STUDY OF AN OPEN CIRCUIT HYDRAULIC POWER SYSTEM WITH COMPACT COOLER-RESERVOIR UNIT

By

Ibrahim Subhi Al-Natour

B.Eng., M.Eng.

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To

My Parents

Wife Kinaz

Daughters Inas, Nuha, Dana

DECLARATION

I herby certify that this material, which I now submit for assessment on the programme of study leading to the award of Doctor of philosophy is entirely my own work and has not been taken from the work of others save and to extend that such work has been cited and acknowledged within the text of my work.

SIGNED————————————————— Date: December 1992

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ABSTRACT

Study of an Open Circuit Hydraulic Power System with Compact Cooler-Reservoir Unit

IBRAHIM SUBHI AL NATOUR BEng, MEng.

In this research, a complete open hydraulic drive mixer system has been designed, instrumented and commissioned, and an extensive programme of experimental tests has been undertaken to 1)- investigate the effectiveness of a cooling unit as an integral part of the open hydraulic system and 2)- validate the mathematical model. The results have shown that the working temperature could be reduced by 40 % by using the integral cooling/reservoir unit and the temperature is always kept below the recommended operation temperature.

A mathematical model for temperature distribution under unsteady state conditions in an open hydraulic systems has been developed to predict pipe wall and fluid temperatures in the system. The thermodynamics processes and heat transfer by convection, conduction and radiation have been taken into account. The developed temperature transient equations are solved by using numerical integration technique which are used widely in computer programming.

A software package has been developed to be used in hydraulic system design. The main advantage of this package is the user friendliness. The simulation results shows a significant difference between the temperatures of the fluid and the pipe wall in the hydraulic systems and demonstrated that this mathematical model is more accurate than those reported elsewhere.

The main results of this investigation is that the hydraulic reservoir has been reduced in the size to about 15 percent of the conventional reservoir in the open hydraulic systems. Furthermore, the experimental results have shown a close agreement with the theoretical results.

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Last but not least, the support and encouragement of my wife and family deserve greater acknowledgements than words can express

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Nomenclature

$\mathbf{A_1}$ inside surface area of hose [m^2]

$\mathbf{A_2}$ outside surface area of hose [m^2]

$\mathbf{A_{1hr}}$ reservoir base area [m^2]

$\mathbf{A_{2vr}}$ one side of fluid vertical area in the reservoir [m^2]

$\mathbf{A_{3vr}}$ another side of fluid vertical area in the reservoir [m^2]

$\mathbf{A_m}$ logarithmic mean surface area of hose [m^2]

$\mathbf{C_{pf}}$ specific heat of fluid [J/(kg.K)]

$\mathbf{C_{ph}}$ specific heat of hose wall [J/(kg.K]

D hose diameter [m]

f dimensionless friction factor

g acceleration of gravity [m/s^2]

$\mathbf{h_a}$ coefficient of heat transfer by natural convection [W/(m^2.K)]

$\mathbf{H_f}$, $\mathbf{H_{ff}}$ head loss [m]

k thermal conductivity of hose wall material [W/(m.K)]

$\mathbf{k_{res}}$ thermal conductivity of reservoir wall material [W/(m.K)]

K constant

L length of hose [m]

$\mathbf{m_1}$ mass flowrate of fluid into reservoir [kg/s.]

$\mathbf{m_2}$ mass flowrate of fluid outlet of reservoir [kg/s]

$\mathbf{M_f}$ mass of fluid in the system [kg]

$\mathbf{M_{fh}}$ mass of fluid within the hos section [kg]

$\mathbf{M_p}$ mass of wall hose material [kg]

$\mathbf{\Delta P}$ pressure drop [bar (10^5 N/m^2)]

Q flowrate [m^3/s]

$\mathbf{Q_1}$ heat flow transferred from fluid to hose wall by forced convection [W/(m.K)]

$\mathbf{Q_2}$ heat flow transferred from hose wall to surrounding atmosphere [W/(m.K)]

$\mathbf{Q_{1R}}$ heat flow transferred from fluid into its surrounding atmosphere at vertical wall of reservoir [W/(m.K)]

$\mathbf{Q_{2R}}$ heat flow transferred from fluid into its surrounding atmosphere at horizontal wall of reservoir [W/(m.K)]

$\mathbf{Q_{3R}}$ heat flow transferred from the surface of fluid in the reservoir to atmosphere [W/(m.K)]

$\mathbf{R_1}$,$\mathbf{R_2}$ hose inside and outside radius [m]

S thickness of wall of reservoir [m]

$\mathbf{T_1}$ temperature of the wall at inside hose surface [K]

$\mathbf{T_2}$ temperature of the wall at outside hose wall surface [K]

$\mathbf{T_a}$ constant surrounding atmosphere temperature [K]

$\mathbf{T_f}$ temperature of fluid [K]

$\mathbf{T_{res}}$ temperature of fluid in reservoir [K]

$\mathbf{T_{wrb}}$ outside wall temperature at bottom of reservoir [K]

$\mathbf{T_{wrv}}$ outside vertical wall temperature of reservoir [K]

W power losses converted into heat energy [Watts]

Greek symbols

ϵ emissivity

ν kinematic viscosity [m^2 /s]

σ surface tension [N/m]

CHAPTER: 1

INTRODUCTION

1.1 power hydraulics means using pressurized fluid in a confined system to accomplish work. Most hydraulic systems use petroleum oil, but often synthetic oil and water based fluid are used for safety reasons. A fluid power system accomplishes two main objectives. First, it provides substantial fluid force to move actuators in locations away from the power source where the two are connected by pipes, tubes, or hoses.

A power source, primarily, is an electric motor or diesel engine coupled to a hydraulic pump, which can be housed in one area to power a cylinder or hydraulic motor at a distance location. Secondly, fluid power systems accomplish highly accurate and precise movement of the actuator with relative ease, this is particularly important in such applications as in the machine tool industry where tolerances are often specified to one ten thousandth of mm and must be repeatable during several million cycles.

Industrial fluid power is a relatively young field of energy transmission and control. Modern hydraulic equipment can economically convert mechanical energy into fluid energy, and with simple components this energy may be regulated to provide direction, speed, and force control. No other type of power transmission provides the range of control of force, speed, and direction that is possible with fluid power transmission Many hydraulic systems seem exceedingly complex, however, their basic design is quite simple. Regardless of the complexity or simplicity of a hydraulic system, each system contains several basic components: (1) a reservoir to hold the fluid supply, (2) connecting lines to transmit the fluid power, (3) a pump to convert input power into fluid power, (4) a pressure control valve to regulate pressure, (5) a directional control valve to control the direction of fluid flow, (6) a flow control device to regulate speed or fluid flow, and (7) an actuator to convert hydraulic power into mechanical motion. Fig.(1) shows a simple hydraulic system which explains the above arrangement.

1.2. Circuit Classification: Any hydraulic system can be classified as open or closed circuits.

1.2.1 Open hydraulic circuit: In an open hydraulic circuit fixed displacement pumps are used. An open hydraulic circuit contains a pump or pumps supplied with liquid from a reservoir, usually at atmospheric pressure. The reservoir can be sealed and pressurized to minimize entry of foreign matter or to assist movement of fluid into the pump inlet. The discharge of the pump is directed through appropriate valves to the hydraulic cylinder or motor, thereby providing the desired linear or rotary force and motion, the returned fluid directed to the reservoir.

Ideally, a large reservoir is used in a conventional open circuit to allow air bubbles and foam to escape from the fluid and to assure that during peak pump demands the oil level will not drop below the suction line. The full flow demand of the pump is supplied through a suction pipeline which has to be of a large diameter in order to prevent cavitation at the pump inlet. The fluid reservoir also acts as a heat dissipator during periods of high heat generation in the system. Fig.(2) shows an example of a typical open hydraulic system.

1.2.2 Closed hydraulic circuit: The key element of the open hydraulic system is the reservoir of significant size where the spent fluid is returned prior to recycling through the pump. A closed hydraulic circuit usually consists of one variable displacement pump which can pump liquid in and out of each port according to the position of the control element and one hydraulic motor whose inlet and outlet ports are connected to the two ports of the pump. Fig. (3) shows an example of a closed circuit. There is always some designed hydrostatic transmission in a closed-loop configuration, therefore, a separate fluid supply has to be provided to make up the leakage, this is usually achieved by using a make-up pump to feed the low pressure side of the loop as shown if Fig. (4)

1.3 Heat Generation in a Hydraulic System: In any hydraulic system it is desirable to maintain the fluid temperature at, or preferably below the recommended or specified maximum working temperature for continuous duty.

A hydraulic system that is allowed to overheat can cause costly seal deterioration and fluid oxidation or breakdown. This results in corrosion and formation of sludge and varnish, which may in turn clog orifices and accelerate valve wear and tear. In some cases, extreme temperatures will cause seizure of valves, pumps, and other components. In addition, oil viscosity will decrease and system operation will probably become erratic. However, it is acknowledged that the ideal operational temperature of industrial oil hydraulic system should not exceed 65 °C, but the recommended maximum operating temperature is below 50 °C because above that level the life of most fluids is shortened.

Fig (5) according to Ref (1) shows the effects of hot oil on system performance.

To design a hydraulic system that will maintain thermal stability, it is necessary to understand how hydraulic systems generate and dissipate heat. Heat is generated in hydraulic system wherever oil flows from higher to lower pressure without doing mechanical work. This means that if a relief valve, for example, is allowing the oil to flow back to the reservoir, and the system pressure is being maintained, the difference in pressure or loss is the difference between the system pressure and the reservoir line pressure.

1.4 Background Literature of Temperature Analysis in Hydraulic Systems:

Thermal transients in a hydraulic system are a fundamental aspect of system performance and generally are concerned with thermodynamic processes which are associated with the power losses converted into heat and also, associated with heat flow transfer and losses during the system operation. Initial studies (2) of temperature analysis in hydraulic systems were carried out on a hydrostatic winch drive where the temperature effects were considered to be vitally important. A cooler had been installed in this system and its thermal characteristics were taken into account, but for simplicity the heat transfer between the fluid and the pipe wall was ignored. However, most conventional studies for temperature change in the hydraulic systems deal with steady state conditions like those given in references (3) to (9) in which heat flows continuously at a uniform rate and unsteady processes were ignored due to their complexities. The transient temperatures in the hydraulic systems have been studied by Ezekiel and Paynter (10) and by those given in Refs. (11) to (14). However, these researchers did not consider heat transfer effects even those associated with coolers.

The following assumptions were usually made in these previous studies:

(1) the system inside and outside pipe wall temperatures are the same as the fluid temperature, and (2) the heat generated by the pressure losses due to pipe friction was negligible . Yang (15,16), Yang and Bowns (17) and Yang (18) have studied the temperature analysis in hydraulic system and they considered the hydraulic system as a closed thermal system and they obtained their experimental results using a closed or partially closed hydrostatic drive propeller system. However, in their study, the heat transfer by conduction along the pipes and reservoir in the system and heat transfer by radiation from the surfaces of reservoir were ignored, which could give rise to error in estimating the temperature distribution in hydraulic systems.

1.5 Background Literature of Digital Simulation in Hydraulic System:

Digital computer simulation technique have the advantage that they can be run on a wide variety of computers and can cope with all kinds of non linearities. However, mathematical modelling and computer programming are time consuming and expensive procedures. In order that the engineer makes the use of digital simulation, a number of general purpose simulation languages and specialist application packages have been developed by others, for example, HASP(hydraulic Automatic Simulation Package (19), ACSL(Advanced Continuous Simulation Language (20), CSMP(Continuous System Modelling Program (21),(22), DSH(Digital Simulation of Hydraulics (23),(24), HOPSAN (25), and GHPS (General Purpose Hydraulic System Simulation Language (26).

It is recognised that the digital computer has the potential to be an important tool in the simulation of physical systems. It is also recognised that in order to utilise this tool to its full advantage, it is desirable that general purpose simulation languages be available which allow the digital computer to be programmed in the same way as an analogue computer. For instance, CSMP provides elements which simulate simple integration, first and second order system behaviour and many non linearities. Not least it provides for the inclusion of user written subroutines.

