \quad (2.9)$$

where  $P_h$  and  $P_r$  are head and rod end gauge pressures and  $A_h$  and  $A_r$  are head and rod end areas [5]. In an ideal system with no losses Eq.(2,9) reduces to

$$P_s A_h = F \quad (2.10)$$

where  $P$  is the system pressure. However in practice, losses will make  $P_h$  less than  $P_s$  and  $P_r$  greater than tank pressure.

The speed depends on the oil supply per time unit and the area. The flow rate required to maintain a given speed is

$$Q = A.v \quad (2.11)$$

Forward and backward speeds are different for the same flow rate because of the difference in effective areas, as well as forces.

Friction losses of cylinder can be taken into account by multiplying the load by a coefficient.

### 2.3.2.2 Motor

Hydraulic motors can be used to convert the controlled fluid power from a valve or pump into rotating mechanical power for driving a load. The fluid flow is changed to rotational velocity or speed, and the pressure is converted into torque to move the load.

A hydraulic motor can be rated by its displacement,  $D_m$ . Motors may be either fixed displacement or variable displacement type. In this study, however, only the former is considered.

For an ideal motor the following equation gives the torque produced by the motor, which is the product of the displacement and the pressure differential.

$$(M_m)_{th} = (P_i - P_o) D_m \quad (2.12)$$

In practice there are some losses: Torque losses are a combination of breakaway torque, coulomb friction, and viscous drag losses [6]. In the model presented here all these losses are taken into account in the mechanical efficiency,  $\eta_m$ . Hence the equation above is modified as follows:

$$M_m = \Delta P_m D_m \eta_m \quad (2.13)$$

The real flow required by the motor for a given output speed is,

$$Q_m = D_m \cdot \Omega_m \cdot \eta_v \quad (2.14)$$

The leakage flow in the motor is considered by the volumetric efficiency,  $\eta_v$ .

### 2.3.3 Pump

Positive displacement pumps serve to create a fluid flow (to displace a volume of fluid) and to allocate the necessary forces to it as required. Pumps are also rated in terms of delivery or displacement,  $D_p$ .

Theoretical flow rate of pump can be determined from :

$$(Q_p)_{th} = \Omega_p \cdot D_p \quad (2.15)$$

Owing to leakage, the actual flow rate is less, which is determined by the volumetric efficiency,  $\eta_v$  :

$$Q_p = \Omega_p \cdot D_p \cdot \eta_v \quad (2.16)$$

Volumetric efficiencies usually lie in the range 80 - 90 per cent, being highest for piston pumps [ 5 ] .

Mechanical power required by the pump is given by

$$W_p = (\Delta P_p \cdot D_p \cdot \Omega_p) / \eta_t \quad (2.17)$$

where  $\eta_t$  is overall pump efficiency and depends on both of efficiencies:

$$\eta_t = \eta_v \cdot \eta_m \quad (2.18)$$

Mechanical efficiencies are usually in the range of 80-95 per cent [ 5 ] .

The majority of positive displacement pumps use either gears, vanes or pistons. Gear pumps are fixed displacement pumps, the others may be either fixed or variable type.

In HST systems, it is a necessary precaution to choose the pump's capacity 10 per cent larger than the largest flow demand [ 4 ] .

### 2.3.4 Pipe Line

Pipe diameters are determined according to the largest flow rate required by the system. In selection of pipes the following recommendations [5], are taken as basis:

- i) Suction lines should not carry fluid at velocities in excess of 1.5 m/s .
- ii) Delivery lines should not carry fluid at velocities in excess of 4.5 m/s .
- iii) Return lines should be of larger diameter than delivery lines. They should never be smaller.

In the closed circuit HST system delivery and return lines may interchange, therefore both of them should be considered as delivery line.

The usage of flexible hoses is also possible in some cases. When hose connections are a necessary operational requirement of a HST system the reinforcing type should be selected, if their advantages are not being exploited, substitution by rigid steel pipes is desirable [7].

### 2.3.5 Valves

Valves fall into three main categories, directional control, flow control and pressure control. In this study only directional control valves and pressure relief valves (as pressure control valve) are considered.

#### 2.3.5.1 Directional Control Valves

Directional control valves are identified by their position and port numbers, in addition to port connections in the mid position such as open centre, closed centre or tandem centre.

While choosing the valve size the following criterion is taken as basis, [8] :

$$N_{dv} \geq \frac{1}{2} N_p \quad (2.19)$$

where

$N_{dv}$  = directional control valve inlet port nominal radius  
 $N_p$  = delivery line nominal diameter

The coefficient of 1/2 is a practical value, otherwise it becomes unpractically overdimensioned.

The pressure losses of directional valves are very hard to calculate from theoretical point of view. However, as most manufacturers release experimental performance curves, the calculation of directional valve loss involves, first curve fitting to manufacturer's curve and then calculation of the loss, applying the updated flow. A better result is yielded by a third-degree polynomial curve fitting. Hence the pressure drop is found from

$$\Delta P_v = C_1 \cdot Q + C_2 \cdot Q^2 + C_3 \cdot Q^3 \quad (2.20)$$

where  $C_1$ ,  $C_2$  and  $C_3$  are the coefficients found by the curve fitting and  $Q$  is the flow rate passing through the valve.

### 2.3.5.2 Pressure Relief Valves

In order to prevent the elements of HST system from damage pressure relief valves should be employed at the necessary points in the circuit. The limit pressure set value of these valves should be selected according to the largest loading level [4].

For choosing the size of relief valves the following criterion is valid [8]:

$$N_{rv} = \frac{1}{2.5} N_p$$

where  $N_{rv}$  is the inlet port nominal radius of pressure relief valve (valve size) and  $N_p$  is the nominal pipe diameter of the delivery line.

### 2.3.6 Filter

The filter serves to reduce the level of dirt in a pressure medium to a reliable level, and thus to protect the individual elements from too much wear.

There are three different filters used in hydraulic systems : Suction, pressure and return line filters. The most used type is the return line filter and it is fitted in the return line.

The filter is selected according to its filtering and flow capacity. A reasonable filtering capacity for the return line filter is 10 micron.

Filter causes pressure drops. Flow versus pressure drop curve is nearly a parabola. By using existing data in the catalog a loss coefficient,  $K_f$ , is evaluated for each filter. Hence the pressure drop in the filter can be determined by

$$P_f = K_f \cdot Q^2 \quad (2.22)$$

### 2.3.7 Circuit Losses

In the previous sections pressure losses in the directional valve and filter were mentioned and some methods were offered to calculate them. In this section the other pressure losses are considered.

Pressure losses in pipes are determined from D'Arcy formula for circular pipe [5] :

$$P_1 = f \frac{L}{d} \frac{\rho}{2} v^2 = \frac{K}{K_1} Q^2 \quad (2.23)$$

where

$$K = \frac{f \cdot L}{d}$$

$$K_1 = \frac{2 A^2}{\rho}$$

$\rho$  = density of fluid

$f$  = friction factor

$L$  = pipe length

$d$  = internal diameter of pipe

$v$  = mean fluid velocity

$Q$  = flow rate in the pipe

$A$  = pipe cross section area

The friction factor  $f$  is a function of Reynolds number ( $Re$ ) and of the pipe roughness. For design purposes it is generally accepted that for



Laminar flow (  $Re < 2500$  )

$$f = 64/Re \quad (2.24)$$

and for turbulent flow (  $Re > 2500$  )

$$f = 0.332/Re^{0.25} \quad (2.25)$$

The value of Reynolds number can be determined from

$$Re = \frac{v \cdot d}{\nu} = \frac{4 Q}{\pi d^2 \nu} \quad (2.26)$$

where  $\nu$  is kinematic viscosity of fluid. Kinematic viscosity usually has a value about  $4.0 \times 10^{-5} \text{ m}^2/\text{s}$  at normal industrial working conditions [5].

Sudden contractions and expansions occur where a pipe is attached to a cylinder, pump or reservoir. In addition there are T and L joints in the piping layout. The loss coefficient  $K$  in these cases is taken as

- i) 0.1 for T joints,
- ii) 0.5 for entry from tank or cylinder to pipe (contraction),
- iii) 1.0 for entry from pipe to tank or cylinder (expansion),
- iv) 1.3 for L joints,

which are experimentally obtained [5].

### 2.3.8 Pump Drive

Hydraulic systems usually employ electrical motors to drive the pump. However, in some HST systems, particularly in mobile applications, the prime mover is inevitably an internal combustion engine.

In a study [9] performed for a closed loop controlled diesel engine, it is concluded that the effect of prime mover droop is relatively insignificant and can be ignored except in extreme cases.

In this study it is assumed that pump drive speed is held constant whichever prime mover is employed. Thus the effect of prime mover dynamics is omitted.

The prime mover should provide the drive power required by the pump. The maximum power requirement is determined by

$$W = \Delta P_p \cdot Q_p \quad (2.27)$$

where

$\Delta P_p$  = maximum pressure differential on the pump

$Q_p$  = maximum pump flow rate

$\Delta P_p$  is calculated by the following equation

$$\Delta P_p = \Delta P_l + \Delta P_t \quad (2.28)$$

where

$\Delta P_l$  = pressure differential corresponding to the load

$\Delta P_t$  = total pressure losses of the circuit

The effect of cylinder differentiation or volumetric efficiency of the motor on the return flow should be taken into account. Hence Eq.(2.27) is modified by an easy manipulation.

### III. COMPUTER-AIDED DESIGN OF HYDROSTATIC TRANSMISSION SYSTEMS

#### 3.1 INTRODUCTION

The design process for hydraulic systems starts with a drawing of the circuit layout. The designer then selects the components needed to form the circuit from manufacturer's catalogues. After the initial design, the prototype is built in order to test the circuit and check the performance characteristics. Depending on the result, the circuit will then be modified and retested until the performance requirements are met. This process requires long lead times and great expense. Computer-aided design techniques can be used to streamline the design process cutting out the need for an inordinate number of modifications to the prototype.

There are generally two steps in the design:

- i) sketching the logical layout,
- ii) sizing the components, making all necessary calculations and verifications.

Since the aim of this study is not to create a program enabling to construct the logical layout, which can be added to the work later, the computer program package assumes that the layout is already created.

There are usually several possible solutions to design problems, and criteria have to be established to enable a decision among these alternatives. Low cost, high reliability, high power/weight ratio, safety and easy maintenance are typical criteria for hydraulic circuits [5]. The efficient use of energy within the system is another important criterion. The written program package minimizes any component size (for a cylinder its volume, for a pump its displacement, for other components their nominal bores) and a result, minimize the cost while maximizing power/weight ratio.

Two basic types of parameters should be considered:

A) Quantities that can be determined with acceptable precision and whose values remain constant or nearly so, for example, oil stiffness.

B) Quantities that are difficult to assess, for example, seal friction and viscous damping. In dealing with the latter the designer relies on the art of circuit design, supported by the wealth of empirical data which have been accumulated over years of experience.

In this work, two main circuits are taken as basis for design procedure. As shown in Fig.3.1 the first one is a general hydraulic circuit. The second is a closed HST circuit. So the study which is mainly devoted to HST systems can be extended general hydraulic systems, e.g. those employing cylinders. Fig.3.2 shows the closed circuit model.

A user friendly, interactive CAD program has been developed. It is assumed that the user is familiar with the subject of hydraulic circuit design, at least with the basic concepts. However, anyone might use the program by assistance of the User's Manual. The basic circuit layout should be provided by the user.

### 3.2 DESIGN ALGORITHM

The programming details are embedded into the computer flow charts which are included in the following sections. A general algorithm of the program package can be summarized as follows:

- 1) Set operating pressure and actuator type.
- 2) Take all necessary actuator data (rpm, torque, etc.).
- 3) Choose a minimum actuator size, calculate the necessary flow rate needed to drive this actuator at the given maximum speed. Calculate return flow rate.
- 4) Using this flow rate and prime mover's speed, choose a minimum pump size. Update the flow rate with this pump's net flow rate. If desired pump is a variable displacement type, use the flow rate computed at the third step.

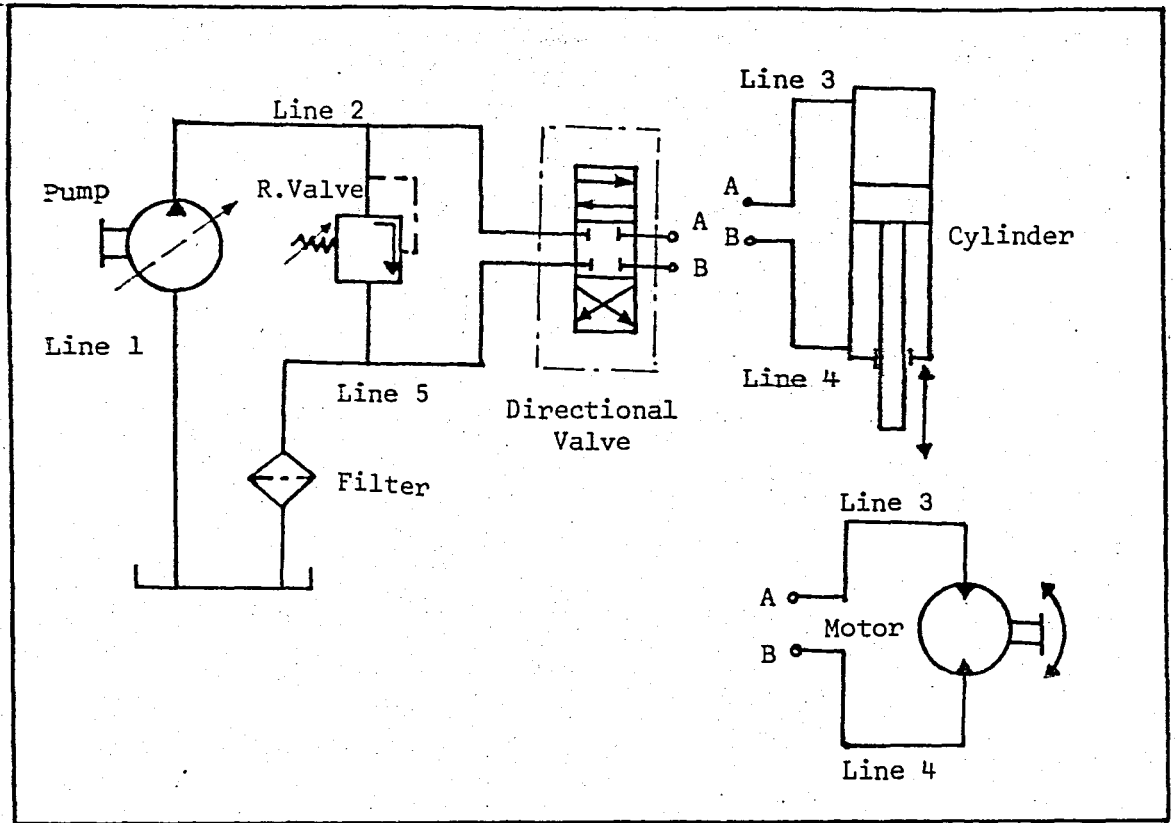


FIGURE 3.1 - Open circuit model

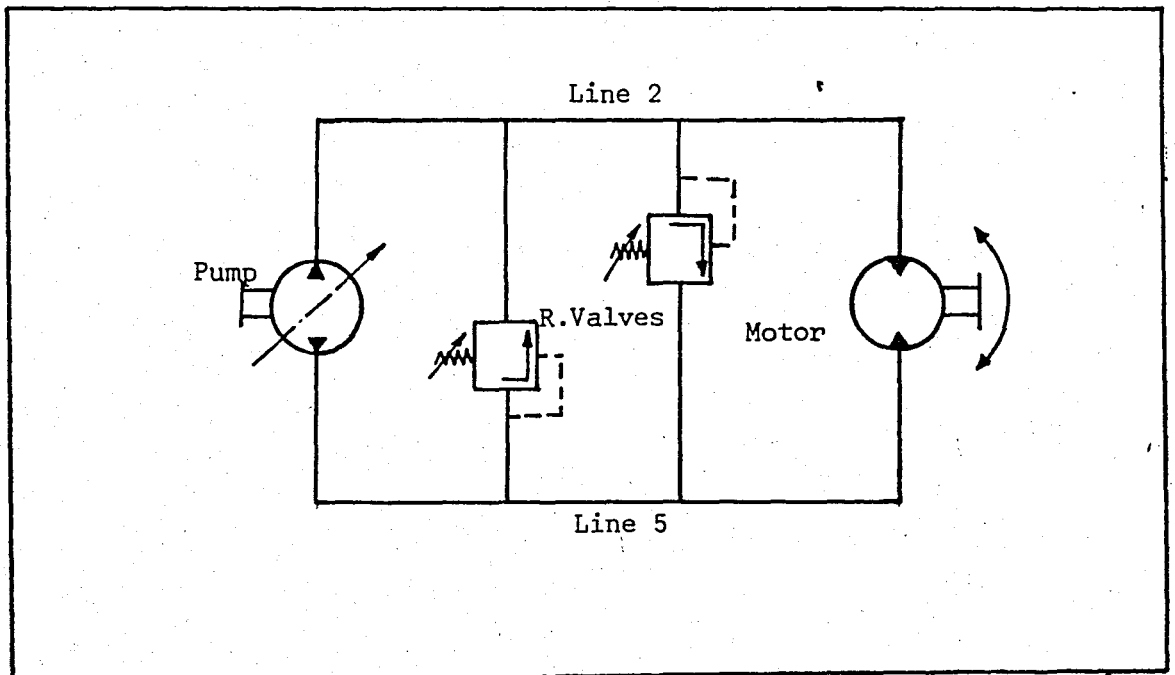


FIGURE 3.2 - Closed circuit model

- 5) Using updated flow rate, choose pipe diameters.
- 6) Using nominal bore of chosen pipe, choose pressure relief valve and directional control valve (if required), calculate pressure loss and loss coefficients of directional valve.
- 7) Choose a filter, using maximum flow rate.
- 8) Enter pipe layout for all lines (number of L,T joints, pipe lengths, pump-tank level difference).
- 9) Calculate pressure losses and coefficients for all lines in turn (joint,pipe,expansion,contraction).
- 10) Calculate pump pressure differential by taking into consideration the sum of losses as well.
- 11) Calculate total pressure loss coefficient.
- 12) Calculate pressure relief valve setting.
- 13) Calculate required power to drive the pump.

The algorithms used for the main program and subprograms are presented with their operational flow charts.

### 3.3 MAIN PROGRAM

The main program fundamentally consists of two sections: the first is "Component Selection Section" and the second is "Pressure Loss Computation Section" which is optional. On the other hand it is designed to permit interactive usage.

In Fig.3.3 the flow chart of the main program, namely CADHST, is given. Printing functions are not shown in order to avoid unnecessary details. Common parameters are given in the forms used in the program, whose meanings can be found from the glossary given in Appendix A.

CADHST writes the output data, comments and prompting commands on the screen and at the same time component specifications are printed on an output file, OUT9. After running, a printout of this file can be taken. In Fig.3.4 all files and programs used by CADHST are listed.

