COMPUTER AIDED DESIGN
OF
FLUID POWER SYSTEMS

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ABSTRACT

This project aims to an integrated computer approach for the design and simulation of hydraulic systems, using components available in market.

An interactive computer software is developed, which draws a graphical layout, selects components and performs steady-state and dynamic calculations on the systems, depending on user-defined inputs, meanwhile allowing redesign or improvement of the system.

The final outputs from printer and plotter can directly be taken to the construction stage, therefore cutting the need for circuit building and testing, decreasing time and money spent during design procedure of hydraulic systems.

The criteria for design, graphics software, development of system dynamic models and mathematical solution methods are explained with an accuracy analysis.

A users' manual is developed to answer any questions that may arise to an inexperienced user.

ÖZET

Bu proje hidrolik sistemlerin Türkiye pazarında bulunan malzeme ile bilgisayar destekli tasarım ve benzetimini amaçlamıştır.

Hidrolik sistemlerin teknik devre çizimini yapan, elemanlarını seçerek, sistem üzerinde sabit ve değişken şartlar altında performans analizi yapabilen, bu arada sistemin en uygun şekillerde boyutlandırılmasını sağlayan bir bilgisayar programı geliştirilmiştir.

Bilgisayardan alınan çıkışlar ve çizimler, doğrudan doğruya sistemin kurulmasında kullanılabilir. Böylece devrenin kurulup test edilmesi ve tekrar tasarlanması işlemleri kolaylaştırılmış, dolayısıyla devrenin tasarımı esnasında harcanan para ve zaman önemli ölçüde azaltılmışdır.

Hidrolik devrelerde tasarım metodları, grafik yazılımı, sistem dinamik modellerinin elde edilmesi ve matematiksel çözüm metodları üzerinde durulmuş, bir de doğruluk analizi eklenmiştir.

İlk kullanımlarda doğabilecek soruları cevaplamak için ise bir kullanıcı kılavuzu yapılarak çalışma tamamlanmıştır.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF SYMBOLS

A	effective area
A_h	head-end area
A_r	rod-end area
A_1	head-end area
A_2	rod-end area
b	damping ratio
B	viscous friction coefficient
B_L	load viscous friction coefficient
B_R	reducer viscous friction coefficient
D	cylinder piston diameter, line diameter
D'	chosen cylinder piston diameter
D_p	pump displacement
D_m	motor displacement
D'_p	chosen pump's displacement
D'_m	chosen motor's displacement
E	young modulus of elasticity for pipe material
E_L	energy loss
F	force
f	variation of force
f_o	natural frequency of oscillations
f_d	frequency of damped oscillations
J	total inertia at motor axis
J_L	load inertia
J_{R1}	reducer inertia at load side
J_{R2}	reducer inertia at motor side

J_M	motor inertia
K_1, K_2	directional valve coefficients
K_s	pump servo control gain
K_q	pump control gain
K_{TOT}	total prosure loss coefficient
K_p	pressure compensation gain
L	delivery line length
M	mass, moment at motor shaft
M_m	total torque at motor shaft
M_{FL}	load friction torque
M_{FR}	reducer friction torque
N	reducer ratio
N_m	motor revolutions in rpm
N_p	pump revolutions in rpm
P_o	outlet pressure
P_i	inlet pressure
P_s	supply pressure
P_L	load-pressure
P_1	head-end pressure
P_2	rod-end pressure
P_{SS}	steady-state pressure
P_{RV}	relief valve set pressure
PR	pressure relief valve set
PP	pressure on the pump
ΔP_V	pressure drop in D.C. valve
ΔP_L	pressure drop in line
ΔP_d	delivery line pressure loss

ΔP_r	return line pressure loss
Q	flow rate through lines
Q_p	pump flow rate
Q_m	motor flow rate
Q_c	cylinder flow rate
Q_R	return flow rate
Q_1	head-end flow
Q_2	rod-end flow
Q_p^*	usefull pump flow rate
S	cylinder stroke
t	pipe thickness
T_m	motor torque
T_p	torque applied at pump shaft
u	pump control input
v	velocity
V	oil volume in delivery line
W	mechanical pump input power
w_p	pump revolutions
w_m	motor revolutions
w_L	load revolutions
w_o	angular frequency of oscillations
y	deviation in cylinder position
Y	cylinder position
β_e	effective bulk modulus of delivery system
β_p	bulk modulus of pipe
β_{e1}	effective bulk modulus at head-end side
β_{e2}	effective bulk modulus at rod-end side

β_{oil}	bulk modulus of oil
β_a	bulk modulus of air
Φ	swash plate angle in radians
ϕ	deviation in swash plate angle
τ_p	pump servo time constant
$\mu_{overall}$	overall efficiency
μ_v	volumetric efficiency
μ_m	mechanical efficiency

I. INTRODUCTION

Fluid power systems are used to transmit rotational or translational power by means of pressurized fluids. They are classified in two main groups depending on the type of fluid used; hydraulics (pressurized liquid), and pneumatics (compressed gas). Hydraulic systems are considered in this project.

The development of the subject, owes a lot to Pascal, who stated the famous law of pressure. It was a century after Pascal, that practical systems were devised by Bramah. He stated that a small force on a long stroke, small diameter piston could produce a very large force on a short stroke, large diameter piston. Bramah appreciated that here was the basis for an efficient means of power transmission using the cycle: mechanical power-hydraulic power-mechanical power.

The unique advantages of hydraulic systems soon re-awakened an interest on the subject. The ability of balancing forces, use of flexible pipes to transmit power through relatively moving parts, and very large force capabilities still remain as the biggest advantages of hydraulics. High force to weight ratio is an important factor, that makes hydraulic equipment the only possible alternative in many situations.

The alternative conversion of fluid power to mechanical power can be achieved by either a rotary motor or a linear actuator, and can be used as a basis for a variety of motions. For example a simple valve and cylinder can produce output motions, which in a wholly mechanical system would require complicated linkages.

The stiffness and braking facility of hydraulic drives is another advantage, and arises due to low compressibility of oil, and as a result, hydraulic drives are little affected by load disturbances, which make them popular in most industries. In a hydraulic system, power is given by:

Power = force x velocity
= pressure x area x velocity
= pressure x flow rate

Control of either pressure or flow rate in a hydraulic system would in turn control power. For this purpose, many methods are developed, simplest of which are the restrictor valves and pressure relief valves. These methods of pressure and flow control are wasteful of power, but are both very popular in modern fluid power systems. When large powers are used and low efficiency can not be tolerated, modern systems make use of variable displacement pumps, to suit the output to demand. Open loop and closed loop control methods are used in many applications, depending on the disturbances acting on the system. These disturbances may take the form of load changes, of variations in flow rate or supply pressure or perhaps of valve wear, and their presence will result in the output speed being different from the desired value. Designing of properly dimensioned hydraulic systems, to meet the requirements, and to simulate existing systems for their performance at various conditions form the basic purposes of this project.

Although they have high efficiency, low operation costs and easy maintenance, the initial cost of hydraulic systems is relatively higher, so they have to be optimally designed and constructed.

The conventional design procedure of an hydraulic system is an iterative process which usually starts by drawing a schematic diagram. It involves as much art as engineering skill. The process blends a knowledge of hydraulic control theory and mathematical modelling with an intuitive "feel" based on past experience. The designer evaluates a circuit by first selecting components and performing hand calculations to estimate its performance. Based on these rough calculations, a prototype system is built, tested to determine actual circuit performance, then re-designed to meet specifications. The procedure continues until satisfactory results are obtained, a time consuming and expensive venture.

Computer-aided design (CAD) techniques can be used to streamline the process. Such a program will use a symbols library containing CETOP symbols for hydraulic components, an engineering data base, containing design information and component characteristics, and a simulation software to allow the designer iterative testing of the simulated circuit, before building the prototype. Therefore an accurate computer simulation of a hydraulic system can substitute for the hardware prototype in all, but the final testing and design stages. Time and money spent during the initial process is reduced dramatically.

The computer program consists of two main sections. First portion is the graphical layout, components selection, steady-state analysis and optimization program. This section designs and analyses a hydraulic system, depending on the loading requirements. The components are either selected from an engineering data library (EDL), containing hydraulic components' data, or a special component may be user defined, such as a valve not occurring in the menu. All components are user defined in case of analysing an existing system. After the final design is obtained, a graphical outlay is drawn, and program performs steady-state analysis on the circuit, and allows optimization if required.

The second portion is the transient analysis and simulation of the designed circuit. Program performs dynamic simulation, due to disturbances acting on the system, and a time history of pressures, flow and speed, at various locations of the circuit is given, together with system dynamic parameters, such as damping ratio, natural frequency, etc. Outputs may be taken in tabular or graphical forms, therefore performance characteristics of hydraulic system under disturbing conditions can be justified and components can be added-or removed, until satisfactory system performance is obtained.

Therefore the integral computer approach for designing hydraulic systems provides a very efficient tool for the designer and allows him to develop better systems, relatively quickly and less expensively.

II. FLUID POWER SYSTEMS

2.1. INTRODUCTION

A fluid power system, consists basically of a fluid power source, a means of distributing the power, a means of controlling the power, and a means of re-converting the fluid power to mechanical power. In this section, basic system designs, control methods and system components are investigated, which are used in designing the computer software.

2.2. FLUID POWER SYSTEM

In a typical fluid power system, a pump, driven by an actuator drives system's fluid. The intake of pump is usually connected to a reservoir. Fluid discharged by pump is used to drive a resisting load by an actuator, which may be in the form of a cylinder or motor. The flow rate, direction and pressure of the pressurized fluid is controlled by valves, downstream the pump.

Valves may be of various types, such as check valves, directional control valves and pressure relief valves.

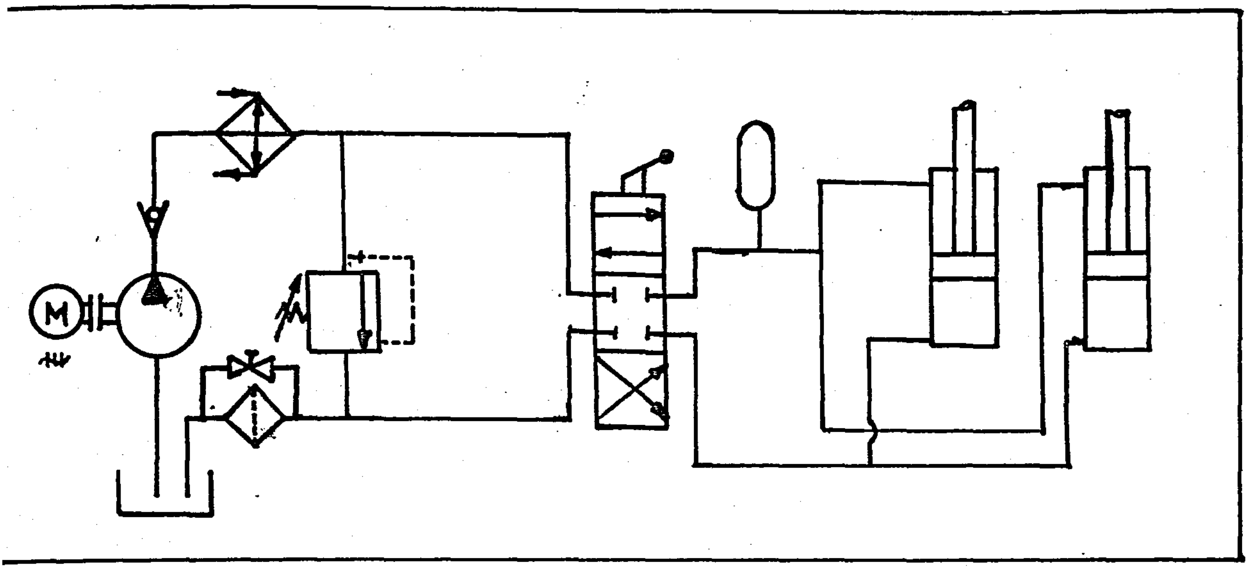

Fig. 2.1. A simple hydraulic system utilizing various components

The working fluid is conditioned by coolers and filters, placed at proper locations in the system. Accumulators are used to smoothout circuit operation and to reduce energy requirements.

Working fluids are selected, depending on the working environment (specially temperature) and equipment used. The most important parameters of a working fluid is its viscosity and bulk modulus, which directly affects circuit efficiency and system stiffness. Figure 2.1. shows a typical hydraulic circuit layout, to drive a translational load.

2.3. CLASSIFICATION OF HYDRAULIC CIRCUITS

Hydraulic systems are designed to meet optimum power transmission at variable or constant speeds and loading conditions. The most important features they offer are:

- remain stalled and undamaged under full load, therefore it is easy to protect against over loads
- they hold stalled at low power loss, therefore can be used for clamping purposes
- hold a preset speed
- provide fast response
- provide dynamic braking
- easy reversing of movement direction

Hydraulic systems may basically be classified as open and closed systems, or depending on the pump and motor type used.

Open circuit systems are usually used to operate at constant speeds and steady-state conditions. In such a system flow is determined by the displacement setting of the pump, which in turn affects the output speed. Flow is almost unaffected by pressure, therefore this is a constant flow system, with variable pressure, as long as displacement is held constant.

The fluid to pump is supplied from a reservoir, and delivered to the control block and actuator inlet. Return flow comes back to the reservoir. Movement direction can be determined by a directional control valve. Figure 2.2. shows a typical open circuit, driving a rotational load.

In the closed circuit configuration motor outlet is connected to the pump inlet. Either the direction of movement, or speed may be varied using a variable displacement, reversible pump. The flow direction of such a pump may be changed by changing the control input, which in turn reverses load movement direction.

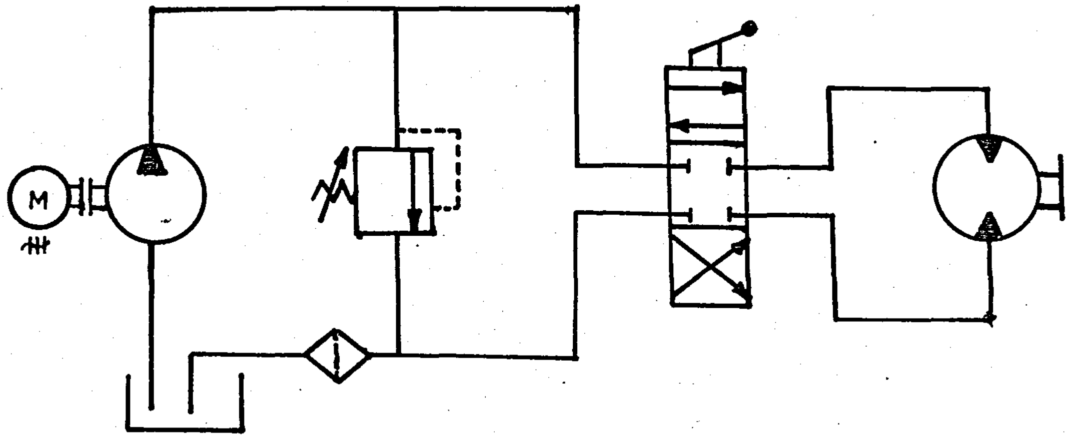

Fig. 2.2. Open circuit system driving a rotational load

One of the most important features of a closed circuit is addition of a charge pump (generally an integral part of the main pump package). Charge pump prevents cavitation by replenishing the closed system with fluid lost through pump and motor leakage and regulates the rated pressure in return line, above a given limit. Control pressure limit is set by the low pressure relief valve, on the discharge side of the charge pump. Back-to-back check valves supply make-up fluid to appropriate low pressure line. At the motor end, there are two pressure relief valves at each direction, which limit fluid pressure in either supply line, and protect the system to shock loads and over running conditions. Figure 2.3. shows a closed system to drive a rotational load.

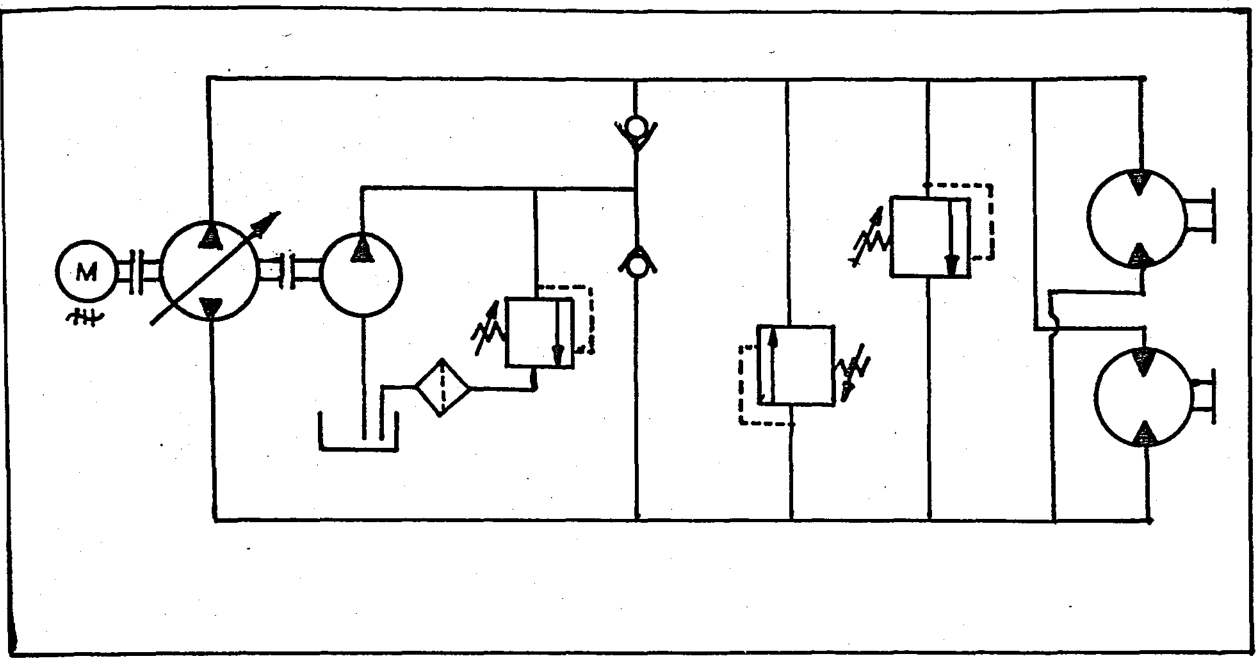

Fig. 2.3. Closed system to drive a rotational load

Another classification is based on pump and motor type used, i.e., fixed or variable displacement pump or motor. In the case of pressure componsation, motor inlet pressure is fed back to pump control servo, and reduces pump flow in case of increasing pressure at load side, due to load disturbance.

In case of hydraulic transmissions (actuator: motor), there may be four combinations of pump and motor. These are fixed displacement pump, fixed displacement motor (PFMF), fixed displacement pump, variable displacement motor (PFMV), variable displacement pump, variable displacement motor (PVMV) and variable displacement pump, constant displacement motor (PVMF) versions. These combinations result in different torque and speed ratios, which are summarized in Table 2.1.

TABLE 2.1. Speed and Torque Ratios

Type of Transmission	N_m/N_p	T_m/T_p
PFMF	Fixed	Fixed
PVMF	Zero to $(D_p/D_m)_{max}$	Infinity to $(D_m/D_p)_{min}$
PFMV	$(D_p/D_m)_{max}$ to Infinity	$(D_m/D_p)_{max}$ to Zero
PVMV	Zero to Infinity	Infinity to Zero

The closed circuit system is more complicated, and involves more components, but in this system, motor shaft can be reversed and/or braked, so when these requirements are vital, closed circuit system becomes an absolute necessity.

2.4. HYDRAULIC SYSTEM ELEMENTS

In this section characteristics of elements which comprise a hydraulic system are investigated and their behaviour equations are developed.

2.4.1. Actuator

Actuators supply necessary power to drive the pump. They are usually in the form of electric motors or internal combustion engines. For all practical purposes, their speed is assumed constant. Actuators are rated in terms of maximum power capacity.

2.4.2. Pump

Pumps used in hydraulic systems are usually of positive displacement type. The types that are used in this project are gear pumps, vane pumps, radial piston pumps, axial piston pumps (constant displacement), axial piston pumps (variable displacement-for open circuits) and axial piston pumps (variable displacement-for closed circuits). Pumps are rated in terms of displacement, maximum pressure and maximum revolutions per minute.

Actual flow delivered by a pump is given by:

$$Q_p = w_p D_p \mu_v \tag{2.1}$$

Mechanical power required by pump is given by:

$$W = \frac{D_p w_p \, (P_o - P_i)}{\mu_m} \tag{2.2}$$

2.4.3. Cylinder

Hydraulic cylinders serve to carry out translational movements and to transfer force. Single and double acting types, with or without cushoning mechanisms are available. They are rated in terms of piston effective area, maximum pressure and stroke.

The force given by a cylinder is:

$$F = P_h A_h - P_r A_r \tag{2.3}$$

For a single acting cylinder, this reduces to:

$$F = P_s A$$

The required flow into the cylinder to maintain a given speed is given by:

$$Q_c = A.v \tag{2.4}$$

Forward and backward speeds are different if effective piston areas at both sides are not equal.

2.4.4. Motor

Hydraulic motors serve to deliver rotational power due to the pressure differential and flow applied to its ports. A hydraulic motor is rated in terms of its displacement, maximum pressure and maximum revolutions per minute.

Actual torque delivered by a motor is given by:

$$T_m = D_m(P_i - P_o)\ \mu_m \tag{2.5}$$

The flow required to drive the motor, at a constant speed is:

$$Q_m = \frac{D_m w_m}{\mu_v} \tag{2.6}$$

2.4.5. Directional Control Valves

Directional control valves are identified by their port numbers and port connections at neutral position. Open - centre, closed-centre and tandem-centre types are available. Directional control valves are sized according to their inlet port nominal radius and maximum pressure.

Pressure loss through a directional control valve depends on the flow rate and is given by a flow rate versus pressure loss curve, supplied by the manufacturer. For design purposes, we can obtain three loss coefficients, by a third degree curve fitting approximation (Appendix D). Then the pressure drop may be computed from:

$$\Delta P_v = C_1 Q + C_2 Q^2 + C_3 Q^3$$

2.4.6. Pressure Relief Valves

Pressure relief valves are employed to limit pressure differential between two locations of a circuit, and prevent system elements from damage due to overloads. Relief valves are sized according to their inlet port nominal radii.

Pressure relief valve setting is determined according to the maximum loading level.

2.4.7. Filter

Hydraulic filters serve to reduce contaminants in the working fluid to a reliable level, and protect individual elements from too much wear. Three types are available, depending on the circuit location that filter is used; suction, pressure and return line filters.

Filters are rated by their maximum flow capacity and filtering size. The pressure drop across a filter, depends on flow rate, and is given by a pressure drop versus flow rate curve, supplied by the manufacturer. For programming purposes, a loss coefficient, K_f may be evaluated, and the pressure drop in the filter can be determined from:

$$\Delta P_f = K_f Q^2$$

(Appendix D)

2.4.8. Lines

Hydraulic pipes, serve to carry pressurized fluid between components in a hydraulic system. Piping consists of two types, hydraulic steel pipes and hydraulic hoses. Hydraulic flexible hoses carry fluid between relatively moving parts. Pipes and hoses are sized according to their nominal bore diameters.

Following requirements should be met, when selecting pipes for hydraulic circuits.

(a) Suction lines to pumps should not carry fluid at velocities in excess of 1.5 m/S, to reduce cavitation at the pump.

(b) Delivery lines should not carry fluid at velocities in excess of 4.5 m/s, to prevent excessive friction and valve closure shocks.

(c) Return lines should be larger than delivery lines to avoid back pressure build-up.

The pressure loss through pipes is computed from:

$$\Delta P_L = f \frac{L}{D} \frac{v^2}{2g} = \frac{K}{K_1} Q^2, \text{ where } \quad K = f \frac{L}{D}, \quad K_1 = \frac{2 A^2}{\rho} \tag{2.7}$$

The friction factor, f is a function of Reynolds number (Re), and pipe roughness. It is assumed to be constant for laminar flow (Re < 2500), and is given by:

$$f = \frac{64}{Re}$$

For turbulent flow (Re < 2500):

$$f = \frac{0.332}{Re^{0.25}}$$

Reynolds number can be determined from:

$$Re = \frac{vD}{\nu} = \frac{4Q}{\pi D^2 \nu}$$

The kinematic viscosity, , usually takes the value of 4.0×10^{-5} m^2/s at normal working conditions.

Sudden contractions, expansions and joints result in pressure loss. The pressure loss through such fittings is given by:

$$P_L = K \frac{\bar{v}^2}{2g}$$, where K is taken as:

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

which are obtained experimentally.

Following figures show examples of pressure loss diagram for a directional control valve and loss coefficient diagram for a pipe bend.

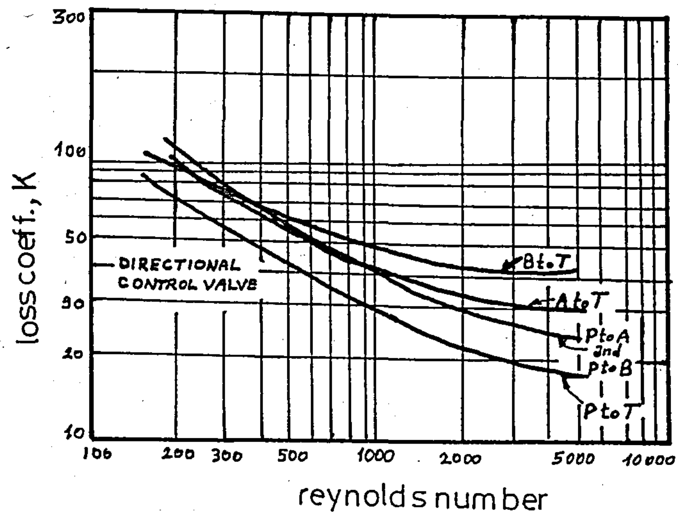

Fig. 2.4. Loss coefficient for a directional control valve

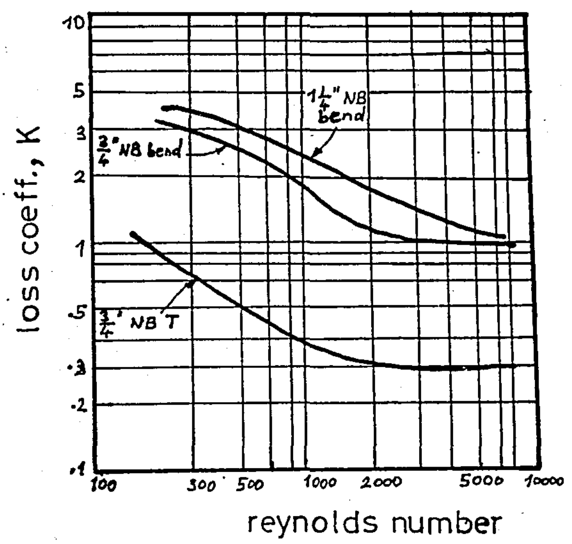

Fig. 2.5. Loss coefficient for bends

III. COMPUTER AIDED DESIGN AND STEADY-STATE ANALYSIS OF FLUID POWER SYSTEMS

3.1. INTRODUCTION

A user friendly, interactive computer program called HYSAN is developed for computer aided design, steady-state analysis and graphical layout of hydraulic systems.