General purpose simulation languages also take into account the inherent difficulties of programming and provide sorting procedures which allow the statements defining the dynamic or static behaviour of each component to be inserted into the programme in any order. However, if these general purpose simulation packages are to be used as tools, the user has to have programming skill. Although there are those that offer some form of block representation for simple mathematical elements and transfer functions, this facility is usually limited and it is common that the user is required to break down his physical systems, and in fact the user works closely to the mathematical environment of the problem and has to be involved with detailed considerations.

Another example is HYTRAN (Hydraulic Transient Analysis),(27) which is one of a group of programmes issued by the MacDonnel Douglas Aircraft Corporation. The other programmes issued are SSFAN(Steady State Flow Analysis)(28), HYTTHA(Hydraulic Transient Thermal Analysis)(29), and HSFR(Hydraulic System Frequency Response). SSFAN calculates the steady state pressure and flows throughout a system under any loading conditions. An iterative procedure is used whereby flows are varied to

obtain a pressure balance throughout the whole system. HSFR calculates the system frequency response using a form of the impedance method.
HYTRAN was developed for the dynamic analysis of aircraft hydraulic systems. This programme allows for distributed parameter models and uses the method of characteristics (30).

Within HYTRAN the programme SSFAN is used to set up initial conditions and the method of specified intervals is used with the method of characteristics and the solution requires both iteration and interpolation. The dynamic models of different systems can be constructed from a model subroutine library and are solved using an explicit fourth order Runge-Kutta (31) integration algorithm. However, the user is required to supply the coding to establish a simulation for a hydraulic circuit and hence it is very difficult for the engineer to use or understand it.

1.6 Need for Improving Performance of Hydraulic System:

In spite of the fact that the industrial fluid power has been established for many years, hydraulic transmission is under constant pressure to improve its performance to face up to the challenges from mechanical and electrical transmission system. This pressure has increased greatly in the last twenty years, basically due to the rapid developments in electrical engineering. A new generation of electric motors has been developed which has a wider range of speed and torque output to suit the load conditions. The competitive features of transmission systems have been examined from points of view which include the system reliability, efficiency, capacity and cost. It is therefore necessary to increase pressure, speeds, capacity, transient response with high reliability and low noise if the industrial fluid power is to keep its position.

In many hydraulic system applications, it is important to reduce the size and energy requirements, especially in mobile applications. For instance, Fig (6) demonstrates a simulated open hydraulic system used for lowering and raising the nose wheel of an aircraft landing gear. In this case the size of the hydraulic reservoir is very important and could be vital when considering the feasibility of a particular system design. Another example illustrated in Fig. (7) which shows a closed hydraulic transmission system applying a two-man operation inside the submarine for the control of the steering and stabilizing system, bow thruster, grappling hook, net winches, mooring winches, and another windlasses.

In both of the above examples, since the hydraulic reservoir in the conventional open hydraulic system requires a large space and the energy consuming boost pump in the closed hydraulic system requires an additional driving power, neither the traditional open hydraulic system nor the closed hydraulic system may be the best design choice under such special working conditions.

1.7 The Present Research and its Objectives:

Primarily, this work is concerned with developing and establishing a theoretical model for temperature analysis under unsteady state conditions in hydraulic system with consideration of the effects of heat transfer by convection, radiation and conduction. This analytical model based on thermodynamic and heat transfer principle has been developed to investigate in detail the temperature distribution in hydraulic systems and to assess the performance of any hydraulic system in order to optimise the processes.

In order to test this model experimentally, a complete open hydraulic mixer system has been designed, instrumented and commissioned.

This system is a combination of compact size of reservoir, control system, cooler arrangement and associated equipment forming the heart of the hydraulic circuit. The hydraulic oil is fed from the pump to the circuit and on to the drive motor, at this point a small amount of the oil flow is used to drive the cooling fan motor. The unit also includes a system control valve which engages the drive motor or allows the oil to recirculate through the cooler, and an adjustable relief valve to protect the hydraulic pump, motor and pipework. When the drive system is on, the returned oil from the motor is fed through the radiator and cooled by the airblast from the fan, then through the filter back into reservoir to be re-used.

In this system the hydraulic oil is kept below its maximum operating temperature no matter how long or hard the system is used within its specification, and the need for the extra cost and capacity of the oil is eliminated.

Usually, a large capacity oil reservoir is fitted in a hydraulic system to slow down the heating effect on the oil. This method is effective for short periods of use, but the oil still overheats if the system is used for long periods or with inefficient pumps or motors. There is also the problem in finding space for the large oil reservoir and the expense of purchasing and carrying the weight of this oil.

The objectives of the present research are:

1- To investigate and develop a mathematical model for temperature distribution in hydraulic systems.

2- To formulate theoretical model to predict effectiveness of the unit.

3- To develop a simulation program to give accurate prediction of the temperature distribution under unsteady state condition in the hydraulic systems in conditions where normal mathematical techniques would insufficient, with the aim of allowing the simulation program for temperature to be used in other more complex hydraulic systems.

4- To study the performance of any existing hydraulic cooling/reservoir system.

5- To develop and modify the design in order to achieve low cost and/ or improved performance (cooling capacity, performance of the hydraulic system in which the unit is used).

6- To investigate the feasibility of providing an inexpensive but efficient oil cooler to small sized hydraulic circuits which will reduce heat problems, reservoir size and will be an integral part of the circuit, ie it will be powered from the circuit flow and pressure.

7- To assess the temperature distribution in a hydraulic system by simulated results and to correlate theoretical and experimental observed results.

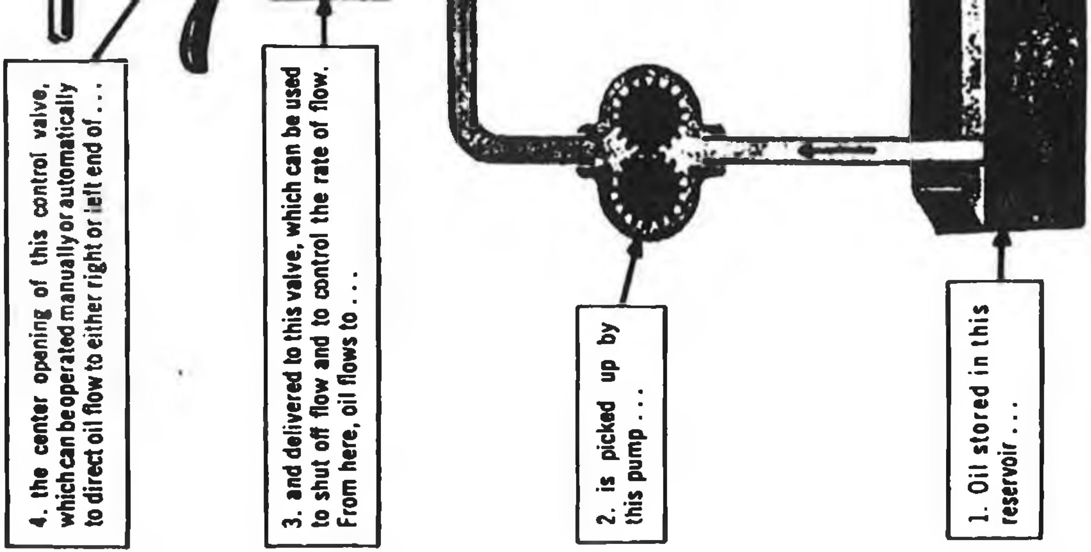
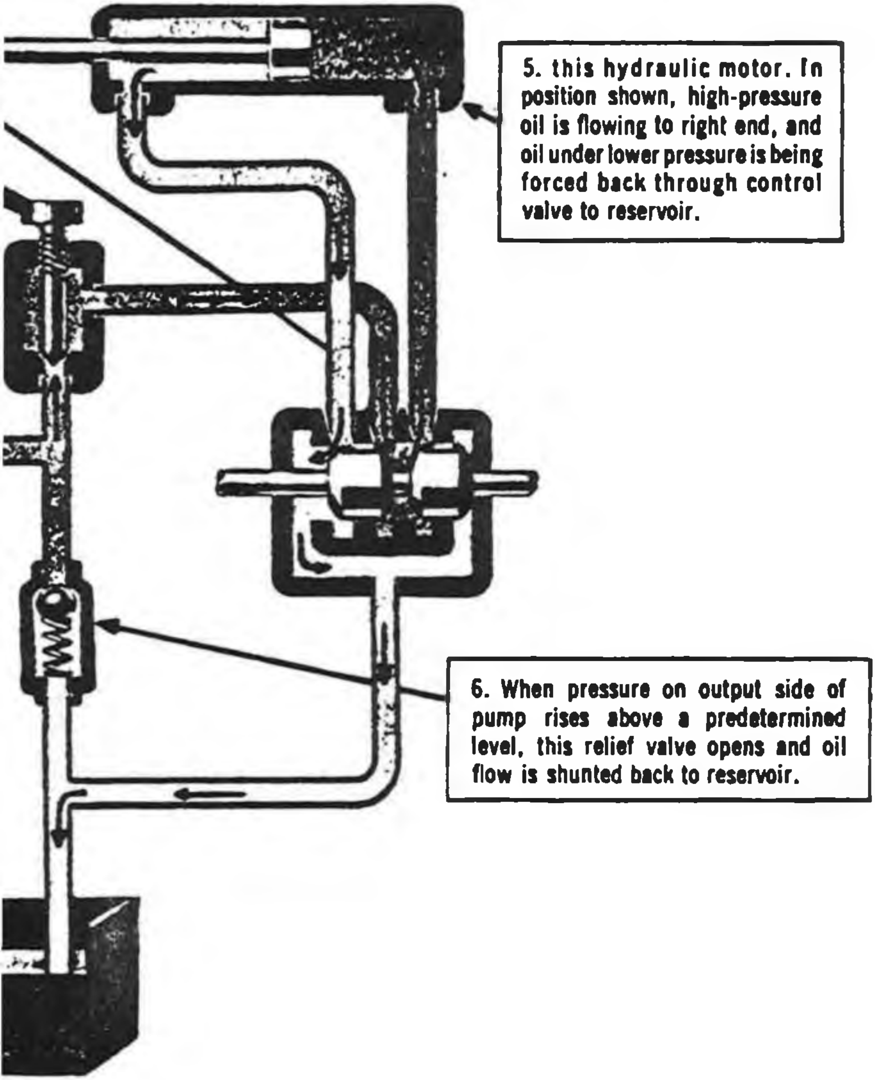


Fig. (1) Simple hydraulic system.


5. this hydraulic motor. In position shown, high-pressure oil is flowing to right end, and oil under lower pressure is being forced back through control valve to reservoir.
6. When pressure on output side of pump rises above a predetermined level, this relief valve opens and oil flow is shunted back to reservoir.

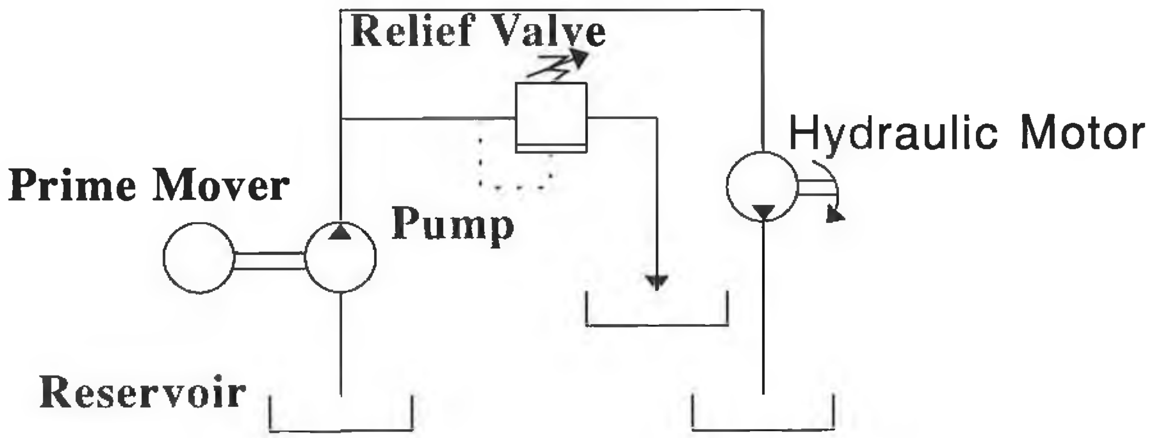


Fig. (2) An example of a typical open hydraulic system.