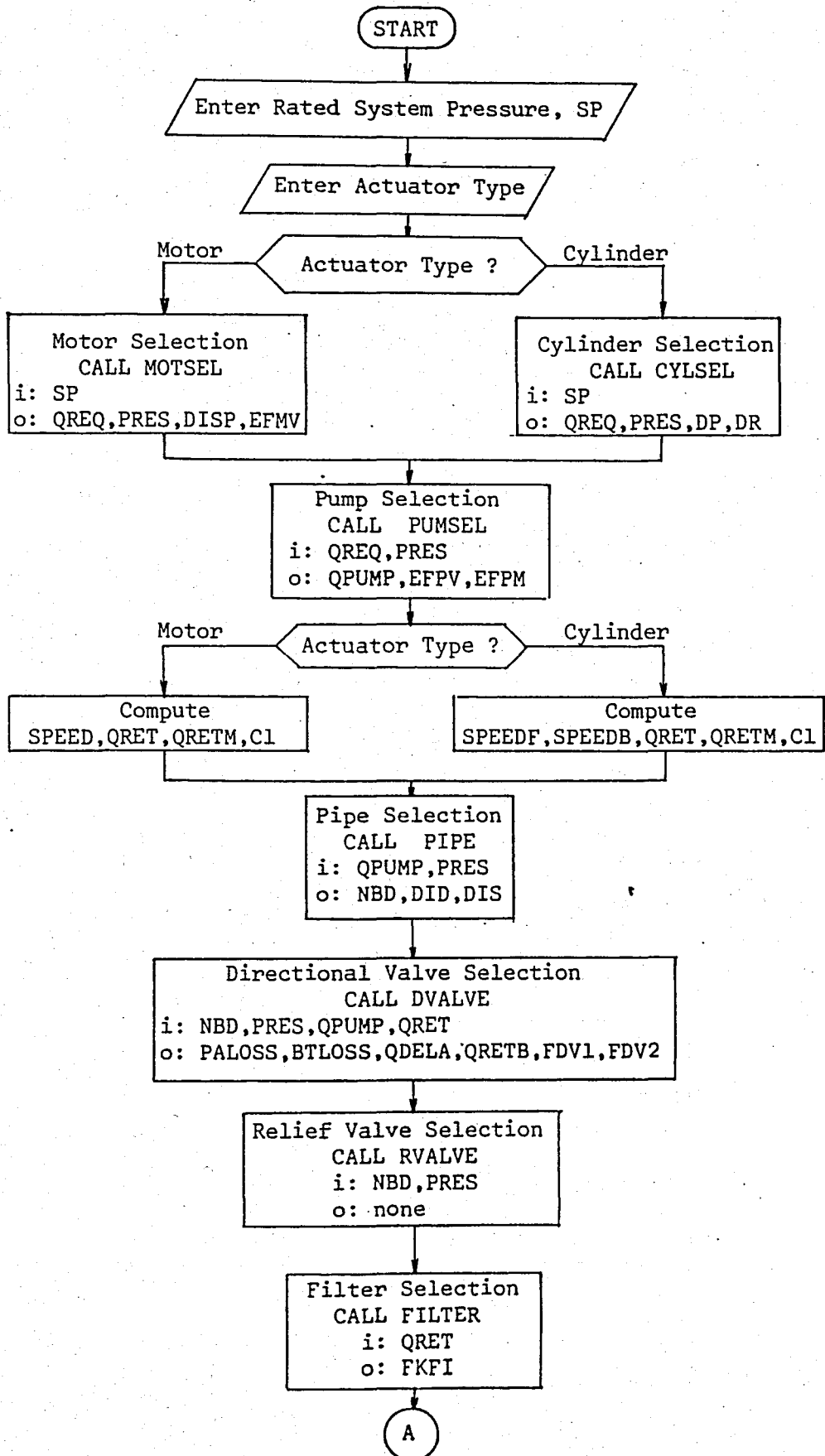
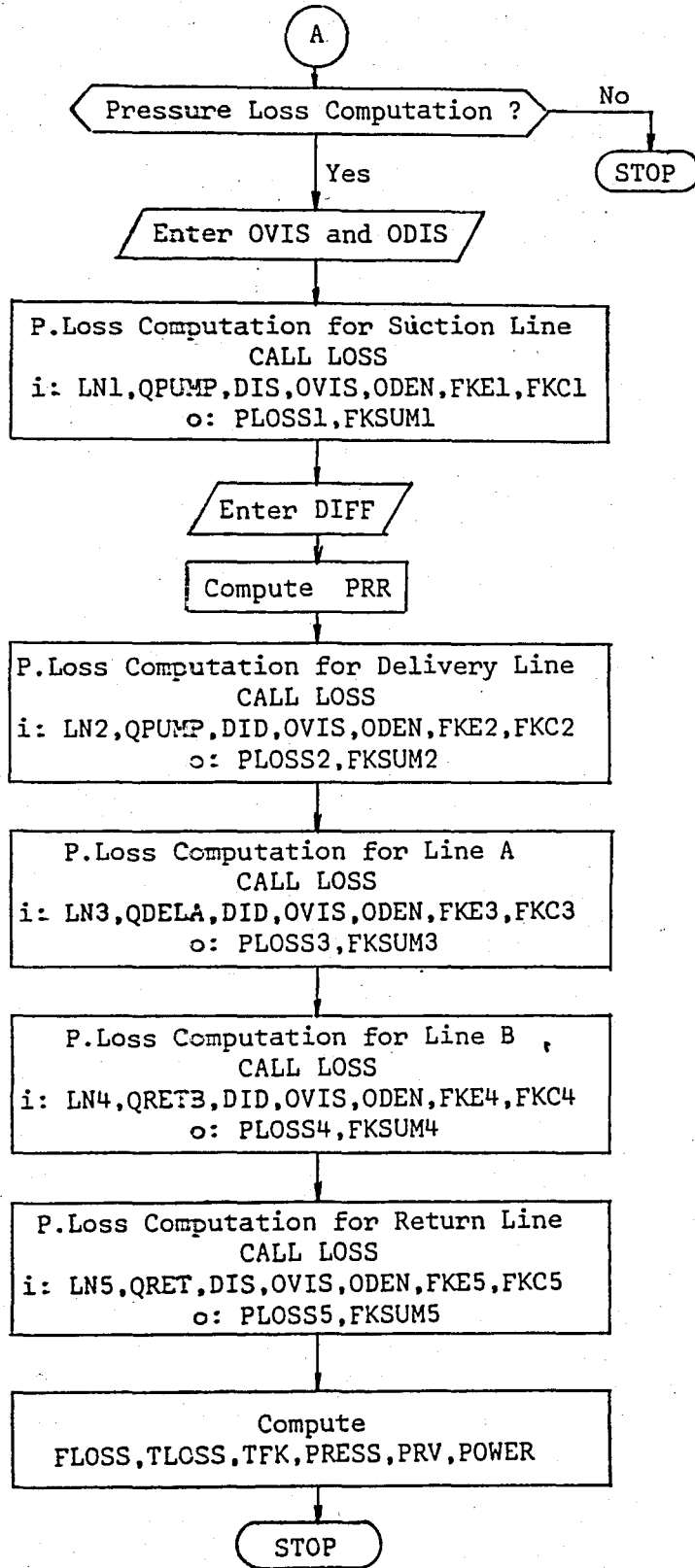


FIGURE 3.3 - General flow chart of CADHST



i: input parameters from the main program to the subroutine  
o: output parameters from the subroutine

FIGURE 3.3 - Continued



Main program : CADHST

Subroutines :

CYLSEL	Cylinder selection
MOTSEL	Motor selection
PUMSEL	Pump selection
PIPE	Pipe selection
DVALVE	Directional valve selection
RVALVE	Relief valve selection
FILTER	Filter selection
LOSS	Pressure loss computation
OKAY	Program controlling

Component data file loading program: FLOAD

Data files (all direct-access):

CYLIN	Cylinder	U.N.:10 (Unit No.)
MOTOR	Motor	U.N.:20
PUMPA	Pump	U.N.:30
DVALF	D.valve	U.N.:40
RVALF	R.valve	U.N.:50
FILTE	Filter	U.N.:60

Input file :

INPUT	On the screen U.N.:5
-------	----------------------

Output files :

OUTPUT	On the screen U.N.:6
OUT1	Component listing file U.N.:1
OUT9	CAD program output file U.N.:9

FIGURE 3.4- The files and their function list

### 3.4. LOSS COMPUTATION

As shown in the flow chart of the main program, Pressure Loss Computation is available in the structure of CADHST and optional.

LOSS is a subroutine used for this task. It has to be called several times to complete the computation, for each line in turn. LOSS is executed for Line 1, Line 2, Line 3, Line 4 and Line 5, which are shown in Fig.3.1 .If a directional valve is not used, the flow rates are made equal to zero earlier, then the pressure losses belonging to these lines become zero.

LOSS computes all losses, including L, T joint losses, pipe losses, contraction and expansion losses. It prompts the user to enter pipe length, numbers of T and L joints, and then computes pressure losses for that line, according to the theory given earlier.

In the main program, pressure losses are calculated and then by taking a summation the total pressure loss corresponding the previously defined operation point is evaluated. On the other hand pressure loss coefficients of loss sources , such as pipe lines, filter and directional valves, are added up and a total pressure loss coefficient is determined. Consequently it is possible to calculate pressure losses for different flow rates by using the following equation:

$$\Delta P_t = K_t \cdot Q^2 \quad (3.1)$$

where  $K_t$  is the total pressure loss coefficient, and  $\Delta P_t$  is the total pressure loss of the circuit corresponding the pump flow rate,  $Q$  ..

Pressure losses of the directional control valve are calculated by a third order polynomial approximation. However, the loss coefficient should be determined by a parabolic approximation in order to use it in the procedure mentioned above. This is done in the subroutine by an easy manipulation.

The return flow is different from the delivery flow because of differentiation of the cylinder or due to the leakage flow in the motor. That is why a modification is done to compensate this difference. The loss coefficients belonging to the lines and components where the return fluid flows are multiplied by

$$C = \left(1 - \frac{D_r^2}{D_p^2}\right)^2 \quad (3.2)$$

in the circuits employing cylinder, where  $D$  is the rod diameter and  $D$  is the piston diameter; or multiplied by the volumetric efficiency of the motor in the circuits employing motor.

### 3.5 COMPONENT SELECTION SUBROUTINES

MOTSEL, CYLSEL, PUMSEL, PIPE, DVALVE, RVALVE and FILTER are written to select motor, cylinder, pump, pipe, directional valve, relief valve and filter respectively. They are also used in an interactive manner.

These subroutines are designed according to hydraulic principles and models given in the previous chapter. They admit data during execution and select appropriate components from the component files. The design criteria are mentioned earlier. They print output data on the screen as well as on the output file, OUT9 . Their prompting commands are given in the User's Manual.

The biggest advantage of these subroutines is that they use direct access files as componentfiles whose superiority is explained in the related section. PIPE does not use a component file, its data is given in itself.

A general flow chart for component selection subroutines is given in Fig.3.5 . All subroutines have the same structure with minor differences, except PIPE. An example of parameters is also given in Fig.3.6 .

### 3.6 COMPONENT FILES

As mentioned earlier the files which are loaded with component data are direct-access files. Their superiority to sequential files is seen in their proper usage, minimizing the time to find an appropriate component. It enables us to design the component files as divided into parts, i.e. data blocks, then it is a convenient way to go directly to the appropriate data block without having to read unnecessary records. For example, cylinder file consists of six data blocks, but only the corresponding data block is processed once the cylinder type and system pressure are entered. This is facilitated by a key-record which takes place on the fourth record

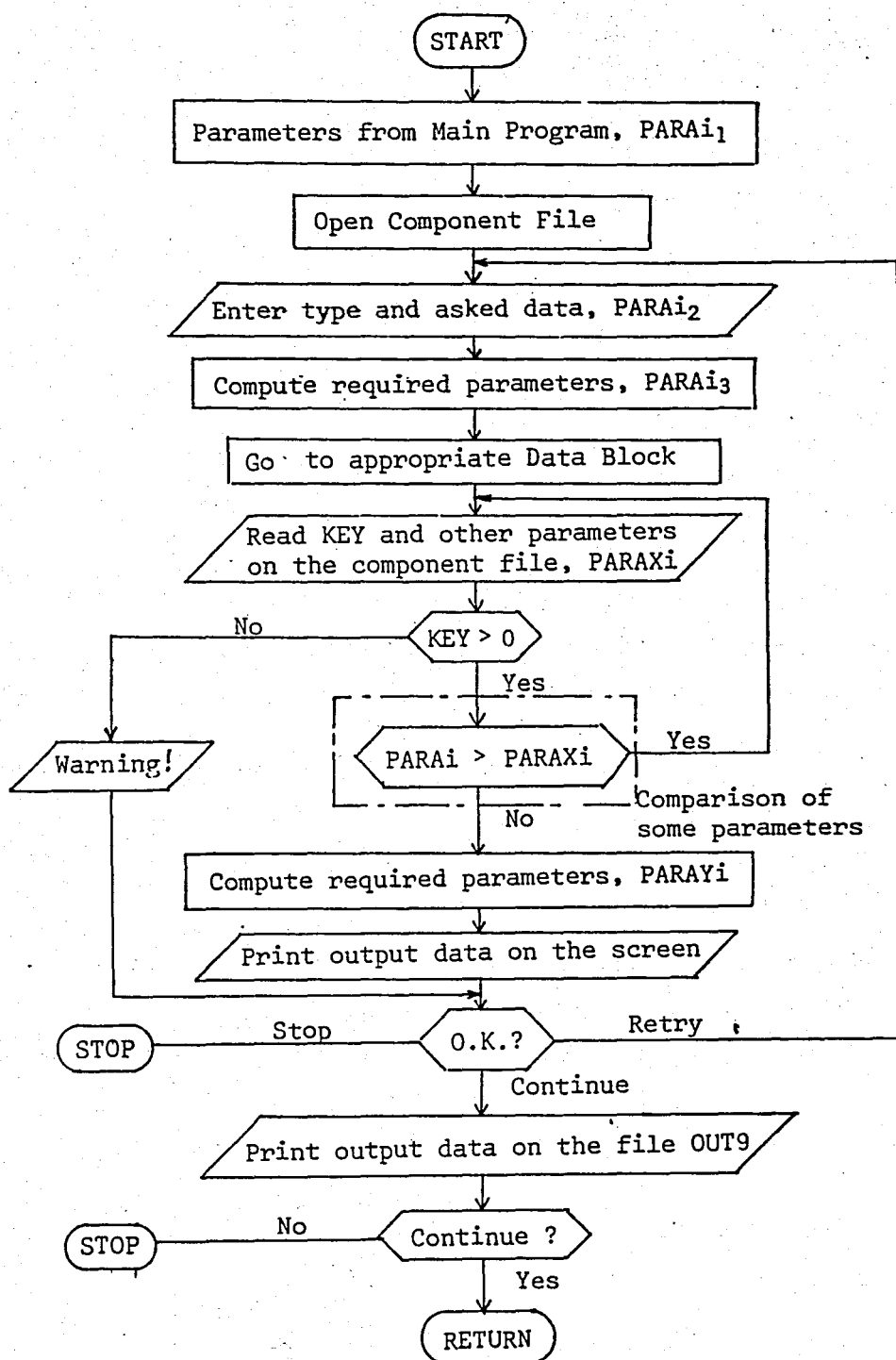


FIGURE 3.5 - General flow chart of component selection subroutines

File Name	: MOTSEL	Motor Selection
PARAi <sub>1</sub>	: SP	Rated system pressure
PARAi <sub>2</sub>	: RPM	Max load speed
	TRQ	Max load torque
PARAi <sub>3</sub>	: EFMV	Volumetric efficiency of motor
	EFMM	Mechanical efficiency of motor
	DISP	Required motor displacement
PARAXi	: DISPX	Motor disp. on the read record
	RPMX	Max motor speed on the read record
	PRESX	Max pressure on the read record
Comparison	: DISP ?	DISPX
	RPM ?	RPMX
	SP ?	PRESX
PARAYi	: DISP	Displacement of the chosen motor
	QREQ	Maximum required flow
	PRES	Pressure required by the load
Printed parameters	: DISP	
	QREQ	
	PRES	
	RPMX	
	PRESX	
	EFMV	
	EFMM	
	RPM	
	TRQ	

FIGURE 3.6- An example parameter list for component selection subroutines

of the component files. The key-record has the beginning record numbers of data blocks, so first it is read and then the beginning record is directly accessed.

File name and file number are written on the first and the second records respectively. The third record is blank. The fourth is the key-record mentioned above and the fifth is also blank.

Record No.	Data					
1	FILE NAME=MOTOR					
2	FILE NO. =20					
3						
4	6	30	40			Key-record
5						
6	1.	6.50	5000.0	250.0	.00065	1.Data Block (Gear Motor)
7						
8						
28	23.	202.70	2300.0	175.	.09000	2.Data Block (Radial P. Motor)
29	0.	0.	0.	0.	0.	
30	1.	108.00	710.0	160.	.02500	
31						3.Data Block (Axial P. Motor)
32						
33						
38	9.	1053.00	160.0	160.0	.08500	
39	0.	0.	0.	0.	0.	
40	1.	9.40	7500.0	350.0	.00080	
41						
42						
43						
59	20.	1000.00	1600.0	350.0	.82400	
60	0.	0.	0.	0.	0.	

FIGURE 3.7- An example direct-access component file

In the component files, data are recorded, beginning from the sixth record. Data records begin with a key-parameter. If the key-parameter is zero, it means that this record is the end of that data block. At the same time the key-parameter shows the number of records. Therefore it can be used as a control parameter to learn the position of the pointer, i.e. the variable determining which record is accessed.

The component files are loaded by a program, FLOAD . By using this program data available from the manufacturer's catalog are loaded on the direct-access component files, but in an arranged manner. FLOAD, at the same time, copies the component files in an output file which is more suitable to take a printed copy. The output file has some minor differences in arrangement but data are completely the same. It is possible to load different files at different times, but a file has to be loaded as a whole at one step.

In Fig.3.7 the structure of motor data file is given as an example. Some component file examples are also given in Appendix D.

### 3.7 USER'S MANUAL

In order to help the user a manual has been prepared. Instructions are loaded on a file and available in the program package. The user first sees a set of instructions on the screen before the execution starts.

The manual is written for the CAD program package as a whole. It comprises the component file loading program FLOAD, the main program CADHST, and all subroutines used by CADHST. It also obeys the order of the algorithm, i.e the commands are written in the order in the programs. Some commands and comments may not be seen on the screen, in that situation the nextstep should be expected.

The User's Manual contains prompting commands and some comments with their meanings. If the command demands a value to be chosen from multiple options, then the menu defined by its number is also offered.

## IV. DYNAMIC BEHAVIOUR

Hydrostatic transmission systems designed without considering the dynamic behaviour may have oscillations which are not in acceptable limits, during the transient response time.

In the present chapter, first a mathematical model for HST systems is derived and then a digital computer program for simulation purposes is presented. The program will give the values of some parameters which are used to evaluate the transient response of a given HST system.

### 4.1 SYSTEM MODELLING

The Variable-Displacement Pump and Fixed-Displacement Motor (PVMF) version of closed circuit HST systems is taken as the basic model. However, it is possible to enlarge the study to other types by some minor modifications in the model and in the computer program.

#### 4.1.1 Motor and Load

The dynamic behaviour of the hydraulic motor and the load connected to the motor via a reductor is expressed as follows:

$$J \cdot \dot{\Omega}_m + B \cdot \Omega_m + M = M_m \quad (4.1)$$

where

$\Omega_m$  = motor speed (refer to Eq.(2.7))

$J$  = inertia moment (refer to Eq.(2.3))

$B$  = viscous friction coefficient (refer to Eq.(2.4))

$M$  = load torque (refer to Eq.(2.5))

$M_m$  = output torque on the motor shaft



where

- $Q_p$  = pump flow rate  
 $Q_m$  = motor flow rate  
 $\beta_e$  = equivalent bulk modulus

Equivalent bulk modulus  $\beta_e$  is calculated by the formula below [5],

$$\frac{1}{\beta_e} = \frac{1}{\beta_o} + \frac{1}{\beta_p} + \frac{s}{\beta_a} \quad (4.8)$$

where  $\beta_o$  is bulk modulus of oil,  $\beta_p$  is of pipe and  $\beta_a$  is of the air in the oil. 's' denotes volumetric percentage of air in the oil.