HYSAN is written in FORTRAN 77 language and its graphics software is designed to meet the requirements of Control Data Systems' (CDC) graphics software, namely TIGS (Terminal Independent Graphics System).

The program HYSAN handles the design procedure in three main steps.

1- Program first creates an interactive graphical layout on the screen (this section may be skipped if not required).

2- After the final graphical layout is obtained, program starts interactive execution of the design steps. It begins from the load section and proceeds to pump, valve, filter, transmission lines and reservoir sections. While doing so, after the selection of each component, program performs steady state calculations on the designed portion of hydraulic circuit and allows re-selection or optimization of previously selected components, if required. Components are either selected from Engineering Data Library (EDL), or user defined. In case of user defined components, program asks to define components' data to be used in design procedure.

3- After selection of all components, program performs steady state calculations of the whole circuit and gives actual pressure, flow and velocity at various locations of the circuit.

Design procedure ends by calculation of pressure losses at all pipes, filters, etc. and pressure relief valve sets, reservoir capacity and overall circuit efficiency under given conditions.

After execution, a printout of components and analysis results is sent to the printer, and the graphical layout is sent to the plotter (APPENDIX A, B).

The device-program interaction and basic program design for HYSAN program is given in Figures 3.1 and 3.2 respectively.

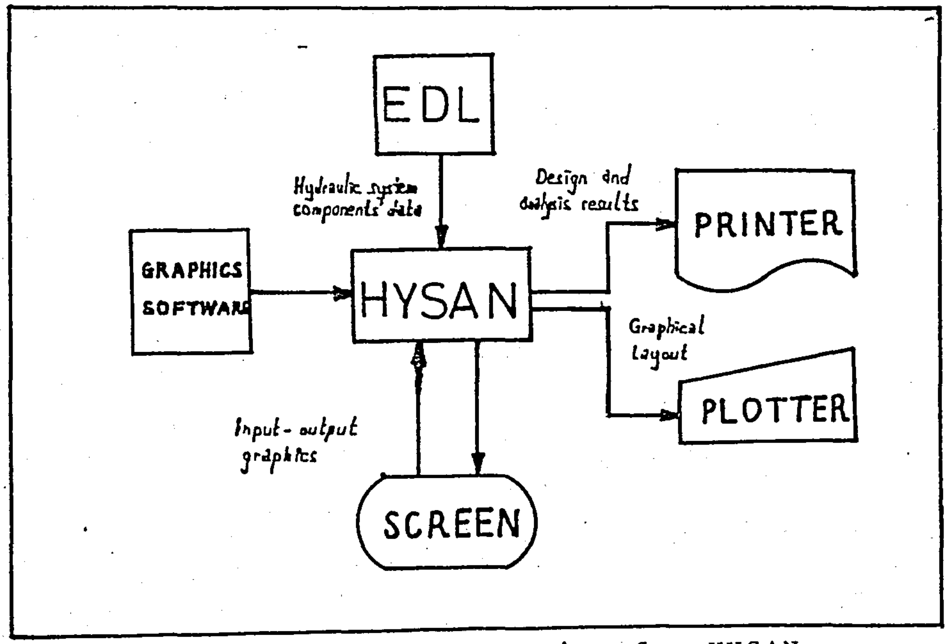

Fig. 3.1. Program-device interaction for HYSAN program

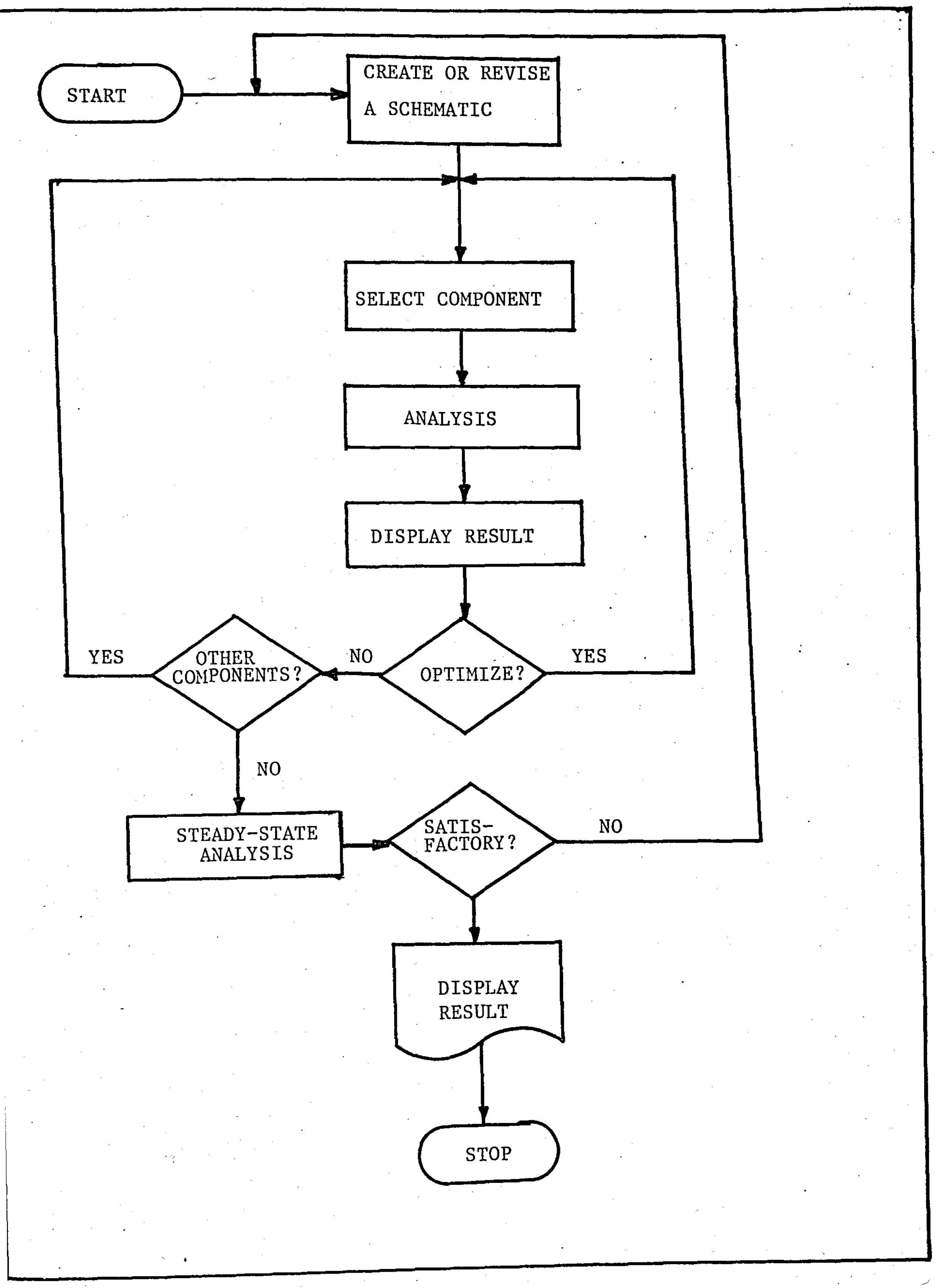

Fig. 3.2. Program main frame for HYSAN

3.2. PROGRAM COMPONENTS

The HYSAN program consists of three sub-systems and a data file loader system. The sub-systems consist of graphics programs, simulation and analysis programs and component data base (EDL).

The main program utilizes nine subroutines for selection of components, six direct access files containing design information of hydraulic components, and 12 graphics subroutines to display hydraulic components on screen. There is a single input file from screen, and three output files, on screen, to printer and to plotter. This organization is shown on tables 3.1. and 3.2.

3.2.1. Graphic Programs

The graphics software designed for this project provides free interaction and symbolic representation of components, using standard CETOP symbols. The interaction consists of selecting different operations by using the tablet command menu, prepared specifically for the design of hydraulic systems.

Components can be positioned on the screen by touching the proper point on the screen, or very accurate positioning may be performed by using the key-board. The component positioning, interactive facility and plotter outputs are organized by the main graphics program, driven by HYSAN.

There are 12 graphics subroutines for components, outputs of which can be seen in Figure 3.3. Plotter output examples may be seen in Appendix B.

TABLE 3.1. Subroutines Utilized by Program HYSAN

Main Program: CAD

Subroutines for Components Selection:

CYLSEL	:	Cylinder selection
MOTSEL	:	Motor selection
PUMSEL	:	Pump selection
PIPE	:	Pipe selection
DVALVE	:	Directional control valve selection
RVALVE	:	Pressure relief valve selection
FILTER	:	Filter selection
LOSS	:	Pressure loss computation
OKAY	:	Program flow control and optimization

Subroutines for Graphics:

RESER	:	Reservoir
FILT	:	Filter
MOTORS	:	Motor
PRVLV	:	Pressure relief valve
COOLER	:	Cooler
ACCUM	:	Accumulator
CYLIND	:	Cylinder
CHKVL	:	Check-valve
PUMP	:	Pump
DCVLV	:	Directional control valve
LINE	:	Fluid and control lines
ACT	:	Actuator

TABLE 3.2. Files Utilized By Program HYSAN

Main Program: CAD

Direct Access Data Files for Components:

CYLIN	: Cylinder file, Unit no: 10
MOTOR	: Motor file, Unit no: 20
PUMPA	: Pump file, Unit no: 30
DVALF	: Directional control valve file, Unit no: 40
RVALF	: Pressure relief valve file, Unit no: 50
FILTE	: Filter file, Unit no: 60

Data Files' Loading Program:

LOADER	: Program to load direct access data files

Input File:

INPUT	: Input file from screen, Unit no: 5

Output Files:

OUTPUT	: Output file on screen, Unit no: 6
OUT 9	: Output file to printer, Unit no: 9
PLOTF	: Output file to plotter

Sub-Files:

EDL	: Data file showing hydraulic components and data loaded in direct access files

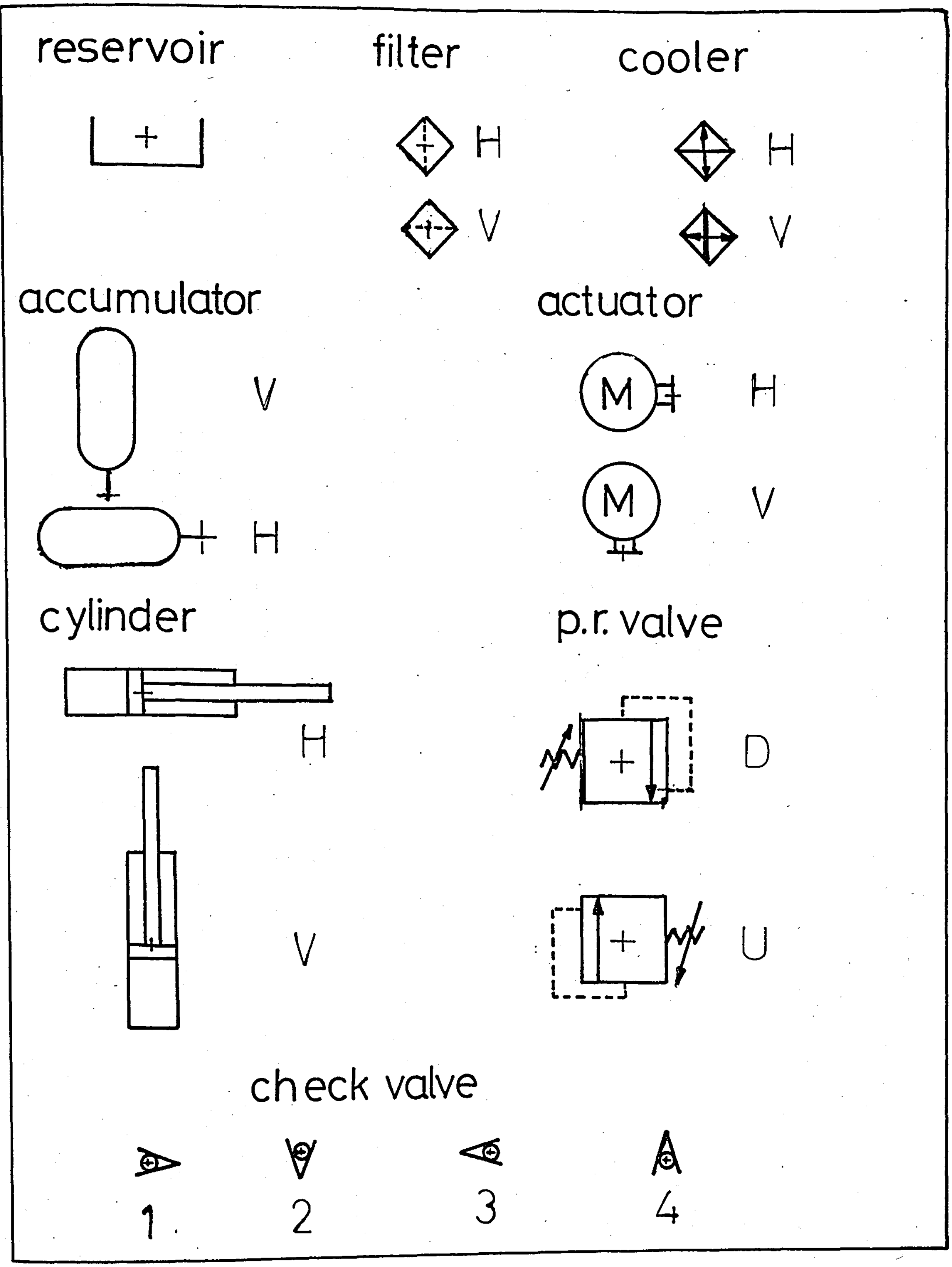

Fig. 3.3. Outputs of graphics subroutines for CETOP symbols

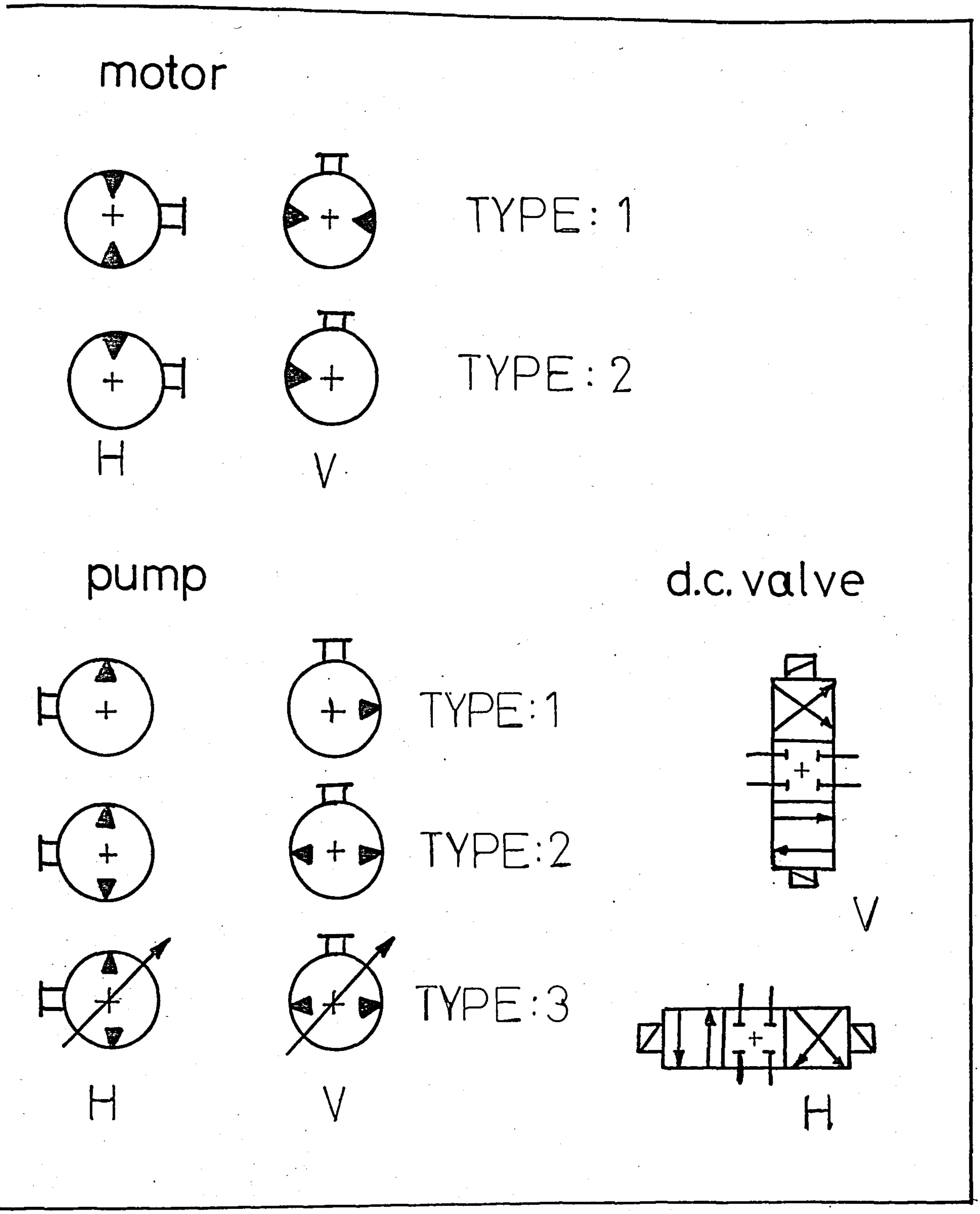

Fig. 3.3. Continued

3.2.2. Simulation Programs

Simulation programs of HYSAN perform sizing of components, steady state analysis, redesign and optimization functions for the design of hydraulic systems.

There are usually several possible solutions to design problems, and some criteria should be chosen to enable the decision among alternative solutions. Low cost, high reliability, high power to weight ratio, safety and easy maintenance are some of these criteria.

For our design programs, decisions are based on minimizing the component size, thus minimizing cost and maximizing power to weight ratio.

Two basic types of data are considered in design of fluid power systems.

1- Quantities that are determined by acceptable precision and whose values remain constant. These values can be obtained from data files, or may be user defined, for example displacement of a pump.

2- Quantities that are difficult to asses, but supported by the wealth of empirical data, accumulated over years of experience, for example seal friction in a cylinder. In this case, most reliable data, obtained from literature are used in the design program.

The design procedure and components selection criteria for HYSAN program is outlined in below steps.

1- Selection of system pressure: Mean system operation pressure is selected by the user. High pressures result in higher efficiency and smaller components, therefore minimize initial cost. Lower pressures yield larger components and higher initial cost, but their operation life and maintenance intervals are much longer.

Recommended main pressure categories are as follows:

Lower pressure rating : 80 bars
Medium pressure rating: 160 bars
Higher pressure rating: 240 bars

2- Actuator Selection

Two actuator types are available, cylinder or motor.

A: Cylinder

Given load requirements: maximum force, maximum velocity, stroke. The required minimum piston diameter will be:

$$D = \frac{\sqrt{4\ F}}{\pi P_s} \tag{3.1}$$

Where F is the maximum load force, and P_s is the supply pressure. Selected cylinders' pressure rating must be greater than system pressure, P_s, and maximum stroke must be larger than required stroke. If selected cylinders piston diameter is D', than required total flow to drive the cylinder at velocity v is:

$$Q = \left(\frac{1}{4\pi} D'^2\right) v\ Nc \tag{3.2}$$

Where Nc is the number of cylinders in parallel. Required load pressure under these conditions is given by:

$$P_L = \frac{4F}{\pi D'^2} \tag{3.3}$$

Which is the minimum pressure to be supplied by the pump, at the load level.

B: Motor

Given load requirements: Maximum rpm and maximum moment at motor level. Motor type is user defined.

Gear motors are cheaper and less efficient, and used for low power applications. Radial piston motors are used in cases where high torque is required at low revolutions. Axial piston motors are most efficient, and capable of delivering high powers at higher torques, but are most expensive. Efficiency figures for motor types are as follows.

Motor Type	Volumetric Efficiency	Mechanical Efficiency
Gear	0.83	0.90
Radial piston	0.90	0.93
Axial piston	0.92	0.94

Required minimum motor displacement is given by:

$$D_m = \frac{1}{\mu_m} \cdot \frac{T}{P_s} \tag{3.4}$$

Selected motors' displacement should be greater than D_m, pressure rating greater than P_s, and maximum revolutions greater than required load revolution, w. Required flow rate to drive the motor at this speed is:

$$\dot{Q} = \frac{D_m' \, w}{\mu_v} \cdot N \tag{3.5}$$

Where D'_m is the displacement of selected motor and N is the number of motors connected in parallel.

Required minimum load pressure to be supplied by pump is:

$$P_L = \frac{1}{\mu_m} \frac{T}{D'_m} \tag{3.6}$$

3- Pump Selection

Knowing required flow rate Q, and supply pressure P_s, a convenient pump type is selected. Prime mover rpm is assumed constant.

Pump type is user defined. Gear pumps are less expensive and have lower efficiency and pressure rating. Vane pumps can operate at low pressures but high flow rates. Radial piston pumps are most effective at high pressures and low flow rates. Axial piston pumps are used where high pressures and high prime mover revolutions are used. Their efficiency is higher and have variable displacement types, including pressure compensation. The axial piston pump type for closed circuits incorporates the feed pump, in a single package.

Pump efficiency figures are listed below.

Pump Type	Volumetric Efficiency	Mechanical Efficiency
Gear	0.85	0.93
Vane	0.85	0.93
Radial piston	0.90	0.94
Axial piston (fixed)	0.93	0.94
Axial piston (var.-open circuits)	0.95	0.95
Axial piston (var.-closed circuits)	0.95	0.95

Under these conditions, chosen pump's minimum displacement must be:

$$D_p = \frac{Q}{w_p} \cdot \frac{1}{\mu_v} \tag{3.7}$$

Pressure rating greater than load pressure P_L, and maximum rpm greater than prime mover's rpm, w_p.

If selected pump has displacement D'_p, actual flow will be:

$$Q_p = D'_p w_p \mu_v \tag{3.8}$$

The updated actuator values, according to the selected pump is:

a) Actuator: Cylinder, updated cylinder speed, V_c:

$$V_c = \frac{4\ Q_p}{\pi_{D'^2}} \cdot \frac{1}{N} \tag{3.9}$$

Return flow rate:

$$Q_R = Q_p \ (1 - \frac{D_r^2}{D_p^2}) \tag{3.10}$$

Where D_r and D_p are rod end and piston end diameters of chosen cylinder.

b) Actuator: Motor, return flow rate is

$$Q_R = Q_p \mu_v \tag{3.11}$$

Actual motor speed is given by:

$$w_m = \frac{Q_p}{D_m} \cdot \frac{\mu_v}{N} \qquad (3.12)$$

4- Pipeline Selection

Pipe lines are selected according to required actual flow rate and SAE standards are taken as a basis. Following table shows this criteria (Table 3.3).

TABLE 3.3. SAE Pipe Diameter Selection Criteria

Flow Rate (m^3/S)	Pipe Size: Suction and Return Line Dia. (N.B) mm	Pipe Size: Delivery Line Dia. (N.B) mm
1.5×10^{-4}	15	8
3.0×10^{-4}	20	10
5.3×10^{-4}	25	15
1.3×10^{-3}	32	20
1.9×10^{-3}	40	25
3.0×10^{-3}	50	32
5.3×10^{-3}	65	40

The inside and outside diameters of chosen pipes can be obtained and outside diameters of chosen pipes can be obtained from Table 3.3, to compute pressure losses.

5- Directional Control Valve Selection

Directional control valves are selected in accordance to delivery and return lines and system pressure. Selection criteria is to take inlet port nominal radius equal to delivery line radius.

TABLE 3.4. Steel Pipe Inner and Outer Diameters

Nominal (mm)	Outside (mm)	Inside (mm)	Max. Working Pressure (Bars)
8	13.7	9.2	464
10	17.1	12.5	384
15	21.3	15.8	369
20	26.7	20.9	306
25	33.4	26.6	288
32	42.2	35.1	240
40	48.3	40.9	217
50	60.3	52.5	184

Directional control valves are classified according to their nominal radii and neutral position port connections. Another important parameter is the loss coefficient of a directional control valve. This parameter is used in pressure loss computations.

Classification of directional control valves,.according to neutral port connections is summarized below:

Valve Type	Neutral Position
E	Closed center
G	Tandem center
H	Open center

The loss coefficients for various port connections of a directional control valve is obtained by a third degree curve fit to the pressure loss versus flow rate diagram, released by the manufacturer. The pressure loss in this case becomes:

$$(\Delta P)_d = K1_d Q + K2_d Q^2 + K3_d Q^3$$

$$(\Delta P)_r = K1_r Q + K2_r Q^2 + K3_r Q^3$$

6- Pressure Relief Valve Selection

Pressure relief valves are selected to fit delivery line nominal bore and to operate at pressures ten percent higher than mean system working pressure.

7- Filter Selection

Filters are used to reduce contaminant level at various locations of the system. In most applications, they are placed on return lines to reduce cost and pressure losses. Filters are sized according to maximum flow rate and filtering size. A filtering rating of 10 microns is convenient for most applications. Pressure loss coefficient for filters is either given by manufacturer or obtained by a second - degree curve fitting to the manufacturer released pressure loss curve (Appendix D).

8- Computation of Pressure Losses

Pressure loss in delivery and return line is computed according to the flow in the line. Loss coefficients for all elements on that line is added to find the total loss coefficient of this line. Loss coefficients are:

For pipes: $K_{pipe} = f \frac{L}{D}$

$f = \frac{64}{Re}$ for laminar flow ($Re < 2500$)

$f = \frac{0.332}{Re^{0.25}}$ for turbulent flow ($Re \geq 2500$)

$$Re = \frac{4Q}{\pi D\nu}$$

For T bends: $K_T = 0.1$
For L bends: $K_L = 1.3$

$K_{TOT} = f\frac{L}{D} + K_T N_T + K_L N_L$, where N_T and N_L are total number of T and L joints in that line respectively. Filter, cooler and directional control valve losses are added if these components exist on that line.