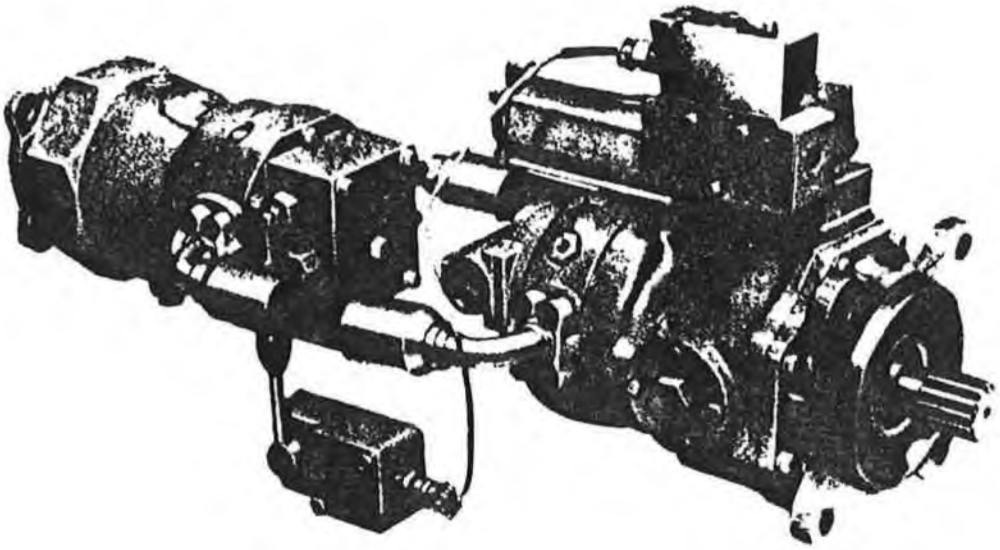


Fig. (3) An example of a typical closed hydraulic circuit.

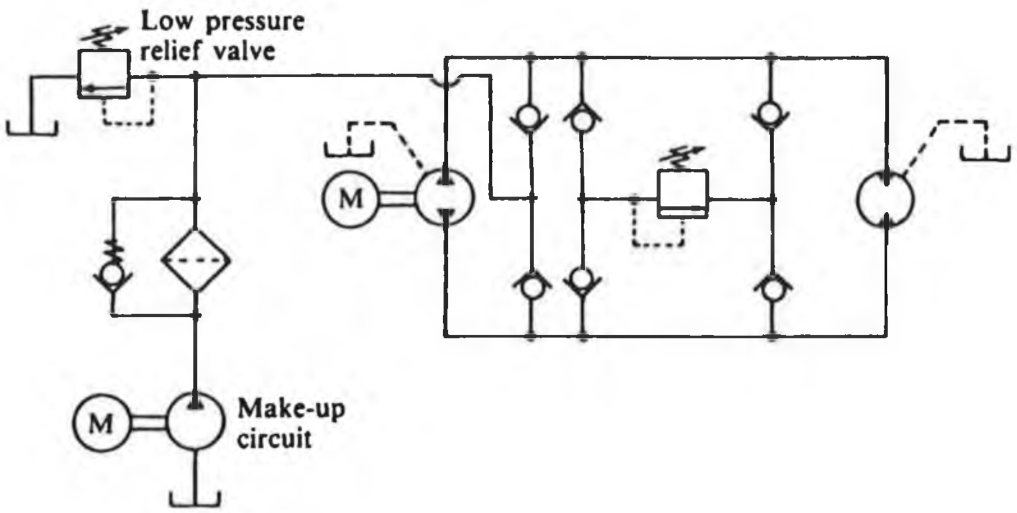


Fig. (4) A closed loop hydraulic transmission with make-up pump.

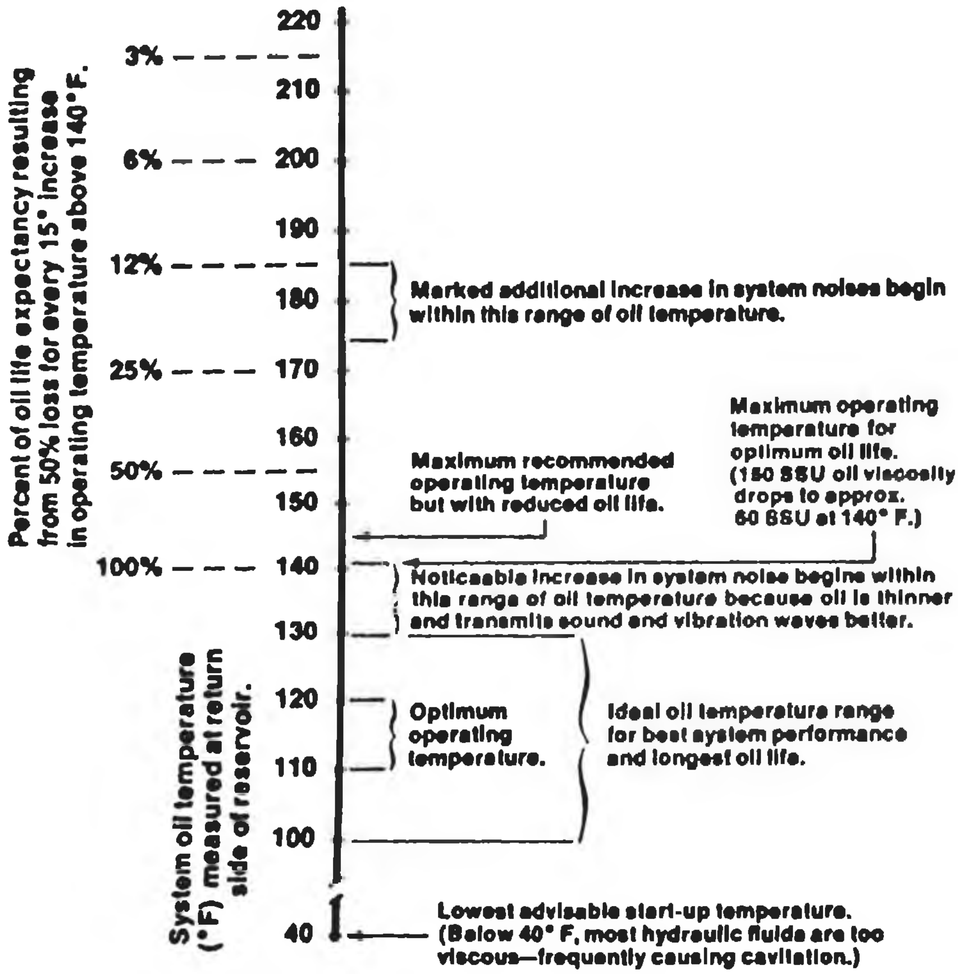


Fig. (5) The effects of hot oil on hydraulic system performance.

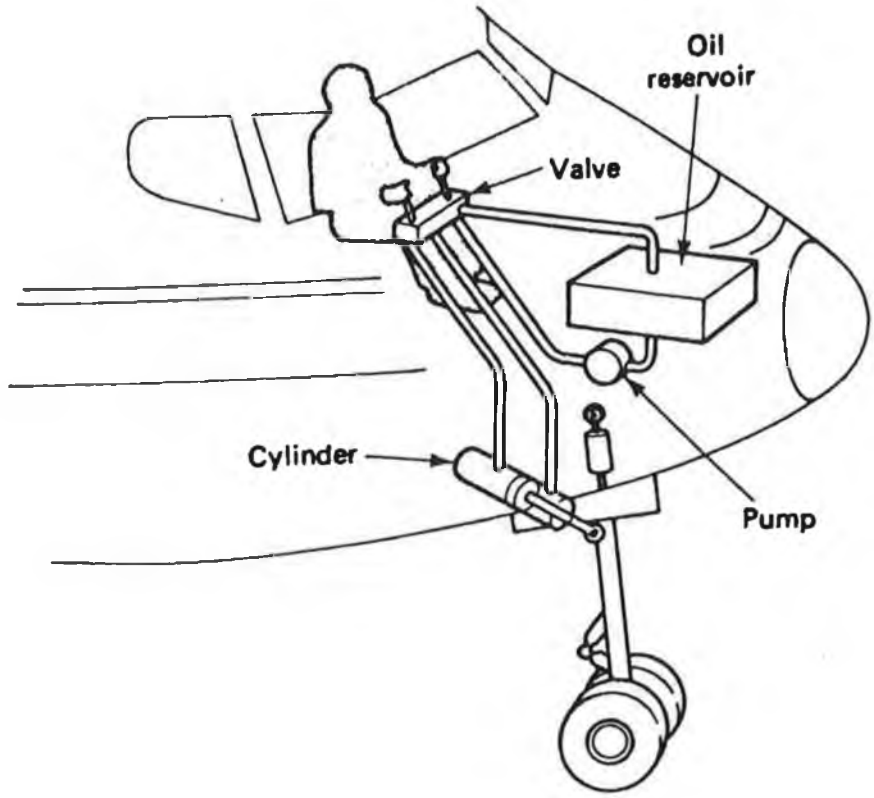


Fig (6) An application of an open simulated hydraulic system used for lowering and raising the nose wheel of an aircraft landing gear.

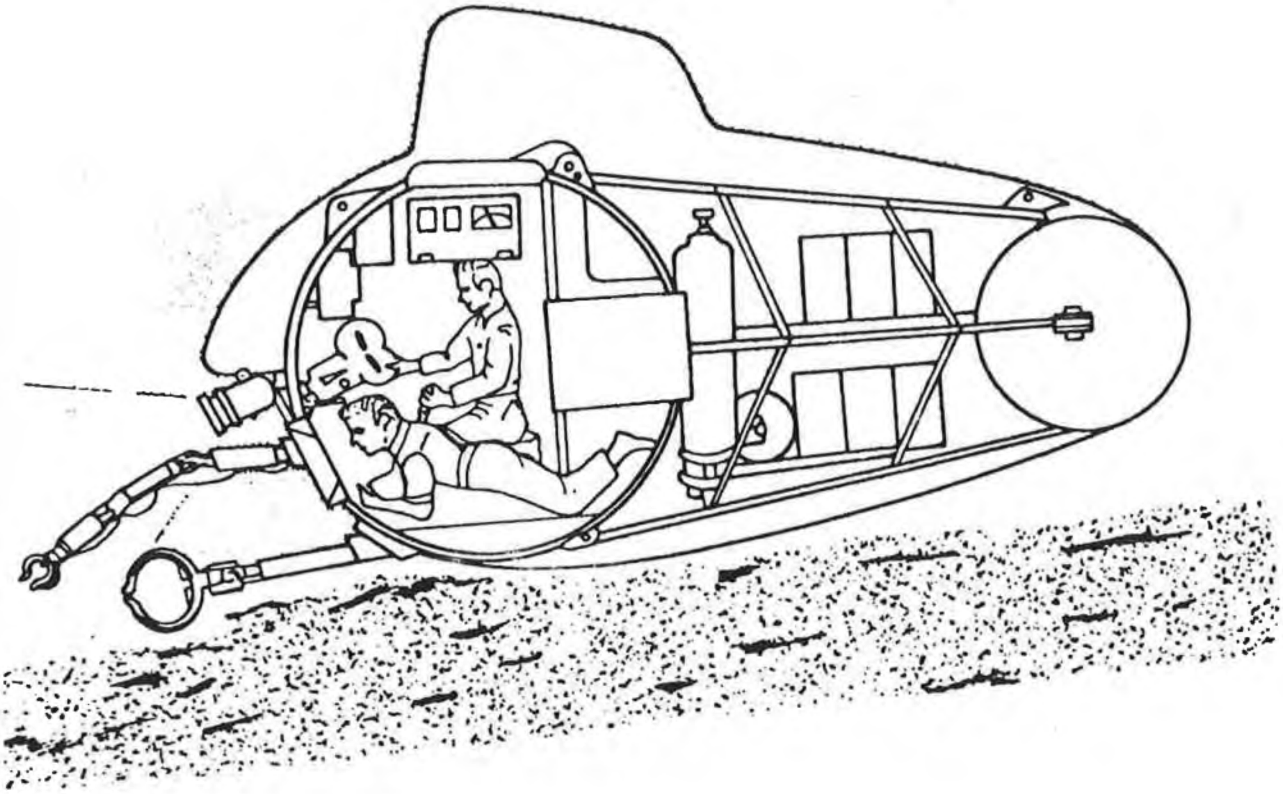


Fig (7) An application of a closed hydraulic transmission system to an undersea submarine research vehicle.