For calculation of  $\beta_a$ ,

- in izothermal operation  $\beta_a = P_i$  (4.9)  
 - in adiabatic operation  $\beta_a = \gamma \cdot P_i$ ,  $\gamma = 1.4$

is taken. Hydraulic steel pipe can be treated as a thin walled cylinder so that

$$\beta_p = (t \cdot E / D) \quad (4.10)$$

where

- $t$  = wall thickness of the pipe  
 $D$  = pipe diameter  
 $E$  = modulus of elasticity of the pipe

The flow rate of the motor is obtained by

$$Q_m = (D_m / \eta_v) \Omega_m \quad (4.11)$$

where the leakage in the motor is taken into account in the volumetric efficiency of the motor,  $\eta_v$ .

Substituting Eq.(4.11) into Eq.(4.7) we have the following equation determining the dynamic behaviour of the pressure in the delivery line :

$$\dot{P}_i = (\beta_e / V) (Q_p - \frac{D_m}{\eta_v} \Omega_m) \quad (4.12)$$

If one writes Eq.(4.12) for small variations about a given operation point,

$$\dot{p}_i = (\beta_e / V) (q_p - \frac{D}{\eta_v} \omega_m) \quad (4.13)$$

is obtained.

#### 4.1.3 Pump and Control Unit

Here, axial piston, variable displacement pumps in either swashplate design (Fig.4.1) or bent-axis design are taken into consideration. In addition, it is assumed that the prime mover speed (pump drive speed) is held constant during the operation.

In a swashplate design pump the flow rate depends on the swivel angle  $\phi$  as in the following equation :

$$Q_p = K_q \cdot \text{tg } \phi \quad (4.14)$$

and in a bent-axis design pump the governing equation for the flow rate is

$$Q_p = K_q \cdot \sin \phi \quad (4.15)$$

where  $K_q$  is a constant depending on pump size, revolution speed, unit conversion coefficients and volumetric efficiency [4] .

The flow rate increases with increasing swivel angle from zero to its maximum value. Swivelling the pump over centre smoothly reverses the pump.

If we make a linearization for small variations about a given operation point the following equation is obtained for either Eq.(4.14) or Eq.(4.15) :

$$q_p = K_q \cdot \phi \quad (4.16)$$

The swivel angle is adjusted with the control variable  $U$ , by the control servomechanism of the pump. In order to eliminate nonlinear terms, writing the equation in a linearized form for small variations about a given point we get

$$\tau_p \cdot \dot{\phi} + \phi = K_s \cdot u \quad (4.17)$$

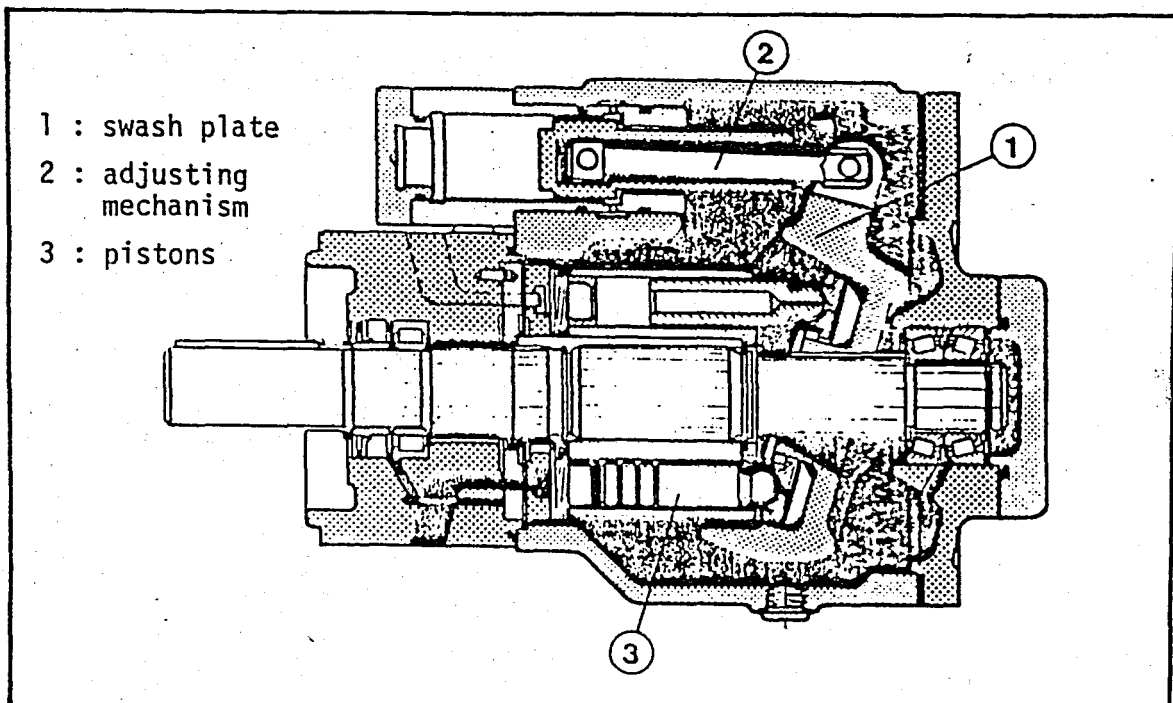


FIGURE 4.1 - A swash plate design axial piston pump with variable displacement (Rexroth A1V)

where  $\tau_p$  is the time constant of the servomechanism and  $K_s$  is a constant for which in Appendix B an evaluation method is given.

Combining Eq.(4.16) and Eq.(4.17) we obtain

$$\dot{q}_p = \frac{1}{\tau_p} (-q_p + K_s K_q u) \quad (4.18)$$

which determines the dynamic behaviour of the pump in basis of flow rate.

The pump can be controlled by various means; e.g. manual (via control lever), electrical (via solenoids), hydraulic (pressure, speed or torque related). In all of these a control quantity is applied; that may be a current, an angle of lever or a pilot pressure. Here the electrical control is taken into consideration since it best fits our study.

Via two solenoids on the electrical control device the adjusting cylinder in the pump is pressurized with an adjusting pressure relative to the pre-selected current, thereby allowing infinite adjustment of the swash plate and thus the displacement. One control solenoid is allocated to each flow direction.

In Appendix B it is explained how the control parameters ( $\tau_p$ ,  $K_S$  and  $K_Q$ ) can be calculated or estimated with data obtained from manufacturer's catalog.

#### 4.1.4 Dynamic Model for HST Systems without Pressure Compensation

The model determining the dynamic behaviour of HST systems about a given operation point is formed by three equations ; Eq.(4.6), Eq.(4.13) and Eq.(4.18) .

Hence rewriting,

$$\dot{\omega}_m = -(B/J)\omega_m + (D_m r_m / J)p_i - (1/J)m \quad (4.19)$$

$$\dot{p}_i = (\beta_e / V) (q_p - (D_m / r_v)\omega_m) \quad (4.20)$$

$$\dot{q}_p = (1/\tau_p) (-q_p + K_S K_Q u) \quad (4.21)$$

These three equations form the dynamic model of HST systems. The signal flow diagram of the system is shown in Fig.4.2 .

If we take  $\omega_m$  ,  $p_i$  and  $q_p$  as the state variables,  $u$  as the control variable, and  $m$  as the disturbance, we obtain the state space representation of the system. That is,

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & 0 \\ a_{21} & 0 & a_{23} \\ 0 & 0 & a_{33} \end{bmatrix} \cdot \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ b_3 \end{bmatrix} \cdot u + \begin{bmatrix} c_1 \\ 0 \\ 0 \end{bmatrix} \cdot m \quad (4.22)$$

where

State variables:  $x_1(t) = \omega_m(t)$

$x_2(t) = p_i(t)$

$x_3(t) = q_p(t)$

Control variable:  $u(t) = u(t)$

Disturbance :  $m(t) = m(t)$

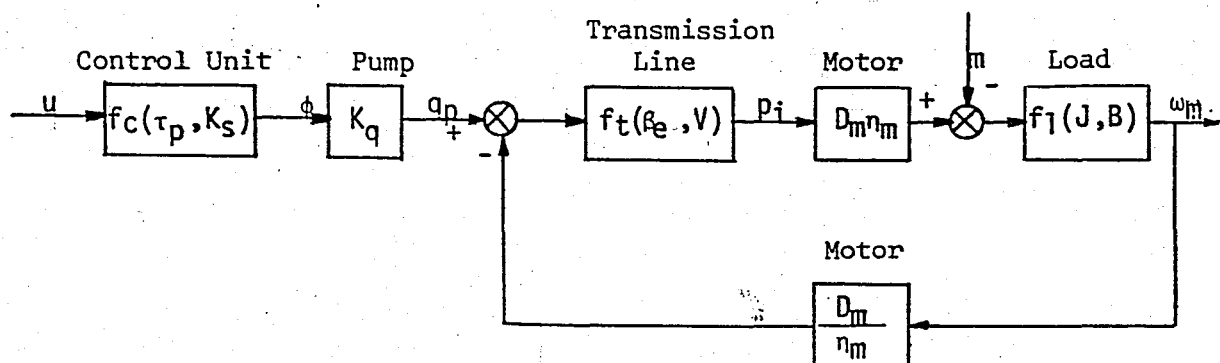


FIGURE 4.2 - The signal flow diagram of HST system

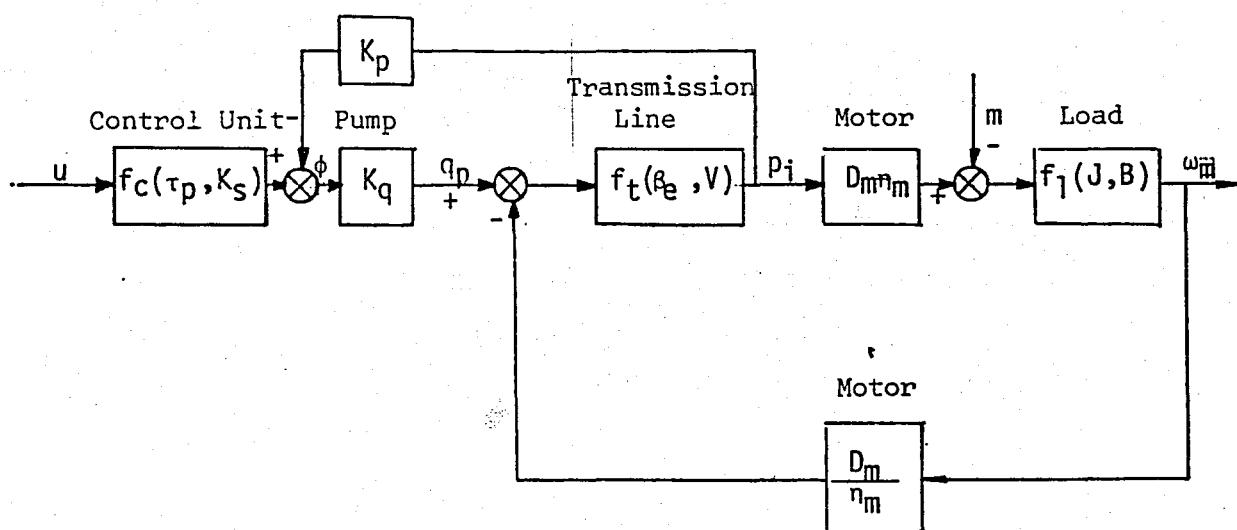


FIGURE 4.3 - The signal flow diagram of the pressure compensated HST system

Nonzero elements of matrix A :

$$a_{11} = -(B/J) \quad , \quad a_{21} = -(D_m \beta_e / V \eta_v) \quad , \quad a_{33} = -(1/\tau_p)$$

$$a_{12} = (D_m \eta_m / J) \quad , \quad a_{23} = (\beta_e / V)$$

Nonzero element of vector B :

$$b_3 = (K_s K_q / \tau_p)$$

Nonzero element of vector C

$$c_1 = -(1/J)$$

or

$$\dot{x}(t) = A x(t) + B u + C m \quad (4.23)$$

#### 4.1.5 Dynamic Model for HST Systems with Pressure Compensation

For inertial type loading , pressure compensation may be done by feeding the motor inlet pressure back into the control variable  $u$  in order to attain a reasonable damping ratio [4] . For this case, the signal flow diagram is shown in Fig.4.3 .

From the system equations and the signal flow diagram the following equation is obtained:

$$q_p(t) = \left[ \left( \frac{K_s}{\tau_p D + 1} \right) u(t) - K_p p_i(t) \right] K_q \quad (4.24)$$

where  $K_p$  is the pressure compensation gain and  $D$  is a symbol which indicates differentiation with respect to time. Opening Eq.(4.24) one has

$$\dot{q}_p = -(1/\tau_p)q_p - (K_p K_q) \dot{p}_i - (K_p K_q / \tau_p) p_i + (K_s K_q / \tau_p) u \quad (4.25)$$

Substituting Eq.(4.20) into Eq.(4.25) the following equation is obtained :

$$\dot{q}_p = \left( \frac{K_p K_q D_m \beta_e}{V \eta_v} \right) \omega_m - \left( \frac{K_p K_q}{\tau_p} \right) p_i - \left( \frac{K_p K_q \tau_p \beta_e}{\tau_p V} \right) q_p + \left( \frac{K_s K_q}{\tau} \right) u \quad (4.26)$$

So, Eq.(4.19), Eq.(4.20) and Eq.(4.26) form the dynamic model of the pressure compensated HST system. We can write them in matrix notation, as the state variables, the control variable and the disturbance the same as those in the preceding case. Hence,

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & 0 \\ a_{21} & 0 & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix} \cdot \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ b_3 \end{bmatrix} \cdot u + \begin{bmatrix} c_1 \\ 0 \\ 0 \end{bmatrix} \cdot m \quad (4.27)$$

where

$$\begin{aligned} a_{11} &= -(B/J) & , & & a_{12} &= (D_m \eta_m / J) & , & & a_{23} &= (\beta_e / V) \\ a_{21} &= -\left(\frac{D_m \beta_e}{V \eta_v}\right) & , & & a_{32} &= -\left(\frac{K_p K_q}{\tau_p}\right) & , & & a_{33} &= -\left(\frac{K_p K_q \beta_e V}{\tau_p V}\right) \\ b_3 &= (K_s K_q / \tau_p) & , & & c_1 &= -(1/J) & , & & a_{31} &= \left(\frac{K_p K_q D_m \beta_e}{V \eta_v}\right) \end{aligned}$$

or

$$\dot{x}(t) = A x(t) + B u(t) + C m(t) \quad (4.28)$$

Consequently we have the same equation as Eq.(4.23), however, some elements of the matrix A are different.

Pressure compensation gain  $K_p$  is explained in Appendix B by giving the method of evaluation.

#### 4.2 SOLUTION OF SYSTEM MATRIX DIFFERENTIAL EQUATION

The form of the dynamic models expressed by Eq.(4.23) and Eq.(4.28) is suitable to analyse the system by means of a digital computer.

The equation to be solved is a linear time-invariant matrix differential equation:

$$\dot{x}(t) = A x(t) + B u(t) + C m(t) \quad (4.29)$$

subject to the initial condition  $x(t_0) = x_0$

The solution of Eq.(4.29) is given by (for the solution procedure see Appendix C, |10| )

$$x(t) = e^{A(t-t_0)} x(t_0) + \int_{t_0}^t e^{A(t-\tau)} B u(\tau) d\tau + \int_{t_0}^t e^{A(t-\tau)} C m(\tau) d\tau \quad (4.30)$$

To enable the use of the digital computer, we could discretize the equation, Taking  $t_0 = 0$ ,

$$x(t) = e^{At} x_0 + \int_0^t e^{A(t-\tau)} B u(\tau) d\tau + \int_0^t e^{A(t-\tau)} C m(\tau) d\tau \quad (4.31)$$

Assuming that  $u$  and  $m$  are constant between any two consecutive sampling instants (for small step sizes,  $T$ ), or  $u(t) = u(kT)$  for the  $k$ th sampling period, then  $x((k+1)T)$  can be obtained as follows (for details see Appendix C)

$$x((k+1)T) = e^{AT} x(kT) + \int_0^T e^{A\tau} B d\tau u(kT) + \int_0^T e^{A\tau} C d\tau m(kT) \quad (4.32)$$

If we define,

$$G(T) = e^{AT} \quad (4.33)$$

$$H(T) = \int_0^T e^{A\tau} B d\tau \quad (4.34)$$

$$Q(T) = \int_0^T e^{A\tau} C d\tau \quad (4.35)$$

then it becomes

$$x((k+1)T) = G(T) x(kT) + H(T) u(kT) + Q(T) m(kT) \quad (4.36)$$

So we get a vector matrix difference equation which represents the solution of Eq.(4.29) only at discrete points of time,  $t = kT$ ,  $k = 0, 1, 2, \dots$



Consequently, the problem reduces to the processing of Eq.(4.36) in the computer. Once  $G(T)$  matrix and  $H(T)$ ,  $Q(T)$  vectors are computed according to  $A$  matrix and  $B$ ,  $C$  vectors, then the values of state variables  $\omega_m$ ,  $p_i$ , and  $q_p$  at every multiple of the step size are obtained, provided that the initial values of variables are given.

### 4.3 SIMULATION AND COMPUTER PROGRAM

#### 4.3.1 General Algorithm

A mathematical model for the dynamic behaviour of the HST systems was given in the previous sections. The model is composed of three coupled differential equations which are written for motor speed, motor inlet pressure and pump flow rate. The model is then reduced to a matrix difference equation to enable the use of the digital computer.

The main object is to determine variations of the state variables mentioned above during the transient response time, according to a step change in the input variable  $U$  or in the torque disturbance  $M$ . The input variable is the current value to be fed into the solenoids of the electrical control unit of the pump. The load torque is the disturbance variable. Both of them, i.e. the current and the torque, may have a step change, alone or together. The system's responses to them can be added since the system is linear.

The composition of the simulation program consists of a few subroutines written for doing necessary computations, and a main program organizing these tasks, which are explained in the following sections.

#### 4.3.2 Main Program

A general flow chart of the main program named HSTSIM is shown in Fig.4.4. HSTSIM is an interactive program which admits data during execution and has some facilities to suit the needs of the inexperienced user. It calls some subroutines: DISCR3, GMPRD, MINV, SMAT, SPARA.