9- Other Computations

Pressure relief valve set pressure is defined to meet a ten percent higher pressure than required pump pressure that is:

PR = (LOAD PRESSURE + DELIVERY LINE LOSSES) * 1.1

The pressure on the pump is load pressure plus losses:

PP = (LOAD PRESSURE + DELIVERY LINE LOSSES)

Power required to drive the pump is given by:

$$W = \frac{PP \cdot Q_p}{\mu_m} \tag{3.13}$$

Required minimum prime mover power is at least ten percent higher than power required to drive the pump, that is:

$$W_{pm} = 1.1\ W \tag{3.14}$$

Recommended minimum reservoir capacity is three times the maximum pump flow rate in liters per minute, w reservoir capacity is in liters.

Overall system efficiency can be calculated as follows.

a) Actuator: Cylinder

$$\mu_{OVERALL} = \frac{(\text{NET PUMP OUTPUT POWER})-(\text{SYSTEM LOSSES})}{(\text{POWER INPUT TO PUMP})} \cdot 0.95$$

The factor of 0.95 is due to cylinder friction.

b) Actuator Motor:

$$\mu_{OVERALL} = \frac{(\text{NET PUMP OUTPUT POWER})-(\text{SYSTEM LOSSES})}{(\text{PUMP INPUT POWER})} \mu_v \mu_m$$

Where μ_v and μ_m are volumetric and mechanical efficiency figures for the motor utilized.

Above analysis completes the components selection and steady-state analysis criteria for HYSAN program. This procedure may be outlined by Figure 3.4.

3.2.3. Engineering Data Library

Engineering data library (EDL), consists of six direct access data files and an indirect access data file, showing the contents of direct access files loaded by LOADER program. The six direct access files and their data recording model is selected to minimize the time of execution for evaluating a proper component size.

The first two records contain file name and file number respectively. Fourth record contains the key that defines the loading structure of that file, so it is the first record that is accessed by the computer, before entering to search the whole file. Computer directly enters to the data block, in accordance to its requirements, given by the main program.

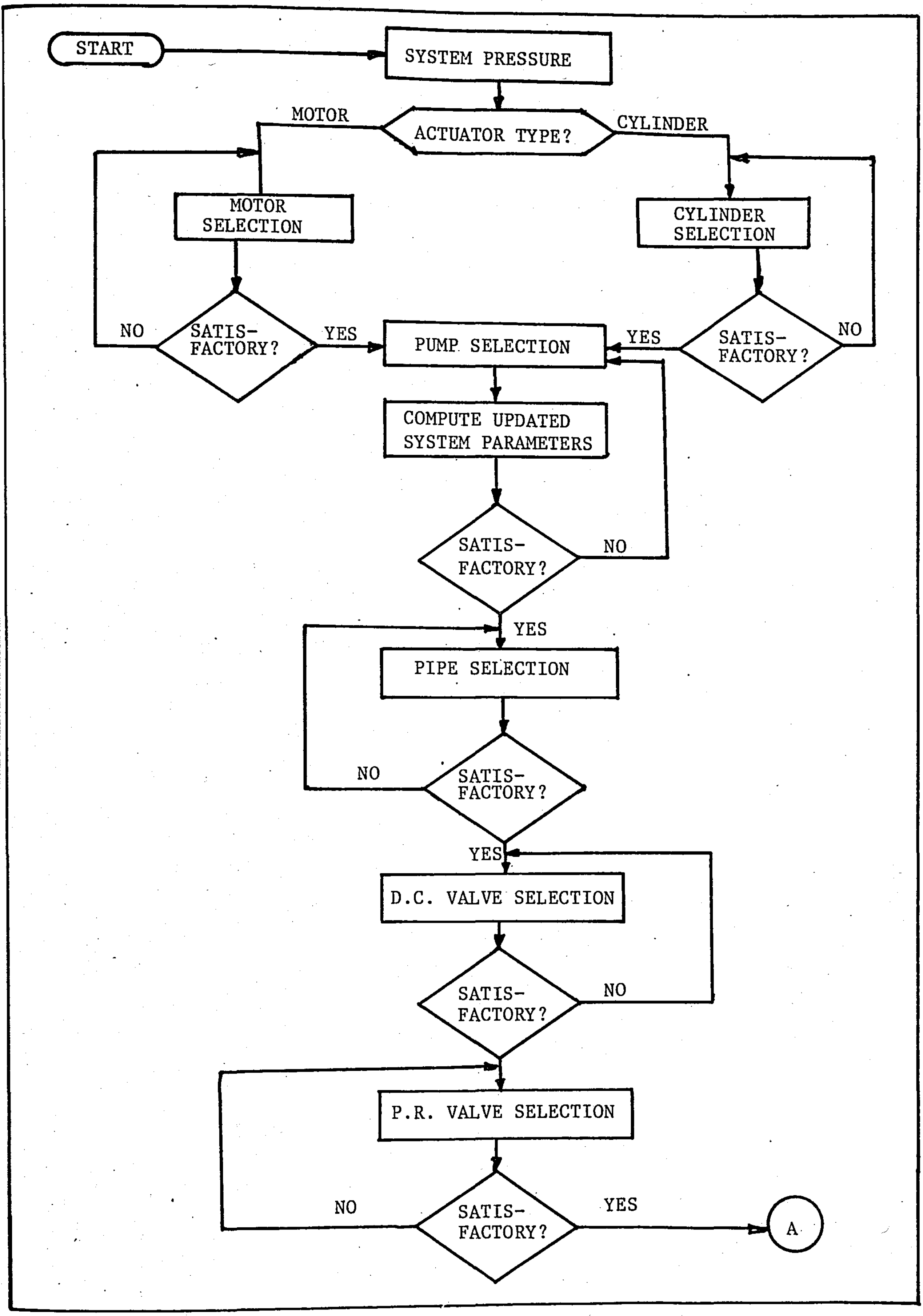

Fig. 3.4. Basic flowchart for HYSAN program simulation software

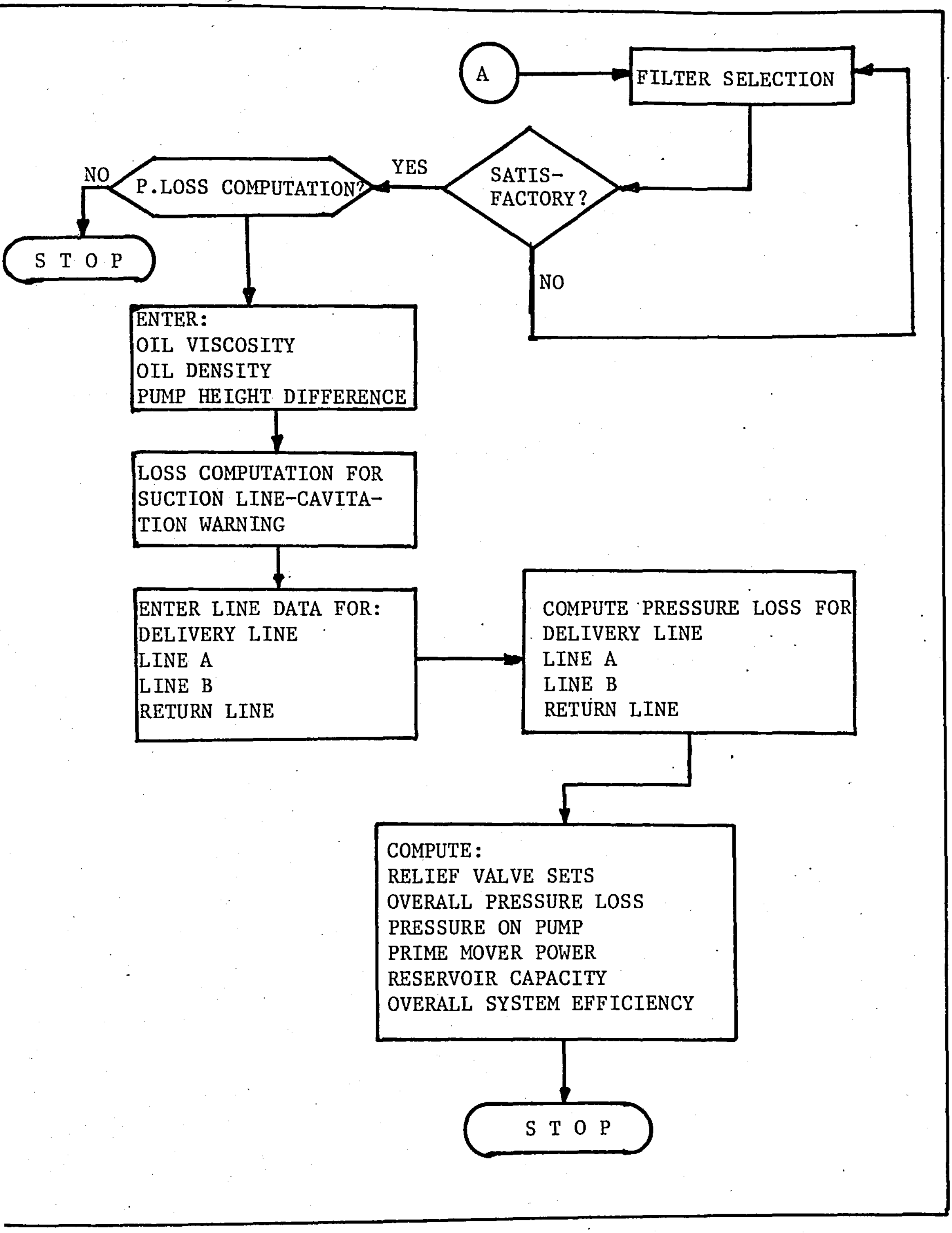

Fig. 3.4. (Continued)

Data files are organised automatically by the file loading program,LOADER,while entering new data or updating the data library.The file organisation of data files is summarized in Table 3.5.

3.3 DATA FILES' LOADING

An interactive computer program called LOADER is developed to load or update component files.This program enables the designer to load the data available from manufacturers' catalogues to engineering data library in an arranged manner.Meanwhile,it copies the direct access files on to an output file to give a printer copy,showing the contents and updated data in these files.

Above programs and files complete the necessary software for HYSAN program to design,sketchand analyse hydraulic systems.

3.4. PROGRAM OPERATION

The operation of HYSAN program depends on the selection of components to obtain output and efficiency figures best fitting the requirements.If results are not satisfactory after a trial,user has the chance to change some of the system parameters,such as mean operation pressure,load parameters or prime mover speed.Therefore HYSAN program reduces the time required for re-selection of components and performance calculations,giving the user chance to make more trials to design a more accurate system.

TABLE 3.5. Table Showing Data Files' Organisation

FILE NAME = CYLINDER

DATA BLOCK NO	DATA LOADED IN THIS BLOCK
1	Cylinder type: A, B 250 bars
2	Cylinder type: C, D, F 250 bars
3	Cylinder type: E 250 bars
4	Cylinder type: A, B 350 bars
5	Cylinder type: C, D, F 350 bars
6	Cylinder type: E 350 bars

FILE NAME = MOTOR

DATA BLOCK NO	DATA LOADED IN THIS BLOCK
1	Gear
2	Radial piston
3	Axial piston

FILE NAME = PUMP

DATA BLOCK NO	DATA LOADED IN THIS BLOCK
1	Gear (Fixed disp.)
2	Vane (Fixed disp.)
3	Radial piston (Fixed disp.)
4	Axial piston (Fixed disp.)
5	Axial piston (Variable disp.)
6	Axial piston (Variable disp.-for closed circuits)

FILE NAME = DC VALVE

DATA BLOCK NO	DATA LOADED IN THIS BLOCK
1	Directional Control Valves, 350 bars max.

FILE NAME = PR VALVE

DATA BLOCK NO	DATA LOADED IN THIS BLOCK
1	Pressure relief valves, 630 bars max.

FILE NAME = FILTER

DATA BLOCK NO	DATA LOADED IN THIS BLOCK
1	Filters, filtering size 10 microns

IV. TRANSIENT ANALYSIS OF FLUID POWER SYSTEMS

4.1. INTRODUCTION

One of the most neglected aspects of conventional circuit design is performance under dynamic conditions. Hydraulic systems designed without considering the dynamic behaviour may fail under disturbing conditions, or may have oscillations which fall outside acceptable limits, during the transient response time.

An interactive computer program called HIDRAN is developed for dynamic simulation of fluid power systems. HIDRAN program predicts the dynamic response of a system under a variety of load demands, and computes flows, pressures, and speed. The output consists of a time history of selected system variables which were disturbed by the controlling input or load change. Outputs may be taken in tabular or graphical forms.

4.2. HIDRAN PROGRAM

HIDRAN program, designed for interactive transient analysis of fluid power systems, performs dynamic simulation of hydraulic systems, due to changes in load and variations of pump control input. The system models used and mathematical methods of solution are presented in following sections.

HIDRAN program consists of a main program and 11 subroutines. There are four main sections in this program. These are system parameters input-output, system modelling, mathematical solution and graphical output sections.

HIDRAN program creates an output file, OUT 8, to be taken from the printer, and a plotter file to obtain analysis results in graphical form from the plotter. Graphical results are also displayed on the screen. The device interaction for HIDRAN program may be outlined in Figure 4.1.

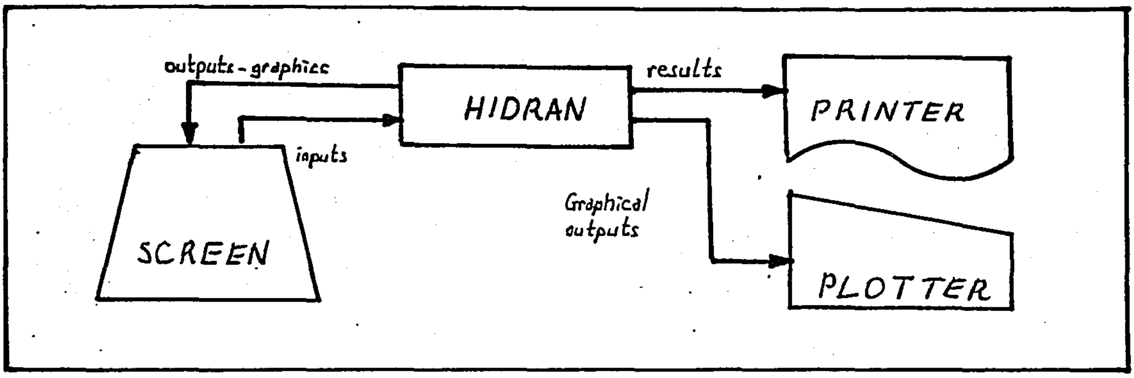

Fig. 4.1. Device interaction of HIDRAN program

The files and subrotuines utilized by HIDRAN program are outlined in Tables 4.1 and 4.2 respectively.

TABLE 4.1. Files Utilized by HIDRAN

Main Program: DYNSIM
Input file: INPUT; Unit no: 5, input from screen
Output files: OUT8, OUTPUT
OUT8: Output file to printer, Unit no: 8
OUTPUT: Output file to screen, Unit no: 6
Plotter files: PLOTF, sent to plotter

There are two graphics subroutines for scaling and graphical display of results, three subroutines for mathematical solution of system equations and six subroutines for system modelling.

TABLE 4.2. Subroutines Utilized by HIDRAN Program

Main Program: DYNSIM

Subroutines for Graphics:

Subroutine AXIS	: Graphical display program
Subroutine G	: Scaling and axis defining program

Subroutines for Mathematical Analysis:

Subroutine DISCR3:	Program to discreatize third degree system of equations
Subroutine MINV	: Program for matrix inversion
Subroutine GMPRD	: Program for matrix multiplication

Subroutines for System Parameters Definition:

Subroutine SYST	: Defines parameters for motor systems
Subroutine SYST1	: Defines parameters for cylinder systems
Subroutine SYST2	: Defines parameters for cylinder systems, analysed due to D.C. valve dynamics

Subroutines for System Modelling:

Subroutine SUBIN	: Creates mathematical model for motor systems
Subroutine SUBIN1:	Creates mathematical model for cylinder systems analysed due to pump-load dynamics
Subroutine SUBIN2:	Creates mathematical model for cylinder systems analysed due to D.C. valve and load dynamics

After displaying results, HIDRAN program computes and displays steady-state variations of selected system variables due to applied inputs. This feature enables the designer to check if program has diverged or not, during execution, since deviations should approach to steady-state values as time increases.

Another output is the computation of system dynamic characteristic values, such as damping ratio, natural frequency, etc. and program gives a warning to the designer, if oscillations may occur behind acceptable limits.

4.3. SYSTEM MODELLING

There are five general system models, analysed by HIDRAN program. These are motor systems with or without pressure compensation, cylinder systems analysed due to pump-load dynamics, with or without pressure compensation and cylinder systems analysed due to directional control valve and load dynamics. The mathematical models for these systems are developed in following sections.

4.3.1. Modeling of Motor Systems Without Pressure Compensation

A general outlay for a motor driver system is shown on Figure 4.2. In such a system, load may be driven, with or without a gearbox. If a gearbox is used, all load data should be shifted to the motor side of the gearbox as explained below, before entering the computer program.

A typical load drive system is shown in Figure 4.3. The parameters shifted to motor axis are given as follows.

$$J = \frac{J_L + J_{R1}}{N^2} + J_M + J_{R2} \tag{4.1}$$

$$B = \frac{B_L}{N^2} + B_R \tag{4.2}$$

$$M = \frac{M_L + M_{FL}}{N} + M_{FR} \qquad (4.3)$$

$$w_m = N w_L \qquad (4.4)$$

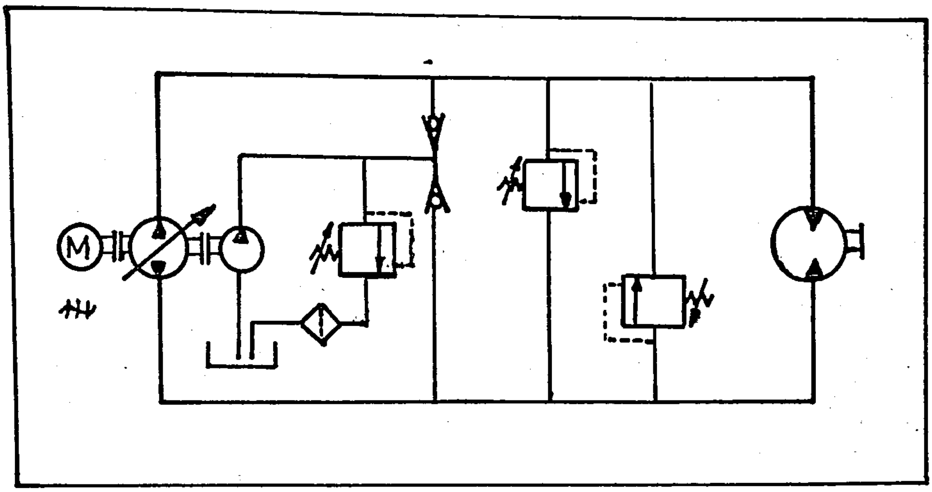

Fig. 4.2. General outlay for a motor driver system

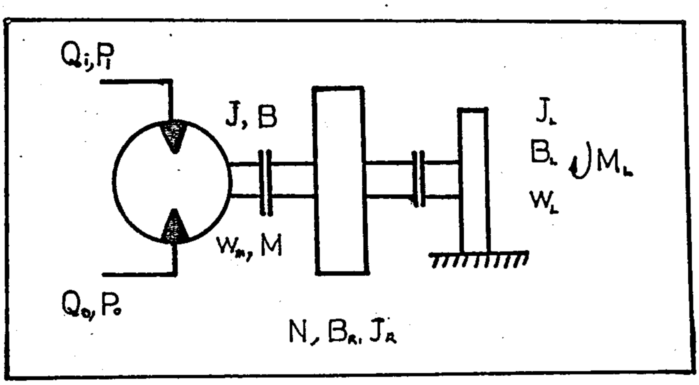

Fig. 4.3. Motor with gearbox

Motor-load equations:

$M_m = D_m(P_i - P_o)\mu_m$ moment developed by motor,

$P_o = \frac{D_m^2 w_m^2}{\mu_v^2} K_R$ motor outlet pressure motor outlet pressure may be assumed atmospheric, since it only depends on the loss at the return lines. Then, combining above equations for motor speed and writing in terms of small variations gives:

$$\dot{w}_m = -\frac{B}{J} w_m + \frac{D_m \mu_m}{J} p_i - \frac{1}{J} M + \frac{D_m \mu_m}{J} P_o \qquad (1)$$

Delivery line:

$\frac{V}{\beta_e} \dot{P}_i = Q_p - Q_m$, compressibility equation for delivery line. Overall bulk modulus of delivery line may be obtained from:

$$\frac{1}{\beta_e} = \frac{1}{\beta_{oil}} + \frac{1}{\beta_p} + \frac{S}{\beta_a} \qquad (4.6)$$

$\beta_p = \frac{tE}{D}$;

$\beta_a = P_i$ for isothermal compression (small pressure variations)

$\beta_a = \gamma P_i$ for adiabatic compression (larger pressure variations)

$\gamma = 1.4$ for air

Motor flow rate: $Q_m = \frac{D_m w_m}{\mu_v}$

Combining above equations for inlet pressure gives:

$$\dot{P}_i = \frac{\beta_e}{V} \left(Q_p - \frac{D_m w_m}{\mu_v}\right) \qquad (2)$$

Pump and control servo:

Q_p = Kq tgΦ for a swash-plate design axial piston pump

Q_p = Kq sinΦ for a bent-axis design axial piston pump

Writing above equations in terms of variations in Φ will give:

q_p = Kq ϕ, where the lower case letters indicate variations in the system variables, from steady-state values.

Pump servo behavior is given by:

$\tau_p \dot{\Phi} + \Phi = K_s u(t)$, where u(t) is the pump control (4.7) input.

Therefore the pump flow in terms of variations will be:

$$\dot{q}_p = \frac{1}{\tau_p} (K_s K_q \; u(t) - q_p) \qquad (3)$$

The dynamic model determining the behaviour of motor systems without pressure compensation may be obtained by combining equations (1), (2) and (3), around a steady-state operation point. The change of outlet pressure depends on the pressure loss change in the return line due to flow changes, and is negligable so canbe ommitted in equation (1). Then the system equations take the form:

$$\dot{w}_m = (-\frac{B}{J}) \; w_m + (\frac{D_m \mu_m}{J}) \; p_i - (\frac{1}{J}) \; M \qquad (1)$$

$$\dot{p}_i = (\frac{\beta_e}{V}) \; q_p - (\frac{D_m \beta_e}{V \; \mu_v}) \; w_m \qquad (2)$$

$$\dot{q}_p = (-\frac{1}{\tau_p})\, q_p + (\frac{K_s K_q}{\tau_p})\, u \qquad (3)$$

Above system may be written in matrix form as:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & 0 \\ a_{21} & 0 & a_{23} \\ 0 & 0 & a_{33} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ b_3 \end{bmatrix} U + \begin{bmatrix} C_1 \\ 0 \\ 0 \end{bmatrix} M$$

Where:

$$x_1 = w_m \qquad a_{11} = \frac{-B}{J} \qquad a_{12} = \frac{D_m \mu_m}{J} \qquad C_1 = \frac{-1}{J}$$

$$x_2 = p_i$$

$$x_3 = q_p \qquad a_{21} = \frac{-D_m \beta_e}{V \mu_v} \qquad a_{23} = \frac{\beta_e}{V}$$

$$a_{33} = \frac{-1}{\tau_p} \qquad b_3 = \frac{K_s K_q}{\tau_p}$$

the control variable is u, and disturbance is M.

4.3.2. Modeling of Motor Systems with Pressure Compensation

In case of pressure compensation, the motor inlet pressure is fed back into the control variable u, in order to attain a reasonable damping ratio. This time the eq'n for pump servo becomes:

$$\tau_p \dot{\Phi} + \Phi = (K_s u - K_p \dot{P}_i)\, K_q \qquad (4.8)$$

Where K_p is the pressure compensation gain of pump.

Therefore the pump flow rate becomes.

$$\dot{q}_p = \frac{1}{\tau_p}\,(-q_p - K_p K_q \tau_p \dot{p}_i - K_p K_q p_i + K_s K_q u) \qquad (4.9)$$

in terms of variations.

Replacing $\dot{p}_i = \frac{\beta_e}{V}\,(q_p - \frac{D_m}{\mu_v} w_m)$ into above eqh and combining with eq'ns (1) and (2) obtained above gives system equations:

$$\dot{q}_p = (\frac{K_p K_q \beta_e D_m}{V\,\mu_v}) w_m - (\frac{K_p K_q}{\tau_p}) p_i - (\frac{K_p K_q \tau_p \beta_e + V}{\tau_p V}) q_p + (\frac{K_s K_q}{\tau_p}) u \qquad (4.10)$$

$$\dot{p}_i = (\frac{\beta_e}{V})\; q_p - (\frac{D_m \beta_e}{V\,\mu_v})\; w_m \qquad (2)$$

$$\dot{w}_m = -(\frac{B}{J})\; w_m + (\frac{D_m \mu_m}{J})\; p_i - (\frac{1}{J})\; M \qquad (1)$$

The state space equations for the motor system with pressure compensation becomes:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & 0 \\ a_{21} & 0 & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix} \cdot \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ b_3 \end{bmatrix} U + \begin{bmatrix} C_1 \\ 0 \\ 0 \end{bmatrix} M$$

Where:

$x_1 = w_m \qquad a_{11} = \frac{-B}{J} \qquad a_{21} = \frac{-\beta_e D_m}{V\,\mu_v} \qquad a_{31} = \frac{K_p K_q \beta_e D_m}{V\,\mu_v}$

$x_2 = p_i$

$x_3 = q_p \qquad a_{12} = \frac{D_m \mu_m}{J} \qquad a_{32} = \frac{-K_p K_q}{\tau_p} \qquad b_3 = \frac{K_s K_y}{\tau_p}$

$a_{23} = \frac{\beta_e}{V} \qquad a_{33} = \frac{-K_p K_q \tau_p \beta_e + V}{\tau_p V} \qquad C_1 = \frac{-1}{J}$

with the control variable u and disturbance M.