CHAPTER: 2

Design and Commissioning of the Experimental Equipment

2.1. Introduction

In order to test the developed mathematical model, a complete open hydraulic mixer system has been designed, instrumented and commissioned in the laboratory of SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING IN DUBLIN CITY UNIVERSITY.

2.2. Experimental Test Rig Components:

Fig (8) and Plate (1) show the selected test rig components required for an open hydraulic drive mixer system which consists of an electric motor, a hydraulic pump, pipelines, control valve, a solenoid directional control valve, a relief valve, a compact sized of hydraulic reservoir, a cooler arrangement and a hydraulic motor coupled with the mechanism of the mixer.

The selection of a gear pump without external drain was made to provide an unit most suitable for test work rather than for any particular mixer hydraulic system application. An external gear motor with an external drain was selected to demonstrate the open circuit design, which is capable of coping with oil circulation in the system due to components flow leakage. The main criterion for the choice of the pump or the hydraulic motor was that they should be capable of easy mounting and / or mechanically as simple as possible.

The choice of the four way three position solenoid directionals control valve was made to allow the reverse of the mixer movement and to unload the pump at the valve central position. All the pipes in the system were chosen as flexible hydraulic hoses to allow the flexibility of arranging installation for the hydraulic transmission system components, and also to mount the mixer at the most convenient operating position.

An adjustable relief valve was selected to limit the maximum system pressure and to protect the hydraulic pump, motor and pipework when the drive is on.

If this system is to be used in mobile or industrial application, it is essential that the hydraulic transmission system should have as small a requirement as possible of the power supply and also the installation space. Therefore, a cooler was selected. The choice of the cooler to be included in this system was made to reduce the size of reservoir in the conventional open system and, to give the opportunity for greater mobility if needed. As a comparison, under the same condition where the electric motor is operated at nominal speed of 1500 r.p.m, the reservoir size in a conventional open system is normally chosen as 6 to 7 times of the system flowrate. This means that a 48 to 56 litres reservoir has to be used.

2.3. Instrumentation of the Experimental Test Rig:

2.3.1. Hydraulic Pump A fixed positive displacement external gear pump has been selected in this system with maximum flowrate of 36 l/min and volume displacement of 24 cm^3/rev. Technical data of this pump is summarised in Table (1). Fig (9) and Plate (2) show an external gear pump with positive displacement in which the fluid is transferred at a constant amount for each cycle of operation. Since the internal pump volume can not be adjusted, the pump is considered to have a fixed displacement.The pumping action occurs in the gear pump when the input drive causes one gear to turn the other. This action, in turn, causes fluid to be displaced from the inlet port to the outlet port in the following manner. The gear teeth seal as one rotates the other. As the teeth part on the suction side of the pump near the low pressure inlet port, they increase the volume of the inlet chamber causing a slight vacuum. The rotating gear teeth transfer fluid, forced in by atmospheric pressure and trapped in the gear teeth, around the outside periphery of the gears to the high pressure outlet chamber. Meshing of the teeth in the outlet chamber reduces the cavity volume by an amount equal to that displaced between the teeth as they mesh. This forces fluid from the outlet cavity and port at system pressure.

2.3.2. Hydraulic Motor: A fixed positive displacement external gear motor with external drain has been selected in this system. This motor has a maximum flowrate of 54 l/min and volume displacement of 36 Cm^3/rev. This external gear positive displacement motor operates in the reverse manner of its pump counterparts. This motor differs from the pump in other respects. Because the case is pressurized from an outside source, an external drain line is provided to protect shaft seals. This drain line is piped directly to the low pressure reservoir as shown in Fig (8) and Plate (8). technical data of the hydraulic motor is given in Table (2). Plate (3) shows a view of the hydraulic motor.

2.3.3. Valves: Hydraulic valves provide the interface between the hydraulic fluid, the control signal and the hydraulic actuators. They are used to control the flowrate, the direction of flow and the pressure of fluid. For instance, the relief valve is used in the present system to first, limit the maximum system pressure which, in turn, protects the system components, piping; and second, limit the maximum output force of the hydraulic system.

All valves, except the directional control valve, which are used in this system are cartridge insert valve type. The advantages of such valves are obvious, because if a control function becomes faulty for some reason, it is a very simple matter to replace it without disturbing any pipework. There is also a saving in the amount of interconnecting pipework. Fig (10) and Plate (8)shows the cartridge insert relief valve which was used in the rig test. All the cartridge insert valves were mounted in valve blocks with screwed port for pipe mounting as shown in Fig (11) and Plate (8). In this case the insert can be replaced without disturbing the fluid connections. Tables (3) to (6) give the technical data of pressure relief valve, flow control valve, check valve and directional control valve respectively.

2.3.4. Pipelines: All pipes in the system were chosen as flexible single wire braid hydraulic hoses, as shown in Fig (12) and Plate (3), which allow the flexibility of arranging installations for the hydraulic transmission system components. Tables (8), (9) and (10) give the technical data of the hose of pressure line, return line and suction line.

2.3.5. Reservoir: A compact size stainless steel reservoir of dimensions (300 mm X 290 mm X 105 mm) has been used in this system. Figure (13) and plate (4) show a view of the reservoir which contains a fluid level and temperature gauge in the form of a transparent 'window', a filler cap which includes a strainer to prevent foreign matter entering when oil is being replenished and a breather to allow movement of air through a filter so that dust is not drawn in as the fluid level fluctuates. The pump intake is fitted with a strainer and the return line is fitted with filter 40 micron filtration with condition indicator, 2.0 bar bypass when filter element is blocked. Table (11) illustrates technical data of the reservoir.

2.3.6.Cooler: A 25 row efficient radiator with a maximum capacity of 15 KW dissipation of heat has been installed in the circuit. Table (12) gives the data of cooler. Plate (5) shows a view of the cooler and Plate (4) shows a view of the cooling unit.

2.4. Instrumentation and Measuring Equipment:

The hose wall temperatures were measured by four thermocouples with a digital output as shown in Plate (6). These thermocouples were firmly stuck onto the surface of hoses. Fig (14) shows the location of four thermocouples. Fluid temperature in the reservoir was measured with a dial mounted gauge driven by a temperature sensitive fluid that is metal encased and immersed in the hydraulic fluid. The fluid level in the reservoir is monitored by a fluid level gauge which was mounted in the reservoir to indicate the level and the amount of make-up fluid necessary to fill the hydraulic system. The temperature and fluid level gauges are illustrated in Plate (4).

The temperatures of the fluid flowing through the hoses were measured by two temperature meters which have been incorporated in the lines of the hydraulic system to record the change of the fluid temperature during the system operation. Plate (7) shows the inline temperature meters.

The pressure is measured using two pressure gauges to check the conditions of the component and the pressure value of the load resistance. These gauges are filled with a dampening fluid such as glycerine to smooth pulsations that might cause damage to the sensitive gear mechanism. Fig (15) and Plate (7) show the pressure gauges.

Flow rate in the system was measured using two inline flow meters as shown in Plate (7) and Fig. (16). This flow rate meter displays the flow rate in gal/min or litres/min. The flow indicator consists of a sharp edged orifice and tapered metering piston which moves in proportion to changes in flow rate. As the flow increases, the pressure difference across the orifice formed by the metering piston and fixed orifice moves the piston against the calibrated spring. Piston movement is directly proportional to flow rate with the sharp edge orifice minimizing the effects of viscosity.

Actual motor, pump and fan speed were obtained using a remote hand digital reading tachometer which measured the speed of the rotating mark on output shaft. The remote hand held digital tachometer is shown in Plate (2).

To get the measurements in different parts of the system, quick-disconnect hose couplings are used where frequent connection are made and broken between components. Plate (7) shows these hose couplings.

2.5. Experimental Test Rig Layout: A general view of the open hydraulic drive mixer system is shown in Plate (1) and schematically, in Fig (8).
The unit was mounted flat against a bracket using four holes of 11 mm diameter, so that the filler cap and filter are on top and that the control valve, filler cap and sight glass are accessible. Mounting details are shown in Fig (17) and Plate (4).

In this experimental rig a 11/8.2 kw Brook electric motor (Cage Pole change two speed Type) was positioned on the experimental bench and was coupled together with the chosen fixed displacement pump without external drains shown in Plate (2). The pump was a flange mounted type and an adapter bracket has been fitted in order to provide this way of mounting as shown in Fig (18). The suction port of pump with 38 mm diameter was connected to the reservoir by a 1.5 m length and 38 mm diameter single wired hydraulic hose.

All hoses except pump suction line, were connected via the control block as shown in Plate (8) and Fig. (19), the 19 mm (3/4 inch) diameter high pressure single wired hydraulic hose with length 1.5 m was connected from the high pressure port of the pump to the 19 mm (3/4) inch B.S.P male fitting on the control block and then 3/4 inch B.S.B male fitting on the control block is connected by a 2 m long, 19 mm (3/4 inch) diameter

single wired hydraulic hose to the pressure port of a 4 way 3 position at tandem centre position of the solenoid directional control valve with a subplate mounted on four bars placed between the hydraulic motor and pump as shown in Plate (3) for the convenience of operation.
The A port of the solenoid directional control valve is connected to one side of hydraulic motor by a 0.3 m length 12.7 mm (1/2 inch) diameter single wired hydraulic hose, while B port of the valve is connected to the other side of the hydraulic motor by a 0.3 m long 12.7 mm (1/2 inch) diameter single wired hydraulic hose. The T port (to low pressure return line) of the directional control valve is connected with 2 m long single wired hydraulic hose of 19 mm (3/4 inch) diameter to the 19 mm (3/4 inch) B.S.P male fitting on the control block.
The hydraulic motor was fixed vertically on a mild steel frame which was welded to the top of the container 2.0 m away from the reservoir. The mechanism of the mixer consists of four blades which is connected to the hydraulic motor by a shaft and placed in the container in 0.3 m of the mixing material to provide a load condition as shown in Plates (3) and (9). Technical data of the mixer is given in Table (13). The location of the mixer in the container is illustrated in Fig (20)
The 12.7 mm (1/2 inch) diameter drain port of the hydraulic motor was connected to 25.7 mm (1/2 inch)B.S.P female fitting on the reservoir by 2 m long, 12.7 mm

(1/2 inch) diameter single wired hydraulic hose.
A detailed listing of technical data for the test rig is given in Table (14).