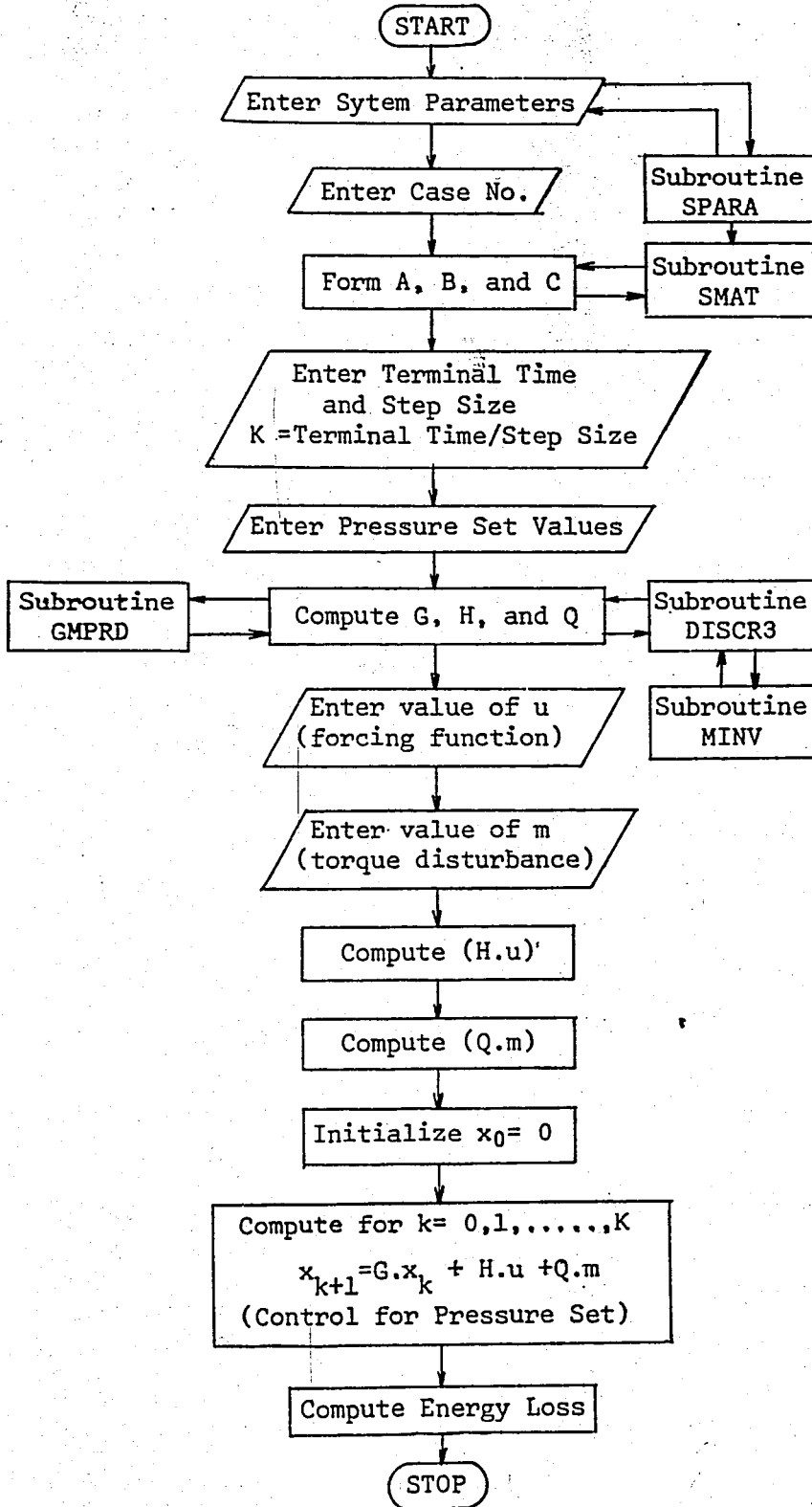


FIGURE 4.4 - General flow chart of the dynamic simulation program

#### 4.3.3 Subroutine for Computing the Discrete-Time System Matrices

In order to achieve a better result, the discrete-time system matrices and vectors should be computed precisely. It depends on the method used. There are some approximation methods to compute  $e^{AT}$  matrix and its integral over the time interval.

Here a subroutine named DISCR3 is used for this purpose. This subroutine computes the discrete-time system matrices corresponding to a continuous system defined by its system matrix  $A$  and its input matrix  $B$  in general, for a given time interval  $T$ . The largest convergent step size is computed by the program and the discrete system matrices are obtained by using doubling formulas. Selection of the largest convergent step size and the corresponding doubling number is based on Van Loan's Criterion [11]. Third order Pade approximations are used to compute matrix exponentials.

DISCR3 calls subroutine MINV which takes the inverse of a matrix and subroutine GMPRD which is used for matrix production. MINV uses standart Gauss-Jordan method.

DISCR3 is called twice by the main program, the first time for computing  $G$  and  $H$ , the second time for computing  $G$  and  $Q$ .

#### 4.3.4 Pressure Relief Valve and Energy Losses

The simulation program, as mentioned earlier, gives deviations of the state variables from an initial steady state operation point for a step change either in the pump control current or in the load torque.

If the system pressure reaches the pressure set value of the pressure relief valve, then relief valve is opened and the excessive flow passes from high pressure side of the circuit, through the valve, to the low pressure line to the pump. It means energy dissipation.

This situation is simulated in the program by a pressure control process. If the pressure is greater than the set value then the pressure is made equal to this set value.

The useful flow rate can be calculated by making the derivative of the pressure equal to zero in Eq.(4.20).

So,

$$q_p^x = (D_m / \eta_v) \omega_m \quad (4.39)$$

is obtained. Hence the energy loss can be determined by integrating the following equation over the time in which the relief valve is open:

$$\dot{E}_l = P_i^x (q_p - q_p^x) \quad (4.40)$$

where

$P_i^x$  = actual pressure set value

$q_p$  = pump flow rate

$q_p^x$  = useful pump flow rate

Integration is made easily by using the trapeze rule as the state variables are already computed for every step size. So we get a comparison value, however, it should be noted that the dynamics of the relief valve is omitted here.

## V. EXAMPLES

## 5.1 EXAMPLE DESIGN FOR A TRANSLATIONAL LOAD DRIVE SYSTEM

In Fig.5.1 the layout of an example hydraulic circuit is shown. According to the load and type specifications entered by the user, the created design program selected the necessary components from the component files and made the necessary calculations. Besides that, the pressure loss computation was also made. This process could be repeated for different oil viscosity values, so the effect of the viscosity number on the pressure losses could be also seen.

The printout taken during the program running is presented in Appendix E. The friction losses of cylinder is taken into account by multiplying the load by 1.15 .

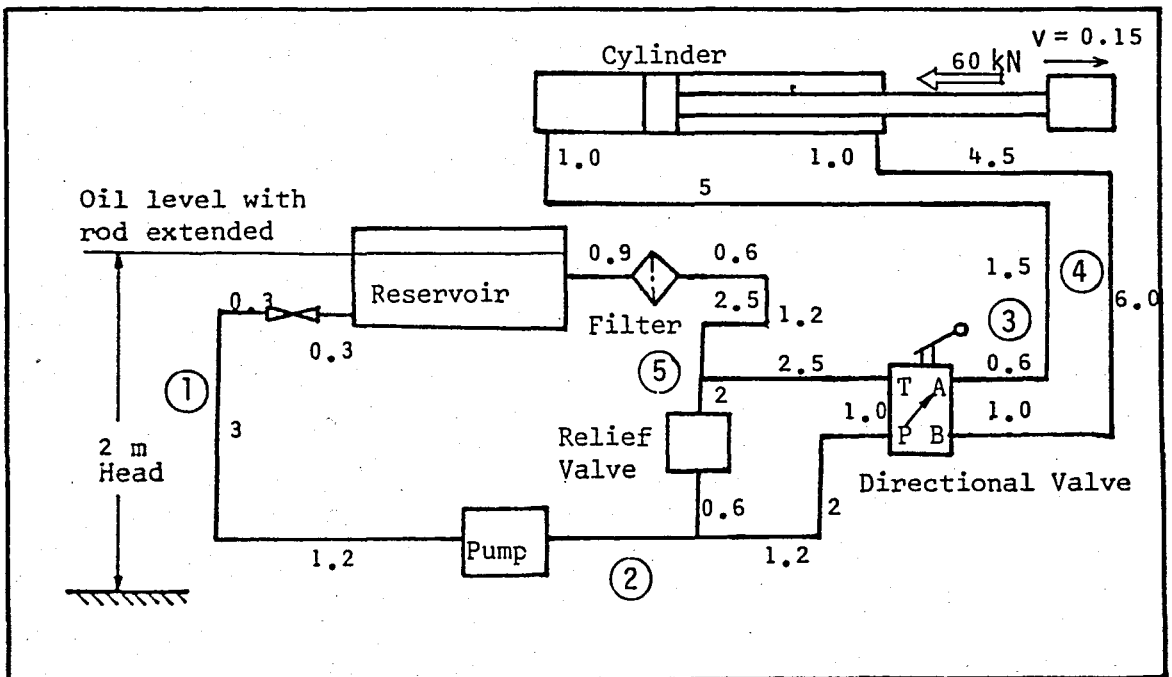


FIGURE 5.1 - Physical layout for the translational load drive system

## 5.2 DESIGN AND SIMULATION EXAMPLES FOR HST SYSTEMS

For a rotational load, a PVMF version of closed circuit HST systems is proposed. Two alternative systems are offered for the same task. The first is a straight-drive system which is composed of a fixed radial piston motor and a variable axial piston pump. In the second system a reductor and a fixed axial piston motor are used.

The printouts of the CAD program are presented in Appendix E. In addition, all parameters used in the CAD and the simulation program are listed in Table 5.1. The parameters used in the simulation program are preceded by an asterisk.

After the design process the simulations of the designed systems were carried out in the computer by the created simulation program. The initial conditions belonging to the actual values of the state variables were calculated at the same load speed in order to make the comparison of the dynamic behaviours of the two alternative systems easy.

The speed of the load at the given operation point is

$$\Omega_1 = 16 \text{ [rd/s]}$$

So the motor speeds are

$$\Omega_{m1} = 16 \text{ [rd/s]}$$

and

$$\Omega_{m2} = 96 \text{ [rd/s]}$$

where the subscript of 1 denotes the first design (straight-drive) and 2 denotes the second design (with reductor). The other parameters were calculated at this operation point. To observe the dynamic behaviours, step changes as large as various percentages of the design control current ( $U_m$ ) were applied at this point. Deviations of motor speed, motor inlet pressure and pump flow rate are illustrated in the figures 5.2 - 5.18. The same process was repeated for the pressure compensated case. Pressures were limited at the relief valve set values and the energy losses were computed. The simulation program was also used to determine the responses of the systems to step changes in the load torque. The effect of the pressure compensation was also observed in this case. The step size was taken as small as 0.001 seconds in all executions.

TABLE 5.1 - Parameter list of designed HST systems

Parameter	Unit	Design No.1	Design No.2
<u>Load Parameters</u>			
$M_l$	Nm	36.0	36.0
$B_l$	Nm/rd/s	20.0	20.0
$J_l$	kg-m <sup>2</sup>	7.9	7.9
$n_l$	rpm	210.0	210.0
$\Omega_l$	rd/s	22.0	22.0
<u>Reductor Parameters</u>			
N		1 (no reductor)	6
$B_r$	Nm/rd/s	0.0	0.01
$J_{r1}$	kg-m <sup>2</sup>	0.0	0.3
$J_{r2}$	kg-m <sup>2</sup>	0.0	0.1
$M_{fr}$	Nm	0.0	0.5
<u>Motor Parameters</u>			
Type		Radial Piston(Fix.)	Axial Piston(Fix.)
x $D_m$	m <sup>3</sup> /rd	$4.743 \times 10^{-5}$	$0.356 \times 10^{-5}$
$J_m$	kg-m <sup>2</sup>	0.1	0.0015
x $n_m$		0.93	0.94
x $n_v$		0.90	0.92
<u>Shifted Parameters</u>			
x J	kg-m <sup>2</sup>	8.0	0.332
x B	Nm/rd/s	20.0	0.566
x M	Nm	36.0	6.5
x $\Omega$	rd/s	22.0	132.0
<u>Rated System Pressure</u>			
P	bars	120	240

TABLE 5.1 - Continued

Parameter	Unit	Design No.1	Design No.2
<u>Pump Parameters</u>			
Type		Swash plate	Bent axis
$D_p$	m <sup>3</sup> /rd	$0.872 \times 10^{-5}$	$0.447 \times 10^{-5}$
$\phi_{\max}$	degree	15.0	25.0
x $K_s$	rd/A	0.67	1.056
x $K_q$	m <sup>3</sup> /rd-s	$4.695 \times 10^{-3}$	$1.526 \times 10^{-3}$
x $\tau_p$	s	0.3	0.1
$U_m$	A	0.3684	0.321
<u>Pipe Line Parameters</u>			
$d_{\text{delivery}}$	m	0.0209	0.0158
$L_{\text{delivery}}$	m	2.5	2.5
x $V$	m <sup>3</sup>	$1.209 \times 10^{-3}$	$0.566 \times 10^{-3}$
<u>Bulk Modulus</u>			
$\beta_p$	N/m <sup>2</sup>	$27.1 \times 10^9$	$34.0 \times 10^9$
$\beta_a$	N/m <sup>2</sup>	$1.68 \times 10^7$	$3.36 \times 10^5$
s	%	1	1
$\beta_o$	N/m <sup>2</sup>	$1.38 \times 10^9$	$1.38 \times 10^9$
x $\beta_e$	N/m <sup>2</sup>	$0.737 \times 10^9$	$0.95 \times 10^9$
<u>Pressure Compensation</u>			
x $K_p$	m <sup>3</sup> /sA	$0.7 \times 10^{-8}$	$0.7 \times 10^{-8}$
x $U_m$	A	0.473	0.467
<u>Relief Valve</u>			
$P_{\text{set}}$	bars	120.0	270.0
<u>Operation Point Values</u>			
$\Omega_m$	rd/s	16.0	22.0
$P_i$	bars	80.708	181.795



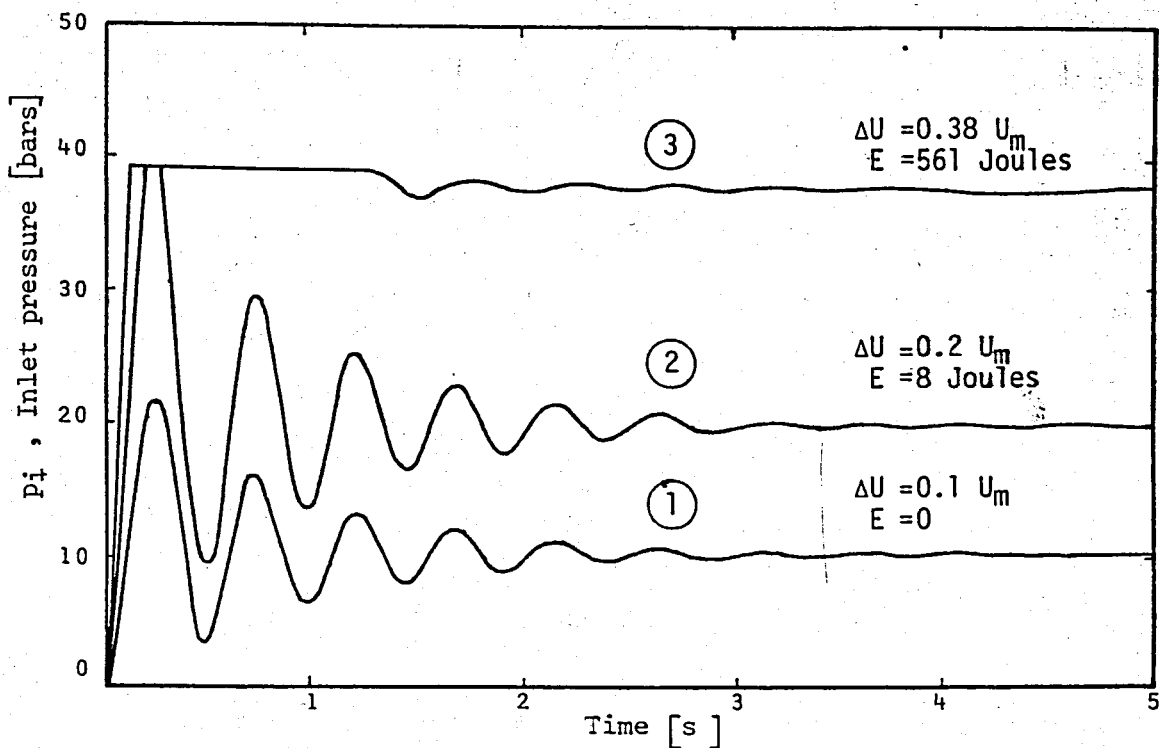


FIGURE 5.2 - Deviations in the inlet pressure  
(Design No.1 , No pressure compensation)

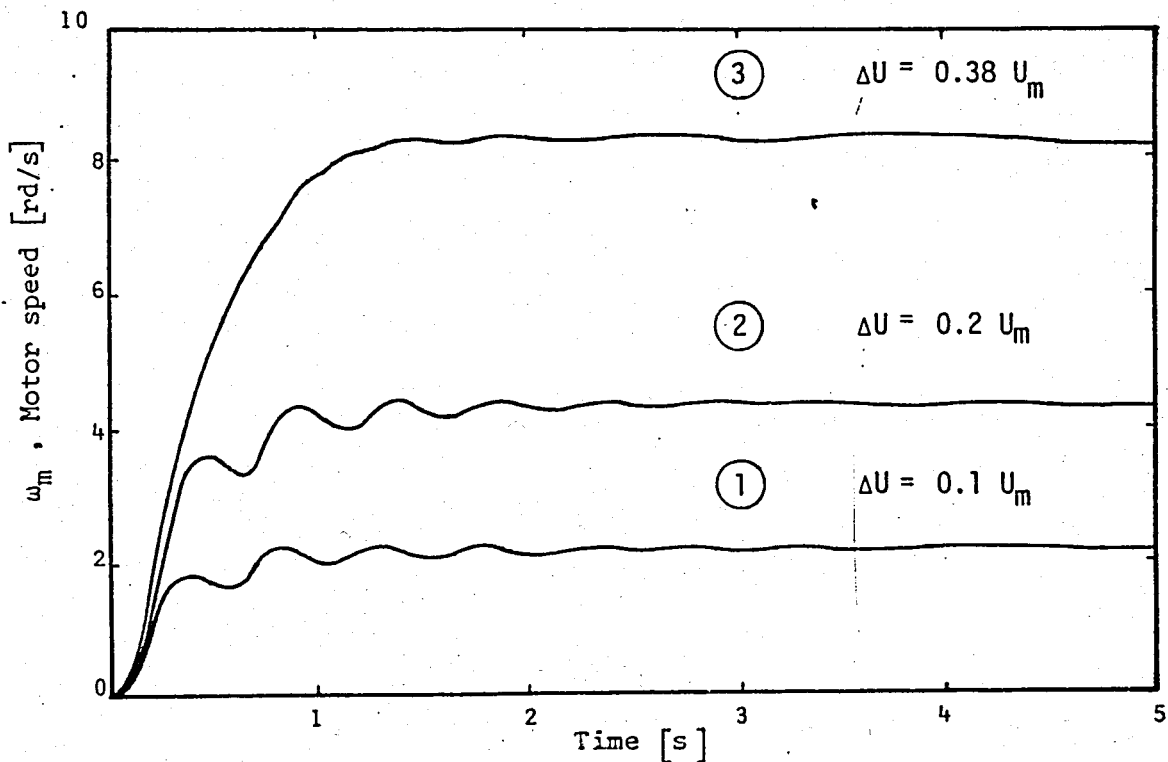


FIGURE 5.3 - Deviations in the motor speed  
(Design No.1 , No pressure compensation)