4.3.3. Characteristic Values for Motor Systems

The system damping ratio is given by:

$$b = \frac{B}{2J} \cdot \frac{1}{D_m \frac{\sqrt{\beta_e}}{\sqrt{VJ}} \left(\frac{\mu_m}{\mu_v}\right)} \simeq \frac{B}{2\ D_m} \sqrt{\frac{V}{J\ \beta_e}} \tag{4.11}$$

for a system without pressure compensation.

$$b = \frac{1}{2\ w_o} \left[\frac{B}{J} + \frac{\beta_e}{V} K_q K_p \right] \tag{4.12}$$

where

$$w_o^2 = \frac{\beta_e}{V} \left[D_m^2 \left(\frac{\mu_m}{\mu_v}\right) + K_q K_p B \right] \tag{4.13}$$

for a system with pressure compensation.

The system is underdamped for $b<1$ and overdamped for $b>1$. For an underdamped system ($b<1$), the natural frequency is:

$$f_o = \frac{D_m}{2\pi} \sqrt{\frac{\beta_e}{JV} \left(\frac{\mu_m}{\mu_v}\right)} \tag{4.14}$$

The frequency of damped oscillations for such a system is:

$$f_d = f_o \sqrt{1-b^2} \tag{4.15}$$

In case of a system with pressure compensation, natural frequency is given by:

$$f_o = \frac{w_o}{2\pi} \tag{4.16}$$

where w_o is defined above.

4.3.4. Steady-State Results for Motor Systems

Steady-state results due to changes in disturbing factors can be obtained by setting the time derivatives to zero in the system equations, and solving simultaneously for state variables.

For a system without pressure compensation, we obtain:

$$\bar{q}_p = K_s K_q u \tag{4.17}$$

$$\bar{w}_m = \frac{K_s K_q u}{D_m} \mu_v \tag{4.18}$$

$$\bar{P}_i = \frac{M}{D_m \mu_m} + \frac{BK_s K_q u}{D_m^2} \frac{\mu_v}{\mu_m} \tag{4.19}$$

For a system with pressure compensation, analysis results in:

$$\bar{P}_i = \frac{M\left[\frac{D_m}{B\mu_v \tau_p} + \frac{K_p K_q \beta_e D_m}{BV\mu_v} - \frac{K_p K_q \beta_e D_m^2 \mu_m}{BV\mu_v}\right] + \frac{K_s K_q}{\tau_p} u}{\left(\frac{K_p K_q}{\tau_p} + \frac{K_p K_q \beta_e D_m^2 \mu_m}{BV\mu_v} + \frac{D_m^2 \mu_m}{B\tau_p \mu_v} - \frac{K_p K_q \beta_e D_m^2 \mu_m}{BV\mu_v}\right)} \tag{4.20}$$

$$\bar{w}_m = \frac{D_m \mu_m}{B} \bar{p}_i - \frac{M}{B} \tag{4.21}$$

$$\bar{q}_p = \frac{D_m^2 \mu_m}{B \ \mu_v} \bar{p}_i - \frac{D_m}{B \ \mu_v} M \tag{4.22}$$

4.3.5. Block Diagram for Motor Systems

Taking the Laplace transformations of system equations for a motor driver hydraulic system without pressure compensation, we obtain:

$$W_m(s) = \frac{1}{Js+B} \left[D_m \mu_m \ P_i(s) - M(s) \right] \tag{4.23}$$

$$P_i(s) = \frac{\beta_e}{Vs} \left[Q_p(s) - \frac{D_m}{\mu_m} w(s) \right] \tag{4.24}$$

$$Q_p(s) = \frac{K_q K_s}{\tau_p s + 1} \tag{4.25}$$

Letting u(s), the control variable and M(s) be the disturbance, acting on the system, we can obtain following block diagram for a system without pressure compensation (Figure 4.4).

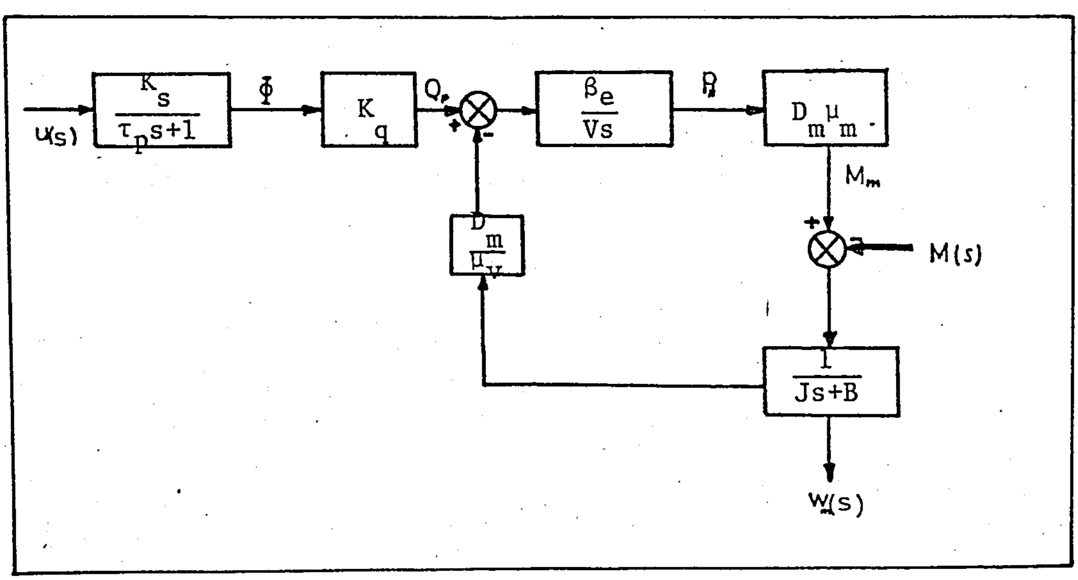

Fig. 4.4. Block diagram for motor systems without pressure compensation

To obtain the block diagram for a system with pressure compensation, we re-write pump equation for pressure compensated pumps, in Laplace transformed form. Other equations remain same. That is:

$$w_m(s) = \frac{1}{Js+B}\left[D_m \mu_m P_i(s) - M(s)\right] \tag{4.23}$$

$$P_i(s) = \frac{\beta_e}{VS}\left[Q_p(s) - \frac{D_m}{\mu_m} w_m(s)\right] \tag{4.24}$$

$$Q_p(s) = K_q\left[\left(\frac{K_s}{\tau_p s+1}\right) u(s) - K_p P_i(s)\right] \tag{4.26}$$

Therefore the block diagram for a system with pressure compensation becomes:

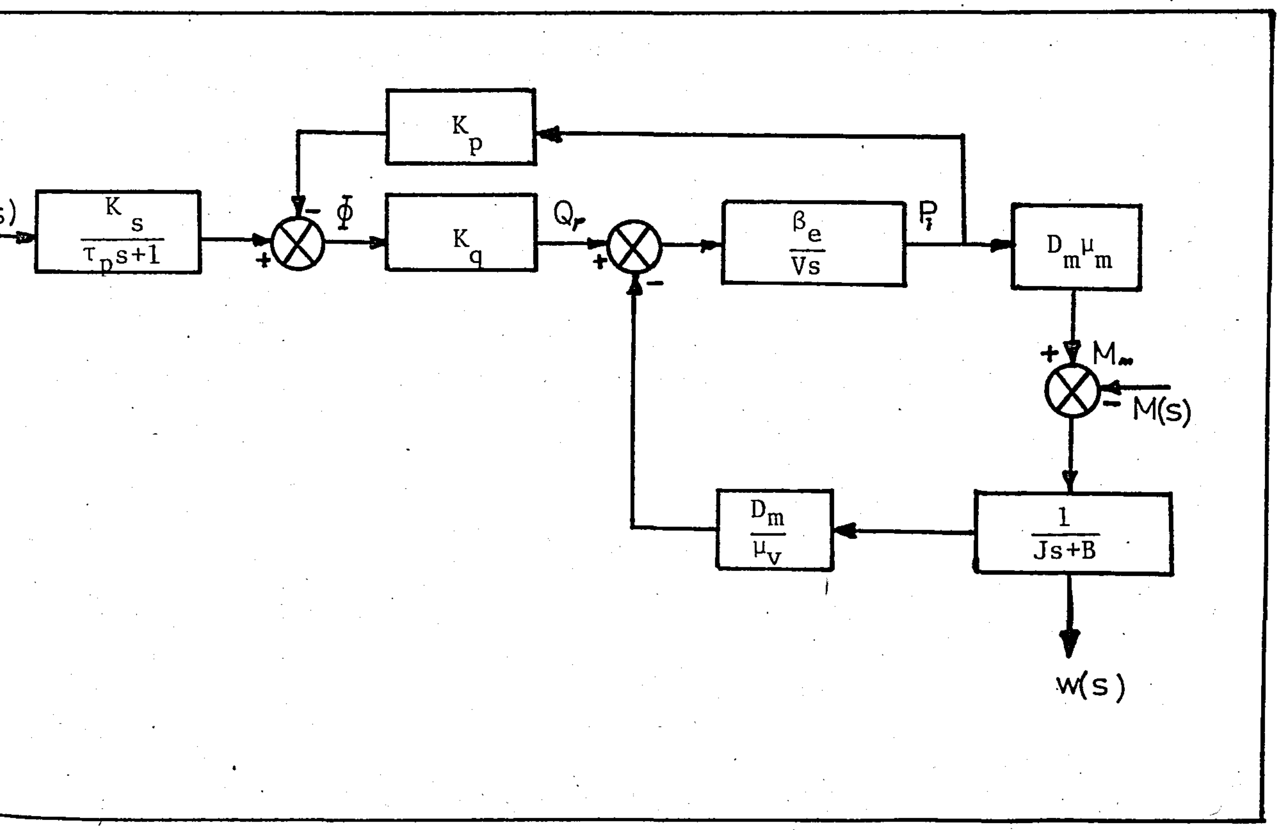

Fig. 4.5. Block diagram for motor systems with pressure compensation

4.3.6. Modeling of Cylinder Systems Without Pressure Compensation

A cylinder system, driving a translational load is shown in Figure 4.6.

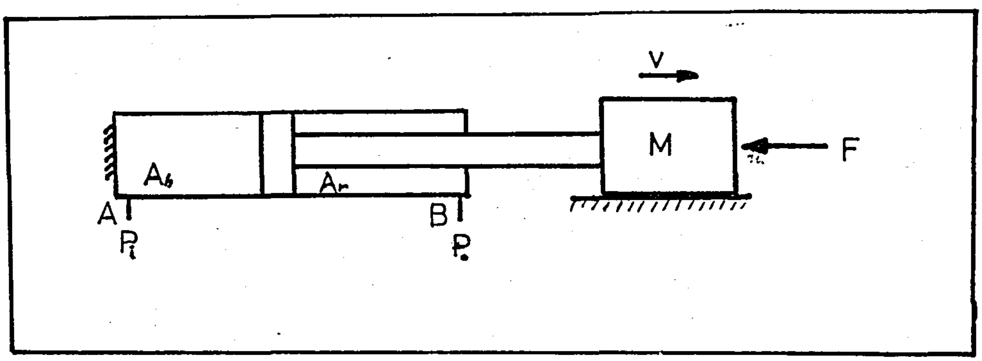

Fig. 4.6. Cylinder driving a translational load

The dynamic model for such a system may be obtained by writing cylinder load equations, and combining with delivery line and pump equations obtained for motor systems.

For the cylinder-load combination:

$$M\dot{v} + Bv + F = F_p \quad ; \quad F_p = A_h P_i - P_o A_r \tag{4.27}$$

combining above equations and neglecting return line pressure loss, the eq'n for cylinder velocity becomes:

$$\dot{v} = \frac{-B}{M} v + \frac{A}{M} P_i - \frac{F}{M} \tag{1}$$

Variation of inlet pressure becomes:

$$\dot{p}_i = (Q_p - Q_c) \frac{\beta_e}{V}, \text{ where } Q_c = A.v \tag{4.28}$$

combining above equations for pressure variation about a steady-state point will give:

$$\dot{p}_i = \frac{\beta_e}{V} (q_p - Av) \tag{2}$$

Pump equation is the same as obtained for motor systems:

$$\dot{q}_p = \frac{1}{\tau_p} (K_s K_q u - q_p) \tag{3}$$

Combining equations (1), (2) and (3) will give the system model.

$$\dot{V} = -(\frac{B}{M}) v + \frac{A}{M} p_i - \frac{F}{M} \tag{1'}$$

$$\dot{p}_i = -(\frac{\beta_e A}{V}) v + \frac{\beta_e}{V} q_p \tag{2'}$$

$$\dot{q}_p = (\frac{-q_p}{\tau_p}) + (\frac{K_s K_q}{\tau_p}) u \tag{3'}$$

Above system may be written in matrix form, convenient for computer solution.

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & 0 \\ a_{21} & 0 & a_{23} \\ 0 & 0 & a_{33} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ b_3 \end{bmatrix} u + \begin{bmatrix} C_1 \\ 0 \\ 0 \end{bmatrix} F$$

Where:

$x_1 = u \qquad a_{11} = \frac{-B}{M} \qquad a_{12} = \frac{A}{M} \qquad a_{23} = \frac{\beta_e}{V} \qquad C_1 = \frac{-1}{M}$

$x_2 = p_i$

$x_3 = q_p \qquad a_{21} = \frac{-\beta_e A}{V} \qquad a_{33} = \frac{-1}{\tau_p} \qquad b_3 = \frac{K_s K_q}{\tau_p}$

4.3.7. Modeling of Cylinder Systems with Pressure Compensation

The dynamic behaviour of a pressure compensated pump is:

$$\dot{q}_p = \frac{1}{\tau_p}\left[-q_p - K_q K_p \tau_p \dot{p}_i - K_p K_q p_i + K_s K_q u\right] \tag{4.9}$$

Substitution of inlet pressure differential,

$$\dot{p}_i = \frac{\beta_e A}{V} v + \frac{\beta_e}{V} q_p$$

will give pump flow for a pressure compensated system as:

$$\dot{q}_p = \frac{1}{\tau_p}\left[-q_p - K_p K_q \tau_p \frac{\beta_e A}{V} v - \frac{K_p K_q \tau_p \beta_e}{V} q_p - K_p K_q p_i + K_s K_q u\right] \tag{4.29}$$

Therefore the system model becomes:

$$\dot{v} = -\left(\frac{B}{M}\right) v + \frac{A}{M} p_i - \frac{F}{M} \tag{1}$$

$$\dot{p}_i = -\left(\frac{\beta_e A}{V}\right) v + \frac{\beta_e}{V} q_p \tag{2}$$

$$\dot{q}_p = \frac{-K_p K_q \beta_e A}{V} v - \frac{K_p K_q}{\tau_p} p_i - \left(\frac{K_p K_q \beta_e}{V} + \frac{1}{\tau_p}\right) q_p + K_s K_q u \tag{4.29}$$

The matrix form expression of above system will give:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & 0 \\ a_{21} & 0 & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ b_3 \end{bmatrix} u + \begin{bmatrix} C_1 \\ 0 \\ 0 \end{bmatrix} F$$

Where:

$x_1 = v \qquad a_{11} = \frac{-B}{M} \qquad a_{12} = \frac{A}{M} \qquad a_{33} = \frac{-K_p K_q \beta_e}{V} - \frac{1}{\tau_p}$

$x_2 = p_i$

$x_3 = q_p \qquad a_{21} = \frac{-\beta_e A}{V} \qquad a_{23} = \frac{\beta_e}{V} \qquad b_3 = K_s K_q$

$a_{31} = \frac{K_p K_q \beta_e A}{V} \qquad a_{32} = \frac{-K_p K_q}{\tau_p} \qquad C_1 = \frac{-1}{M}$

with the control variable u and disturbance F.

4.3.8. Steady-State Results for Cylinder Systems

Steady-state results may be obtained by equating the time derivatives in system equations to zero, and solving simultaneously for system variables.

Steady-state results for a system without pressure compensation gives:

$$\bar{q}_p = K_s K_q u \tag{4.30}$$

$$\bar{v} = \frac{K_s K_q u}{A} \tag{4.31}$$

$$\bar{p}_i = \frac{F}{A} + \frac{B}{A^2} K_s K_q u \tag{4.32}$$

Steady state analysis results for a system with pressure compensation is:

$$\bar{p}_i = \frac{F\left[\frac{K_p K_q \beta_e A}{BV} + \frac{K_p K_q \beta_e A^2}{BV} + \frac{A}{B\tau_p}\right] + K_s K_q u}{\left(\frac{2K_p K_q \beta_e A^2}{BV} + \frac{K_p K_q}{\tau_p} + \frac{A^2}{B\tau_p}\right)} \tag{4.33}$$

$$\bar{q}_p = \frac{A^2}{B} \bar{p}_i - \frac{A}{B} F \tag{4.34}$$

$$\bar{V} = \frac{1}{A} \bar{q}_p \tag{4.35}$$

4.3.9. Block Diagram for Cylinder Systems

Taking the Laplace transformation of system equations for a cylinder driver hydraulic system, without pressure compensation we obtain:

$$v(s) = \frac{-B}{Ms} v(s) + \frac{A}{Ms} p_i(s) - \frac{F(s)}{Ms} \tag{4.36}$$

$$p_i(s) = \frac{-\beta_e A}{Vs} v(s) + \frac{\beta_e}{Vs} Q_p(s) \tag{4.37}$$

$$Q_p(s) = \frac{-1}{\tau_p s} Q_p(s) + \frac{K_s K_q}{\tau_p s} u(s) \tag{4.38}$$

And the block diagram becomes as shown in Figure 4.7.

The block diagram for cylinder systems with pressure compensation may be obtained the same way as follows.

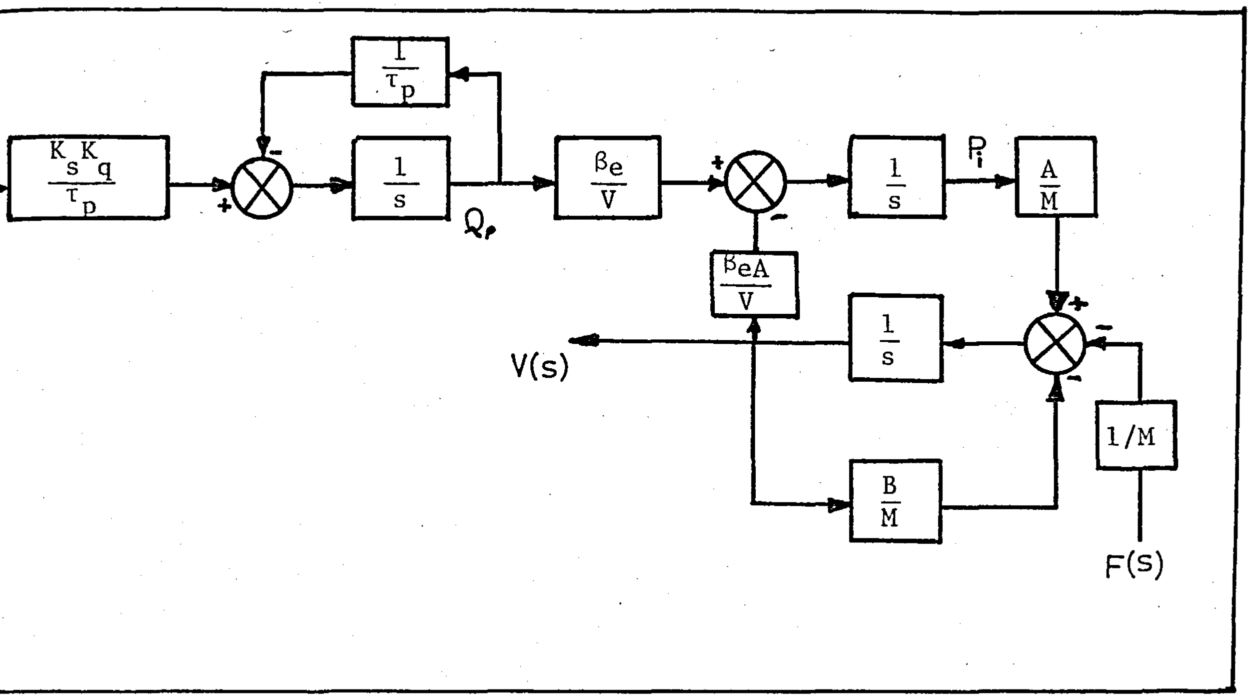

Fig. 4.7. Block diagram for a system without pressure compensation

$$v(s) = \frac{-B}{Ms} V + \frac{A}{Ms} p_i - \frac{F(s)}{Ms} \tag{4.39}$$

$$p_i(s) = \frac{-\beta_e A}{Vs} v + \frac{\beta_e}{Vs} q_p \tag{4.40}$$

$$q_p(s) = \frac{-K_p K_q \beta_e A}{Vs} v - \frac{K_p K_q}{\tau_p s} p_i -$$

$$- \left(\frac{K_p K_q \beta_e}{Vs} + \frac{1}{\tau_p s}\right) q_p + \frac{K_s K_q}{s} u(s)$$

The block diagram may be drawn as in Figure 4.8.

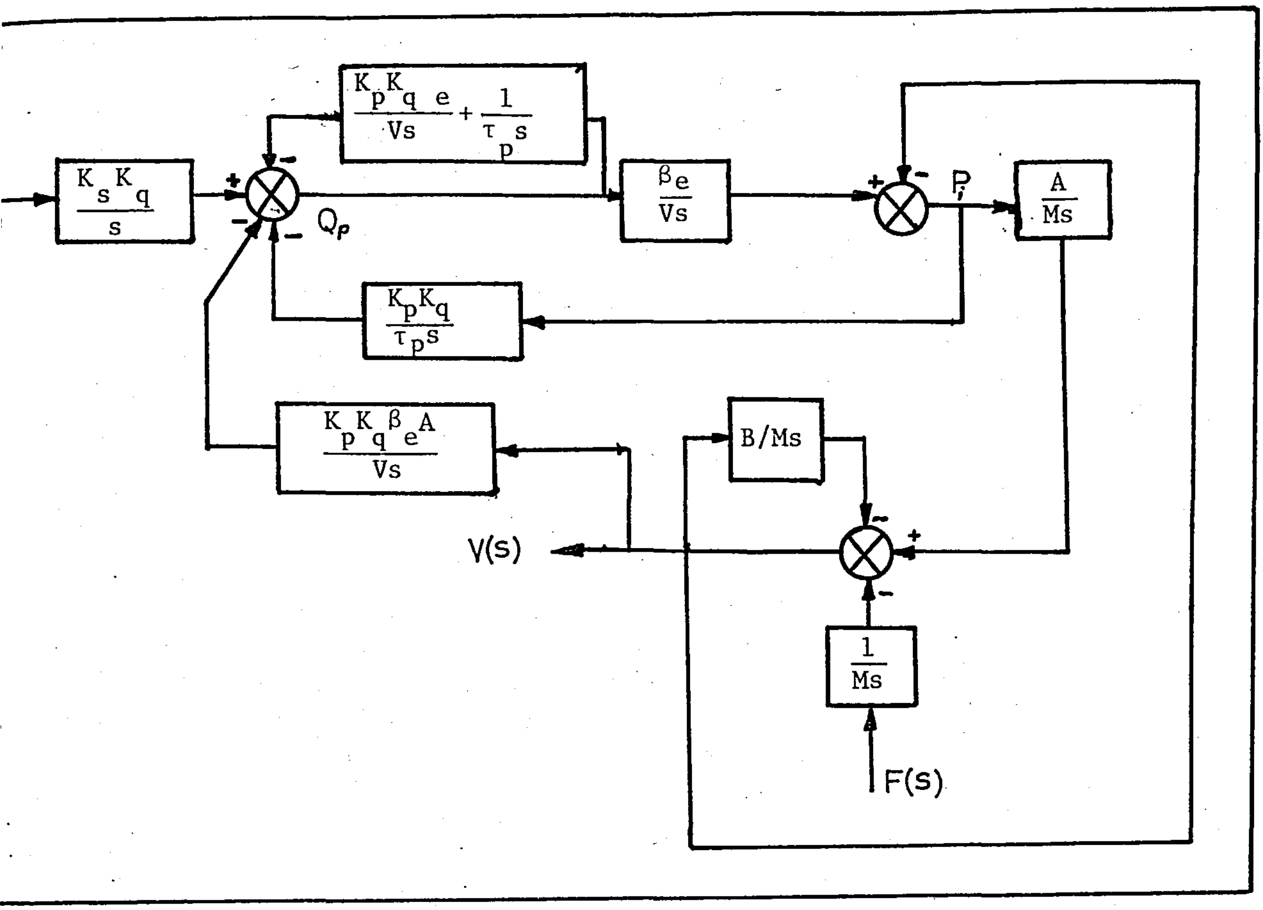

Fig. 4.8. Block diagram for a cylinder system with pressure compensation

4.3.10. Modeling Of A Cylinder Driver System Analysed Due to Directional Control Valve Dynamics

HIDRAN program analyses cylinder systems in two ways, either due to load changes during the operation, or due to the valve position change at time zero. Flow and pressure at two sides of cylinder and cylinder velocity are selected outputs for such a system.

In the analysis of this system, supply pressure to directional control valve, P_s, is assumed constant. Cylinder utilized is a single acting cylinder with rod at one side of piston. Directional valve coefficients are taken as equal for $P_s \rightarrow A$ and $B \rightarrow P_t$ position and $P_s \rightarrow B$ and $A \rightarrow P_t$ position, which is the case for most directional control valve designs.

The load-cylinder and directional control valve combination is shown in Figure 4.9.