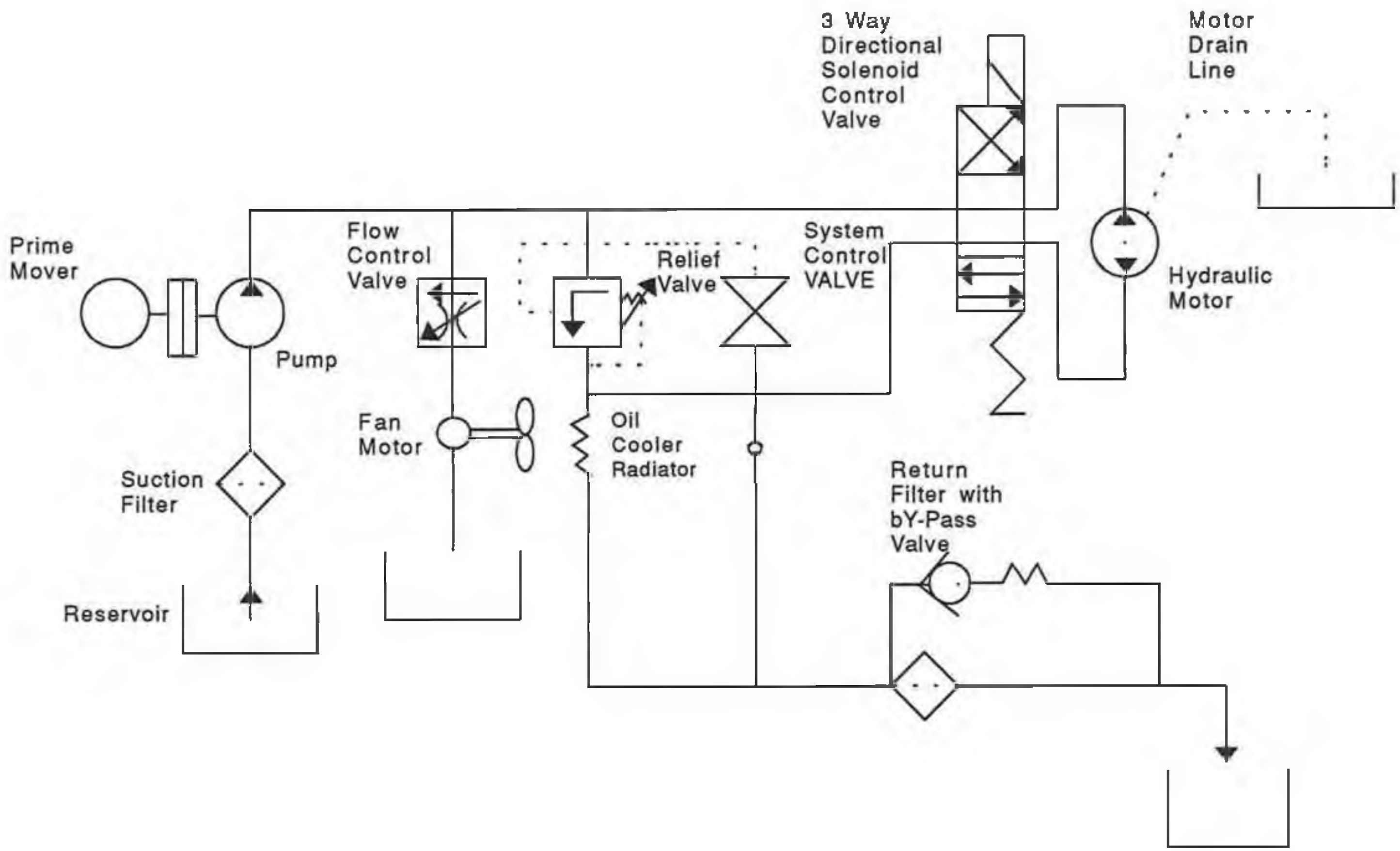


Fig (8) An open hydraulic drive mixer test rig

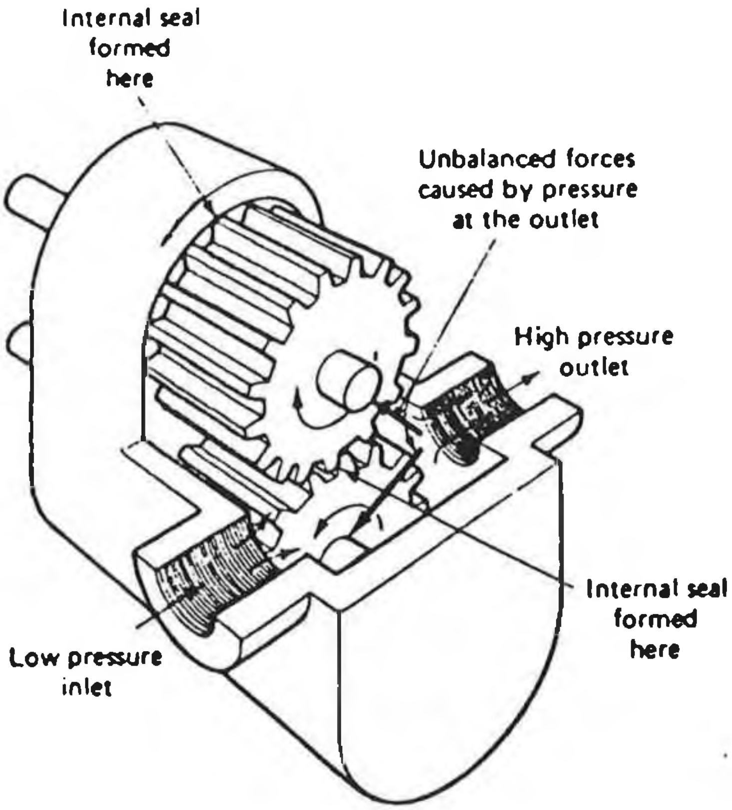


Fig (9) External hydraulic gear pump

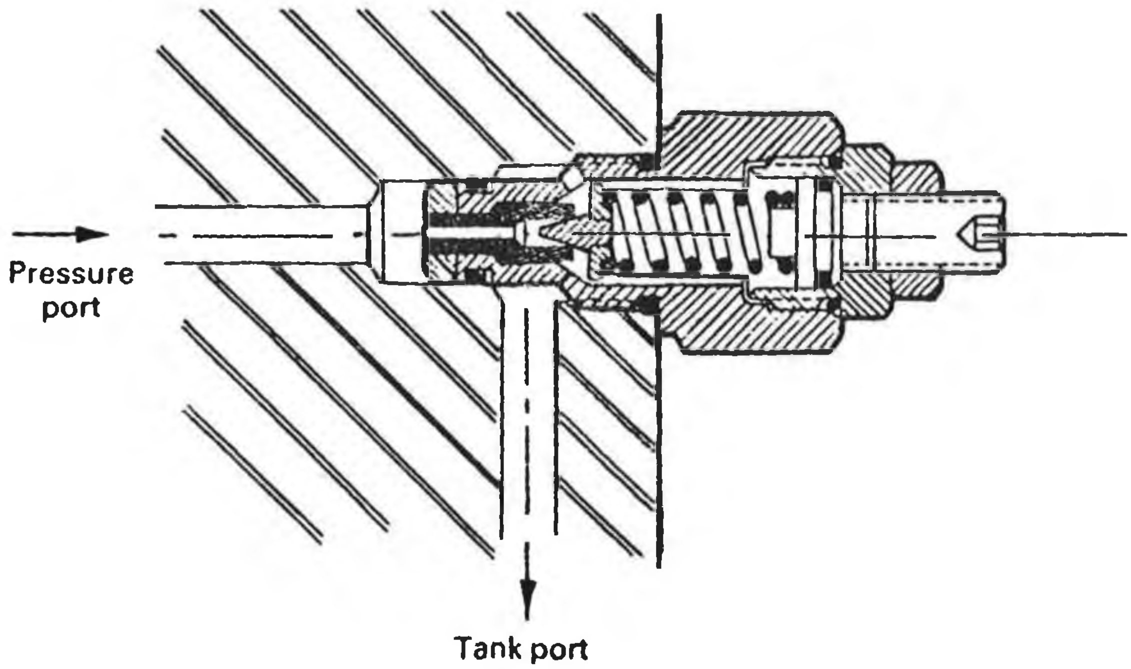


Fig (10) The cartridge insert relief valve

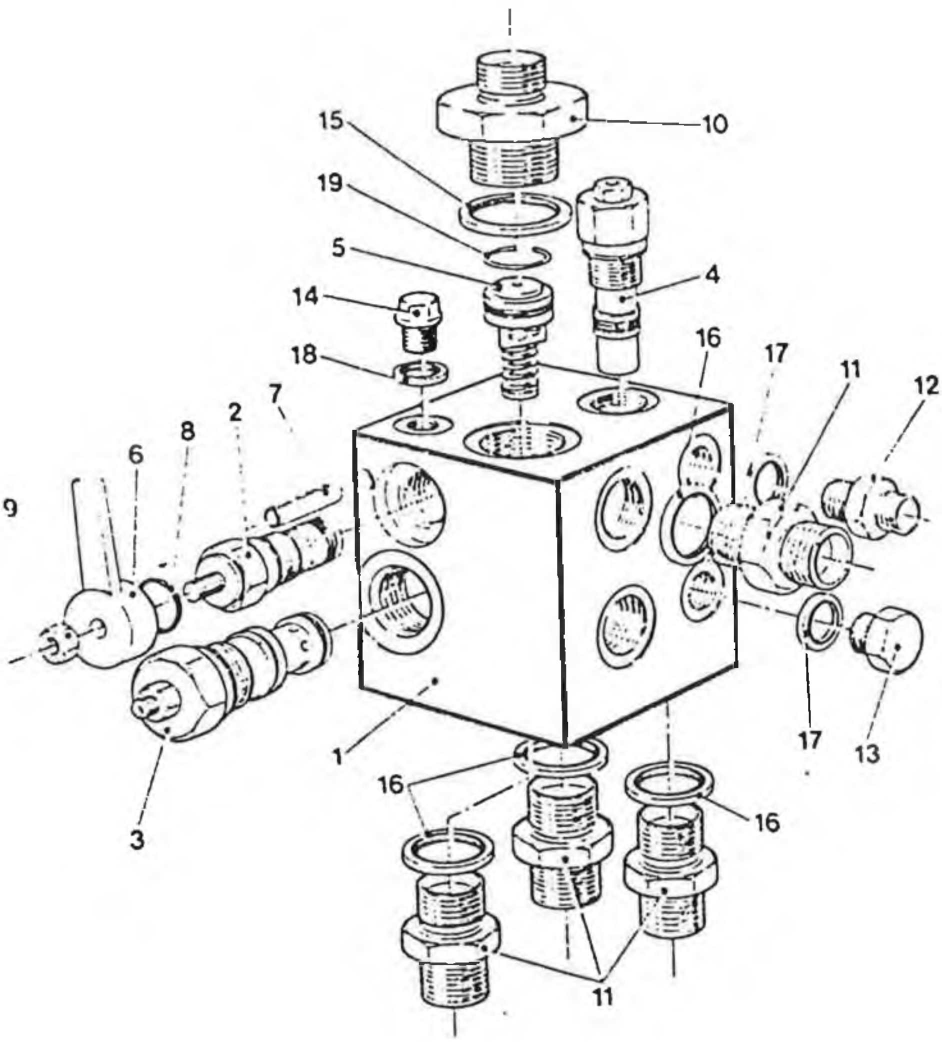


Fig (11) Valves block assembly

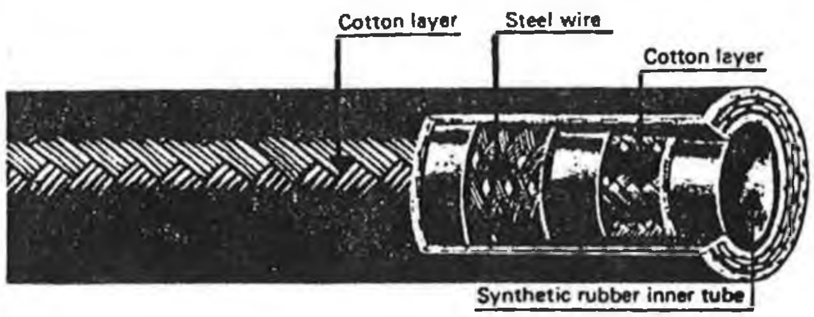