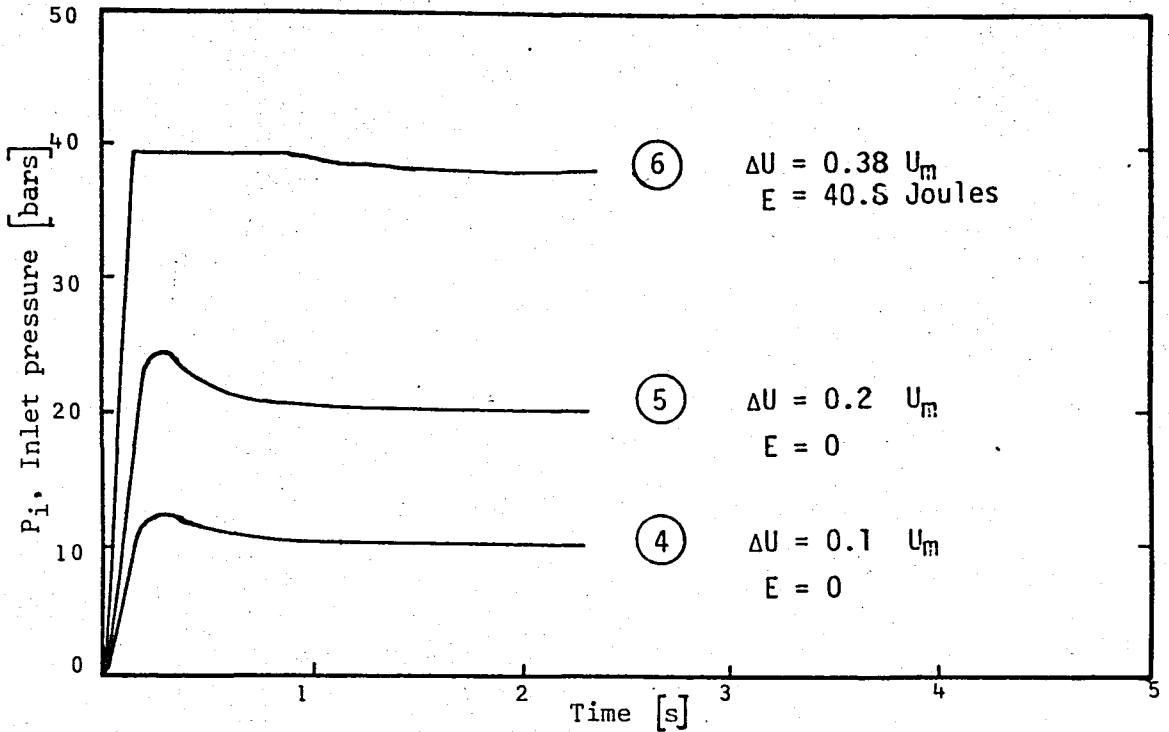


FIGURE 5.4 - Deviations in the inlet pressure  
(Design No.1, Pressure compensated case)

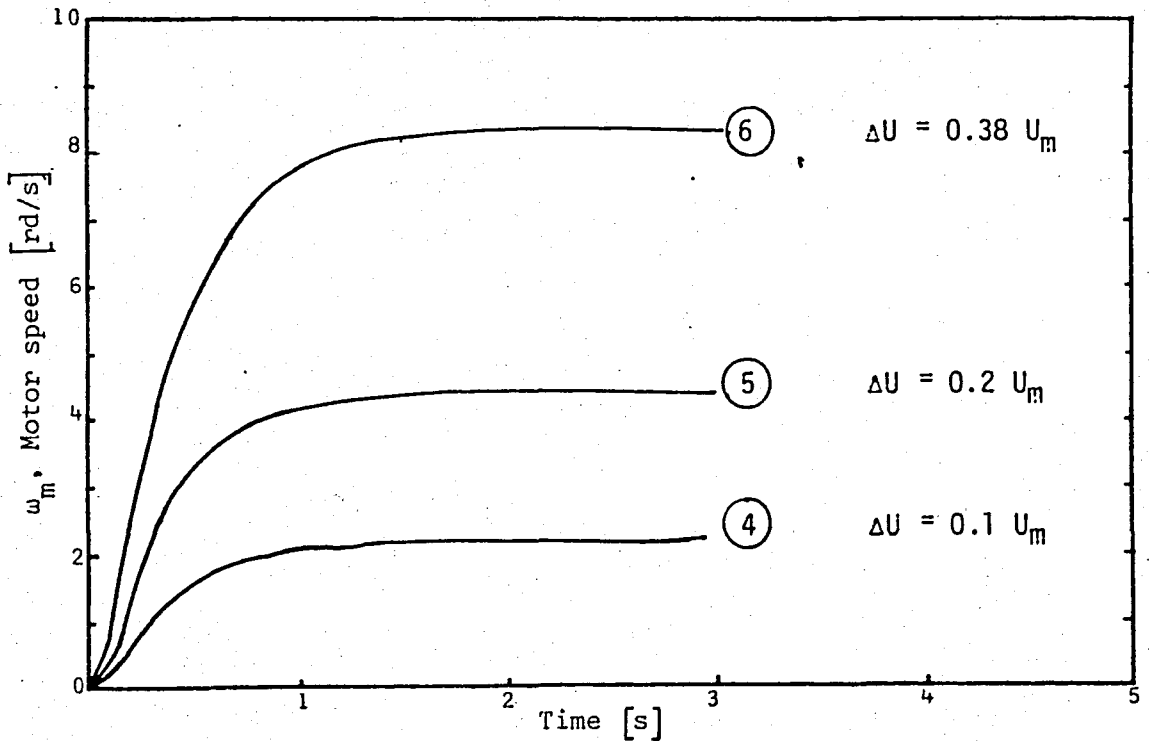


FIGURE 5.5 - Deviations in the motor speed  
(Design No.1, Pressure compensated case)

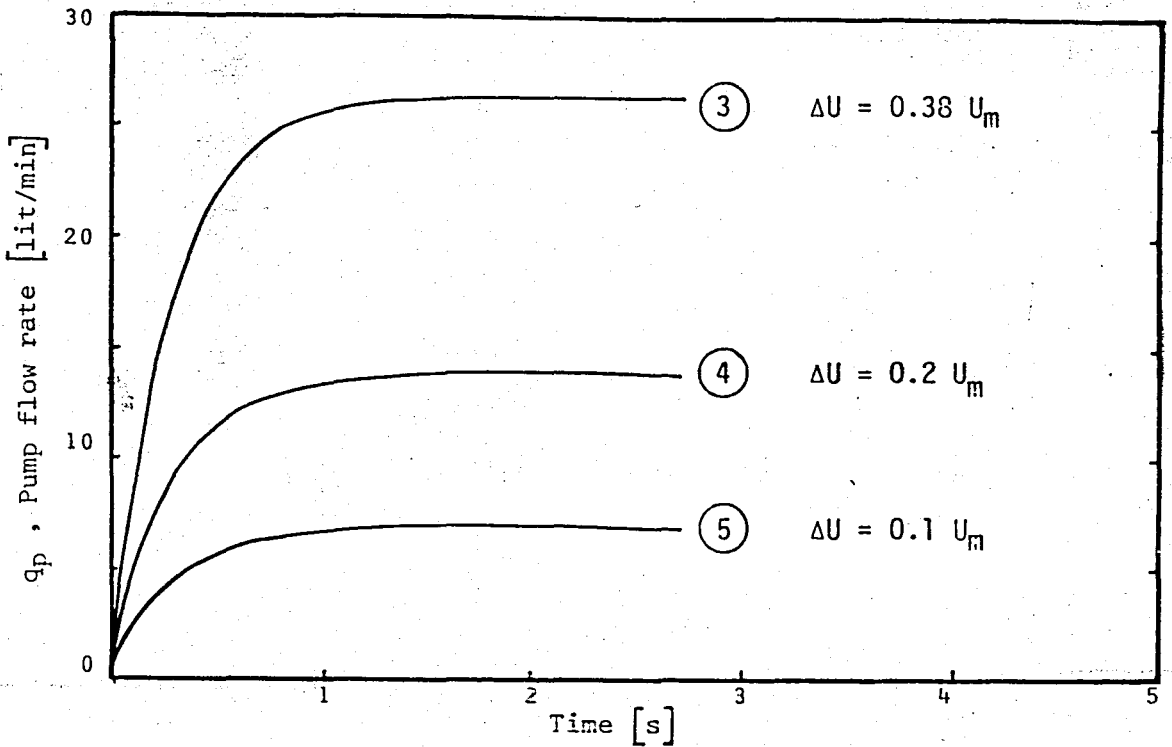


FIGURE 5.6 - Deviations in the pump flow rate  
(Design No.1, No pressure compensation)

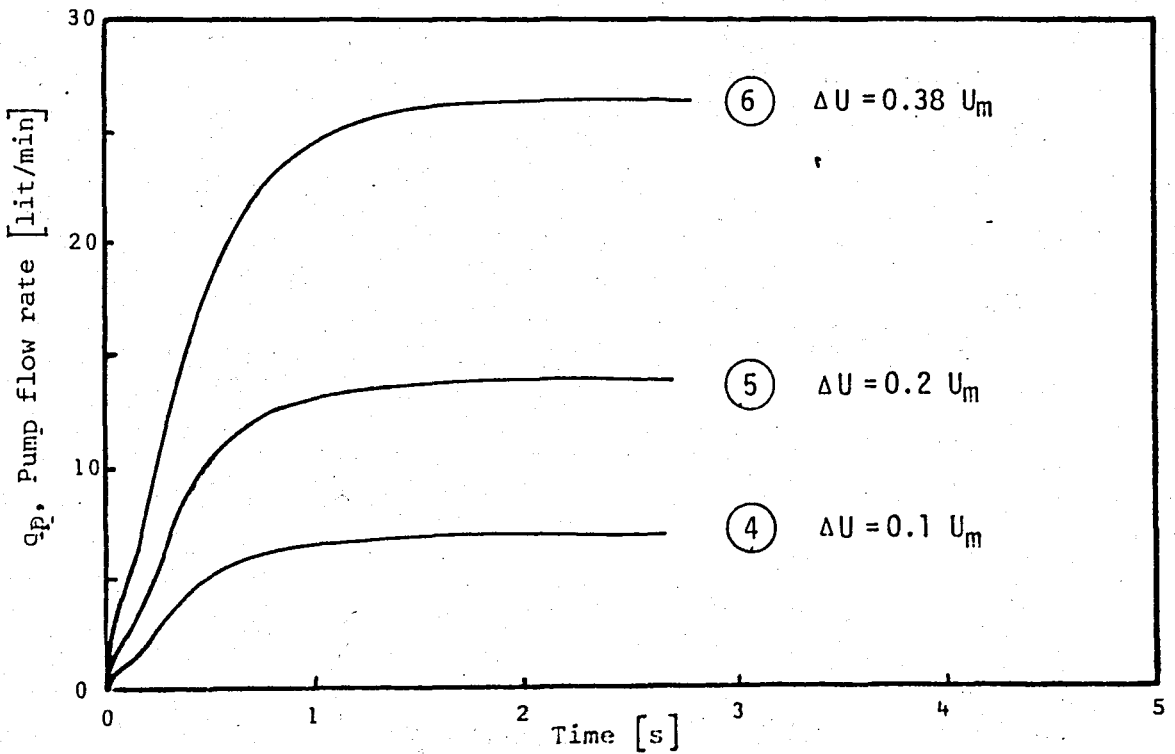


FIGURE 5.7 - Deviations in the pump flow rate  
(Design No.1, Pressure compensated case)

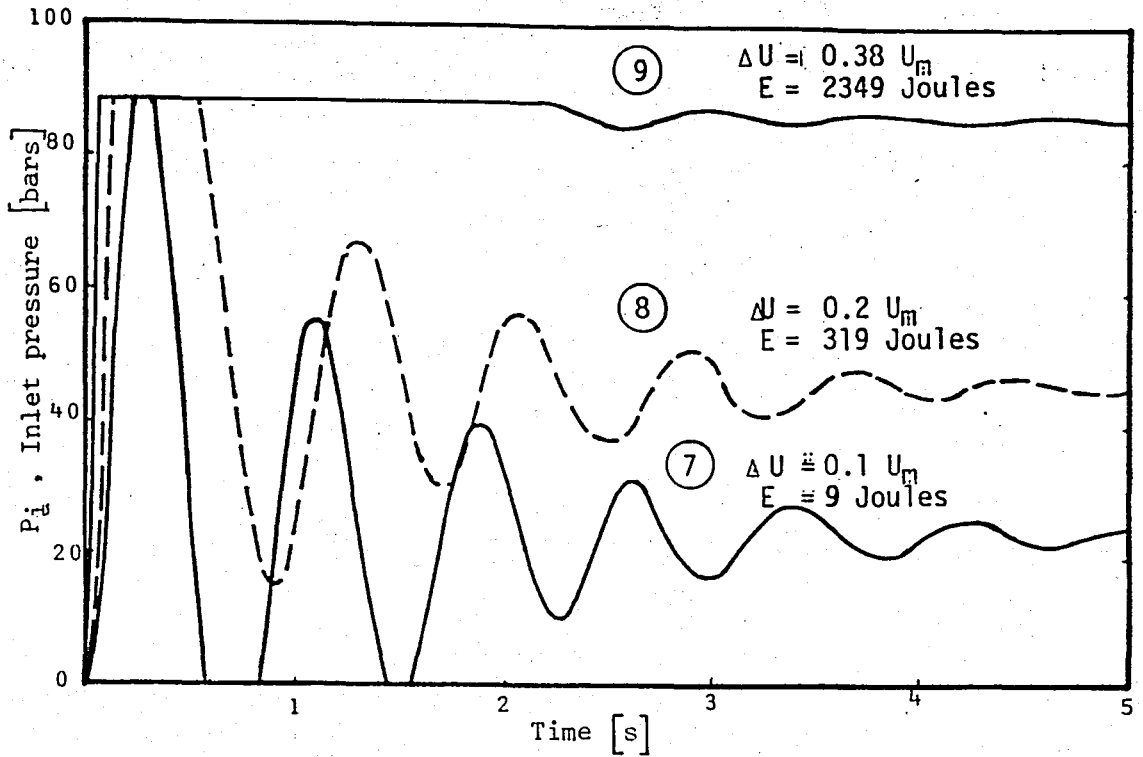


FIGURE 5.8 - Deviations in the inlet pressure  
(Design No.2, No pressure compensation)

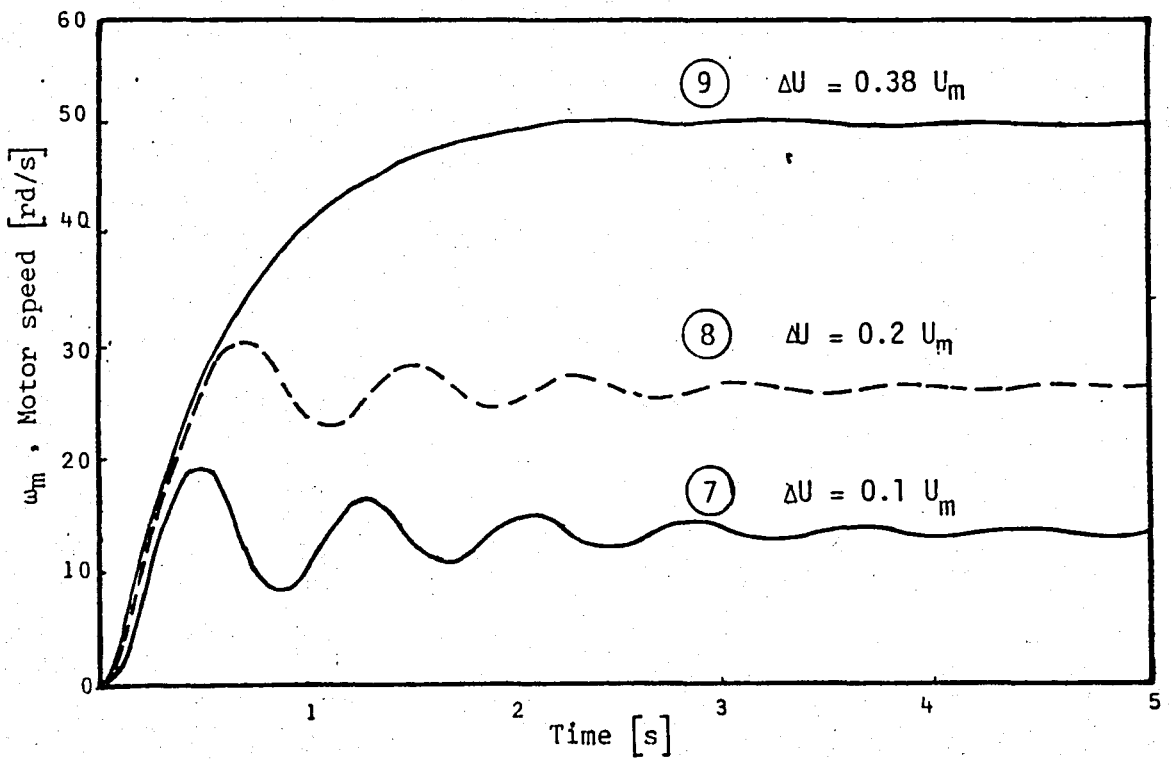


FIGURE 5.9 - Deviations in the motor speed  
(Design No.2, No pressure compensation)

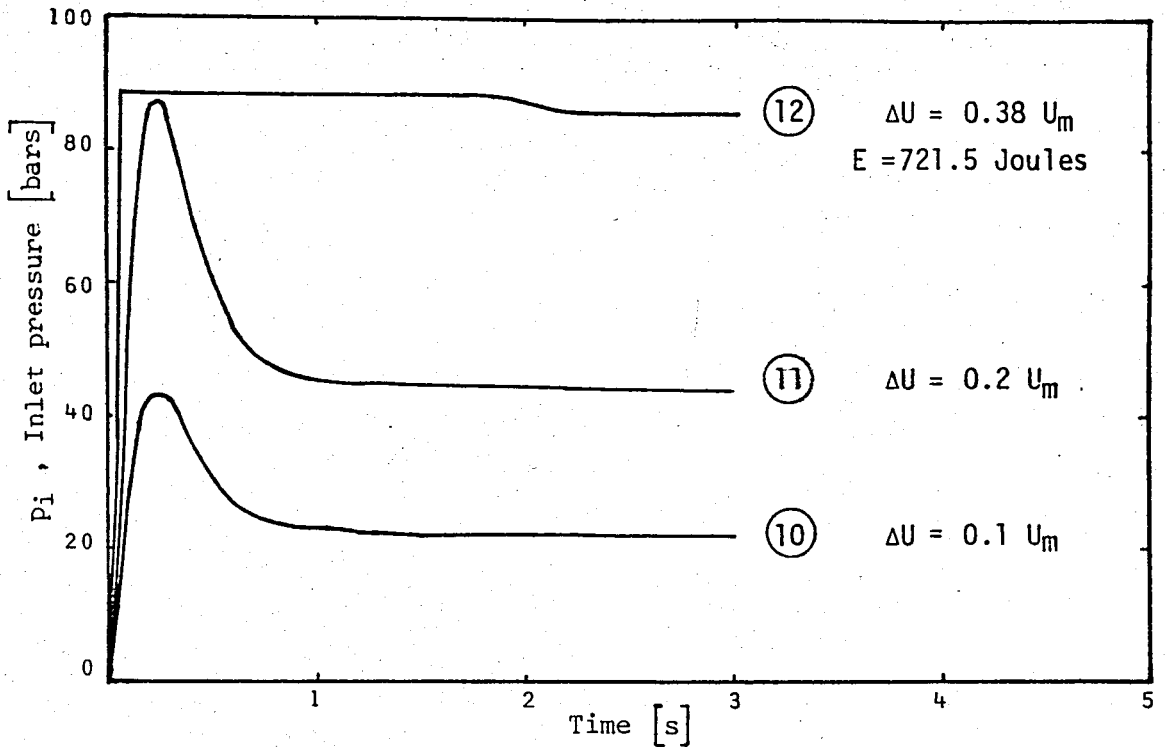


FIGURE 5.10 - Deviations in the inlet pressure  
(Design No.2, Pressure compensated case)