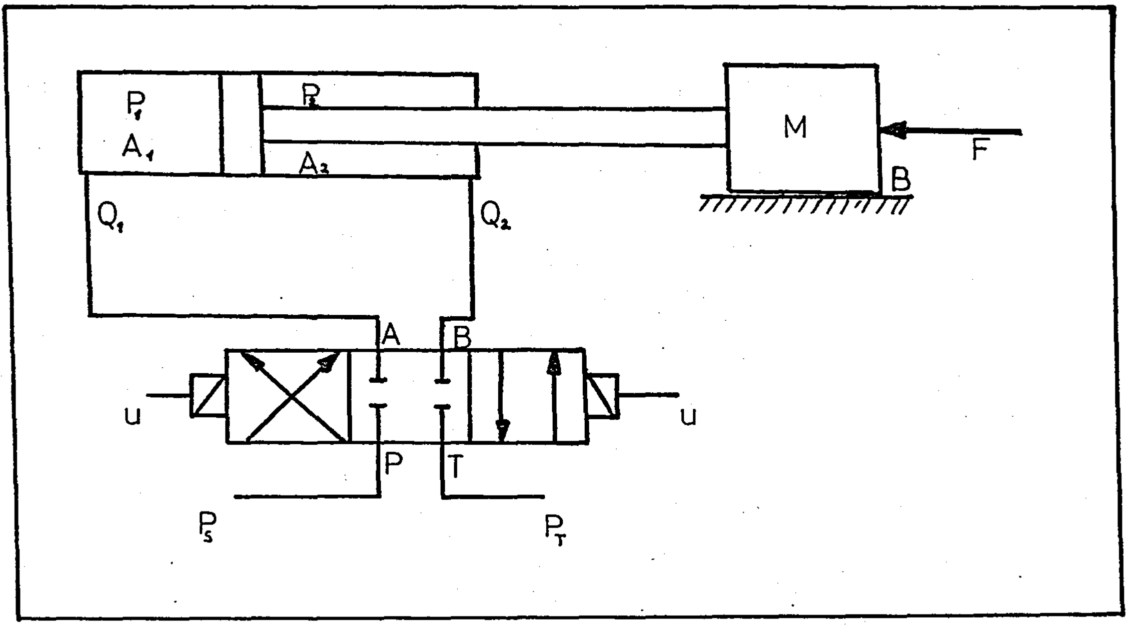

Fig. 4.9. Cylinder-load and directional control valve combination

The dynamic model for such a system may be obtained by the following analysis on cylinder, load and directional control valve sections. For the load:

$$\ddot{Y} = - \frac{B}{M} \dot{Y} + \frac{A_1}{M} P_1 - \frac{A_2}{M} P_2 - \frac{F}{M} \tag{4.42}$$

take $A_2 = \alpha A_1$, $\alpha \leq 1$, then

$$\ddot{Y} = - \frac{B}{M} \dot{Y} + \frac{A_1}{M} (P_1 - \alpha P_2) - \frac{F}{M} \tag{1}$$

For chamber one of cylinder:

$$\dot{P}_1 = \frac{\beta_{e1}}{V_1}\left[Q_1 - A_1\dot{Y}\right] \tag{4.43}$$

Letting Y_1 and Y_2 be the initial cylinder positions from head and rod ends, we can write:

$$\dot{P}_1 = \frac{\beta_{e1}}{Y_1}\left[\frac{Q_1}{A_1} - \dot{Y}\right] \tag{2}$$

For chamber two of cylinder:

$$\dot{P}_2 = \frac{\beta_{e2}}{V_2}\left[Q_2 + A_2\dot{Y}\right] = \frac{\beta_{e2}}{Y_2}\left[\frac{Q_2}{A_2} + \dot{Y}\right] \tag{3}$$

For flow through directional control valve, we can write: $u = +1$; valve position: $P \to A/B \to T$

$$Q_1 = K_1 \operatorname{sgn}(P_s - P_1)\sqrt{(P_s - P_1)\operatorname{sgn}(P_s - P_1)} \tag{4.44}$$

$$Q_2 = K_2 \operatorname{sgn}(P_T - P_2)\sqrt{(P_T - P_2)\operatorname{sgn}(P_T - P_2)} \tag{4.45}$$

$u = 0$; valve position: closed

$$Q_1 = Q_2 = 0$$

$u = -1$, valve position: $P \to B/A \to T$

$$Q_1 = K_2 \operatorname{sgn}(P_T - P_1)\sqrt{(P_T - P_1)\operatorname{sgn}(P_T - P_1)} \tag{4.46}$$

$$Q_2 = K_1 \operatorname{sgn}(P_s - P_2)\sqrt{(P_s - P_2)\operatorname{sgn}(P_s - P_2)} \tag{4.47}$$

Assuming no reverse flow, we can re-write above flow equations in a simpler form:

$$u = +1 : Q_1 = K_1 \sqrt{P_s - P_1} \tag{4.48}$$

$$Q_2 = -K_2 \sqrt{P_2 - P_T} \tag{4.49}$$

$$u = 0 : Q_1 = Q_2 = 0$$

$$u = -1 : Q_1 = -K_2 \sqrt{P_1 - P_T} \tag{4.50}$$

$$Q_2 = K_1 \sqrt{P_s - P_2} \tag{4.51}$$

Therefore the state model becomes:

$$\ddot{y} = -\frac{B}{M}\dot{Y} + \frac{A_1}{M}(P_1 - \alpha P_2) - \frac{F}{M} \tag{1''}$$

$$\dot{P}_1 = -\frac{\beta_{e1}}{Y}\dot{Y} + \frac{\beta_{e1}}{A_1 Y} f_1 (P_1)\, u \tag{2''}$$

$$\dot{P}_2 = \frac{\beta_{e2}}{S-Y}\dot{Y} - \frac{\beta_{e2}}{\alpha(S-Y)A_1} f_2 (P_2)\, u \tag{3''}$$

with the initial conditions: $Y(0) = Y_{10}$; $\dot{Y}(0) = 0$

$P_1(0) = P_T = P_{atm}$

$P_2(0) = P_T = P_{atm}$

and

$$\left.\begin{aligned} f_1 (P_1) &= K_1 \sqrt{P_s - P_1} \\ f_2 (P_2) &= K_2 \sqrt{P_2 - P_T} \end{aligned}\right\} \text{for } u = +1$$

$$\left.\begin{aligned} f_1 (P_1) &= K_2 \sqrt{P_1 - P_T} \\ f_2 (P_2) &= K_1 \sqrt{P_s - P_2} \end{aligned}\right\} \text{for } u = -1$$

$$f_1 (P_1) = f_2 (P_2) = 0 \text{ for } u = 0$$

Above system can be linearized about steady-state values of Y, P_1 and P_2, to obtain a convenient form for computer solution. The system can directly be written in terms of small deviations from the assumed linearization values.

$$\ddot{y} = -\frac{B}{M}\dot{y} + \frac{A_1}{M}p_1 - \alpha\frac{A_1}{M}p_2 - \frac{f}{M} \tag{4.52}$$

$$\dot{p}_1 = -\frac{\beta_{e1}}{Y_L}\dot{y} + \left[\frac{\beta_{e1}}{Y_L A_1}u_L \cdot \frac{\partial f(p_1)}{\partial p_1}\right]_{p_1=P_{1L}} \cdot p_1$$

$$+ \left(\frac{\beta_{e1}}{Y_L}\frac{f\ (p_{1L})}{A_1}\right) u \tag{4.53}$$

$$\dot{p}_2 = \frac{\beta_{e2}}{S-Y_L}\dot{y} - \left[\frac{\beta_{e2}}{S-Y_L} \cdot \frac{u_L}{\alpha A_1} \cdot \frac{\partial f(P_2)}{\partial P_2}\right]_{p_2=P_{2L}} \cdot P_2$$

$$- \left(\frac{\beta_{e2}}{S-Y_L} \cdot \frac{f_2(p_{2L})}{\alpha A_1}\right) u \tag{4.54}$$

where $Y = Y-Y_L$, $p_1 = P_1-P_{1L}$, $p_2 = P_2-P_{2L}$, $f = F-F_L$. The linearized state model has the form:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & 0 \\ a_{31} & 0 & a_{33} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} + \begin{bmatrix} 0 \\ b_1 \\ b_2 \end{bmatrix} u + \begin{bmatrix} C \\ 0 \\ 0 \end{bmatrix} f$$

The coefficients are:

$$a_{11} = \frac{-B}{M} \quad , \quad a_{12} = \frac{A_1}{M} \quad , \quad a_{13} = \frac{A_2}{M} \quad , \quad C = \frac{-1}{M} \quad ,$$

$$a_{21} = \frac{-\beta_{e1}}{Y_L} \quad , \quad a_{31} = \frac{\beta_{e2}}{S-Y_L}$$

for $u = +1$ and $u_L = +1$:

$$a_{22} = \frac{-\beta_{e1}}{Y_L} \cdot \frac{1}{A_1} \cdot \frac{K_1 \sqrt{P_s - P_{1L}}}{2(P_s - P_{1L})}$$

$$a_{33} = \frac{\beta_{e2}}{S-Y_L} \cdot \frac{1}{A_2} \cdot \frac{K_2 \sqrt{P_{2L} - P_T}}{2\ (P_{2L} - P_T)}$$

$$b_1 = \frac{\beta_{e1}}{Y_L} \cdot \frac{K_1 \sqrt{P_s - P_{1L}}}{A_1}$$

$$b_2 = \frac{\beta_{e2}}{S-Y_L} \cdot \frac{K_2 \sqrt{P_{2L} - P_T}}{A_2}$$

For $u = 0$ and $u_L = 0$:

$$a_{22} = a_{33} = b_1 = b_2 = 0$$

For $u = -1$ and $u_L = -1$:

$$a_{22} = \frac{-\beta_{e2}}{Y_L} \cdot \frac{1}{A_1} \cdot \frac{K_2 \sqrt{P_{1L}-P_T}}{2\,(P_{1L}-P_T)}$$

$$a_{33} = \frac{\beta_{e2}}{S-Y_L} \cdot \frac{1}{A_2} \cdot \frac{K_1 \sqrt{P_s-P_{2L}}}{2\,(P_s-P_{2L})}$$

$$b_1 = \frac{\beta_{e1}}{Y_L} \cdot \frac{K_2 \sqrt{P_{1L}-P_T}}{A_1}$$

$$b_2 = \frac{\beta_{e2}}{S-Y_L} \cdot \frac{K_1 \sqrt{P_s-P_{2L}}}{A_2}$$

which completes the dynamic model. The steady state values used in linearization can be obtained by setting time derivatives zero in the system equation, and solving for P_{1L}, P_{2L} and V_L. Doing so, we get:

$$V_L = \frac{\frac{B}{A_1} - \sqrt{(\frac{B}{A_1})^2 - 4\,\{(\frac{A_1}{K_2})^2 + \alpha(\frac{A_2}{K_1})^2\}\,\{P_T - P_s - \frac{F}{A_1}\}}}{2\,\{(\frac{A_1}{K_2})^2 + \alpha(\frac{A_2}{K_1})^2\}} \qquad (4.55)$$

$$P_{1L} = P_T + \left(\frac{V_L}{K_2}\right)^2 A_1^2 \tag{4.56}$$

$$P_{2L} = P_s - \left(\frac{A_2}{K_1}\right)^2 V_L^2 \tag{4.57}$$

which are the steady-state values, to be used in linearization of system equations. Steady-state piston position is user defined (measured at head-end side of piston).

4.3.11. Block Diagram for Directional Control Valve Dynamics

Above state model may be transformed into Laplace domain to obtain:

$$s\dot{Y}(s) = a_{11}\dot{Y}(s)+a_{12}P_1(s)+a_{13}P_2(s)+CF(s) \tag{4.58}$$

$$sP_1(s) = a_{21}\dot{Y}(s) + a_{22}P_1(s) + b_1u(s) \tag{4.59}$$

$$sP_2(s) = a_{31}\dot{Y}(s) + a_{33}P_2(s) + b_2u(s) \tag{4.60}$$

This system will lead to the block diagram shown in Figure 4.10.

The transfer function for this system may be expressed by:

$$\dot{Y}(s) = \frac{a_{12}b_1(s+a_{33}) + a_{13}b_2(s+a_{22})}{(s+a_{11})(s+a_{22})(s+a_{33})+a_{12}a_{21}(s+a_{33})+a_{13}a_{31}(s+a_{22})} \tag{4.61}$$

where the coefficients depend on valve position and are defined above.

4.4. MATHEMATICAL SOLUTION METHOD

The matrix form of dynamic models expressed in above analysis is suitable to be solved by a digital computer.

The equation system to be solved is a linear, time - invariant matrix differential equation in the form of:

$$\dot{\vec{X}}(t) = \underline{\underline{A}}\vec{X}(t) + \underline{B}U(t) + \underline{C}M(t)$$

Subject to the initial condition: $\vec{X}(t_o) = \vec{X}_o$.

The solution to above equation is given by (see Appendix E:)

$$\vec{X}(t) = e^{\underline{\underline{A}}(t-t_o)}\,\vec{X}(t_o) + \int_{t_o}^{t} e^{\underline{\underline{A}}(t-\tau)}\underline{B}\,U(\tau)d\tau \; +$$

$$+ \int_{t_o}^{t} e^{\underline{\underline{A}}(t-\tau)}\underline{C}\,M(\tau)d\tau \qquad (4.62)$$

To enable the use of a digital computer, above continuous time system is discretized by taking $t_o = 0$.

Assuming that U and M are constant between two consecutive sampling instants, we can write:

$U(t) = U(kT)$

$\vec{X}(t) = X((k+1)\,T)$, where T is a sampling period. Then we obtain equation (4.62) in discrete form.

$$\bar{X}((k+1)T) = e^{\underline{\underline{A}}t}\,\vec{X}(kT) + \int_0^T e^{\underline{\underline{A}}\tau}\,\underline{B}\,d\tau\,U(kT) + $$

$$+ \int_0^T e^{\underline{\underline{A}}\tau}\,\underline{C}\,d\tau\,M(kT) \qquad (4.63)$$

If we define; the three matrices $\underline{\underline{G}}$, $\underline{\underline{H}}$ and $\underline{\underline{Q}}$ as:

$$\underline{\underline{G}}(T) = e^{\underline{\underline{A}}T}$$

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

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

above equation becomes: (equation 4.63)

$$\vec{X}((k+1)T) = \underline{\underline{G}}(T)\vec{X}(kT) + \underline{\underline{H}}(T)\;U(kT) + \underline{\underline{Q}}(T)\;M(kT)$$

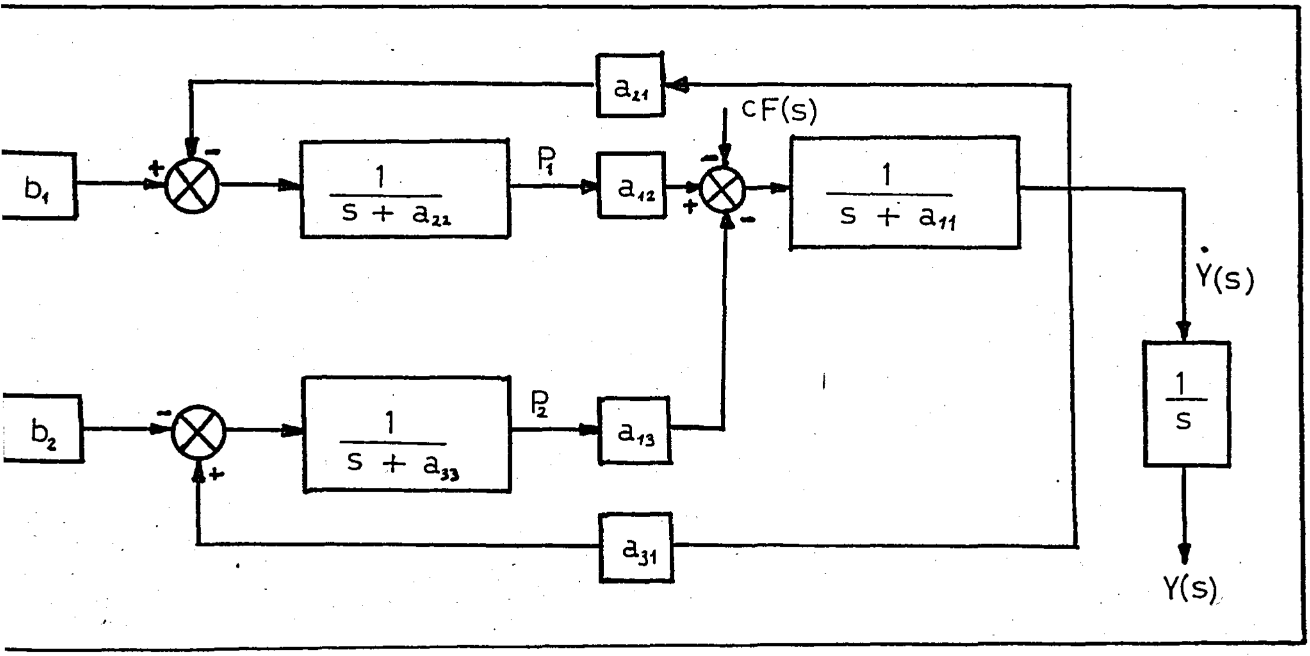

Fig. 4.10. Block diagram for directional control valve dynamics

Which is a vector matrix difference equation, representing the solution of system equations, only at the discrete points of time, $t = kT$, $k = 0,1,2,\ldots$

Consequently the problem reduces to forming of the coefficient matrices $\underline{\underline{G}}(T)$, $\underline{\underline{H}}(T)$ and $\underline{\underline{Q}}(T)$, from the system matrix $\underline{\underline{A}}$, and vectors $\underline{B}$ and $\underline{C}$, and processing of above equation in the computer. Given the initial values for the state variables, the values at every multiple of step size are obtained as outputs.

In the HIDRAN program, a subroutine called DISCR3, developes G, H and Q matrices, and the integrals over the time interval. The method used is doubling formulae method, and largest convergent step size and corresponding doubling number is based on Van Loan's Criterion*.

Subroutine GMPRD, obtains a matrix product, by conventional methods, and MINV uses standart Gauss-Jordan method for the inversion of a matrix.

The simulation results can be taken at any discrete time points as required.

4.5. PRESSURE RELIEF VALVE OPERATION AND ENERGY LOSSES DURING TRANSIENT RESPONSE TIME

The Program HIDRAN, asks for the steady-state system pressure and pressure relief valve sets (if any) at various locations of the hydraulic system. During the computation of system variables, it checks for the maximum value that pressure differential can take, that is

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*Van Loan C.F., "Computing Integrals Involving the Matrix Exponential", IEEE Trans. on Autom. Control, AC-23, pp.395-404, 1978.

$$P_{max} = (P_{RV} - P_{SS}) \qquad (4.64)$$

which is the difference between relief valve set and steady - state pressure. If system pressure reaches set value of the relief valve, valve opens, and excessive flow passes from the high pressure side of the circuit to low pressure side, which results in energy dissipation.

In such a situation, usefull flow rate can be calculated by setting pressure differential to zero, in the pump flow rate equation. That is:

$$Q_p^* = \frac{D_m}{\mu_v} w_m \qquad (4.65)$$

Energy loss can be determined by integrating the equation

$$\dot{E}_L = P_{RV} (Q_p - Q_p^*) \qquad (4.66)$$

over the time interval in which relief valve is open. Integration is performed by using the trapeze rule. The only inaccuracy in these computations may result due to the omission of relief valve dynamics. This situation is analysed and following figure is obtained.

The time versus opening graph of pressure relief valve was as follows (Fig. 4.12).

It is seen from the curve that inaccuracy of HIDRAN program results because of relief valve opening delay and at relatively small simulation time.

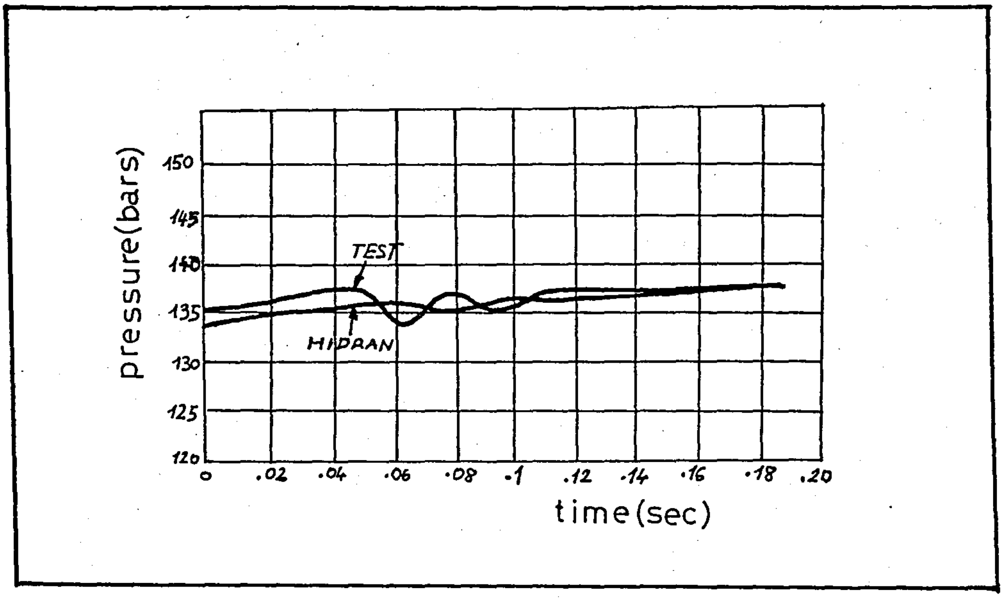


Fig. 4.11. Pressure curve at pressure relief valve inlet

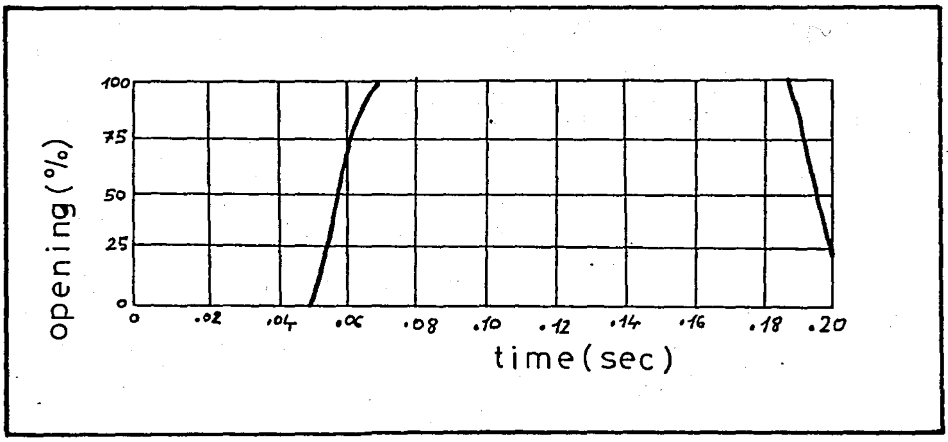


Fig. 4.12. Time vs. opening graph of pressure relief valve

4.6. ACCURACY OF HIDRAN PROGRAM

The results obtained by HIDRAN program are numerical solutions to system of differential equations developed to describe system dynamics. Results are, therefore subject to errors inherent in numerical solution methods, but can be kept small enough to have a minimal impact, less than 5%, on the final results.

Other forms of inaccuracy are the omission of directional control valve and pressure relief valve dynamics, but these fall well beyond the accuracy limits required by most applications. A typical circuit test data is compared to HIDRAN results, obtained for the same system and load disturbance conditions. The system was a motor system, operating at 120 bars. Load disturbance was 10 newton-meters (Fig. 4.13).