Fig (12) Structure of a single flexible wire braid hydraulic hose

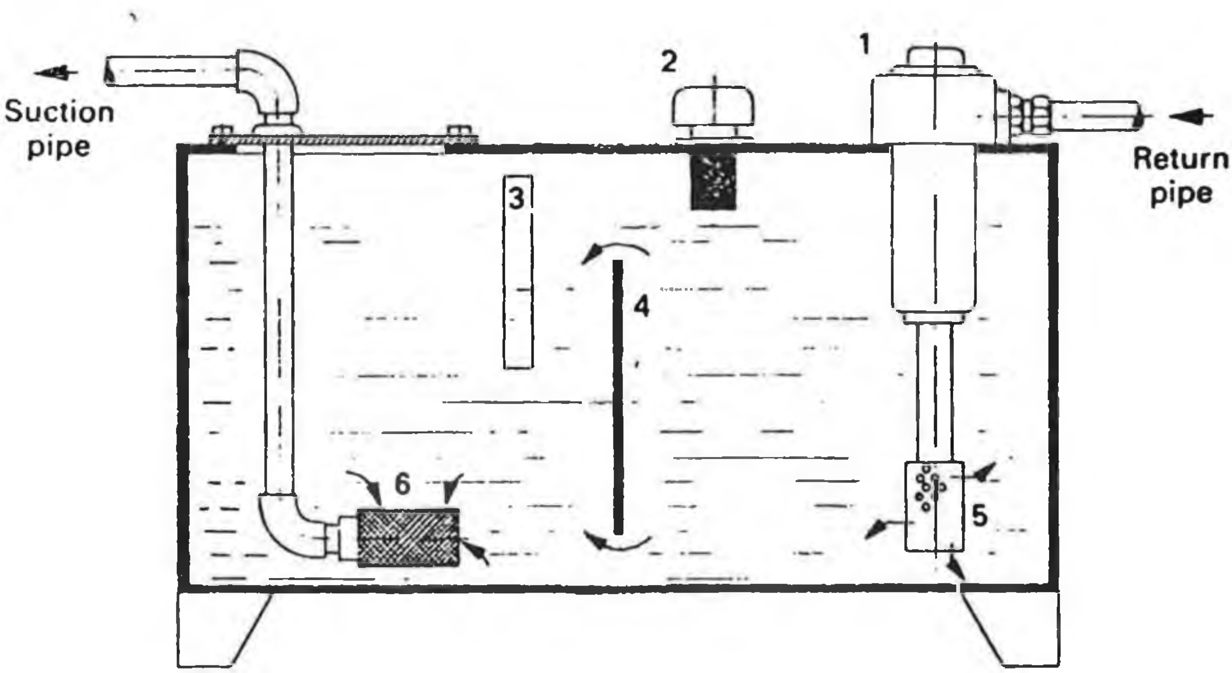


Fig (13) Structure of the hydraulic reservoir

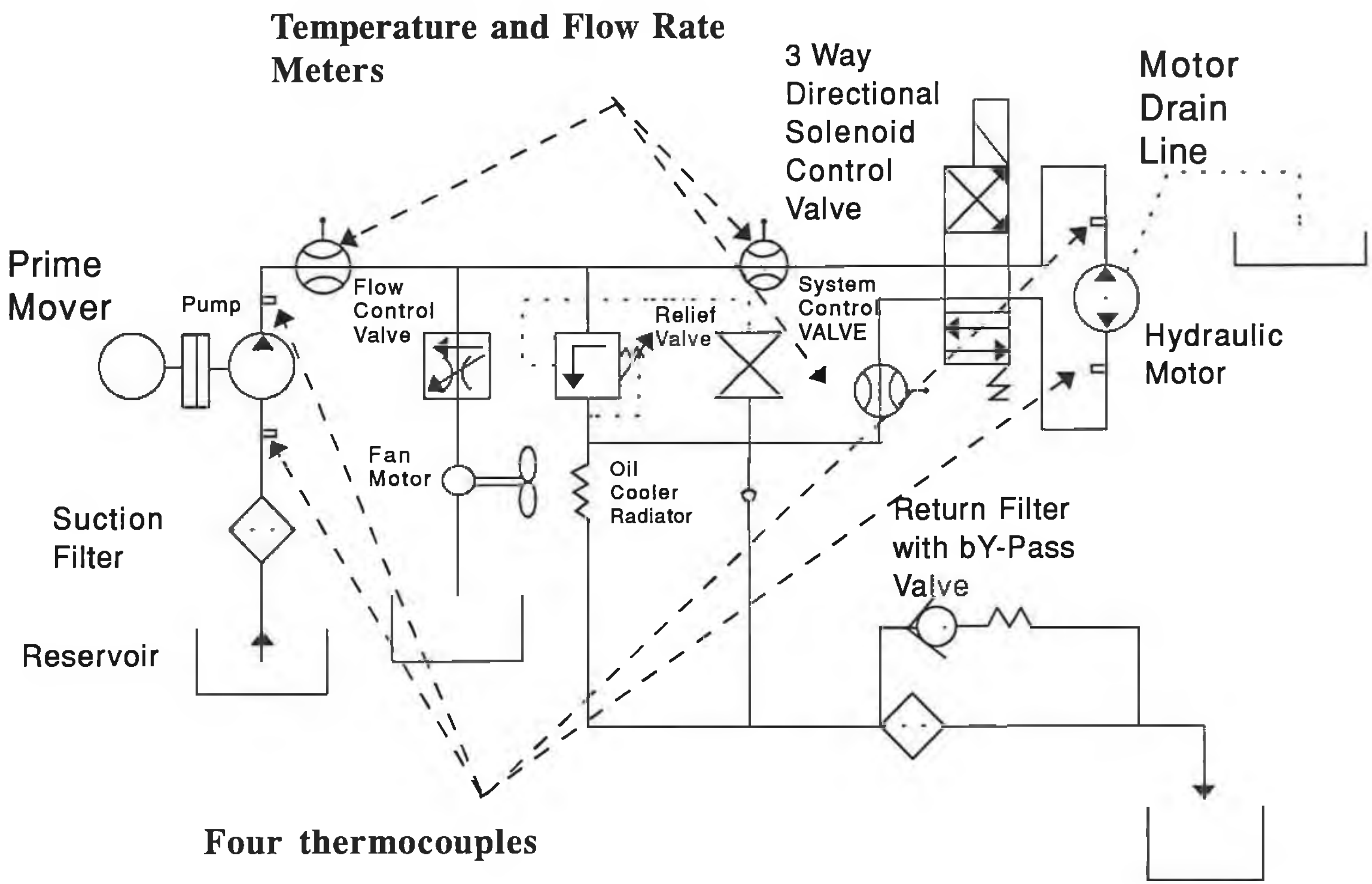


Fig. (14) Position of the measurements equipment

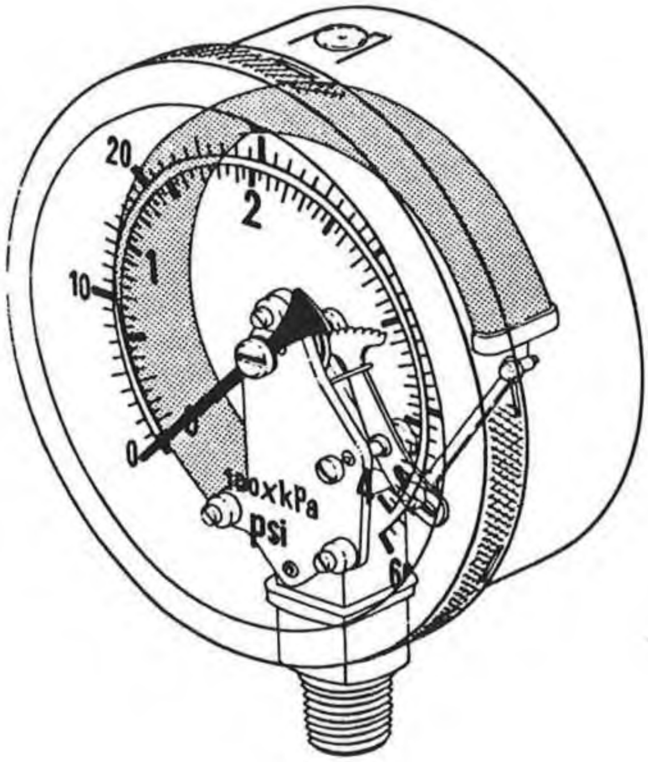
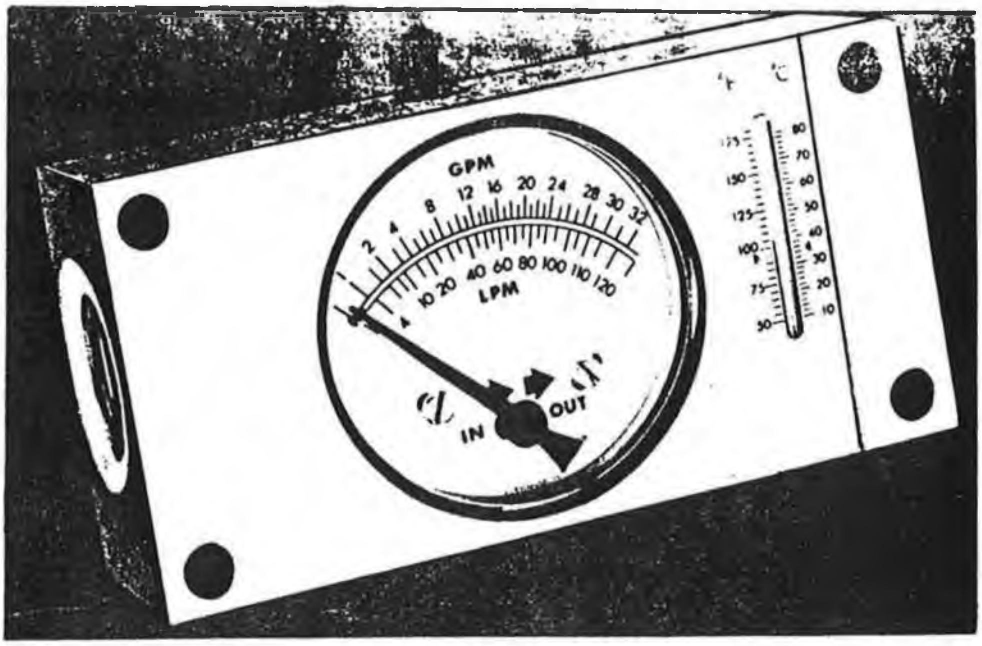
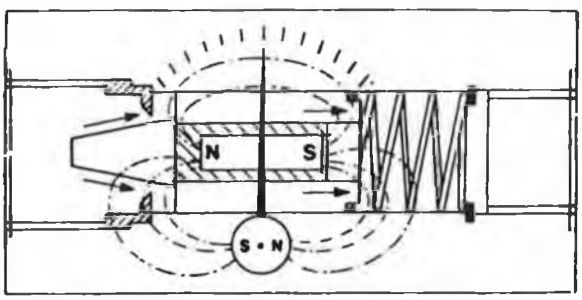
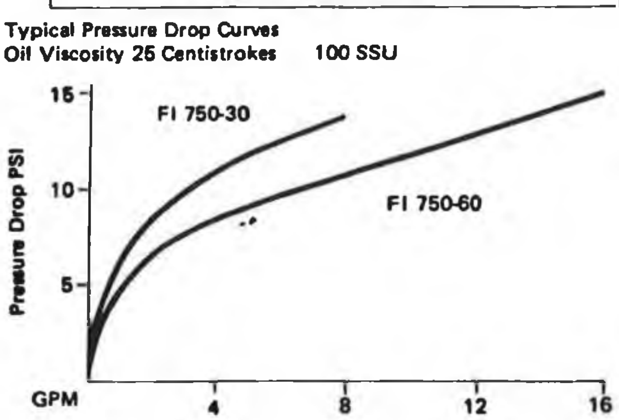