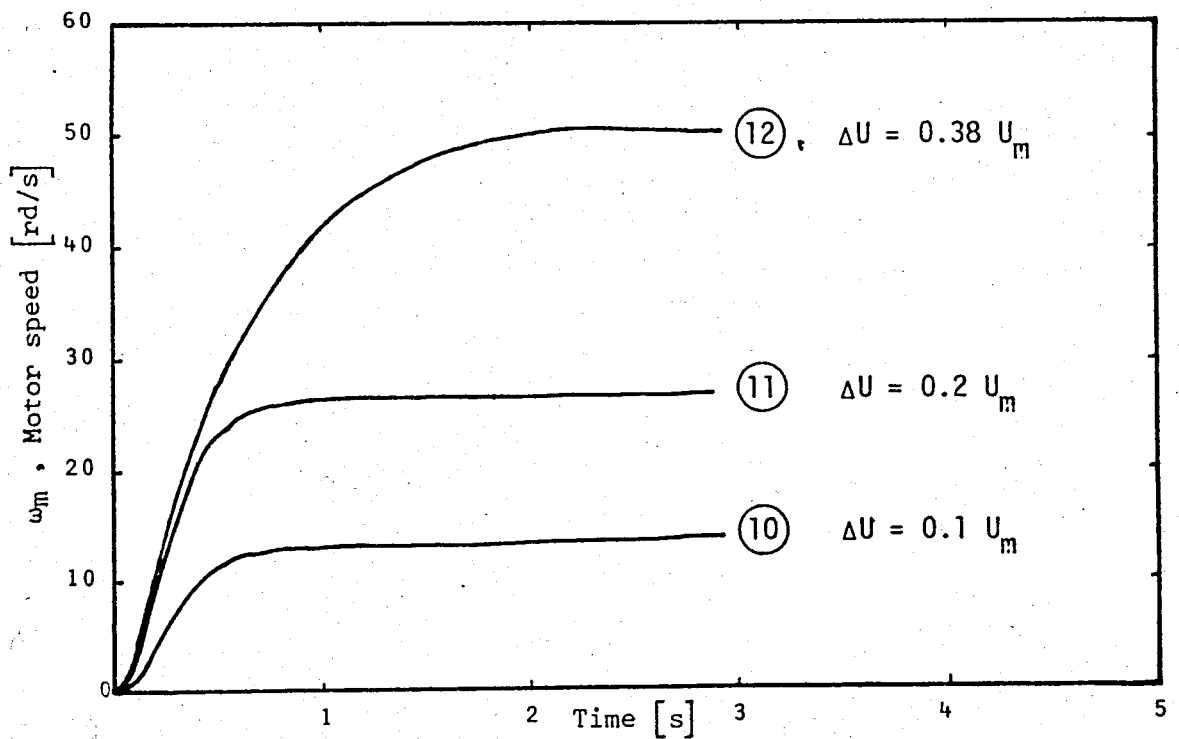


FIGURE 5.11 - Deviations in the motor speed  
(design No.2, Pressure compensated case)

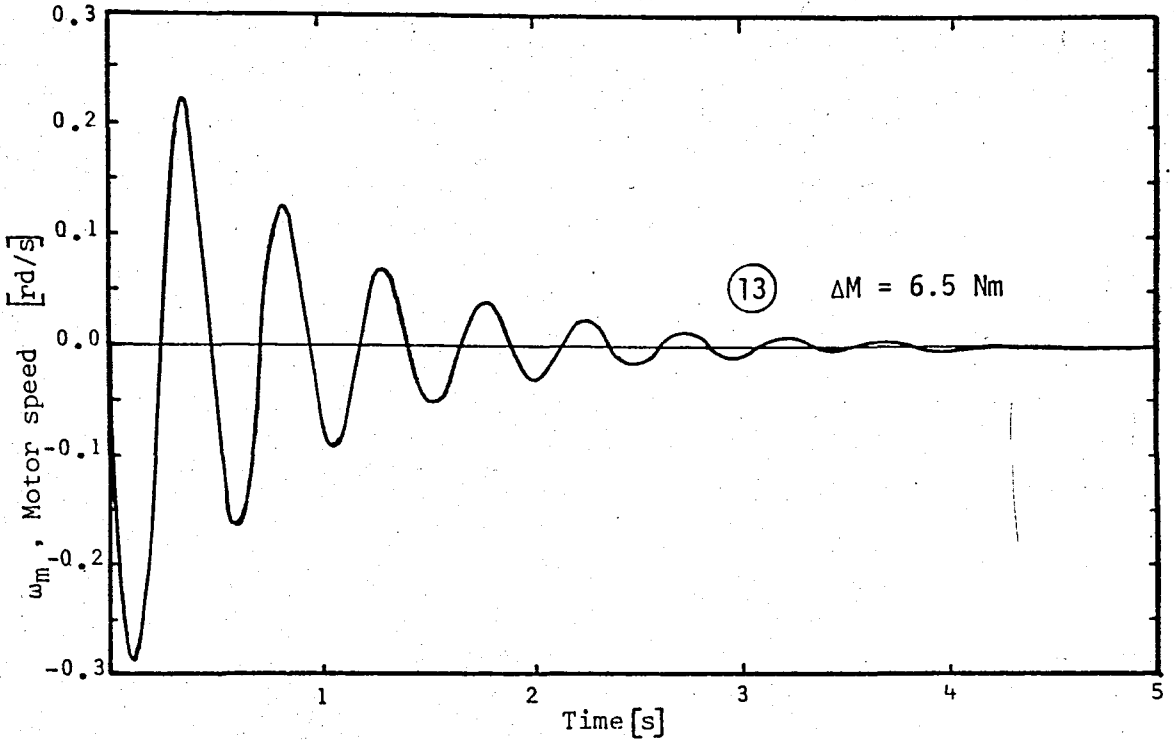


FIGURE 5.12 - Dynamic response of the motor speed to a step change in the load torque (Design No.1, No pressure compensation)

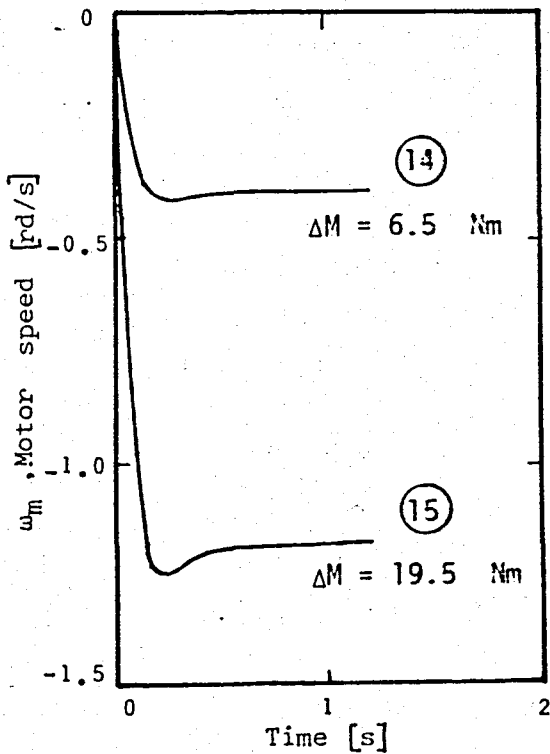


FIGURE 5.13 - Dynamic response of the motor speed to a step change in the load torque (Design No.1, Pressure Compensated)

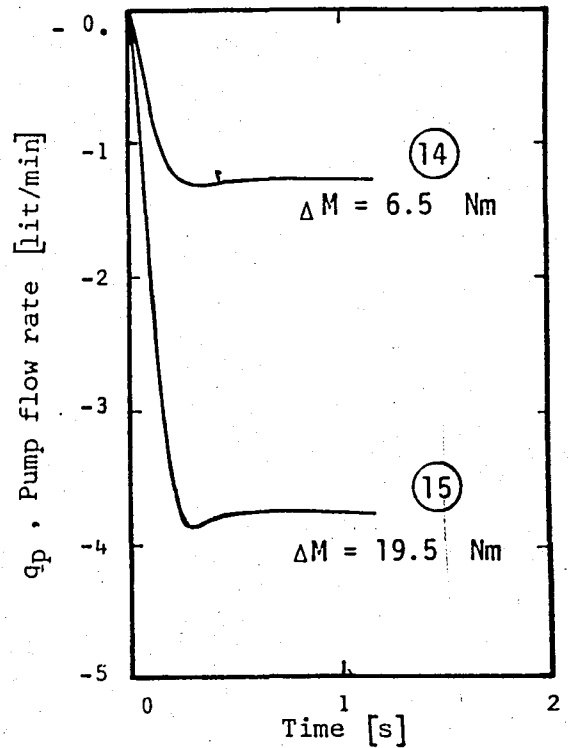


FIGURE 5.14 - Dynamic response of the pump flow rate to a step change in the load torque (Design No.1, Pressure Compensated)

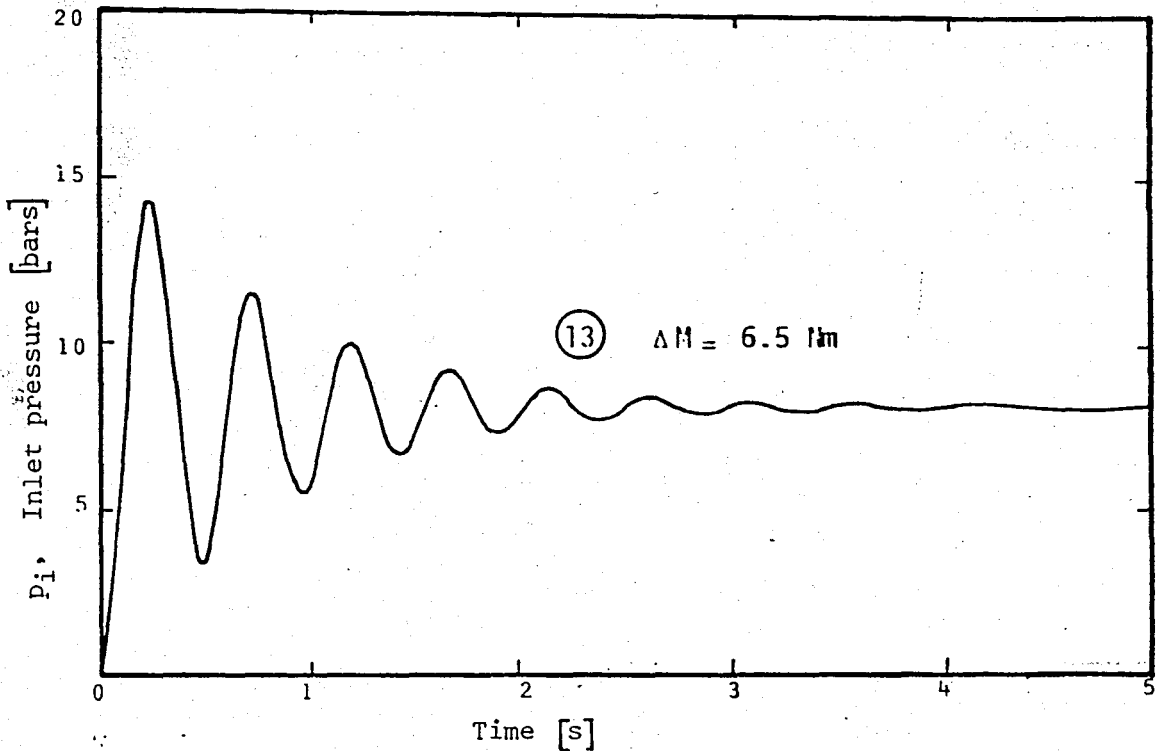


FIGURE 5.15 - Dynamic response of the inlet pressure to a step change in the load torque (Design No.1, No pressure compensation)

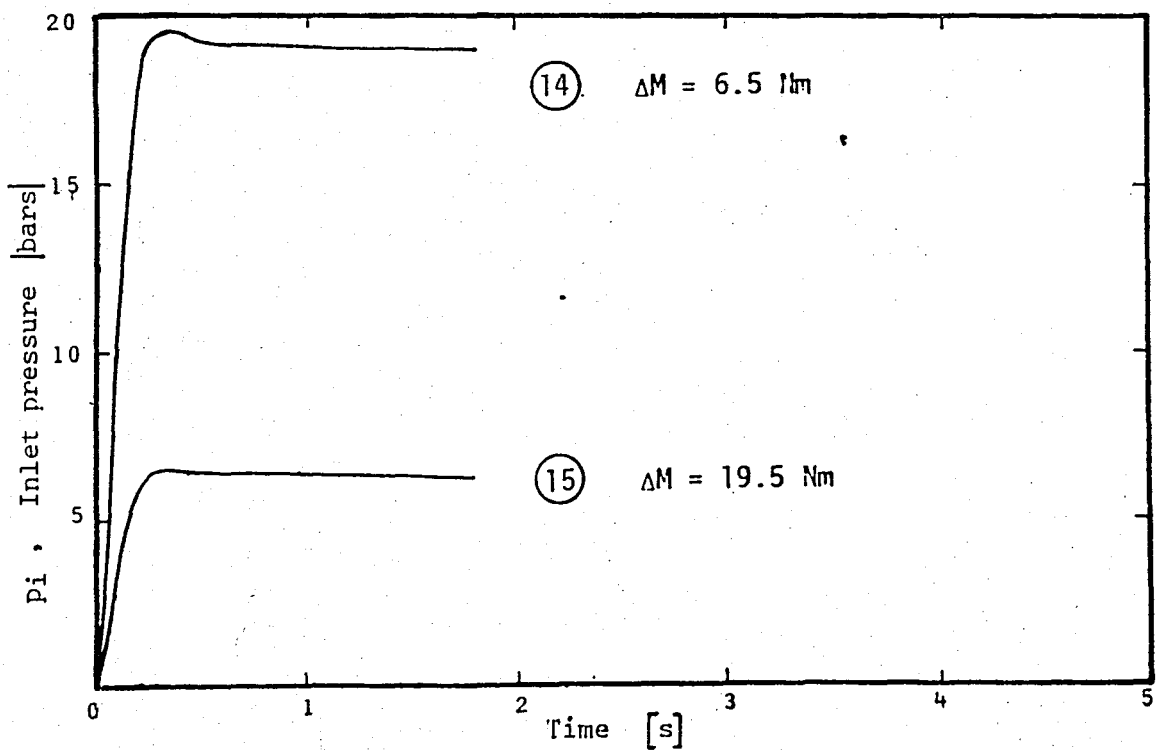


FIGURE 5.16 - Dynamic response of the inlet pressure to a step change in the load torque (Design No.1, Pressure compensated case)

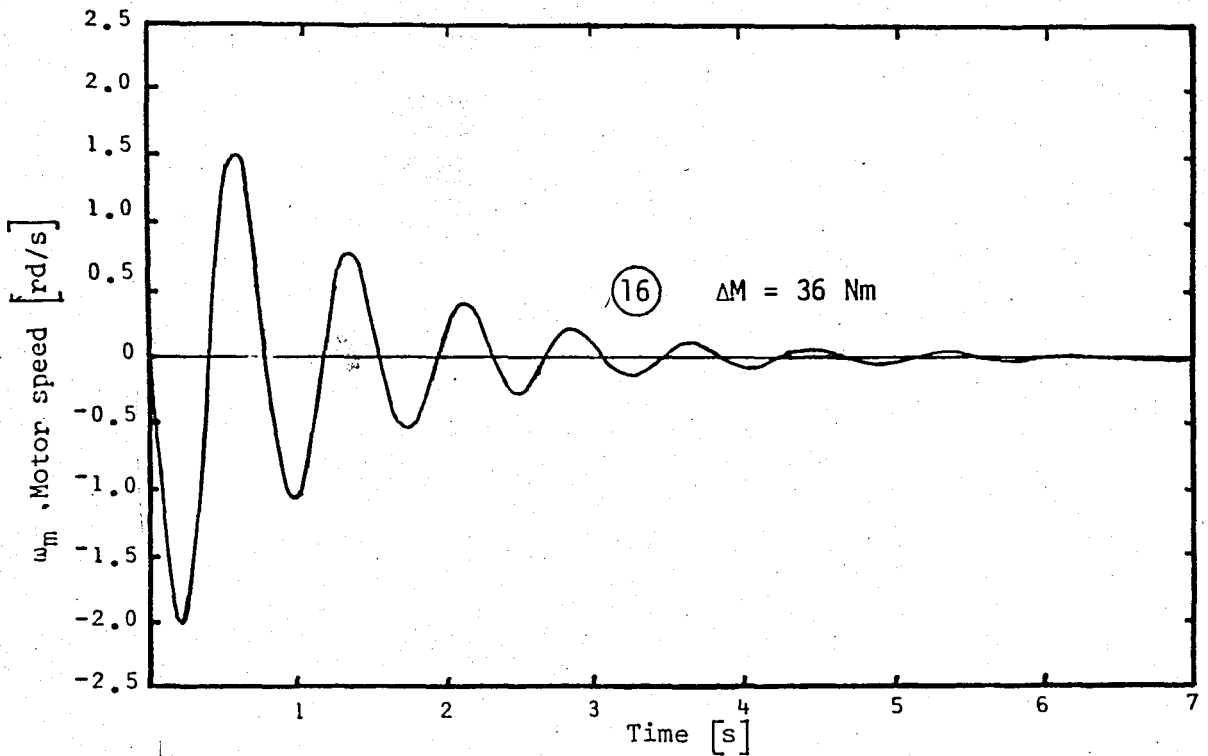


FIGURE 5.17 - Dynamic response of the motor speed to a step change in the load torque (Design No.2 , No pressure compensation)

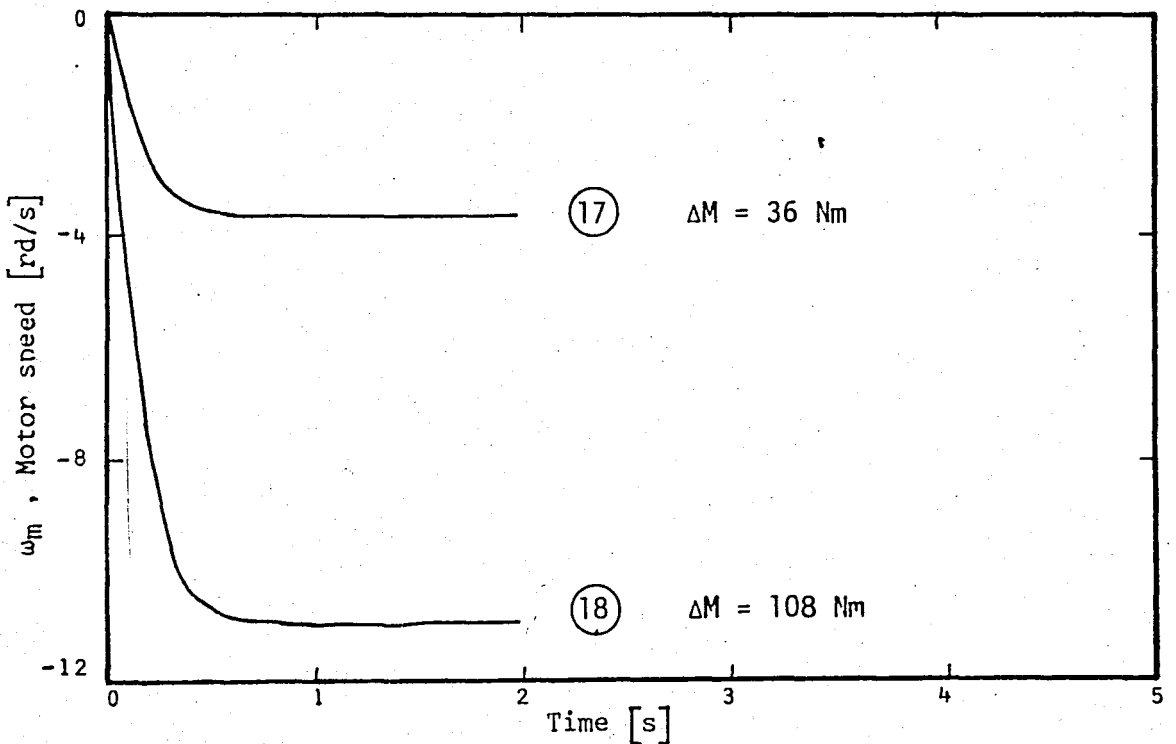


FIGURE 5.18 - Dynamic response of the motor speed to a step change in the load torque (Design No.2, Pressure compensated case)



## VI. CONCLUSION

Computer aided design and dynamic simulation techniques are very useful tools in the design of hydrostatic transmission systems. The use of the digital computer makes it possible first to design a HST system and then to simulate it in a few minutes. So the designer can obtain the best solution in a shorter time and in a more reliable way.