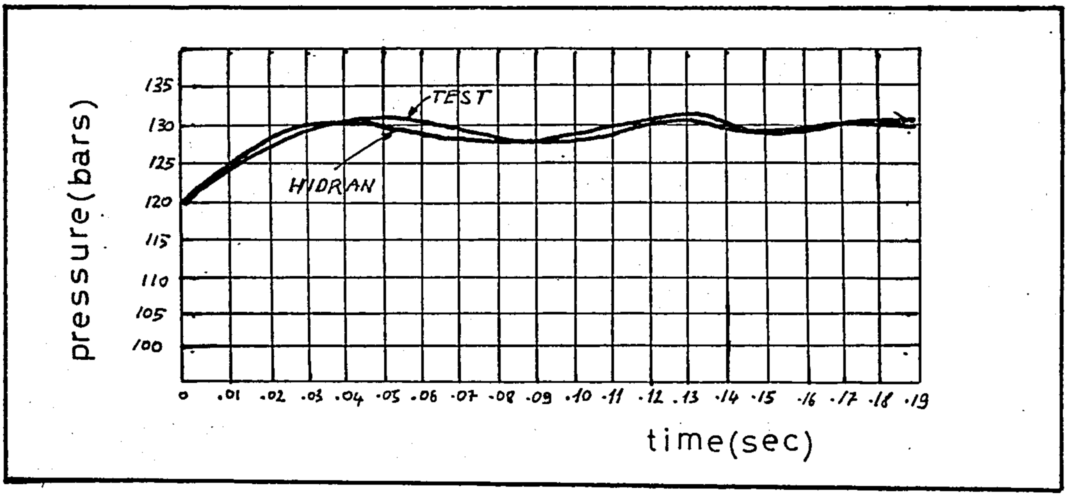
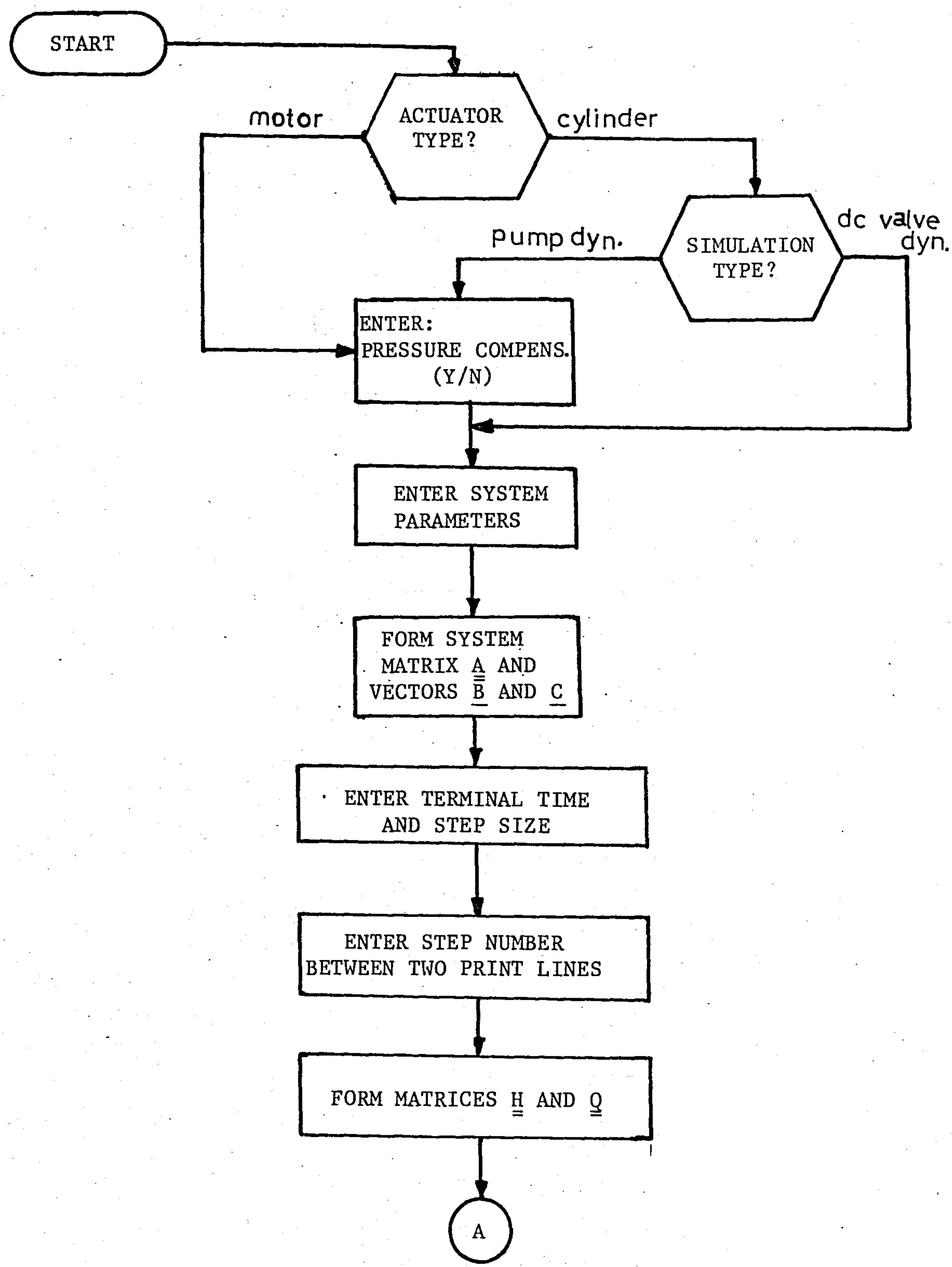
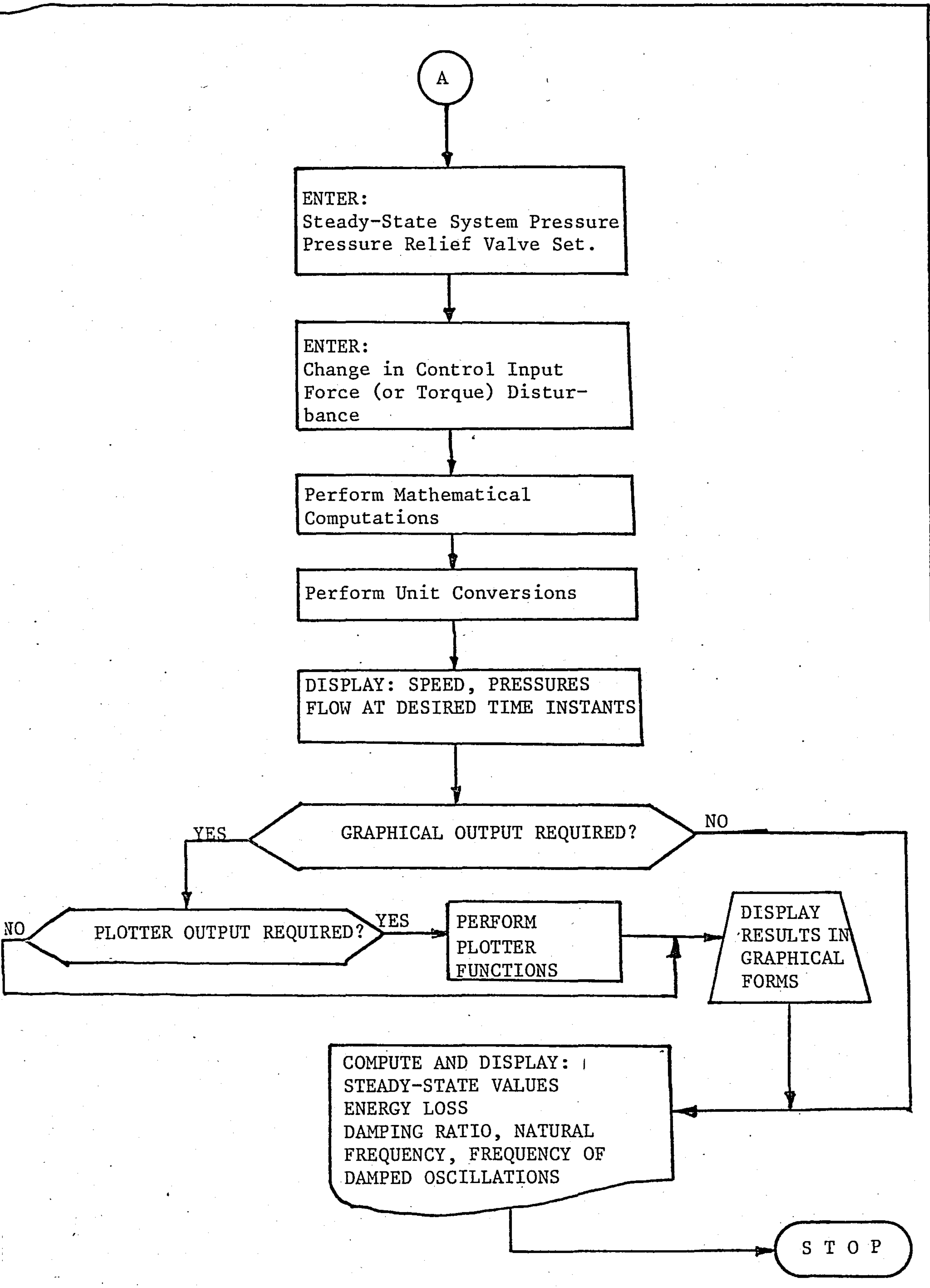


Fig. 4.13. Comparison of HIDRAN with test results

Based on this comparison, it may be concluded that HIDRAN program predicts pressures, speeds and flows, with a reasonable accuracy. The difference is mainly due to the technique used in solving differential equations during the simulation process and due to assumptions made to simplify complex mathematical models of hydraulic components. These assumptions speed up the execution of the program, produce more stable outputs, and require less detailed input data. A general flowchart of HIDRAN program is given in Figure 4.14.



ig. 4.14. General flowchart for HIDRAN program



ig. 4.14. Continued

V. CONCLUSIONS

Hydraulic systems should be optimally designed and tested before going into service because of their relatively higher initial cost and longer operation life.

Computer Aided Design techniques can take the place of conventional design by giving a much quicker and more accurate solution to the problem. The designer has the possibility to simulate the circuit, and investigate the dynamical behaviour, before building up and testing the system, therefore he can add or improve components without constructing a prototype.

The capability and accuracy of CAD program largely depends on the data files of components. Using files with insufficient number of components, program results in inaccurately dimensioned systems.

Consideration of dynamical behaviour of hydraulic systems is a necessity for modern design. Observations on the dynamic response of several solutions to a design problem enables the designer to select the best configuration.

From the dynamic simulation results, we can conclude that larger systems show a faster response to load changes and are more stable. Pressure compensation can be used to increase stability and damping ratio of a system, but results in almost constant torque (or force) transmission and reduced speeds at increased load (Appendix C).

The dynamic simulation results, obtained from CAD program are compared to actual test results, and it may be concluded that accuracy figures are well beyond the limits required for most applications. The dynamics of prime mover and pressure relief valve may be added to the program to obtain much higher accuracy in special applications.

APPENDIX A - SCREEN OUTPUTS FOR HYSAN AND HIDRAN PROGRAMS

Screen outputs and inputs entered by user during the execution of HYSAN and HIDRAN programs are given in below outputs.

The system designed by HYSAN was a four cylinder system, operating at 200 bars, driven by a constant displacement pump.

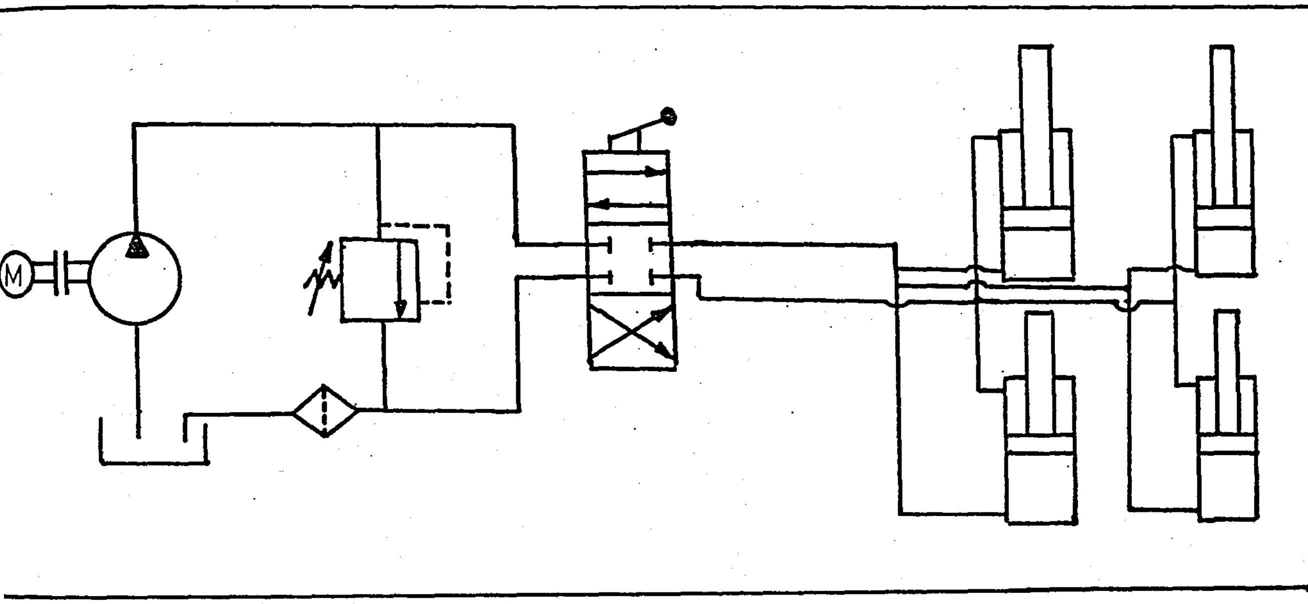


Fig.A1. System designed by HYSAN to indicate screen outputs

Note that the inputs after the (?) sign were entered by the user.

```
CADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCAD
F                                                          F
P                                                          P
S                                                          S
F                  COMPUTER AIDED DESIGN                   F
P                           OF                             P
S              FLUID POWER CONTROL SYSTEMS                 S
F                                                          F
P                                                          P
S                                                          S
CADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCADCAD



NTER JOB NO.(LESS THAN 100)
24

   JOB NO.=24

RAPHICS SOFTWARE REQUIRED?(0,1)
IF REQUIRED ENTER 1.IF NOT ENTER 0
0


NTER SYSTEM PRESSURE(BARS)

ECOMMENDED PRESSURE RATINGS
 LOWER PRESSURE RATING     80 BARS
 MEDIUM PRESSURE RATING   160 BARS
 HIGH PRESSURE RATING     240 BARS

200


   RATED SYSTEM PRESSURE=200.0  BARS

NTER ACTUATOR TYPE

 ACTUATOR TYPE  ENTERING VALUE
   CYLINDER            1
   MOTOR               2

1



                        **CYLINDER SELECTION**


NTER TYPE:
CYLINDER TYPE     ENTERING VALUE
  A.OR B               1
  C.OR D.OR F          2
  E                    3

1
NTER NO. OF CYLINDERS IN PARALLEL
```

```
*ENTER MAX LOAD.FRW VELOCITY.STROKE
   MAX. LOAD(N):

? 150000
   FRW. VELOCITY(M/S)

? .1
   STROKE(MM)

? 10000 .
 SELECTION TYPE?
 1:SELECTION FROM MENU
 2:PREVIOUSLY SELECTED
? 1

      ASKED SPCS:
           NO. OF CYLINDERS IN PARALLEL=  4
           MAX LOAD=150000.000  N
           MAX VELOCITY=   .100  M/S
           STROKE=  1000.0  MM
           TYPE=A.B

      CHOSEN CYLINDER'S SPCS:
           PISTON DIAMETER=100.0  MM
           ROD DIAMETER= 70.0  MM
           MAX STROKE=  1250.0  MM
           MAX PRESSURE=350.0  BARS

      CALCULATED SPCS:
           NECESSARY FLOW=   188.496  LIT/MIN
           LOAD PRESSURE=  191.0  BARS


*ENTER CONTROL KEY(1.2.3)
 1:RESULTS OK.CONTINUE/2:RETRY CYLINDER SELECTION/3:STOP
? 1



                                **PUMP SELECTION**


*ENTER PUMP TYPE:
  PUMP TYPE             ENTERING VALUE
  GEAR                         1
  VANE (FIXED)                 2
  RADIAL PISTON (FIXED)        3
  AXIAL PISTON (FIXED)         4
  AXIAL PISTON (VARIABLE) 5 (FOR OPEN CIRCUITS)
  AXIAL PISTON (VARIABLE) 6 (FOR CLOSED CIRCUITS)
? 4
*ENTER PRIME MOVER'S RPM
? 1800
 SELECTION TYPE?
 1:SELECTION FROM MENU
 2:ALREADY SELECTED PUMP
? 1

      ASKED SPCS:
           TYPE=AXIAL PISTON
           PRIME MOVER'S RPM= 1800.0

      CHOSEN PUMP'S SPCS:
```

```
          MAX RPM= 3000.0
          MAX PRESSURE=  400.0  BARS
          VOL.EFF.= .93
          MEC.EFF.= .94

     CALCULATED SPCS:
          MAX THEO. FLOW=     225.00  LT/MIN
          ACTUAL PUMP FLOW=     209.25  LT/MIN


ENTER CONTROL KEY(1.2.3)
1:RESULTS OK.CONTINUE/2:RETRY PUMP SELECTION/3:STOP
 1


     UPDATED PARAMETERS:

          ACTUAL FORWARD VELOCITY=        .111  M/S

          ACTUAL BACKWARD VELOCITY=        .218  M/S

          RETURN FLOW=   106.718  LT/MIN


ENTER CONTROL KEY(1.2.3)
1:RESULTS OK.CONTINUE/2:RETRY PUMP SELECTION/3:STOP
 1



                                 **PIPE SELECTION**
SELECTION TYPE?
1:SELECTION FROM MENU
2:ALREADY SELECTED PIPES
? 1

     PIPE LINE SPCS:
          SUCTION AND RETURN LINE DIAMETER(NOMINAL)=65  MM
               INNER DIAMETER=66.00  MM
               OUTER DIAMETER=74.20  MM
          DELIVERY LINE DIAMETER(NOMINAL)=40  MM
               INNER DIAMETER=40.90  MM
               OUTER DIAMETER=48.30  MM


*ENTER CONTROL KEY(1.2.3)
 1:RESULTS OK.CONTINUE/2:RESULTS OK.CONTINUE/3:STOP
? 1



                                 **DC VALVE SELECTION**


*ENTER DC VALVE TYPE:
  DC VALVE TYPE               ENTERING VALUE
  E (CLOSED CENTER)                 1
  G (TANDEM CENTER)                 2
  H (OPEN CENTER)                   3
  NO DC VALVE REQUIRED              4

? 1

     DC VALVE SPCS:
          TYPE=E
          INLET PORT NOMINAL RADIUS=NG22
```

```
        MAX PRESSURE=  350.0  BARS


NTER CONTROL KEY(1.2.3)
:RESULTS OK.CONTINUE/2:RETRY DC VALVE SELECTION/3:STOP
1



                              **PR VALVE SELECTION**

   PR VALVE SPCS:
        MAX PRESSURE=  315.0  BARS
        INLET PORT NOMINAL RADIUS=NG16


NTER CONTROL KEY(1.2.3)
:RESULTS OK.CONTINUE/2:RESULTS OK.CONTINUE/3:STOP
1



                              **FILTER SELECTION**

   CHOSEN FILTER'S SPCS:
        MAX FLOW= 660.00  LT/MIN
        FILTERING SIZE=10.0  MIC


NTER CONTROL KEY(1.2.3)
1:RESULTS OK.CONTINUE/2:RESULTS OK.CONTINUE/3:STOP
 1

NTER CONTROL-KEY FOR P.LOSS COMPUTATION:

IF PRESSURE LOSS COMPUTATION IS REQUIRED.ENTER 1.
IF NOT ENTER 0

 1



                              **P.LOSS COMPUTATION**


NTER OIL VISCOSITY(M2/S)
  DEFAULT OIL VISCOSITY= 0.4 E -4  M2/S
IF THIS DEFAULT VALUE IS SUFFICIENTLY TRUE.ENTER 1
THERWISE ENTER REAL VISCOSITY NUMBER

 1

    OIL VISCOSITY=  .400E-04  M2/S


NTER OIL DENSITY(KG/M3)
  DEFAULT OIL DENSITY= 858.2 KG/M3
IF THIS DEFAULT VALUE IS SUFFICIENTLY TRUE.ENTER 1.
OTHERWISE ENTER REAL DENSITY VALUE

 1

    OIL DENSITY= 858.20  KG/M3
```

```
                  **PRESSURE LOSS COMPUTATION FOR LINE    1**

TER PIPE LENGTH
4
TER NO.OF T JOINTS

TER NO.OF L JOINTS


  SUCTION LINE P.LOSS=     .00326  BARS
       SUCTION LINE P.LOSS COEFF.=       268.


TER THE HEIGHT DIFFERENCE BETWEEN PUMP AND TANK (M)
    (NEGATIVE IF PUMP IS ABOVE)
.03

  PRESSURE AT THE PUMP INLET=     .99422  BARS
ATTENTION:NEGATIVE PRESSURE AT PUMP INLET



                  **PRESSURE LOSS COMPUTATION FOR LINE    2**

TER PIPE LENGTH
2
TER NO.OF T JOINTS

TER NO.OF L JOINTS
:

  DELIVERY LINE P. LOSS=     .18895  BARS
       DELIVERY LINE P.LOSS COEFF.=     15535.

  DC VALVE P.LOSS(P-A)=   2.05301  BARS
       DC VALVE P.LOSS COEFF.(P-A)=    168796.



                  **PRESSURE LOSS COMPUTATION FOR LINE    3**

NTER PIPE LENGTH
2
NTER NO.OF T JOINTS
)
NTER NO.OF L JOINTS
3

  LINE A P.LOSS=     .21616  BARS
       LINE A P.LOSS COEFF.=     17772.



                  **PRESSURE LOSS COMPUTATION FOR LINE    4**

NTER PIPE LENGTH
2
NTER NO.OF T JOINTS
)
NTER NO.OF L JOINTS
3
```

```
INE B P.LOSS(RETURN)=     .05238  BARS
    LINE B P.LOSS COEFF.=     16558.

C VALVE P.LOSS(B-T)=     .92578  BARS
    DC VALVE P.LOSS COEFF.(B-T)=    292643.



                **PRESSURE LOSS COMPUTATION FOR LINE    5**

 PIPE LENGTH

 NO.OF T JOINTS

 NO.OF L JOINTS

ETURN LINE P.LOSS=     .00691  BARS
    RETURN LINE P.LOSS COEFF.=      2185.

ET.LINE FILTER P.LOSS=     .01265  BARS
    RET.LINE FILTER P.LOSS COEFF.=      4000.

OTAL PRESSURE LOSSES=   3.45585  BARS
    TOTAL P.LOSS COEFF.=   284136.  BAR/(M3/S)2

HE PRESSURE ON THE PUMP= 193.95244  BARS

R VALVE SET PRESSURE= 213.34769  BARS

OWER REQUIRED BY THE PUMP=  71.95842  KW
ECOMMENDED MINIMUM PRIME MOVER POWER=   79.2KW
ECOMMENDED MINIMUM RESERVOIR CAPACITY= 627.8 LITERS
VERALL SYSTEM EFFICIENCY=81. PERCENT

 IS COMPLETED.PROGRAM STOP!
OP
```

The system analysed by HIDRAN was a motor system operating at 200 bars. The disturbances acting on the system were a step increase of pump control current, and motor torque. The computer outputs and inputs entered by user are given in next pages.

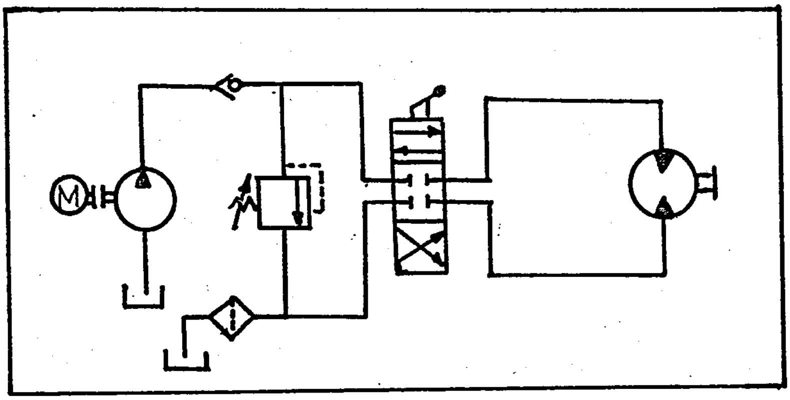


Fig.A2 Motor system analysed by HIDRAN

```
8
NTER ACTUATOR TYPE
CTUATOR TYPE  ENTERING VALUE
YLINDER          1
TOR              2
2
NTER CASE NO.(1,2)
 SYSTEM WITHOUT PRESSURE COMPENSATION
 SYSTEM WITH PRESSURE COMPENSATION


                         **SYSTEM PARAMETERS**

NTER VALUES OF THE FOLLOWING SYSTEM PARAMETERS
OTOR INERTIA=? (KG-M2)
.003
OAD INERTIA=? (KG-M2)
2
OAD VISCOUS FRICTION COEFF=? (N-M/RD-S)
4
OTOR DISPLACEMENT=? (CM3/REV)
120
OTOR MECH. EFFIC.=? (&)
.95
OTOR VOL. EFFIC.=? (&)
.93
ELIVERY LINE LENGTH=? (M)
4
ELIVERY LINE INNER DIA.=? (MM)
40.9
ULK MOD. OF HYDRAULIC FLUID=? (N/M2)
26e9
ULK MOD. OF PIPE=? (N/M2)
30e9
OL. PERCENT OF ENTRAINED AIR=? (%)
1
YSTEM PRESSURE=? (BARS)
200
IME CONST. OF PUMP SERVO=? (S)
.3
UMP SERVO CONTROL GAIN=? (M3/S-RD)
.01
UMP CONTROL CURRENT GAIN=? (RD/A)
.1
NTER TERMINAL TIME AND STEP SIZE
2,.01


    STEP SIZE= .010000  S
    TERMINAL TIME=   2.00  S
    NUM.OF ITERATIONS=     200
NTER STEP NUMBER BETWEEN TWO PRINTOUT LINES
8
NTER STEADY STATE SYSTEM PRESSURE(BARS)
200
NTER PRESSURE RELIEF VALVE SET(BARS)
300
NTER CHANGE IN CONTROL INPUT
.1
NTER TORQUE DISTURBANCE (N-M)
125.5
```

RIATIONS IN SYSTEM VARIABLES DUE TO APPLIED INPUTS :

TIME(S)	SPEED(RPM)	PRES.(BARS)	FLOW(LT/MIN)
.010	-1.200	.1312	.197
.080	-7.253	7.5501	1.40
.160	-4.971	24.6535	2.48
.240	8.284	41.2760	3.30
.320	27.674	49.2349	3.94
.400	45.502	45.7278	4.42
.480	55.401	33.8247	4.79
.560	55.254	20.1406	5.07
.640	47.648	11.1949	5.29
.720	38.046	10.3583	5.46
.800	31.848	16.7296	5.58
.880	31.922	26.1840	5.68
.960	37.660	33.7785	5.76
1.040	45.807	36.1863	5.81
1.120	52.395	33.0142	5.86
1.200	54.727	26.5503	5.89
1.280	52.462	20.2932	5.92
1.360	47.435	17.1309	5.94
1.440	42.485	18.0769	5.95
1.520	39.974	22.0712	5.96
1.600	40.762	26.7838	5.97
1.680	44.022	29.8943	5.98
1.760	47.886	30.1778	5.98
1.840	50.472	27.9169	5.99
1.920	50.771	24.5457	5.99
2.000	48.992	21.8072	5.99

APHICAL OUTPUT REQUIRED(0,1)?
REQUIRED ENTER 1.IF NOT ENTER 0

STEADY STATE DEVIATIONS DUE TO INPUTS:

STEADY STATE DEVIATION IN MOTOR SPEED= 46.500 RPM
STEADY STATE DEVIATION IN MOTOR PRESSURE= 24.790 BARS
STEADY STATE DEVIATION IN PUMP FLOW RATE= 6.000 LT/MIN

ENERGY LOSS= .000 JOULES

SYSTEM DAMPING RATIO= .1097

NOTE:UNDERDAMPED SYSTEM-OSCILLATIONS POSSIBLE

SYSTEM NATURAL FREQUENCY= 1.4488 HZ

FREQUENCY OF DAMPED OSCILLATIONS= 1.4400HZ

) OF DYNAMIC SIMULATION-PROGRAM STOP
STOP

0.972 CP SECONDS EXECUTION TIME.

APPENDIX B - COMPLETE EXAMPLES OF STEADY-STATE DESIGN BY HYSAN.

The complete examples of steady-state design by HYSAN are given in following pages. The systems are a cylinder and a motor system.

The cylinder system was designed to move a maximum of 20,000 newtons force, at a minimum velocity of 0.3 m/s, using a single cylinder. System was driven by a constant displacement axial piston pump.

The plotter and printer outputs showing the circuit diagram, component selection and steady-state analysis results are given.

The motor system was designed to deliver a maximum torque of 160 newton-meters, at 1500 rpm. System pressure was 160 bars, driven by a variable displacement axial piston pump.

The plotter and printer outputs of HYSAN program, showing circuit layout and design results are given in following pages.