Fig (15) Pressure gauge



Operation





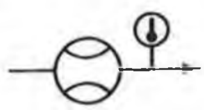


Fig (16) Flowrate and temperature meter

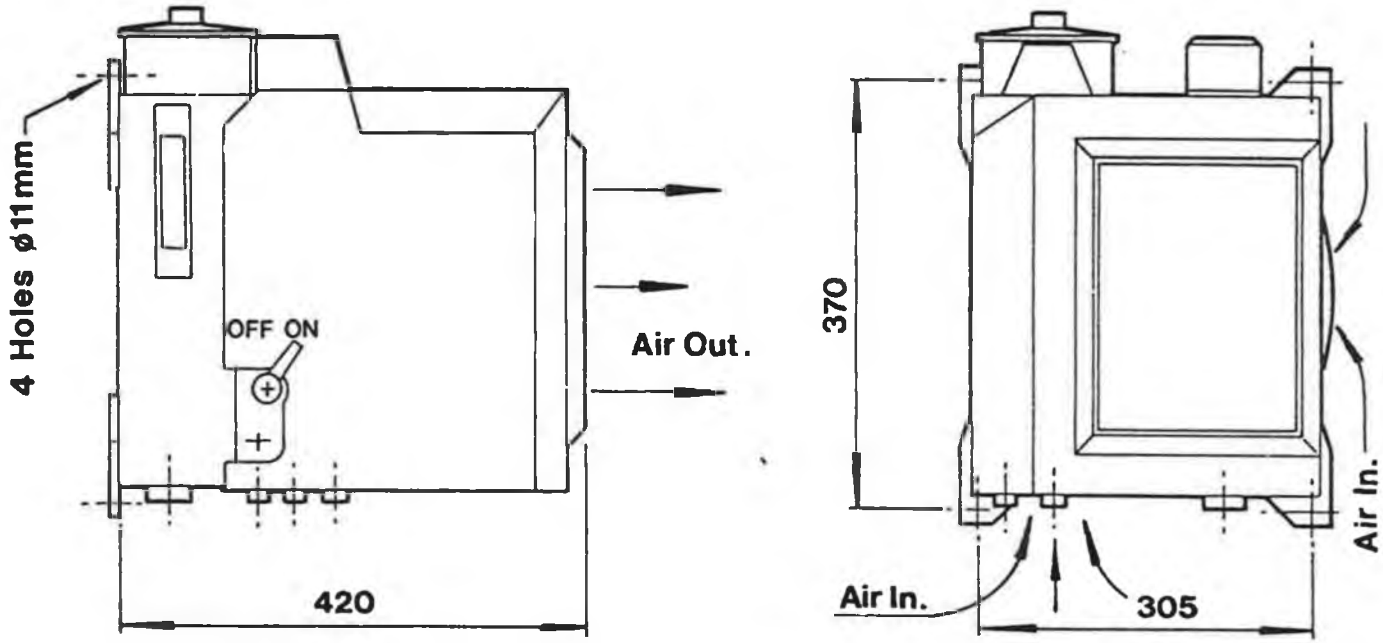


Fig (17) Mounting detail of the unit.

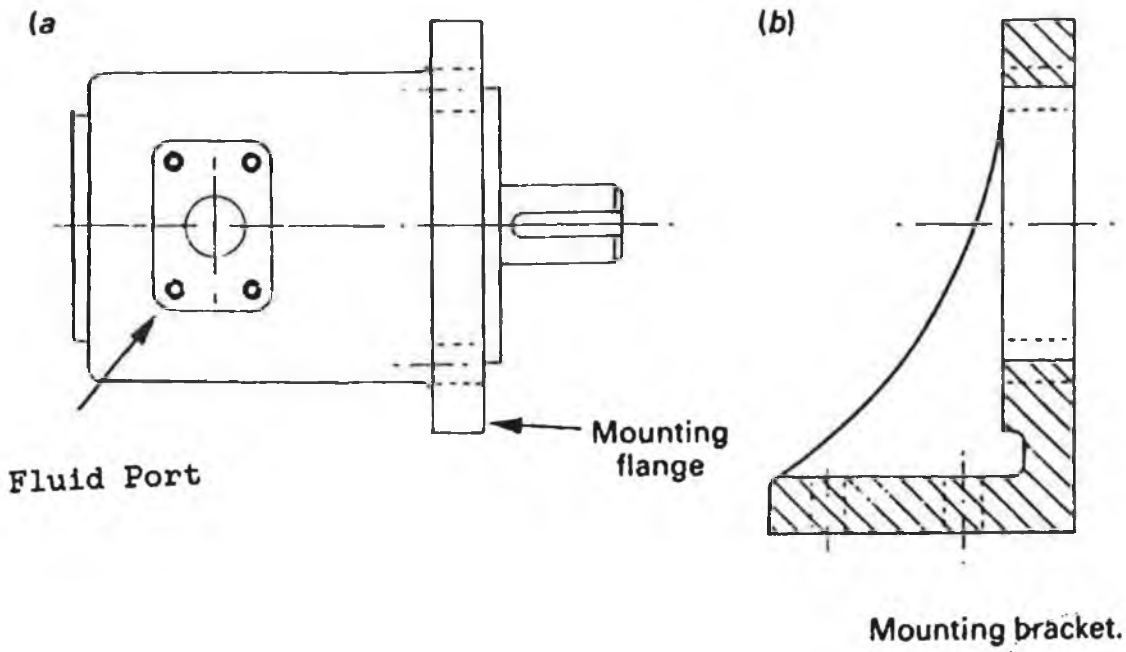


Fig (18) The pump with front mounting flange and bracket

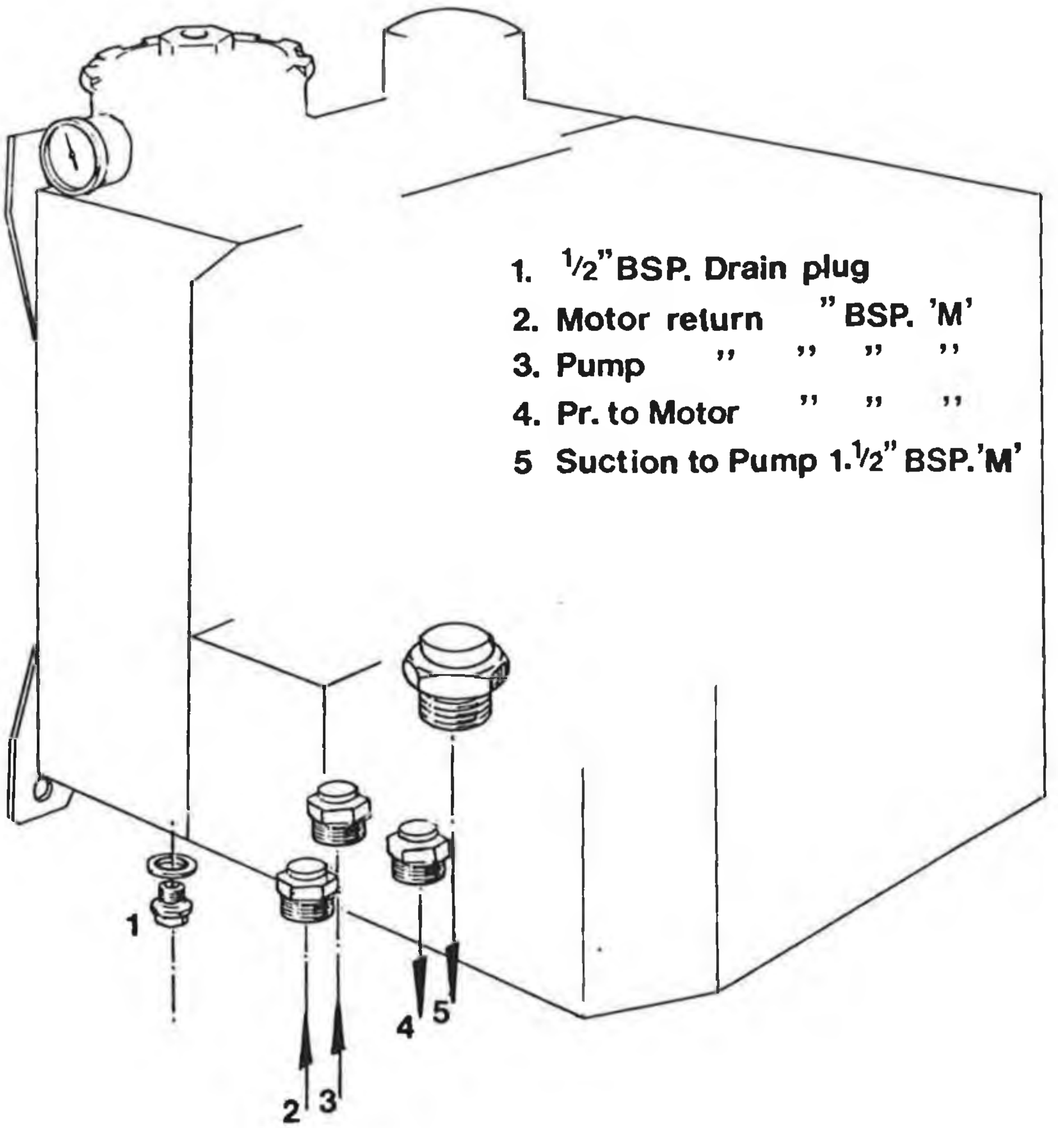


Fig (19) Position of drain, pump and hydraulic motor port in the control block

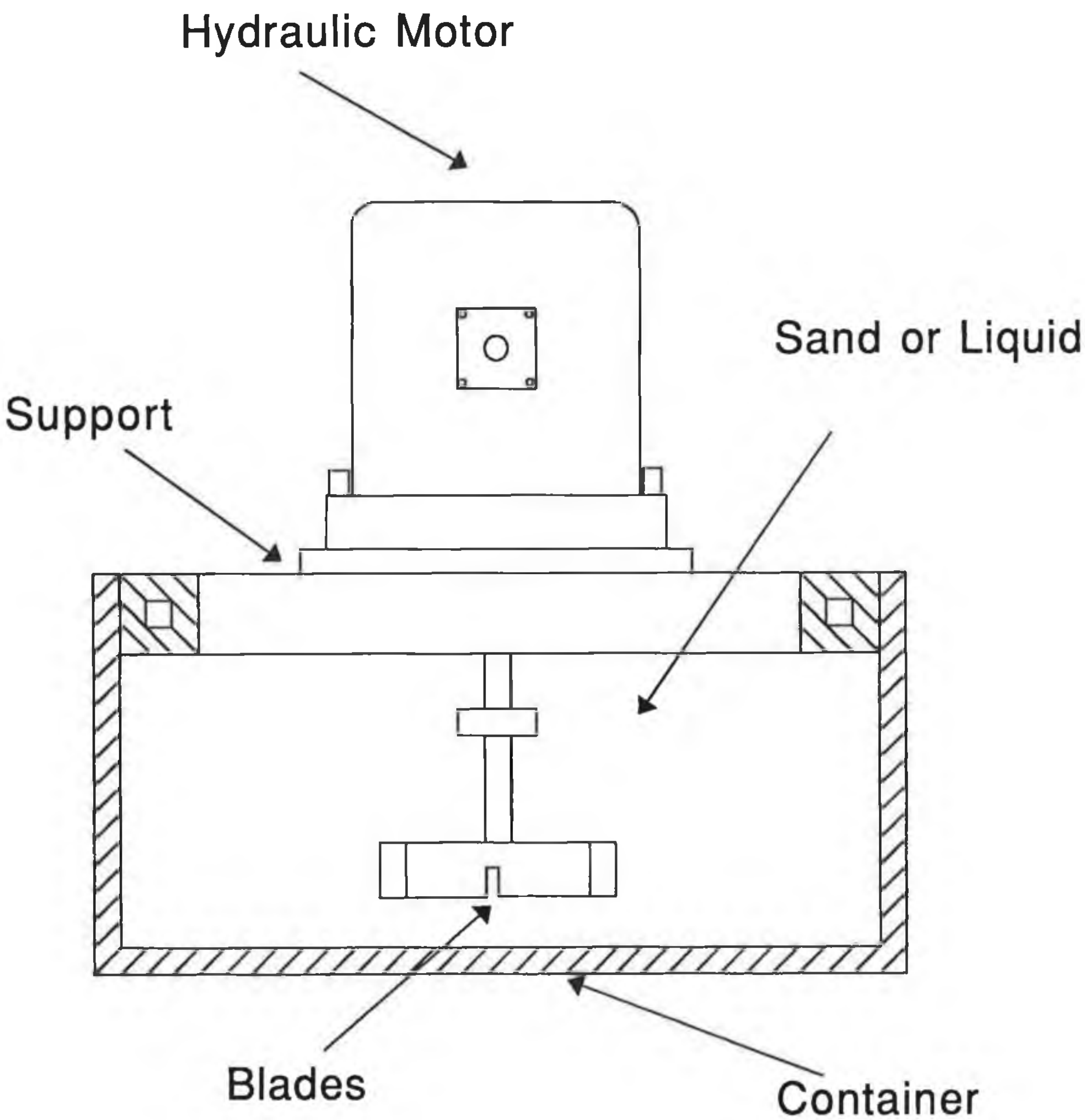


Fig. (20) The location of the mixer inside the container

Name of Component	Pump	Unit
Type: External Gear 1LA30DE		
Maximum Flowrate at 1500 rpm	36	L/min
Displacement	24	Cm^3/rev
Continuous Maximum Pressure	220	Bar
Maximum Speed	3000	r.p.m