Some advantages of the CAD program over the conventional design process can be summarized as follows:

- i) reduces the design time,
- ii) makes all calculations which are sometimes omitted in the conventional design procedure,
- iii) makes possible to see all possible solutions together,
- iv) effects of some important parameters, such as oil viscosity number, can be easily seen.

The success of the CAD program heavily depends on the component files. With small and insufficient files, the program results in wrongly chosen components. Use of direct access files reduces the run time which is important especially when huge component files are used.

Since computer aided design of hydraulic circuits is in fact a very large field, the work presented here is only an introductory research. However, the scope and effectiveness of the program package can be easily extended so that highly complex hydraulic circuits can be designed by the CAD program.

Consideration of the dynamic behaviour is a necessity for HST systems. Simulation of HST system in the digital computer makes the CAD program complete. Usually exist several solutions to a design program. Observation of the dynamic behaviours helps the designer to choose the best solution.

The results obtained from the simulation program are outlined as follows :

i) HST systems may have oscillations during the transient response time. If the dynamic behaviour is not tolerable, then the pressure compensation may be applied. Pressure compensation reduces excessive overshoots.

ii) Pressure compensation also causes energy losses to reduce during the relief valve operation.

iii) Variations in the load torque causes some oscillations in the systems without pressure compensation, but does not affect the steady state value of the speed. However, if pressure compensation is applied then it causes a reduction in the steady state value of the motor speed.

iv) If the deviation in the motor speed is not tolerable, then by means of the closed loop control it may be reduced or completely overcome.

The dynamics of the prime mover and the pressure relief valve is omitted in the dynamic model. The study can be extended so that they are also considered.

## A P P E N D I C E S

## APPENDIX A

## GLOSSARY OF COMMON PARAMETERS USED IN CADHST

BTLOSS	Pressure loss of directional valve
DID	Internal diameter of delivery line pipe
DIFF	Height difference between pump and tank
DIS	Internal diameter of suction (return) line pipe
DISP	Motor displacement (geometric delivery)
DP	Diameter of cylinder piston
DR	Diameter of piston rod of cylinder
FKCi	Contraction loss coefficient FKC1=0.5, FKC2=0., FKC3=0., FKC4=0.5, FKC5=0.
FDV1	Directional valve p.loss coefficient (P → A)
FDV2	Directional valve p.loss coefficient (B → T)
FKEi	Expansion loss coefficient FKE1=0., FKE2=0., FKE3=1., FKE4=0., FKE5=1.
FKFI	Filter loss coefficient
FKSUMi	P.loss coefficients of fluid lines
FLOSS	P.loss of filter
LNi	Line code number LN1: suction line LN2: delivery line LN3: line A LN4: line B LN5: return line
NBD	Delivery line nominal diameter
ODIS	Density of oil used in the circuit
OVIS	Viscosity of oil used in the circuit

PALOSS	P.loss of directional valve (P A)
PLOSSi	P.loss of line i
POWER	Maximum power required by the pump
PRES	Updated system pressure
PRESS	Pump pressure differential
PRR	Pressure at pump inlet
PRV	Pressure setting value of relief valve
TFK	Total p.loss coefficient (modified)
QDELA	Flow rate in Line A
QPUMP	Flow rate required by pump
QREQ	Maximum flow rate required by actuator
QRET	Flow rate in return line
QRETB	Flow rate in line B
SP	Rated system pressure
SPEED	Updated motor speed
SPEEDB	Updated backward velocity of cylinder
SPEEDF	Updated forward velocity of cylinder
TLOSS	Total pressure loss in the circuit

## APPENDIX B

## EVALUATION OF PUMP CONTROL COEFFICIENTS

Here, the evaluation is done for the electrical control, but the procedure is almost the same for the other control techniques.

B.1 EVALUATION OF  $K_q$  AND  $K_s$ 

In Fig.B.1 the control graph for the electrical control is illustrated. The input parameter is the electrical current.

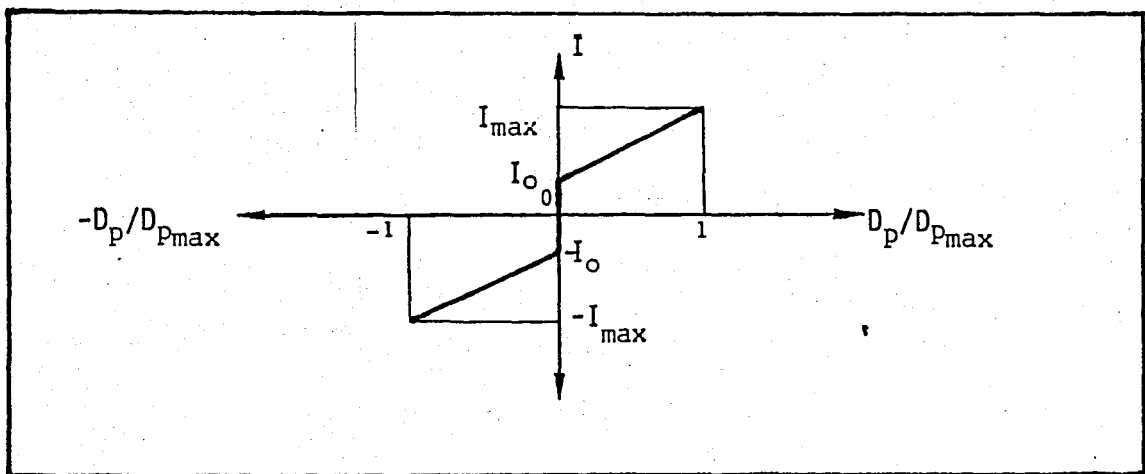


FIGURE B.1 - Control graph of the electrical control

It is clear from the figure that

$$U_{\max} = I_{\max} - I_0 \quad (\text{B.1})$$

On the other hand for the swash plate design pump,

$$Q_{P\max} = K_q \cdot \text{tg } \phi_{\max}$$

or

$$K_q = Q_{P\max} / \text{tg } \phi_{\max} \quad (\text{B.2})$$

At the same time,

$$Q_{pmax} = K_s \cdot K_q \cdot U_{max} = K \cdot U_{max}$$

or

$$K = Q_{pmax} / U_{max} \quad (B.3)$$

where

$$K = K_s \cdot K_q \quad (B.4)$$

From Eq.(B.2) and Eq.(B.3)

$$K_s = \text{tg } \phi_{max} / U_{max} \quad (B.5)$$

So, Eq.(B.2) and Eq.(B.5) give us the pump control coefficients, namely  $K_s$  and  $K_q$ .

For the bent-axis design pump the following equations can be written :

$$K_q = Q_{pmax} / \sin \phi_{max} \quad (B.6)$$

$$K_s = \sin \phi_{max} / U_{max} \quad (B.7)$$

As an example, for a swashplate design pump (Rexroth A4V 90) the necessary parameters are given as :

$$Q_{pmax} = 2.25 \times 10^{-3} \text{ [m}^3/\text{s]} \quad , \quad \phi_{max} = 15^\circ$$

$$U_{max} = 600 - 200 = 400 \text{ [mA]} = 0.4 \text{ [A]}$$

By using Eq.(B.2)

$$K_q = 8.397 \times 10^{-3} \text{ [m/rd s]}$$

and by using Eq.(B.5)

$$K_s = 0.67 \text{ [rd/A]}$$

are obtained. Also,

$$K = K_s \cdot K_q = 5.625 \times 10^{-3} \text{ [m /sA]}$$

## B.2 EVALUATION OF TIME CONSTANT $\tau_p$

The time constant of the pump control servomechanism can be estimated by

$$\tau_p \geq \frac{T}{4} \quad (\text{B.8})$$

where  $T$  is the minimum swivel time and given in the catalog. Eq.(B.8) reflects from the fact that in a first degree control system the output (here swivel angle) reaches 98 percentage of its steady-state value in a time of  $4\tau_p$  for a step change in the output parameter (here solenoid current) .

## B.3 EVALUATION OF PRESSURE COMPENSATION GAIN $K_p$

Pressure compensation gain  $K_p$  can be evaluated by the equations below which are obtained from an analytic study [4]. In the mentioned study , omitting the time constant  $\tau_p$ , the natural frequency and the damping ratio are given by

$$\omega_0^2 = \frac{\beta}{JV} \left\{ D_m^2 \frac{\eta_m}{\eta_v} + K_q K_p B \right\} \quad (\text{B.9})$$

and

$$\zeta = \frac{1}{2\omega_0} \left\{ \frac{B}{J} + \frac{\beta_e}{V} K_q K_p \right\} \quad (\text{B.10})$$

Using these equations and desired damping ratio value a reasonable pressure compensation gain  $K_p$  value can be obtained.



## APPENDIX C

## SOLUTION OF VECTOR-MATRIX EQUATIONS

## C.1 SOLUTION OF TIME-INVARIANT VECTOR MATRIX DIFFERENTIAL EQUATIONS

Consider the following vector-matrix differential equation

$$\dot{x}(t) = A x(t) + B u(t) + C m(t), \quad x(t_0) = x_0 \quad (C.1)$$

where

$A = (n \times n)$  constant matrix

$B = (n \times r)$  constant matrix

$C = (n \times s)$  constant matrix

$x = (n)$  vector

$u = (r)$  vector

$m = (s)$  vector

(Notice that  $n=3$ ,  $r=1$ , and  $s=1$  in our case)

By writing Eq.(C.1) as

$$\dot{x}(t) - A x(t) = B u(t) + C m(t)$$

and multiplying both sides of the equation by  $e^{-At}$ , we obtain

$$e^{-At} [\dot{x}(t) - A x(t)] = e^{-At} [B u(t) + C m(t)]$$

or

$$\frac{d}{dt} [e^{-At} x(t)] = e^{-At} [B u(t) + C m(t)]$$

Integrating the preceding equation between  $t_0$  and  $t$  we get

$$e^{-At} x(t) = e^{-At_0} x_0 + \int_{t_0}^t e^{-A(t-\tau)} [B u(\tau) + C m(\tau)] d\tau$$

or

$$x(t) = e^{A(t-t_0)} x_0 + \int_{t_0}^t e^{A(t-\tau)} [B u(\tau) + C m(\tau)] d\tau \quad (C.2)$$

This is the solution to Eq.(C.1). Notice that if the initial time is given as zero instead of  $t_0$ , then we have

$$x(t) = e^{-At} x_0 + \int_0^t e^{-A(t-\tau)} [B u(\tau) + C m(\tau)] d\tau \quad (C.3)$$

$$x_0 = x(0)$$

## C.2 DISCRETIZATION OF VECTOR-MATRIX DIFFERENTIAL EQUATION

A continuous time system which is described by a vector-matrix differential equation can be approximated by a vector-matrix difference equation. The discrete-time solutions must be valid at equally spaced sampling instants. In discretizing the continuous-time system it is assumed that the input or the forcing functions to change only at these same instants.

Consider the system which is described by Eq.(C.1). The formal solution of Eq.(C.1) is Eq.(C.3).

Taking  $T$  as the sampling period, at  $t = (k+1)T$ , and  $t = kT$  we have

$$x((k+1)T) = e^{A(k+1)T} x_0 + e^{A(k+1)T} \int_0^{(k+1)T} e^{-A\tau} [B u(\tau) + C m(\tau)] d\tau \quad (C.4)$$

and

$$x(kT) = e^{AkT} x_0 + e^{AkT} \int_0^{kT} e^{-A\tau} [B u(\tau) + C m(\tau)] d\tau \quad (C.5)$$

Multiplying Eq.(C.5) by  $e^{AT}$  and subtracting from Eq.(C.4), we obtain

$$\begin{aligned} x((k+1)T) &= e^{AT} x(kT) + e^{A(k+1)T} \int_{kT}^{(k+1)T} e^{-A\tau} [B u(\tau) + C m(\tau)] d\tau \\ &= e^{AT} x(kT) + e^{AT} \int_0^T e^{-At} B dt u(kT) \\ &\quad + e^{AT} \int_0^T e^{-At} C dt m(kT) \end{aligned}$$

or

$$x((k+1)T) = G(T) x(kT) + H(T) u(kT) + Q(T) m(kT) \quad (\text{C.6})$$

where

$$G(T) = e^{AT} \quad (\text{C.7})$$

$$H(T) = e^{AT} \int_0^T e^{-At} B dt = \int_0^T e^{A\tau} B d\tau \quad (\text{C.8})$$

$$Q(T) = e^{AT} \int_0^T e^{-At} C dt = \int_0^T e^{A\tau} C d\tau \quad (\text{C.9})$$

$G = (n \times n)$  matrix

$H = (n \times r)$  matrix

$Q = (n \times s)$  matrix

Consequently, the continuous-time system described by Eq.(C.1) is approximated by Eq.(C.6) which is a vector-matrix equation.

APPENDIX D

COMPONENT DATA FILE EXAMPLES

\*\*\*\*\*  
 FILE NAME=CYLINDER  
 FILE NO.=1

DATA BLOCK NO.= 1  
 CYLINDER TYPE:A,B 250 BARS

	DP	DR	ST	ST	ST	ST	ST
1	40.	20.	45.	65.	100.	155.	280.
2	40.	28.	230.	275.	340.	450.	700.
3	50.	28.	143.	175.	225.	315.	505.
4	50.	36.	342.	395.	465.	625.	955.
5	63.	36.	214.	260.	330.	445.	705.
6	63.	45.	44.	515.	620.	795.	1215.
7	80.	45.	285.	340.	420.	565.	890.
8	80.	56.	555.	640.	775.	990.	1500.
9	100.	56.	365.	435.	545.	725.	1125.
10	100.	70.	719.	815.	960.	1255.	1880.
11	125.	70.	474.	560.	685.	910.	1410.
12	125.	90.	973.	1110.	1330.	1695.	2510.
13	140.	90.	805.	930.	1130.	1450.	2180.
14	140.	100.	1070.	1230.	1470.	1870.	2780.
15	160.	100.	850.	965.	1165.	1510.	2290.
16	160.	110.	1110.	1240.	1490.	1905.	2840.
17	200.	125.	1080.	1240.	1500.	1935.	2920.
18	200.	140.	1465.	1685.	2010.	2560.	3810.

DATA BLOCK NO.= 2  
 CYLINDER TYPE:C,D,F 250 BARS

	DP	DR	ST	ST	ST	ST	ST
1	40.	20.	460.	510.	610.	770.	1130.
2	40.	28.	990.	1130.	1300.	1630.	2330.
3	50.	28.	760.	805.	1005.	1265.	1805.
4	50.	36.	1325.	1495.	1695.	2155.	3095.
5	63.	36.	1010.	1140.	1340.	1670.	2410.
6	63.	45.	1650.	1870.	2170.	2670.	3870.
7	80.	45.	1258.	1395.	1645.	2045.	2995.
8	80.	56.	2027.	2285.	2645.	3275.	4745.
9	100.	56.	1553.	1755.	2065.	2565.	3705.
10	100.	70.	2530.	2895.	3305.	4105.	5605.
11	125.	70.	1955.	2210.	2580.	3210.	4640.
12	125.	90.	3374.	3780.	4410.	5460.	7780.
13	140.	90.	2960.	3325.	3905.	4755.	6905.
14	140.	100.	3715.	4185.	4875.	6015.	8555.
15	160.	100.	3184.	3520.	4080.	5060.	7300.
16	160.	110.	3910.	4280.	5000.	6220.	8860.
17	200.	125.	3970.	4440.	5180.	6440.	9240.
18	200.	140.	5070.	5710.	6640.	8210.	11790.

DATA BLOCK NO.= 3  
 CYLINDER TYPE:F 250 BARS

	DP	DR	ST	ST	ST	ST
1	40.	20.	138.	165.	210.	280.
2	40.	28.	385.	440.	530.	750.
3	50.	28.	272.	315.	380.	500.
4	50.	36.	536.	610.	700.	920.
5	63.	36.	384.	440.	530.	690.
6	63.	45.	630.	780.	920.	1150.
7	80.	45.	487.	565.	670.	865.
8	80.	56.	647.	965.	1140.	1430.
9	100.	56.	610.	730.	890.	1090.
10	100.	70.	1067.	1230.	1430.	1790.
11	125.	70.	772.	890.	1050.	1350.
12	125.	90.	1437.	1620.	1920.	2400.
13	140.	90.	1220.	1400.	1660.	2100.
14	140.	100.	1580.	1830.	2100.	2650.
15	160.	100.	850.	965.	1165.	1510.
16	160.	110.	1210.	1240.	1490.	1905.
17	200.	125.	1650.	1800.	2140.	2720.
18	200.	140.	2160.	2400.	2820.	3500.

DATA BLOCK NO. = 4  
CYLINDER TYPE: A, B 350 BARS

	DP	UR	ST	ST	ST	ST	ST	ST	ST
1	63.	45.	320.	370.	420.	500.	600.	760.	1200.
2	80.	56.	420.	470.	540.	620.	760.	970.	1480.
3	100.	70.	550.	610.	700.	810.	970.	1250.	1870.
4	125.	90.	740.	830.	930.	1080.	1300.	1670.	2480.
5	140.	100.	820.	920.	1040.	1200.	1440.	1840.	2750.
6	160.	110.	850.	950.	1070.	1240.	1490.	1910.	2840.
7	200.	140.	1170.	1310.	1470.	1700.	2020.	2570.	3820.
8	220.	160.	1440.	1580.	1790.	2050.	2430.	3100.	4550.
9	250.	180.	1630.	1810.	2030.	2300.	2760.	3430.	5110.

DATA BLOCK NO. = 5  
CYLINDER TYPE: C, D, F 350 BARS

	DP	UR	ST	ST	ST	ST	ST	ST	ST
1	63.	45.	1360.	1490.	1650.	1860.	2160.	2660.	3860.
2	80.	56.	1680.	1830.	2020.	2270.	2630.	3260.	4730.
3	100.	70.	2100.	2270.	2520.	2830.	3300.	4100.	5870.
4	125.	90.	2790.	3030.	3330.	3760.	4390.	5440.	7760.
5	140.	100.	3080.	3340.	3680.	4160.	4850.	5990.	8590.
6	160.	110.	3220.	3420.	3770.	4230.	4950.	6150.	8810.
7	200.	140.	4160.	4550.	5020.	5660.	6590.	8160.	11740.
8	220.	160.	4980.	5380.	5980.	6730.	7830.	9580.	13880.
9	250.	180.	5600.	6090.	6730.	7580.	8830.	10730.	15530.