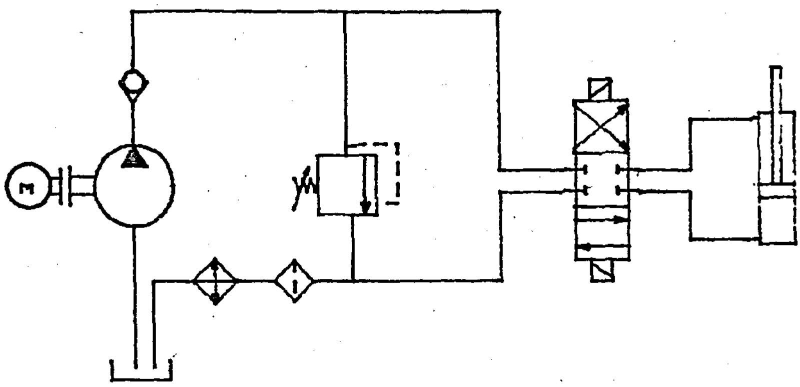


Fig. B1. Plotter output of cylinder system designed by HYSAN

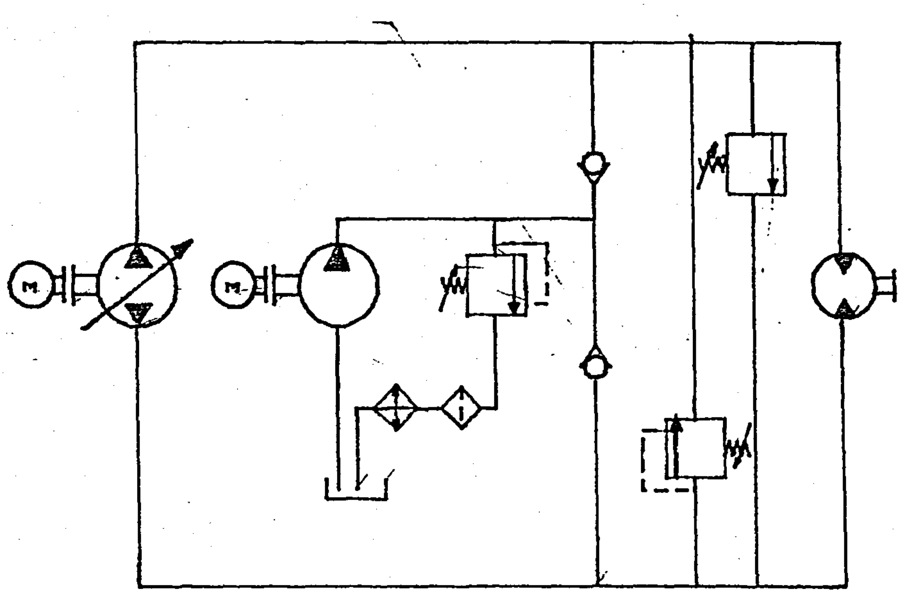


Fig. B2. Plotter output of motor system designed by HYSAN

```
                           CADCADCADCADCADCADCADCADCADCADCADCADCADCAD
                           F
                           P
                           S
                           F                       COMPUTER AIDED DESIGN
                           P                                OF
                           S                   FLUID POWER CONTROL SYSTEM
                           F
                           P
                           S
                           CADCADCADCADCADCADCADCADCADCADCADCADCADCAD

NO.= 1

D SYSTEM PRESSURE= 80.0  BARS


                     **CYLINDER SELECTION**

ED SPCS:
  NO. OF CYLINDERS IN PARALLEL=  1
  MAX LOAD= 20000.000  N
  MAX VELOCITY=   .300  M/S
  STROKE=   200.0  MM
  TYPE=A,B

SEN CYLINDER'S SPCS:
  PISTON DIAMETER= 63.0  MM
  ROD DIAMETER= 36.0  MM
  MAX STROKE=  214.0  MM
  MAX PRESSURE=250.0  BARS

CULATED SPCS:
  NECESSARY FLOW=     56.111  LIT/MIN
  LOAD PRESSURE=   64.2  BARS


                      **PUMP SELECTION**

FD SPCS:
  TYPE=AXIAL PISTON
  PRIME MOVER'S RPM= 1800.0
```

```
CHOSEN PUMP'S SPCS:
     DISPLACEMENT=     54.800  CM3/REV
     MAX RPM= 3750.0
     MAX PRESSURE=  350.0  BARS
     VOL.EFF.= .93
     MEC.EFF.= .94

CALCULATED SPCS:
     MAX THEO. FLOW=        93.64  LT/MIN
     ACTUAL PUMP FLOW=        91.74  LT/MIN


UPDATED PARAMETERS:

     ACTUAL FORWARD VELOCITY=         .490  M/S

     ACTUAL BACKWARD VELOCITY=         .728  M/S

     RETURN FLOW=     61.781  LT/MIN



                         **PIPE SELECTION**

PIPE LINE SPCS:
     SUCTION AND RETURN LINE DIAMETER(NOMINAL)=40  MM
          INNER DIAMETER=40.90  MM
          OUTER DIAMETER=43.30  MM
     DELIVERY LINE DIAMETER(NOMINAL)=25  MM
          INNER DIAMETER=26.60  MM
          OUTER DIAMETER=33.40  MM



                         **DC VALVE SELECTION**

DC VALVE SPCS:
     TYPE=E
     INLET PORT NOMINAL RADIUS=NG16
     MAX PRESSURE=  350.0  BARS



                         **PR VALVE SELECTION**

PR VALVE SPCS:
     MAX PRESSURE=  630.0  BARS
     INLET PORT NOMINAL RADIUS=NG10



                         **FILTER SELECTION**

CHOSEN FILTER'S SPCS:
     MAX FLOW= 160.00  LT/MIN
     FILTERING SIZE=10.0  MIC



                         **P.LOSS COMPUTATION**

OIL VISCOSITY=  .400E-04  M2/S

OIL DENSITY= 858.20  KG/M3
```

```
     SUCTION LINE P.LOSS COEFF.=       2678.

PRESSURE AT THE PUMP INLET=     .97644  BARS
     ATTENTION:NEGATIVE PRESSURE AT PUMP INLET

DELIVERY LINE P. LOSS=      .21535  BARS
     DELIVERY LINE P.LOSS COEFF.=    92124.

DC VALVE P.LOSS(P-A)=      .67202  BARS
     DC VALVE P.LOSS COEFF.(P-A)=   287431.

LINE A P.LOSS=     .32952  BARS
     LINE A P.LOSS COEFF.=   140965.

LINE B P.LOSS(RETURN)=      .15112  BARS
     LINE B P.LOSS COEFF.=   142532.

DC VALVE P.LOSS(B-T)=      .59614  BARS
     DC VALVE P.LOSS COEFF.(B-T)=   562270.

RETURN LINE P.LOSS=     .03035  BARS
     RETURN LINE P.LOSS COEFF.=    18906.

RET.LINE FILTER P.LOSS=     .07422  BARS
     RET.LINE FILTER P.LOSS COEFF.=    70000.

TOTAL PRESSURE LOSSES=   2.05841  BARS
     TOTAL P.LOSS COEFF.=   880565.  BAR/(M3/S)2

THE PRESSURE ON THE PUMP=  65.94268  BARS


PR VALVE SET PRESSURE=  72.53695  BARS


POWER REQUIRED BY THE PUMP=  10.72565  KW




RECOMMENDED MINIMUM PRIME MOVER POWER=   11.3KW


RECOMMENDED MINIMUM RESERVOIR CAPACITY= 275.2 LITERS


OVERALL SYSTEM EFFICIENCY=40. PERCENT




 IS COMPLETED



1.02.UCLP, UU, P03    ,       0.225KLNS.
```

```
CADCADCADCADCADCADCADCADCADCADCADCADCADC
F
P
S
F                  COMPUTER AIDED DESIG
P                          OF
S           FLUID POWER CONTROL SYST
F
P
S
CADCADCADCADCADCADCADCADCADCADCADCADCADC

OB NO.= 2

RATED SYSTEM PRESSURE=160.0  BARS


                         **MOTOR SELECTION**

SKED SPCS:
     NO. OF MOTORS IN PARALLEL= 1
     MAX LOAD TORQUE=    160.000  N-M
     MAX SPEED=1500.00  RPM
     TYPE=AXIAL PISTON

HOSEN MOTOR'S SPCS:
     DISPLACEMENT=    80.00  CM3/REV
     MAX SPEED=  3350.0  RPM
     MAX PRESSURE=    350.0  BARS
     VOL.EFF.= .92
     MEC.EFF.= .94
     MOTOR INERTIA=     .01500  KG-M2

ALCULATED SPCS:
     NECESSARY FLOW=    130.435  LT/MIN
     LOAD PRESSURE= 133.635  BARS


                         **PUMP SELECTION**

SKED SPCS:
     TYPE=AXIAL PISTON(VAR.)
     PRIME MOVER'S RPM= 2000.0
```

CHOSEN PUMP'S SPCS:
    DISPLACEMENT=   90.000 CM3/REV
    MAX RPM= 2000.0
    MAX PRESSURE= 400.0 BARS
    VOL.EFF.= .95
    MEC.EFF.= .95

CALCULATED SPCS:
    MAX THEO. FLOW=   180.00 LT/MIN
    ACTUAL PUMP FLOW=   130.43 LT/MIN

UPDATED PARAMETERS:
    ACTUAL MOTOR SPEED= 1500.0 RPM
    RETURN FLOW=   120.000 LT/MIN

PIPE SELECTION

PIPE LINE SPCS:
    SUCTION AND RETURN LINE DIAMETER(NOMINAL)=50 MM
        INNER DIAMETER=52.50 MM
        OUTER DIAMETER=60.30 MM
    DELIVERY LINE DIAMETER(NOMINAL)=32 MM
        INNER DIAMETER=35.10 MM
        OUTER DIAMETER=42.20 MM

PR VALVE SELECTION

PR VALVE SPCS:
    MAX PRESSURE= 315.0 BARS
    INLET PORT NOMINAL RADIUS=NG16

FILTER SELECTION

CHOSEN FILTER'S SPCS:
    MAX FLOW= 160.00 LT/MIN
    FILTERING SIZE=10.0 MIC

P.LOSS COMPUTATION

OIL VISCOSITY= .400E-04 M2/S

OIL DENSITY= 858.20 KG/M3

SUCTION LINE P.LOSS=   .03592 BARS
    SUCTION LINE P.LOSS COEFF.=   7600.

PRESSURE AT THE PUMP INLET=   .96577 BARS
    ATTENTION:NEGATIVE PRESSURE AT PUMP INLET

DELIVERY LINE P. LOSS=   .17705 BARS
    DELIVERY LINE P.LOSS COEFF.=   37404.

DC VALVE P.LOSS(P-A)=   .00000 BARS
    DC VALVE P.LOSS COEFF.(P-A)=   0.

LINE A-P.LOSS=   .00000 BARS

```
LINE B P.LOSS(RETURN)=      .00000  BARS
     LINE B P.LOSS COEFF.=          0.

DC VALVE P.LOSS(B-T)=     .00000  BARS
     DC VALVE P.LOSS COEFF.(B-T)=          0.

RETURN LINE P.LOSS=     .02973  BARS
     RETURN LINE P.LOSS COEFF.=      7432.

RET.LINE FILTER P.LOSS=      .26000  BARS
     RET.LINE FILTER P.LOSS COEFF.=     70000.

TOTAL PRESSURE LOSSES=      .43678  BARS
     TOTAL P.LOSS COEFF.=    103002.  BAR/(M3/S)2

THE PRESSURE ON THE PUMP= 134.14711  BARS


PR VALVE SET PRESSURE= 147.56182  BARS


POWER REQUIRED BY THE PUMP=  30.69723  KW



RECOMMENDED MINIMUM PRIME MOVER POWER=   33.8KW


RECOMMENDED MINIMUM RESERVOIR CAPACITY= 391.3 LITERS


OVERALL SYSTEM EFFICIENCY=78. PERCENT



.D IS COMPLETED



18.53.UCLP, BU, P33    ,        0.212KLNS.
```

APPENDIX C - DYNAMIC SIMULATION OF SYSTEMS DESIGNED IN APPENDIX B

The dynamic simulation of systems designed by HYSAN in APPENDIX B, were performed by HIDRAN program.

The cylinder system utilized a constant displacement pump, therefore the flow was constant. The disturbance was a step increase of cylinder load. The change in cylinder speed and load pressure are given as outputs in tabular and graphical forms.

For the motor system, the disturbances were in the form of a step increase in pump control current, and load torque. Outputs are taken both in tabular and graphical forms. It is seen that the system shows slight oscillations, before reaching the steady-state condition.

The same system was analysed by using a pressure compensated pump. Results of printer and plotter outputs are given.

It is seen that pressure compensation results in reduced speed, although pump control input was increased, and load pressure nearly remains constant, which means a constant torque transmission.

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JOB NO.= 1

TUATOR TYPE:CYLINDER

MULATION DUE TO PUMP-LOAD DYNAMICS

YSTEM WITHOUT PRESSURE COMPENSATION

** SYSTEM PARAMETERS **

MASS= 2000.000(KG)

VISCOUS FRICTION COEFF.= 1300.000(N/M-S)

NDER EFFECTIVE AREA= .003(M2)

CTIVE BULK MOD.OF DELIVERY SYSTEM=*********(N/M2)

. OIL VOLUME IN DELIVERY LINE= 1.667(LT)

SERVO TIME CONST = .300(S)

MP SERVO CONTROL GAIN= .010(M3/S-RD)

MP CONTROL CURRENT GAIN= .010(RD/A)

STEP SIZE= .010000 S
TERMINAL TIME= 1.00 S
NUM.OF ITERATIONS= 100

STEADY STATE SYSTEM PRESSURE= 30.000 BARS

PRESSURE RELIEF VALVE SET= 100.000 BARS

CHANGE IN CONTROL INPUT= .0000

FORCE DISTURBANCE=1000.00000 N

ARIATIONS IN SYSTEM VARIABLES DUE TO APPLIED INPUTS :

TIME(S)	SPEED(M/S)	PRES.(BARS)	FLOW(LT/MIN)
.010	-.005	.4775	0.
.020	-.008	1.7639	0.
.040	-.007	5.0895	0.
.060	.002	6.3079	0.
.080	.008	4.1093	0.
.100	.006	.2500	0.
.120	-.003	.3087	0.
.140	-.009	2.3568	0.
.160	-.005	5.7574	0.
.180	.004	5.3234	0.
.200	.008	3.0188	0.
.220	.003	.4578	0.
.240	-.005	.9473	0.
.260	-.009	3.9109	0.
.280	-.002	6.0652	0.
.300	.006	5.0567	0.
.320	.007	2.0328	0.
.340	.009	.3371	0.
.360	-.007	1.3132	0.
.380	-.007	4.3007	0.
.400	.001	6.0022	0.
.420	.007	4.1223	0.
.440	.006	1.2634	0.
.460	-.002	.5748	0.
.480	-.008	2.7842	0.
.500	-.004	5.4301	0.
.520	.004	5.6038	0.
.540	.008	3.1465	0.
.560	.003	.7382	0.
.580	-.005	1.1152	0.
.600	-.007	3.7343	0.
.620	-.002	5.7422	0.
.640	.006	4.9437	0.
.660	.007	2.2510	0.
.680	.001	.6429	0.
.700	-.006	1.3699	0.
.720	-.006	4.5493	0.

.760	.007	4.1216	0.
.780	.005	1.5388	0.
.800	-.002	.3207	0.
.820	-.007	2.7311	0.
.840	-.004	5.1402	0.
.860	.003	5.3982	0.
.880	.007	3.2497	0.
.900	.003	1.0830	0.
.920	-.004	1.2757	0.
.940	-.007	3.5860	0.
.960	-.002	5.4518	0.
.980	.005	4.3316	0.
1.000	.006	2.4379	0.

STEADY STATE DEVIATIONS DUE TO INPUTS:

STEADY STATE DEVIATION IN PISTON SPEED= .000 M/S

STEADY STATE DEVIATION IN CYLINDER PRESSURE= 3.208 BARS

STEADY STATE DEVIATION IN PUMP FLOW RATE= .000 LT/MIN

ENERGY LOSS= .000 JOULES

D OF DYNAMIC SIMULATION-PROGRAM STOP
59.54.UCLP, BU, P03 , 0.201KLNS.

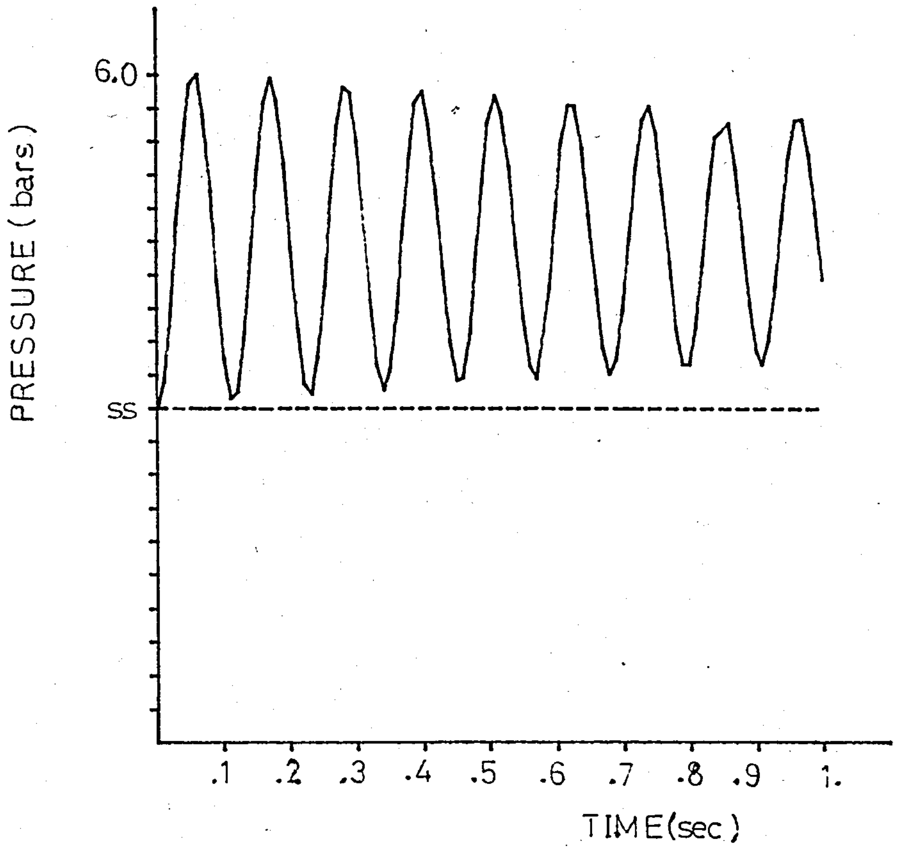

Fig.C1. Plotter output of pressure variation computed by HIDRAN

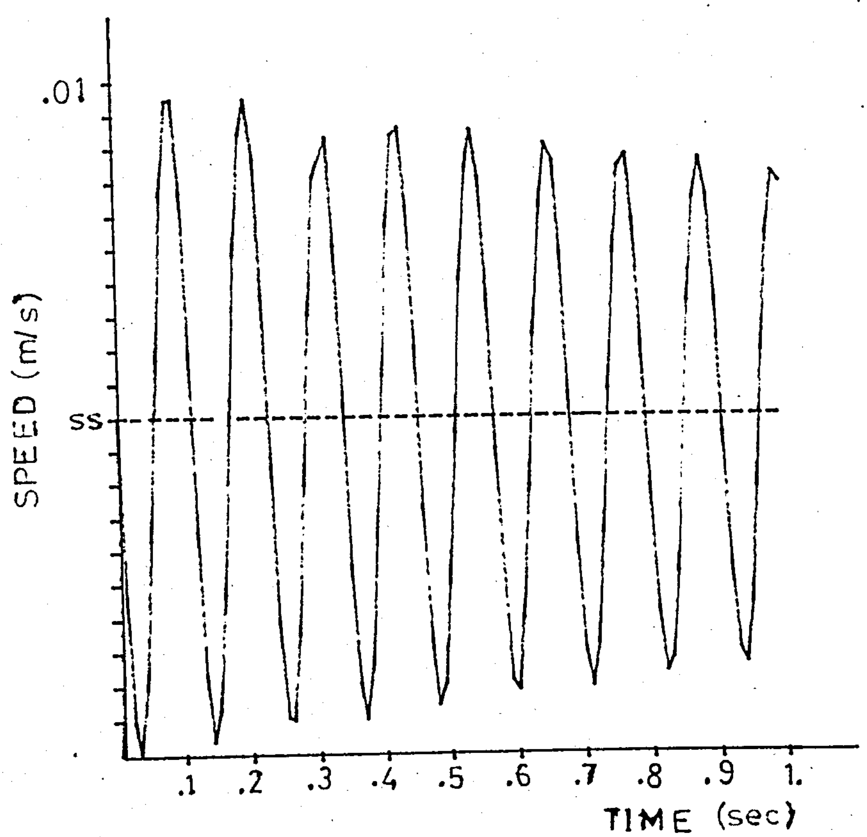

Fig.C2. Plotter output of speed variation computed by HIDRAN

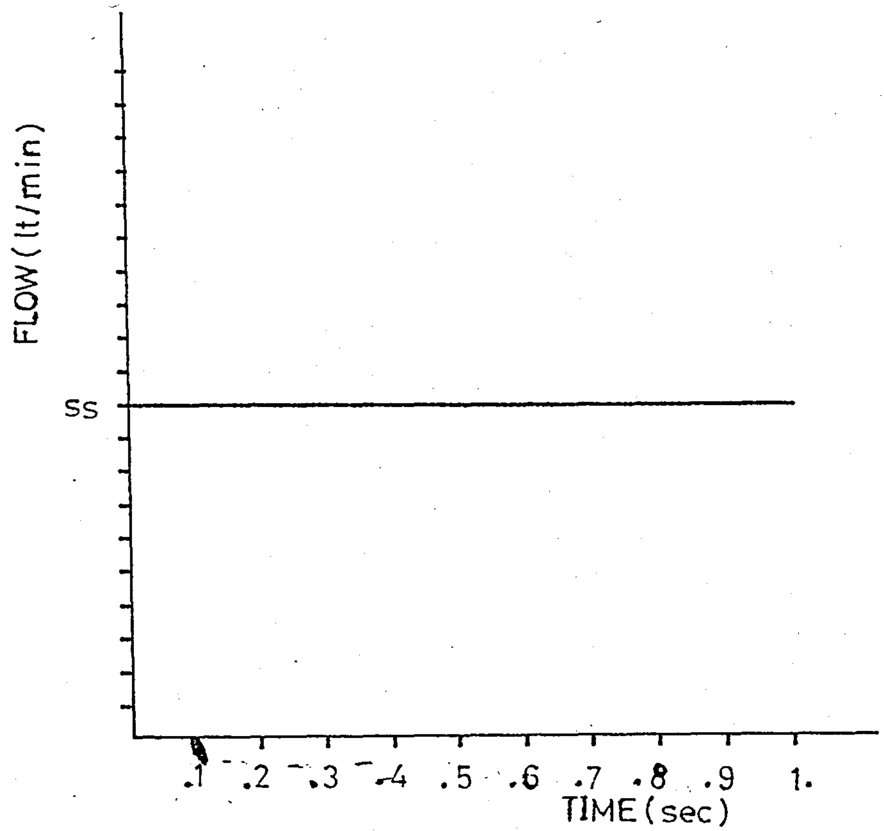

Fig.C3 Plotter output of flow variation computed by HIDRAN

The cylinder system analysed by HIDRAN is equipped with a constant displacement pump,therefore pump flow is constant. (Fig.C3)A load increase of 1000 newtons sets the system in oscillations.Load pressure shows an oscillatory increase of six bars maximum,(Fig.C1) while speed shows a slight oscillatory variation,since pump flow is constant.(Fig.C2)

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JOB NO.= 2

ACTUATOR TYPE:MOTOR

SYSTEM WITHOUT PRESSURE COMPENSATION

SYSTEM PARAMETERS

'OTAL INERTIA= 3.0075 (KG-M2)

'ISCOUS FRIC. COEFF.= 20.000 (N-M/RD-S)

IOTOR DISPLACEMENT= 80.000 (CM3/REV)

IOTOR MECH. EFFIC.= .94(&)

IOTOR VOL. EFFIC.= .92(&)

'OTAL OIL VOL. IN DELIVERY LINE= 2.4127 (LT)

IVERALL BULK MOD. OF DELIVERY SYSTEM= .16572E+10(N/M2)

'UMP SERVO TIME CONST.= .2000(S)

PUMP SERVO CONTROL GAIN= .10000E-01(M3/S-RD)

PUMP CONTROL CURRENT GAIN= .10000E-01(RD/A)

STEP SIZE= .010000 S
TERMINAL TIME= 2.00 S
NUM.OF ITERATIONS= 200

STEADY STATE SYSTEM PRESSURE= 133.000 BARS

PRESSURE RELIEF VALVE SET= 200.000 BARS

CHANGE IN CONTROL INPUT= .1000

TORQUE DISTURBANCE= 10.00000 N-M

VARIATIONS IN SYSTEM VARIABLES DUE TO APPLIED INPUTS :

TIME(S)	SPEED(RPM)	PRES.(BARS)	FLOW(LT/MIN)
.010	-.307	.0323	.293E-01
.040	-1.092	.4867	.109
.080	-1.804	1.7846	.198
.120	-2.134	3.6590	.271
.160	-2.109	5.8952	.330
.200	-1.778	8.3062	.379
.240	-1.199	10.7366	.419
.280	-.437	13.0637	.452
.320	.445	15.1974	.479
.360	1.388	17.0770	.501
.400	2.342	18.6684	.518
.440	3.264	19.9594	.533
.480	4.123	20.9558	.545
.520	4.895	21.6769	.555
.560	5.564	22.1516	.563
.600	6.126	22.4144	.570
.640	6.579	22.5027	.575
.680	6.928	22.4540	.579
.720	7.181	22.3040	.583
.760	7.350	22.0853	.586
.800	7.448	21.8263	.588
.840	7.486	21.5508	.590
.880	7.479	21.2776	.592
.920	7.438	21.0212	.593
.960	7.374	20.7915	.595
1.000	7.297	20.5947	.595
1.040	7.214	20.4337	.596
1.080	7.131	20.3089	.597
1.120	7.054	20.2187	.597
1.160	6.985	20.1597	.598
1.200	6.926	20.1280	.598
1.240	6.878	20.1189	.598
1.280	6.841	20.1277	.598
1.320	6.815	20.1498	.599
1.360	6.799	20.1810	.599
1.400	6.791	20.2175	.599
1.440	6.790	20.2562	.599
1.480	6.795	20.2944	.599

[illegible]	[illegible]	[illegible]	[illegible]
1.560	6.814	20.3632	.599
1.600	6.827	20.3912	.599
1.640	6.840	20.4143	.599
1.680	6.852	20.4325	.599
1.720	6.864	20.4458	.599
1.760	6.875	20.4548	.599
1.800	6.884	20.4600	.599
1.840	6.891	20.4621	.599
1.880	6.897	20.4616	.599
1.920	6.901	20.4591	.599
1.960	6.904	20.4553	.599
2.000	6.905	20.4507	.599

STEADY STATE DEVIATIONS DUE TO INPUTS:

STEADY STATE DEVIATION IN MOTOR SPEED= 6.900 RPM

STEADY STATE DEVIATION IN MOTOR PRESSURE= 20.430 BARS

STEADY STATE DEVIATION IN PUMP FLOW RATE= .600 LT/MIN

ENERGY LOSS= .000 JOULES

SYSTEM DAMPING RATIO= .5406

NOTE:UNDERDAMPED SYSTEM-OSCILLATIONS POSSIBLE

SYSTEM NATURAL FREQUENCY= .9789 HZ

FREQUENCY OF DAMPED OSCILLATIONS= .8235HZ

END OF DYNAMIC SIMULATION-PROGRAM STOP

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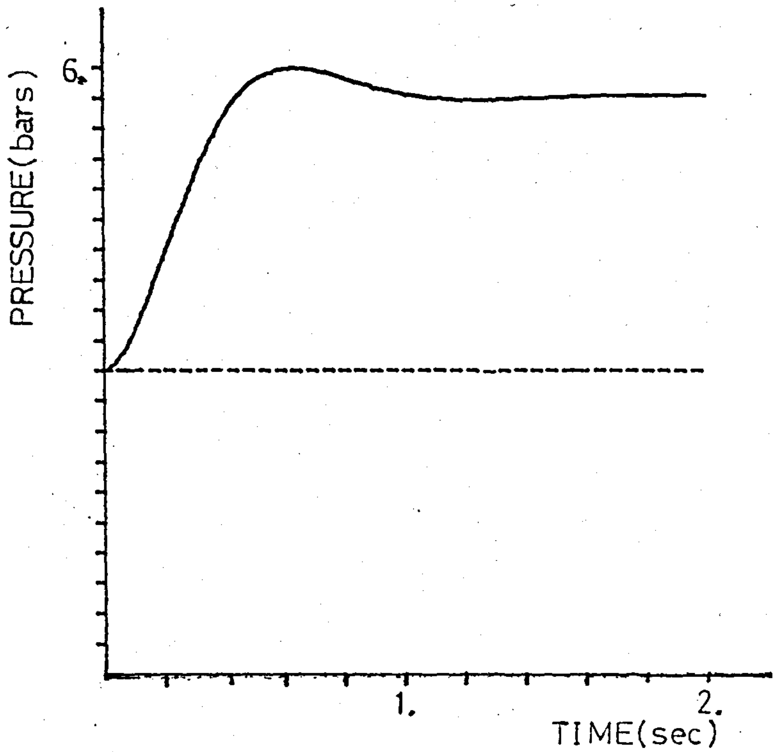

Fig.C4. Plotter output for pressure variation in motor system

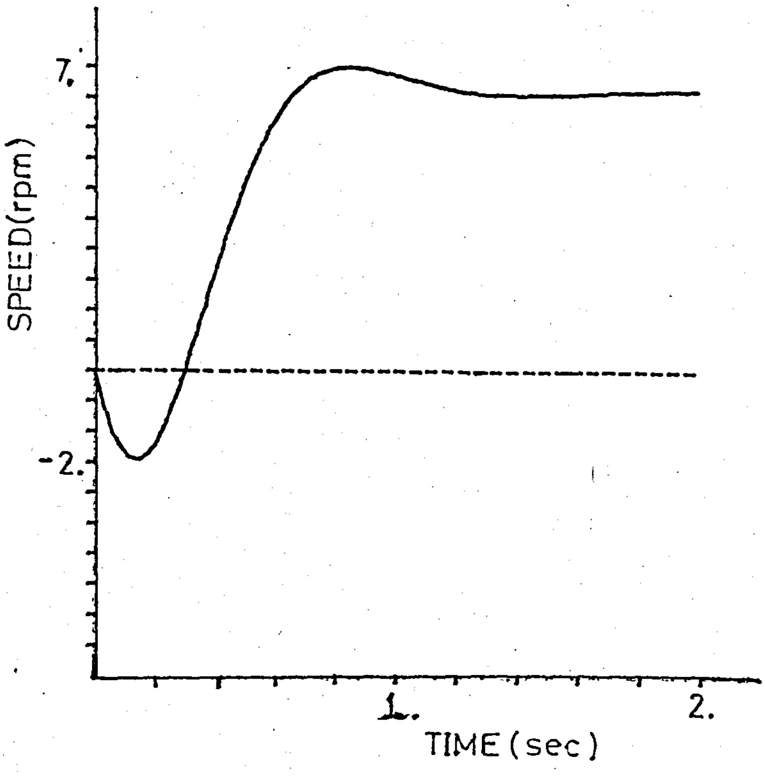

Fig.C5. Plotter output for speed variation in motor system

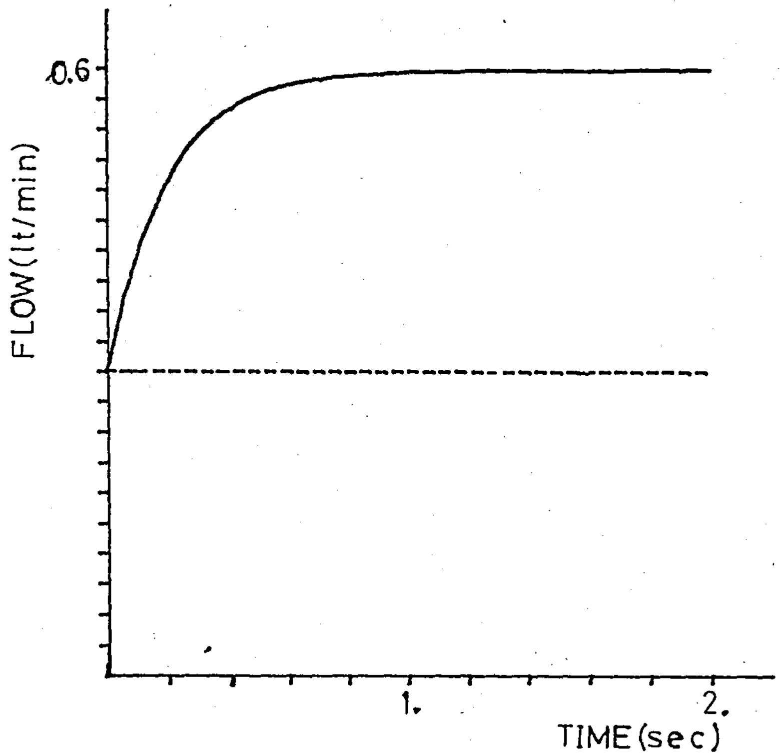

Fig.C6 Plotter output for flow variation in motor system

The motor system is analysed with respect to load change of
newton-meters, and a control input increase of 0.1Amp. The
ad pressure showed a slight oscillatory increase of six bars,
e to the load increase, while speed first decreased, due to the
mpressibility of delivery system, but later increased because
increasing pump flow. Pump flow increased by 0.6 lt/min, due
increased control input.

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JOB NO.= 3

ACTUATOR TYPE:MOTOR

SYSTEM WITH PRESSURE COMPENSATION

SYSTEM PARAMETERS

```
OTAL INERTIA=  3.0075      (KG-M2)

ISCOUS FRIC. COEFF.=  20.000     (N-M/RD-S)

OTOR DISPLACEMENT=  80.000     (CM3/REV)

OTOR MECH. EFFIC.=  .94(?)

OTOR VOL. EFFIC.=  .92(3)

OTAL OIL VOL. IN DELIVERY LINE=  2.4127     (LT)

VERALL BULK MOD. OF DELIVERY SYSTEM=  .16572E+10(N/M2)

[illegible]
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MP SERVO CONTROL GAIN= .10000E-01(M3/S-RD)

MP CONTROL CURRENT GAIN= .10000E-01(RD/A)

ESSURE COMPENSATION GAIN= .10000E-02(M3/S-A)

STEP SIZE= .010000 S
TERMINAL TIME= 2.00 S
NUM.OF ITERATIONS= 200

STEADY STATE SYSTEM PRESSURE= 133.000 BARS

PRESSURE RELIEF VALVE SET= 200.000 BARS

CHANGE IN CONTROL INPUT= 1.0000

TORQUE DISTURBANCE= 10.00000 N-M

ARIATIONS IN SYSTEM VARIABLES DUE TO APPLIED INPUTS :

TIME(S)	SPEED(RPM)	PRES.(BARS)	FLOW(LT/MIN)
.010	-.307	.0000	-.267E-01
.040	-1.115	.0000	-.970E-01
.080	-1.970	.0000	-.171
.120	-2.625	.0000	-.228
.160	-3.126	.0001	-.272
.200	-3.511	.0001	-.305
.240	-3.806	.0001	-.331
.280	-4.032	.0001	-.351
.320	-4.205	.0001	-.366
.360	-4.337	.0001	-.377
.400	-4.439	.0001	-.386
.440	-4.517	.0001	-.393
.480	-4.577	.0001	-.398
.520	-4.623	.0001	-.402
.560	-4.658	.0001	-.405
.600	-4.684	.0001	-.407
.640	-4.705	.0001	-.409
.680	-4.721	.0001	-.411
.720	-4.733	.0001	-.412
.760	-4.742	.0001	-.412
.800	-4.749	.0001	-.413
.840	-4.755	.0001	-.413
.880	-4.759	.0001	-.414
.920	-4.762	.0001	-.414
.960	-4.765	.0001	-.414
1.000	-4.766	.0001	-.414
1.040	-4.768	.0001	-.415
1.080	-4.769	.0001	-.415
1.120	-4.770	.0001	-.415
1.160	-4.770	.0001	-.415
1.200	-4.771	.0001	-.415
1.240	-4.771	.0001	-.415
1.280	-4.772	.0001	-.415
1.320	-4.772	.0001	-.415
1.360	-4.772	.0001	-.415

1.440	-4.772	.0001	-.415
1.480	-4.772	.0001	-.415
1.520	-4.772	.0001	-.415
1.560	-4.772	.0001	-.415
1.600	-4.772	.0001	-.415
1.640	-4.772	.0001	-.415
1.680	-4.773	.0001	-.415
1.720	-4.773	.0001	-.415
1.760	-4.773	.0001	-.415
1.800	-4.773	.0001	-.415
1.840	-4.773	.0001	-.415
1.880	-4.773	.0001	-.415
1.920	-4.773	.0001	-.415
1.960	-4.773	.0001	-.415
2.000	-4.773	.0001	-.415

STEADY STATE DEVIATIONS DUE TO INPUTS:

STEADY STATE DEVIATION IN MOTOR SPEED= -4.775 RPM

STEADY STATE DEVIATION IN MOTOR PRESSURE= .000 BARS

STEADY STATE DEVIATION IN PUMP FLOW RATE= -.415 LT/MIN

ENERGY LOSS= .000 JOULES

SYSTEM DAMPING RATIO= .0566

NOTE:UNDERDAMPED SYSTEM-OSCILLATIONS POSSIBLE

SYSTEM NATURAL FREQUENCY=********* HZ

END OF DYNAMIC SIMULATION-PROGRAM STOP
5.40.32.UCLP, EU, P03 , 0.210KLNS.

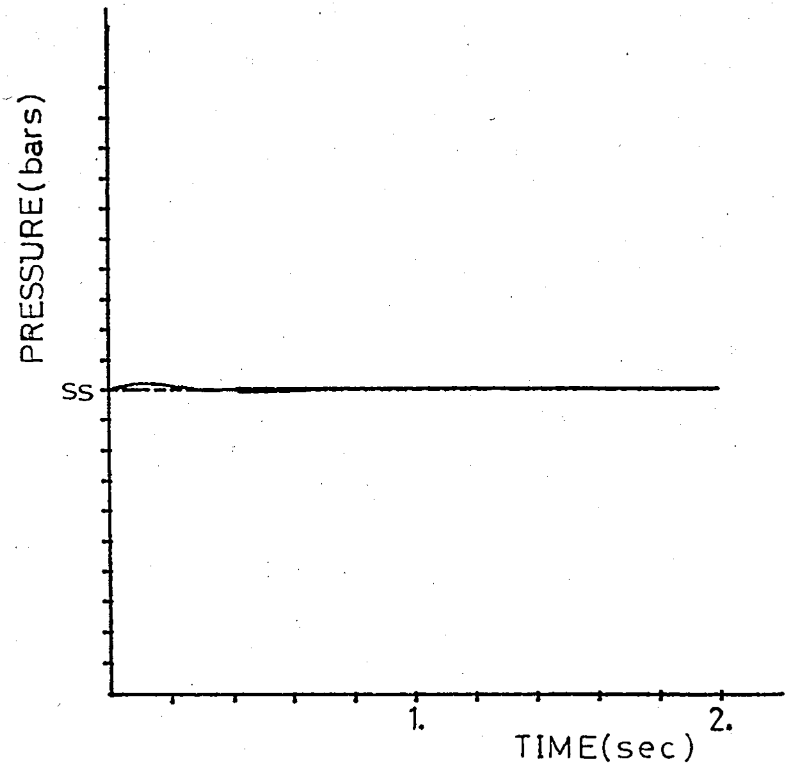

Fig.C7.

Plotter output of pressure variation in motor system with pressure compensation

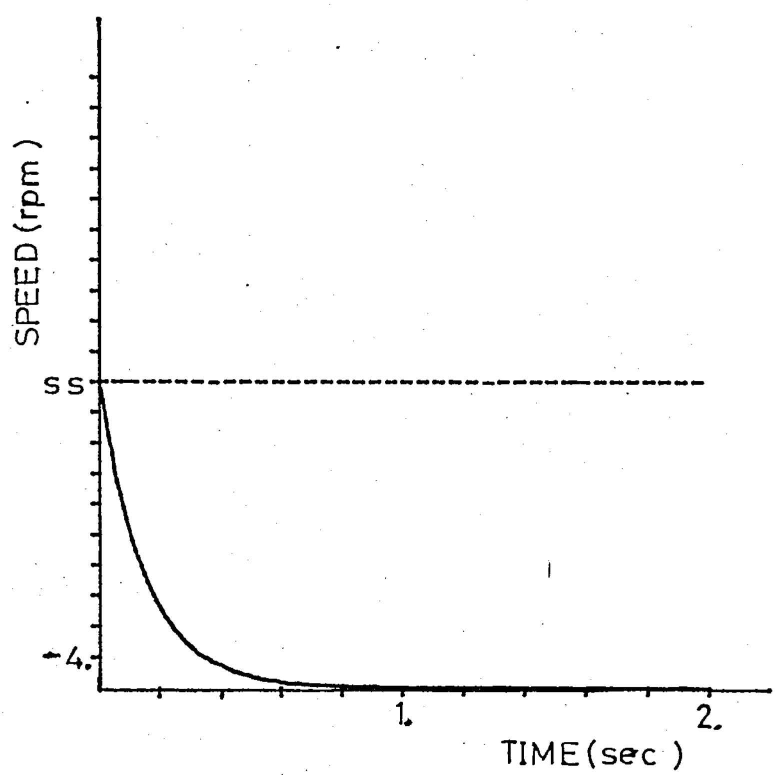

Fig.C8.

Plotter output of speed variation in motor system with pressure compensation

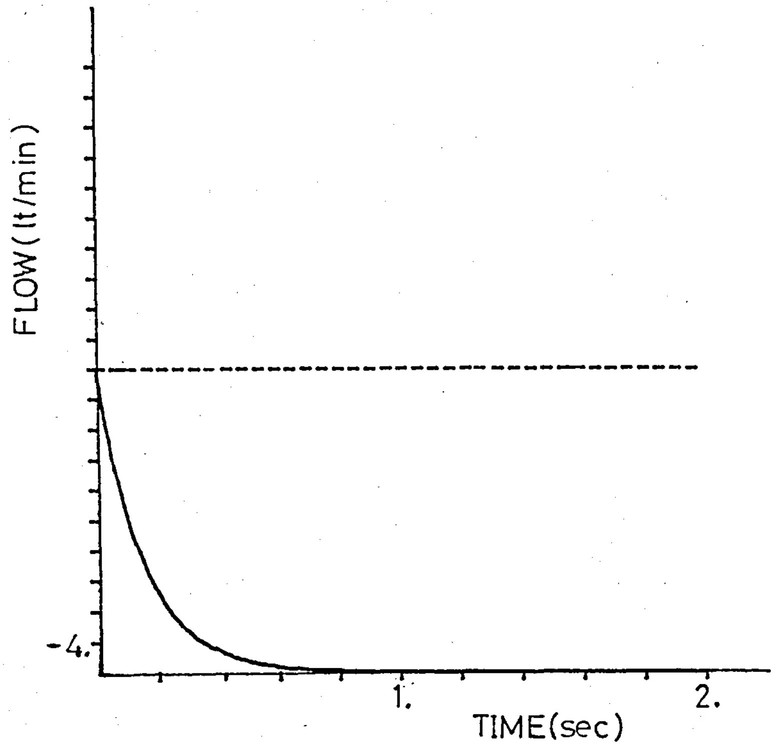

Fig.C9 Plotter output of flow variation in motor system with pressure compensation

The pressure compensated motor system is analysed for a control input of 1.0 Amp. and a load disturbance of ten newton meters. It may be concluded that, althaugh load disturbance was a high value, still system showed a relatively quick response, due to pressure compensation facility of the pump. Load pressure remains almost constant, as expected, which results in a constant torque transmission. (Fig.C7) Speed decreased since pump flow decreased rapidly because of the increasing pressure.

APPENDIX D - EVALUATION OF DIRECTIONAL CONTROL VALVE LOSS COEFFICIENTS

To evaluate the loss coefficients, we fit a third degree curve to the pressure loss versus flow rate curve of directional control valve.

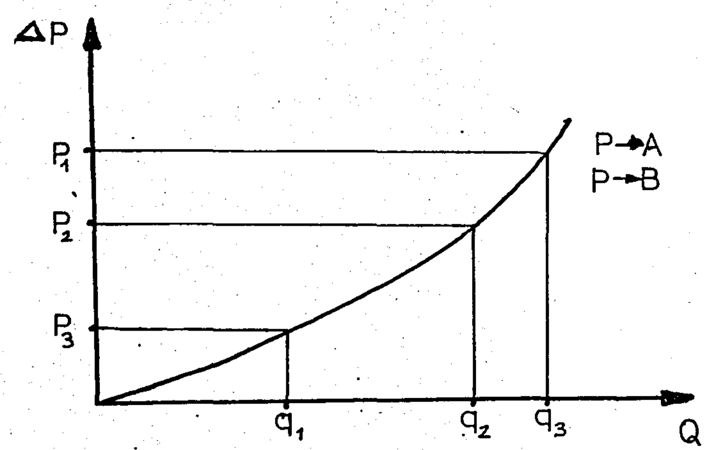

Fig. D1.

Following equations may be written for arbitrary taken points on the loss curve of directional control valve.

$$c_1q_1 + c_2q_1^2 + c_3q_1^3 = p_1$$

$$c_1q_2 + c_2q_2^2 + c_3q_2^3 = p_2$$

$$c_1q_3 + c_2q_3^2 + c_3q_3^3 = p_3$$

above linear system may be solved for c_1, c_2 and c_3 to obtain the valve coefficients.

These coefficients may be loaded in data files, for later use in design problems.

Same method applies for coolers and filters.

APPENDIX E - SOLUTION OF VECTOR MATRIX EQUATIONS

Considering the following matrix differential equation.

$$\dot{\vec{X}}(t) = \underline{\underline{A}}\ \vec{X}(t) + \underline{B}\ U(t) + \underline{C}\ M(t)$$

Subject to initial condition: $\vec{X}(t_o) = \vec{X}_o$.

Above equation may be written in the form of:

$$\dot{\vec{X}}(t) - \underline{\underline{A}}\vec{X}(t) = \underline{B}U(t) + \underline{C}M(t)$$

Multiply both sides by $e^{-\underline{\underline{A}}t}$ to obtain:

$$e^{-\underline{\underline{A}}t}\left[\dot{\vec{X}}(t) - \underline{\underline{A}}\vec{X}(t)\right] = e^{-\underline{\underline{A}}t}\left[\underline{B}U(t) + \underline{C}M(t)\right]$$

in other terms:

$$\frac{d}{dt}\left[e^{-\underline{\underline{A}}t}\ \vec{X}(t)\right] = e^{-\underline{\underline{A}}t}\left[\underline{B}U(t) + \underline{C}M(t)\right]$$

above differential equation may be integrated between $t_o = 0$ and t.

$$e^{-\underline{\underline{A}}t}\ \vec{X}(t) = e^{-\underline{\underline{A}}t}\int_o^t e^{\underline{\underline{A}}(t-\lambda)}\left[\underline{B}U(\lambda)+\underline{C}M(\lambda)\right]d\lambda + \vec{X}_o$$

with initial condition: $\vec{X}_o = \vec{X}(o)$

Above continuous-time system can be approximated by discrete-time solutions at equally spaced sampling instants. The input and disturbance functions are assumed constant during a sampling instant.

Letting $t_o = (k+1)T$ and $t_2 = kT$ we have:

$$\vec{X}((k+1)T) = e^{\underline{\underline{A}}(k+1)T}\,\vec{X}_o + e^{\underline{\underline{A}}(k+1)T}\int_o^{(k+1)T} e^{-\underline{\underline{A}}\lambda}\left[\underline{B}U(\lambda)+Cm(\lambda)\right]d\lambda$$

and

$$\vec{X}(kT) = e^{\underline{\underline{A}}kT}\,\vec{X}_o + e^{\underline{\underline{A}}kT}\int_o^{kT} e^{-\underline{\underline{A}}\lambda}\left[\underline{B}U(\lambda)+\underline{C}m(\lambda)\right]d\lambda$$

Multiplying above eq'n by $e^{\underline{\underline{A}}T}$ and subtructing from the equation above, we obtain:

$$\vec{X}((k+1)T) = e^{\underline{\underline{A}}T}\,\vec{X}(kT) + e^{\underline{\underline{A}}T}\int_o^{T} e^{-\underline{\underline{A}}t}\,\underline{B}\,U(kT)\,dt +$$

$$+\, e^{\underline{\underline{A}}t}\int_o^{T} e^{-\underline{\underline{A}}t}\,\underline{C}\,m(kT)\,dt$$

or

$$\vec{X}((k+1)T) = \underline{\underline{G}}(T)\,\vec{X}(kT) + \underline{\underline{H}}(T)\,U(kT) + \underline{\underline{Q}}(T)\,m(kT)$$

where

$$\underline{\underline{G}}(T) = e^{\underline{\underline{A}}T}$$

$$\underline{\underline{H}}(T) = e^{\underline{\underline{A}}T}\int_o^{T} e^{-\underline{\underline{A}}t}\,\underline{B}\,dt$$

$$\underline{\underline{Q}}(T) = e^{\underline{\underline{A}}T}\int_o^{T} e^{-\underline{\underline{A}}t}\,\underline{C}\,dt$$

Therefore the matrix equation is approximated by a vector matrix difference equation, convenient for computer solution.

APPENDIX F - EVALUATION OF PUMP CONTROL COEFFICIENTS KP, KQ AND KS

Following method can be used to evaluate control coefficients for a variable displacement axial piston pump, used in hydraulic systems.

To evaluate K_q and K_s, obtain control graph from manufacturer cathalogue. In the case of electrical control such a graph will look like:

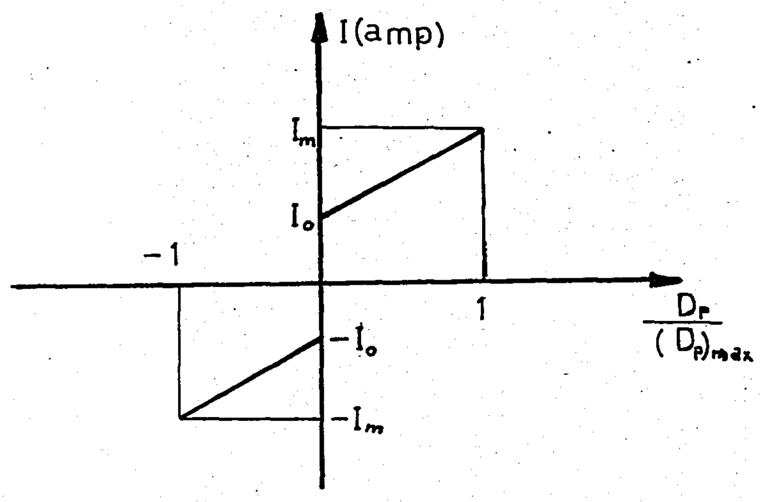

Fig.F1. Control graph for axial piston pump

From the figure: $U_{max} = I_{max} - I_o$, which is the maximum change in control input.

For a swash-plate design axial piston pump:

$$(Q_p)_{max} = K_q tg\Phi_{max}.$$

For a bent-axis design axial piston pump:

$$(Q_p)_{max} = K_q \sin\Phi_{max}$$

Therefore:

$$K_q = \frac{(Q_p)_{max}}{tg\Phi_{max}} \text{ for swash-plate design} \tag{1}$$

$$K_q = \frac{(Q_p)_{max}}{\sin\Phi_{max}} \text{ for bent-axis design} \tag{1^o}$$

Since

$$(Q_p)_{max} = K_s K_q U_{max}$$

for both designs, we can obtain

$$K_s = \frac{tg\Phi_{max}}{U_{max}} \text{ for swash-plate design} \tag{2}$$

$$K_s = \frac{\sin\Phi_{max}}{U_{max}} \text{ for bent-axis design} \tag{2^o}$$

Therefore equations (1), (1'), (2) and (2') give the control coefficients for axial piston pumps.

To obtain a reasonable pressure compensation gain K_p, for a damping ratio b, following analysis can be made:

$$w_o^2 = \frac{\beta_e}{JV}\left[D_m^2 \left(\frac{\mu_m}{\mu_v}\right) + K_q K_p B\right] \tag{3}$$

for a pressure compensated axial piston pump. Knowing the desired damping ratio, b, we can obtain required pressure compensation gain, K_p from:

$$b = \frac{1}{2w_o}\left[\frac{B}{J} + \frac{\beta_e}{V} K_p K_q\right] \tag{4}$$

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