DATA BLOCK NO. = 6  
CYLINDER TYPE: E 350 BARS

	DP	UR	ST	ST	ST	ST	ST	ST	ST
1	63.	45.	530.	590.	670.	770.	900.	1140.	1700.
2	80.	56.	670.	740.	830.	940.	1120.	1400.	2090.
3	100.	70.	850.	940.	1050.	1200.	1410.	1780.	2610.
4	125.	90.	1140.	1250.	1400.	1600.	1890.	2370.	3450.
5	140.	100.	1260.	1380.	1550.	1760.	2090.	2620.	3830.
6	160.	110.	1260.	1400.	1550.	1780.	2110.	2670.	3900.
7	200.	140.	1700.	1830.	2100.	2400.	2840.	3570.	5240.
8	220.	160.	2070.	2270.	2540.	2900.	3400.	4300.	6200.
9	250.	180.	2340.	2570.	2860.	3240.	3820.	4750.	7000.

\*\*\*\*  
 FILE NAME=JUTOP  
 FILE NO.=20

DATA BLOCK NO.= 1  
 MOTOR TYPE: GEAR

	DISP	KPI.	PRESS	INERTIA
1	6.50	5000.0	250.0	0.00000
2	8.60	3500.0	250.0	0.00000
3	12.10	4000.0	250.0	0.00000
4	16.20	3000.0	175.0	0.00000
5	20.90	3000.0	250.0	0.00000
6	22.40	2500.0	125.0	0.00000
7	23.40	3200.0	225.0	0.00000
8	25.90	2900.0	200.0	0.00000
9	30.10	3900.0	175.0	0.00000
10	32.60	3000.0	150.0	0.00000
11	35.10	3300.0	150.0	0.00000
12	37.60	3100.0	140.0	0.00000
13	40.00	3000.0	210.0	0.00000
14	50.00	2700.0	210.0	0.00000
15	63.00	2700.0	180.0	0.00000
16	70.00	2700.0	160.0	0.00000
17	80.00	2700.0	140.0	0.00000
18	100.00	2700.0	110.0	0.00000
19	109.00	2300.0	210.0	0.00000
20	129.90	2300.0	210.0	0.00000
21	150.70	2300.0	210.0	0.00000
22	181.90	2300.0	175.0	0.00000
23	202.70	2300.0	175.0	0.00000

DATA BLOCK NO.= 2  
 MOTOR TYPE: RADIAL PISTON

	DISP	KPI.	PRESS	INERTIA
1	103.00	710.0	160.0	0.00000
2	155.00	630.0	160.0	0.00000
3	202.00	500.0	160.0	0.00000
4	298.00	400.0	160.0	0.00000
5	401.00	315.0	160.0	0.00000
6	609.00	250.0	160.0	0.00000
7	702.00	220.0	160.0	0.00000
8	803.00	200.0	160.0	0.00000
9	1053.00	160.0	160.0	0.00000

DATA BLOCK NO.= 3  
 MOTOR TYPE: AXIAL PISTON

	DISP	KPI.	PRESS	INERTIA
1	9.40	7500.0	350.0	.00080
2	11.00	6000.0	350.0	.00120
3	10.00	6000.0	400.0	.00150
4	22.70	5600.0	350.0	.00200
5	28.10	4750.0	350.0	.00250
6	32.00	4750.0	400.0	.00400
7	54.80	3750.0	350.0	.00750
8	63.00	3750.0	400.0	.01100
9	80.00	3350.0	350.0	.01500
10	90.00	3350.0	400.0	.02000
11	107.00	3000.0	350.0	.02400
12	125.00	3000.0	400.0	.03000
13	160.00	2650.0	350.0	.04500
14	180.00	2650.0	400.0	.06000
15	200.00	2500.0	350.0	.08800
16	250.00	2500.0	350.0	.08800
17	350.00	2240.0	350.0	.16000
18	500.00	2000.0	350.0	.27000
19	710.00	1600.0	350.0	.82400
20	1000.00	1600.0	350.0	.82400

\*\*\*\*\*

FILE NAME=PUMP  
FILE NO.=30

DATA BLOCK NO.= 1

PUMP TYPE:	GLAR(FIX.)	DISP	RPM	PRESS
		3.50	5500.00	250.0
		4.50	5000.00	250.0
		6.50	4000.00	250.0
		8.60	3200.00	250.0
		12.10	4000.00	250.0
		16.20	4800.00	250.0
		20.90	3600.00	250.0
		22.40	2200.00	175.0
		23.40	3200.00	250.0
		25.90	2900.00	250.0
		30.10	3900.00	210.0
		32.60	3600.00	210.0
		35.10	3100.00	175.0
		37.60	3100.00	175.0

DATA BLOCK NO.= 2

PUMP TYPE:	VANU(FIX.)	DISP	RPM	PRESS
		10.00	2000.00	175.0
		16.00	2400.00	175.0
		26.00	1400.00	175.0
		36.00	2400.00	175.0
		50.00	1800.00	175.0
		63.00	1800.00	175.0
		80.00	1800.00	175.0
		90.00	1800.00	60.0
		100.00	1800.00	60.0

DATA BLOCK NO.= 3

PUMP TYPE:	RADIAL PISTON(FIX.)	DISP	RPM	PRESS
		1.51	2000.00	700.0
		2.14	2000.00	700.0
		2.59	2000.00	700.0
		3.39	2000.00	700.0
		3.57	2000.00	700.0
		4.32	2000.00	700.0
		4.82	2000.00	700.0
		5.02	2000.00	700.0
		5.83	2000.00	710.0
		7.14	2000.00	700.0
		8.13	2000.00	700.0
		8.63	2000.00	700.0
		9.71	2000.00	700.0
		16.07	2000.00	700.0
		19.43	2000.00	700.0



DATA BLOCK NO. = 4

PUMP TYPE: AXIAL PISTON(FIX.)

	DISP	RPM	PRESS
1	9.47	7500.00	350.0
2	12.00	6000.00	350.0
3	16.00	6000.00	400.0
4	22.70	5000.00	350.0
5	28.00	4750.00	350.0
6	32.00	4750.00	400.0
7	54.00	3750.00	350.0
8	63.00	3750.00	400.0
9	80.00	3350.00	350.0
10	90.00	3350.00	400.0
11	107.00	3000.00	350.0
12	120.00	3000.00	400.0
13	160.00	2050.00	350.0
14	180.00	2050.00	400.0
15	200.00	2500.00	350.0
16	250.00	2500.00	350.0
17	350.00	2240.00	350.0
18	500.00	2000.00	350.0
19	700.00	1600.00	350.0
20	1000.00	1600.00	350.0

DATA BLOCK NO. = 5

PUMP TYPE: AXIAL PISTON(VAR.)

	DISP	RPM	PRESS
1	20.00	4750.00	350.0
2	28.00	3600.00	350.0
3	40.00	3750.00	350.0
4	54.00	3000.00	350.0
5	58.00	3350.00	350.0
6	70.00	3000.00	350.0
7	80.00	2750.00	350.0
8	107.00	2450.00	350.0
9	117.00	2050.00	350.0
10	160.00	2100.00	350.0
11	250.00	1850.00	350.0
12	350.00	1650.00	350.0
13	500.00	1500.00	350.0
14	700.00	1200.00	350.0

DATA BLOCK NO. = 6

PUMP TYPE: AXIAL PISTON(VAR.)

	DISP	RPM	PRESS
1	12.00	6000.00	350.0
2	20.00	4750.00	350.0
3	40.00	3700.00	400.0
4	54.00	3750.00	350.0
5	50.00	3400.00	400.0
6	71.00	3200.00	400.0
7	90.00	2900.00	400.0
8	107.00	1800.00	350.0
9	120.00	2000.00	400.0
10	150.00	2500.00	350.0
11	200.00	2000.00	400.0
12	350.00	2240.00	350.0
13	500.00	2000.00	350.0
14	700.00	1600.00	350.0
15	1000.00	1200.00	350.0

\*\*\*\*\*  
 FILE NAME=DC VALVE  
 FILE NO.=40

DATA BLOCK NO.= 1

	NG	TYPE	PRESS	PA(1)	PA(2)	PA(3)	BT(1)	BT(2)	BT(3)
1	0.	1.	315.0	.27E-01	.24E-02	-.58E-05	.17E-01	.24E-02	-.10E-04
2	0.	2.	315.0	.30E-01	.19E-02	.67E-05	.12E-01	.40E-02	-.21E-04
3	0.	3.	315.0	.26E-01	.14E-02	.29E-05	.26E-01	.14E-02	.25E-05
4	10.	1.	315.0	.19E-02	.16E-02	-.26E-05	-.21E-01	.22E-02	-.86E-05
5	10.	2.	315.0	.88E-02	.61E-03	.30E-05	.17E-02	.12E-02	.28E-05
6	10.	3.	315.0	.87E-02	.65E-03	-.17E-05	-.52E-02	.16E-02	-.19E-05
7	10.	1.	350.0	-.29E-02	.12E-03	-.93E-07	.10E-02	.14E-03	.27E-19
8	10.	2.	350.0	-.29E-01	.52E-03	-.13E-05	.40E-03	.11E-03	.12E-06
9	10.	3.	350.0	-.83E-03	.13E-03	-.17E-06	.13E-01	.90E-04	.60E-06
10	22.	1.	350.0	-.17E-02	.57E-04	-.95E-08	.45E-02	.34E-04	.48E-07
11	22.	2.	350.0	-.17E-02	.57E-04	-.95E-08	.45E-02	.34E-04	.48E-07
12	22.	3.	350.0	-.17E-02	.57E-04	-.95E-08	.19E-02	.40E-04	.33E-07
13	25.	1.	350.0	.77E-03	.18E-04	.33E-08	.18E-03	.29E-04	-.83E-09
14	25.	2.	350.0	.16E-03	.29E-04	-.83E-09	-.38E-02	.64E-04	-.51E-07
15	25.	3.	350.0	-.38E-02	.64E-04	-.51E-07	-.38E-02	.64E-04	-.51E-07

\*\*\*\*\*  
 FILE NAME=PK VALVE  
 FILE NO.=50

DATA BLOCK NO.= 1

	NG	PRESS
1	0.	030.0
2	10.	030.0
3	10.	315.0
4	22.	315.0
5	25.	315.0

\*\*\*\*\*  
 FILE NAME=FILICK  
 FILE NO.=00

DATA BLOCK NO.= 1

	FLOW	K #L10	MICRON
1	30.00	20.0	10.00
2	60.00	5.0	10.00
3	110.00	1.50	10.00
4	160.00	.71	10.00
5	240.00	.51	10.00
6	330.00	.37	10.00
7	660.00	.54	10.00
8	990.00	.37	10.00
9	1300.00	.51	10.00

APPENDIX E

SAMPLE RUNS



\*\*CYLINDER SELECTION\*\*

\*ENTER TYPE(MENU 3)

? :

\*ENTER MAX LOAD,FRW VELOCITY,STROKE

? 69000.0,15,1200

ASKED SPCS:

MAX LOAD= 69000.000 N

MAX VELOCITY= .150 M/S

STROKE= 1200.0 MM

TYPE=A,B

CHOSEN CYLINDER'S SPCS:

PISTON DIAMETER=100.0 MM

ROD DIAMETER= 70.0 MM

MAX STROKE= 1200.0 MM

MAX PRESSURE=250.0 BARS

CALCULATED SPCS:

NECESSARY FLOW= 70.686 LIT/MIN

LOAD PRESSURE= 87.9 BARS

\*ENTER CONTROL-KEY (MENU 11)

? :

\*\*\*PUMP SELECTION\*\*

ENTER PUMP TYPE (MENU 5)

? 1

ENTER PRIME MOVER'S RPM

? 1450

ASKED SPCS:

TYPE=VANE

PRIME MOVER'S RPM= 1450.0

CHOSEN PUMP'S SPCS:

DISPLACEMENT= 63.000 CM3/REV

MAX RPM= 1300.0

MAX PRESSURE= 175.0 BARS

VOL.EFF.= .85

MEC.EFF.= .93

CALCULATED SPCS:

MAX THEO. FLOW= 91.350 LT/MIN

ACTUAL PUMP FLOW= 77.648 LT/MIN

ENTER CONTROL-KEY (MENU 13)

? 1

UPDATED PARAMETERS:

ACTUAL FORWARD VELOCITY= .165 M/S

ACTUAL BACKWARD VELOCITY= .323 M/S

RETURN FLOW= 39.600 LT/MIN

ENTER CONTROL-KEY (MENU 14)

? 1

\*\*PIPE SELECTION\*\*

PIPE LINE SPCS:

SUCTION AND RETURN LINE DIAMETER(NOMINAL)=32 MM  
INNER DIAMETER=35.10 MM  
OUTER DIAMETER=42.20 MM  
DELIVERY LINE DIAMETER(NOMINAL)=20 MM  
INNER DIAMETER=20.90 MM  
OUTER DIAMETER=26.70 MM

\*ENTER CONTROL-KEY (MENU 15)

? 1

\*\*DC VALVE SELECTION\*\*

\*ENTER DC VALVE TYPE(MENU 6 )

? 2

DC VALVE SPCS:

TYPE=C  
INLET PORT NOMINAL RADIUS=NG10  
MAX PRESSURE= 315.0 BARS

\*ENTER CONTROL-KEY (MENU 16)

? 1

\*\*PR VALVE SELECTION\*\*

PR VALVE SPCS:

MAX PRESSURE= 630.0 BARS  
INLET PORT NOMINAL RADIUS=NG10

\*ENTER CONTROL-KEY (MENU 17)

? 1

\*\*FILTER SELECTION\*\*

CHOSEN FILTER'S SPECS:  
MAX FLOW= 130.00 LT/MIN  
FILTERING SIZE=10.0 MIC

\*ENTER CONTROL-KEY (MENU 18)

? 1

\*ENTER CONTROL-KEY FOR P.LOSS COMPUTATION(0.1)

? 1

\*\*P.LOSS COMPUTATION\*\*

\*ENTER OIL VISCOSITY

? 1

OIL VISCOSITY= .400E-04 M2/S

\*ENTER OIL DENSITY

? 1

OIL DENSITY= 858.20 KG/M3

\*\*PRESSURE LOSS COMPUTATION FOR LINE 1\*\*

\*ENTER PIPE LENGTH

? 4.8

\*ENTER NO.OF T JOINTS

? 0

\*ENTER NO.OF L JOINTS

? 2



SUCTION LINE P.LOSS= .08103 BARS  
SUCTION LINE P.LOSS COEFF.= 48385.

\*ENTER THE HEIGHT DIFFERENCE BETWEEN PUMP AND TANK  
(NEGATIVE IF PUMP IS ABOVE)

? 2

PRESSURE AT THE PUMP INLET= 1.08735 BARS

\*\*PRESSURE LOSS COMPUTATION FOR LINE 2\*\*

\*ENTER PIPE LENGTH

? 5.4

\*ENTER NO.OF T JOINTS

? 1

\*ENTER NO.OF L JOINTS

? 2

DELIVERY LINE P. LOSS= .67712 BARS  
DELIVERY LINE P.LOSS COEFF.= 404308.

DC VALVE P.LOSS(P-A)= 5.76551 BARS  
DC VALVE P.LOSS COEFF.(P-A)= 3442591.

\*\*PRESSURE LOSS COMPUTATION FOR LINE 3\*\*

\*ENTER PIPE LENGTH

? 8.1

\*ENTER NO.OF T JOINTS

? 0

\*ENTER NO.OF L JOINTS

? 3

LINE A P.LOSS= 1.06758 BARS  
LINE A P.LOSS COEFF.= 637452.

\*\*PRESSURE LOSS COMPUTATION FOR LINE 4\*\*

\*ENTER PIPE LENGTH

? 12.5

\*ENTER NO.OF T JOINTS

0

? \*ENTER NO.OF L JOINTS

? 3

LINE B P.LOSS(RETURN)= .57463 BARS

LINE B P.LOSS COEFF.= 1548723.

DC VALVE P.LOSS(B-T)= 2.12301 BARS

DC VALVE P.LOSS COEFF.(B-T)= 4873715.

\*\*PRESSURE LOSS COMPUTATION FOR LINE 5\*\*

\*ENTER PIPE LENGTH

? 09.7

\*ENTER NO.OF T JOINTS

? 1

\*ENTER NO.OF L JOINTS

? 4

RETURN LINE P.LOSS= .07157 BARS

RETURN LINE P.LOSS COEFF.= 164300.

RET.LINE FILTER P.LOSS= .03049 BARS

RET.LINE FILTER P.LOSS COEFF.= 70000.

TOTAL PRESSURE LOSSES= 10.40992 BARS

TOTAL P.LOSS COEFF.= 6215739. BAR/(M3/S)2

THE PRESSURE ON THE PUMP= 96.84233 BARS

PR VALVE SET PRESSURE= 100.52662 BARS

POWER REQUIRED BY THE PUMP= 13.47593 KW

\*CAD IS COMPLETED.PROGRAM STOP!



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COMPUTER AIDED DESIGN  
OF  
HYDROSTATIC TRANSMISSION SYSTEMS

JOB NO. = 3

RATED SYSTEM PRESSURE = 240.0 BARS

**\*\*MOTOR SELECTION\*\***

ASKED SPCS:

MAX LOAD TORQUE = 81.212 N-M  
MAX SPEED = 1260.0 RPM  
TYPE = AXIAL PISTON

CHOSEN MOTOR'S SPCS:

DISPLACEMENT = 22.70 CM<sup>3</sup>/REV  
MAX SPEED = 5000.0 RPM  
MAX PRESSURE = 350.0 BARS  
VOL. EFF. = .92  
MEC. EFF. = .94  
MOTOR INERTIA = 0.0200 KG-m<sup>2</sup>

CALCULATED SPCS:

NECESSARY FLOW = 31.089 LT/MIN  
LOAD PRESSURE = 239.137 BARS

**\*\*PUMP SELECTION\*\***

ASKED SPCS:

TYPE = AXIAL PISTON (VAR.)  
PRIME MOVER'S RPM = 1450.0

CHOSEN PUMP'S SPCS:

DISPLACEMENT = 28.100 CM<sup>3</sup>/REV  
MAX RPM = 4750.0  
MAX PRESSURE = 350.0 BARS  
VOL. EFF. = .95  
MEC. EFF. = .95

CALCULATED SPCS:

MAX THEO. FLOW = 40.745 LT/MIN  
ACTUAL PUMP FLOW = 31.089 LT/MIN

UPDATED PARAMETERS:

ACTUAL MOTOR SPEED = 1260.0 RPM  
RETURN FLOW = 28.602 LT/MIN

**\*\*PIPE SELECTION\*\***

PIPE LINE SPCS:

SUCTION AND RETURN LINE DIAMETER (NOMINAL) = 25 MM  
INNER DIA. = 26.60 MM  
OUTER DIA. = 33.40 MM  
DELIVERY LINE DIAMETER (NOMINAL) = 15 MM  
INNER DIA. = 15.80 MM  
OUTER DIA. = 21.30 MM

**\*\*PR VALVE SELECTION\*\***

PR VALVE SPCS:

MAX PRESSURE = 630.0 BARS  
INLET PORT NOMINAL RADIUS = NG 6

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