Table (1) Technical data for hydraulic pump

Name of Component	Hydraulic	
Type: External Gear MPLA54	Motor	Unit
Maximum Flowrate at 1500 r.p.m	54	L/min
Displacement	36	Cm^3/rev
Continuous Maximum Pressure	220	Bar
Maximum Speed	2800	r.p.m

Table (2) technical data for hydraulic motor

Name of Component Type: Cartridge Ventable STERLING A4B125	Pressure Relief valve	Unit
Flow	200	L/min
Pressure Range	20 to 350	Bar
Operating Temperature	-30 to +90	°C

Table (3) Technical data for pressure relief valve

Name of Component Type:Cartridge STERLING	Check Valve	Unit
Maximum Flow	100	L/min
Maximum Pressure	350	Bar
Cracking Pressure	0.1	Bar
Operating Temperature	-30 to 90	°C

Table (4) Technical data for check valve

Name of Component	Flow Control Valve	Unit
Type: Flow Regulator Restr-ictive STERLING J2A60		
Flow	2-40	L/min
Pressure Range	20-350	Bar
Operating Temperature	-30 to 90	°C

Table (5) Technical data for flow control valve

Name of Component	Directional Control Valve	Unit
4 Way 3 Position Solenoid		
Maximum Flowrate	100	L/min
Maximum Pressure	250	Bar

Table (6) Technical data for directional control valve

Name of Component	Electric Motor	Unit
Type: 3 Phase A.C. Cage Pole Change 2 Speed BROOK MOTOR D160M		
Maximum Horse Power	11/8.2	KW
Speed	3000/1500	r.p.m

Table (7) Technical data for the electric motor

Name of Component	Pressure Pipeline	Unit
Material	Rubber Hose	
Length	3.5	m
Pipe Internal Diameter	3/4	inch

Table (8) Technical data for pressure pipeline

Name of Component	Pressure Return Line	Unit
Material	Rubber Hose	
Internal Diameter	3/4	inch
Length	2	m

Table (9) Technical data for pressure return line

Name of Component	Pressure Suction Line	Unit
Material	Rubber Hose	
Length	1.5	m
Internal Diameter	1.5	inch

Table (10) Technical data for pressure suction line

Name of component Type: Reservoir top is not open directly to its surrounding atmosphere	Reservoir	Unit
Material	Stainless	
Width of Reservoir Base	105	mm
Length of Reservoir Base	300	mm
Height of Reservoir	340	mm
Capacity of Reservoir	8	Litre

Table (11) Technical data for hydraulic reservoir

Name of Component Type: Hydraulic Oil Radiator	Cooler	Unit
Maximum flowrate	80	L/min
Maximum Heat Dissipation	15	KW
Volumetric Size	3	Litre

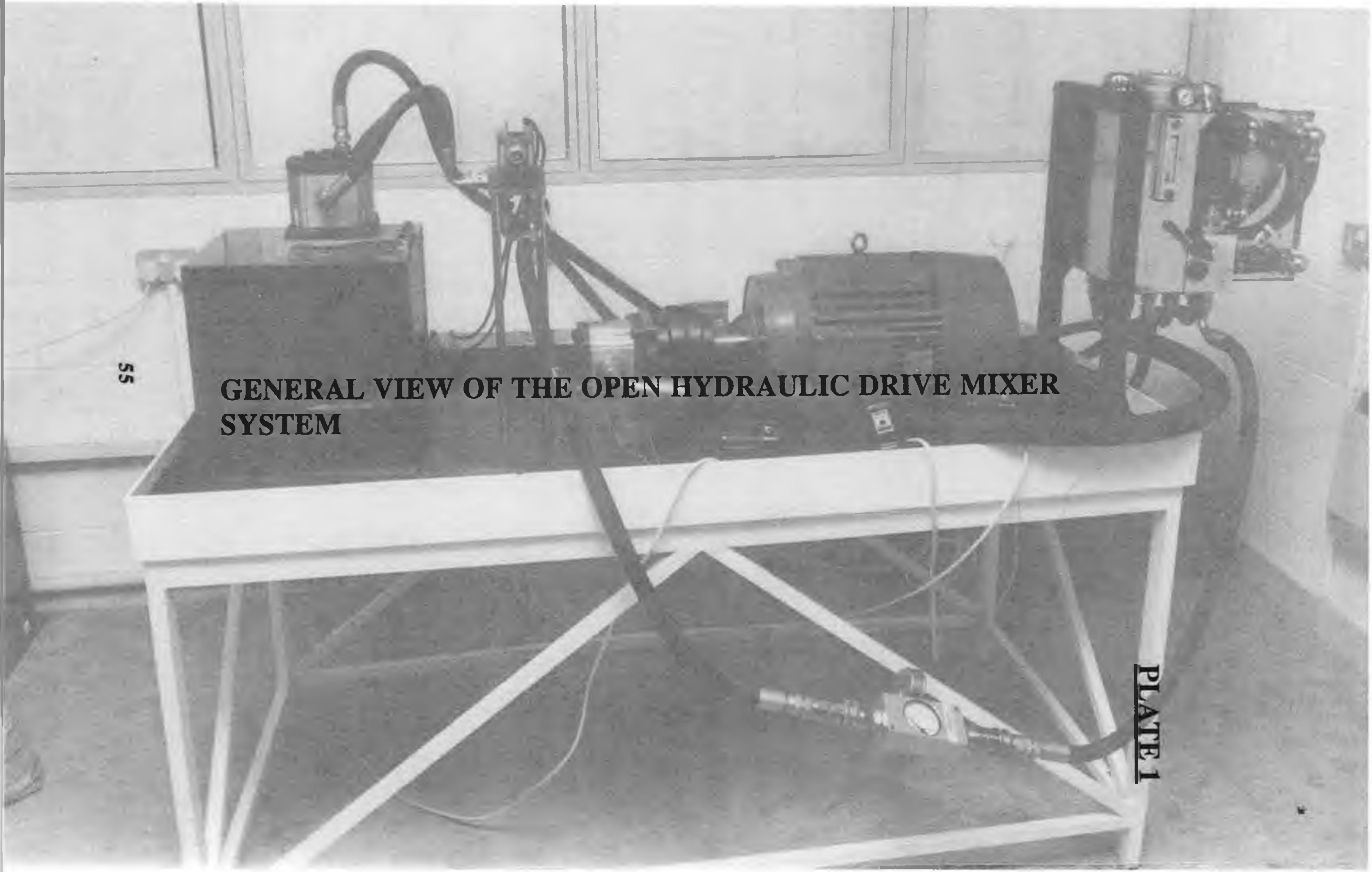
Table (12) Technical data for hydraulic oil cooler

Name of Component	Mixer	Unit
Diameter	300	mm
Initial Angular Velocity	0	r.p.m
Number of Blades	4	

Table (13) Technical data for the mixer

No	Component Name	Technical	Data	Unit
1	Electric Motor	Horse Power at	8.2	KW
3	Hydraulic Motor With External Drain	Maximum Displacement	36	Cm^3/rev.
		Maximum Pressure	220	Bar
4	High Pressure Pipeline	Hose Diameter	3/4	inch
5	Low Pressure Pipeline	Hose Diameter	3/4	inch
6	Suction Line	Hose Diameter	1.5	inch
7	Relief Valve	Maximum Flowrate	200	L/min
		Maximum Pressure	350	Bar
8	Check Valve	Maximum Flowrate	100	L/min
		Maximum Pressure	350	Bar
9	Flow Control Valve	Maximum Flowrate	40	L/min
		Maximum Pressure	350	Bar
10	4 way 3 Position Solenoid Directional Control Valve	Maximum Flowrate	100	L/min
		Maximum Pressure	300	Bar
11	Reservoir			
12	Cooler	Maximum Flowrate	90	L/min
		Heat Dissipation	15	KW
		Volumetric Size	3	L

Table (14) Technical data for experimental test rig



GENERAL VIEW OF THE OPEN HYDRAULIC DRIVE MIXER SYSTEM

PLATE 1

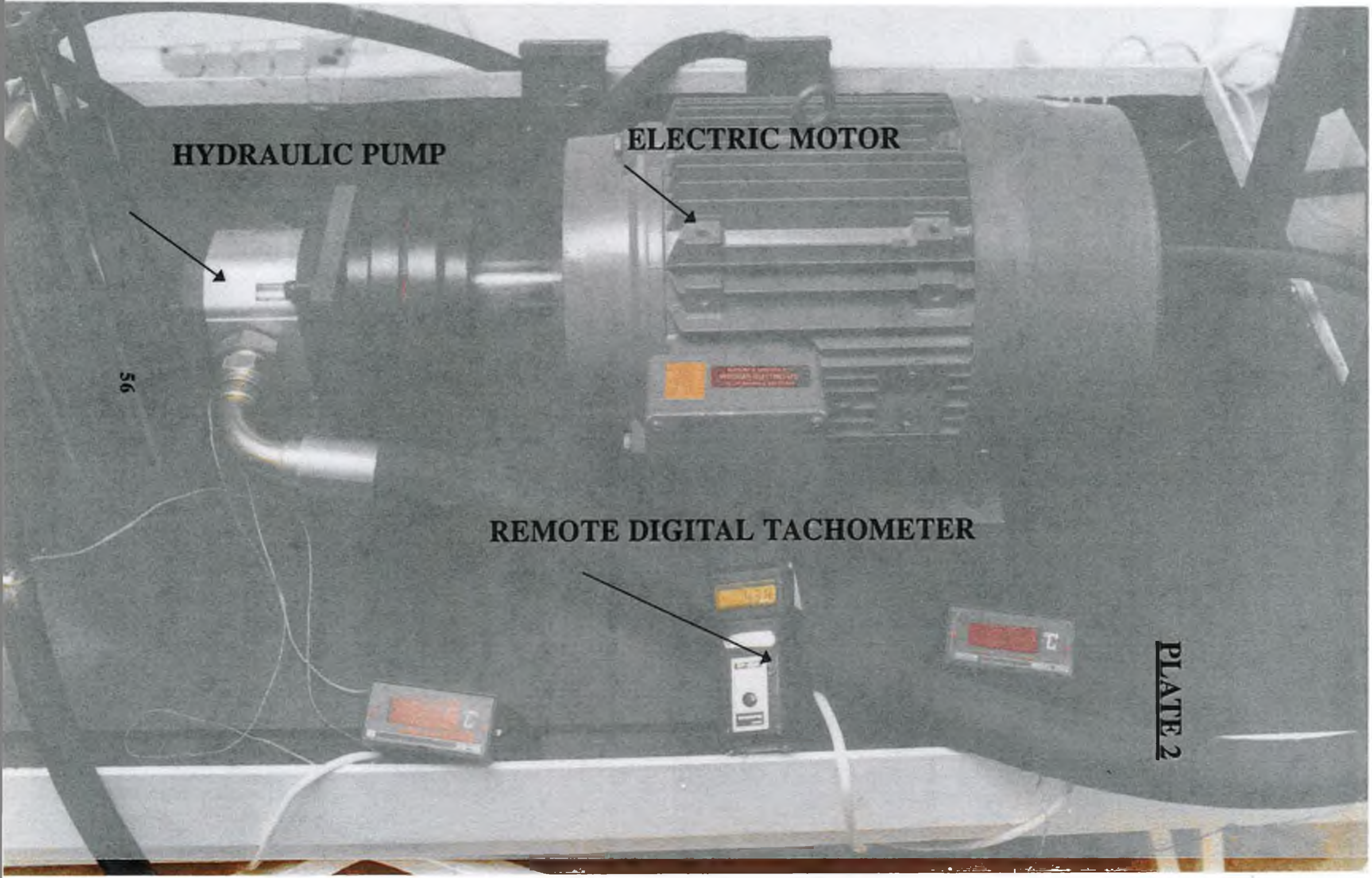


PLATE 2

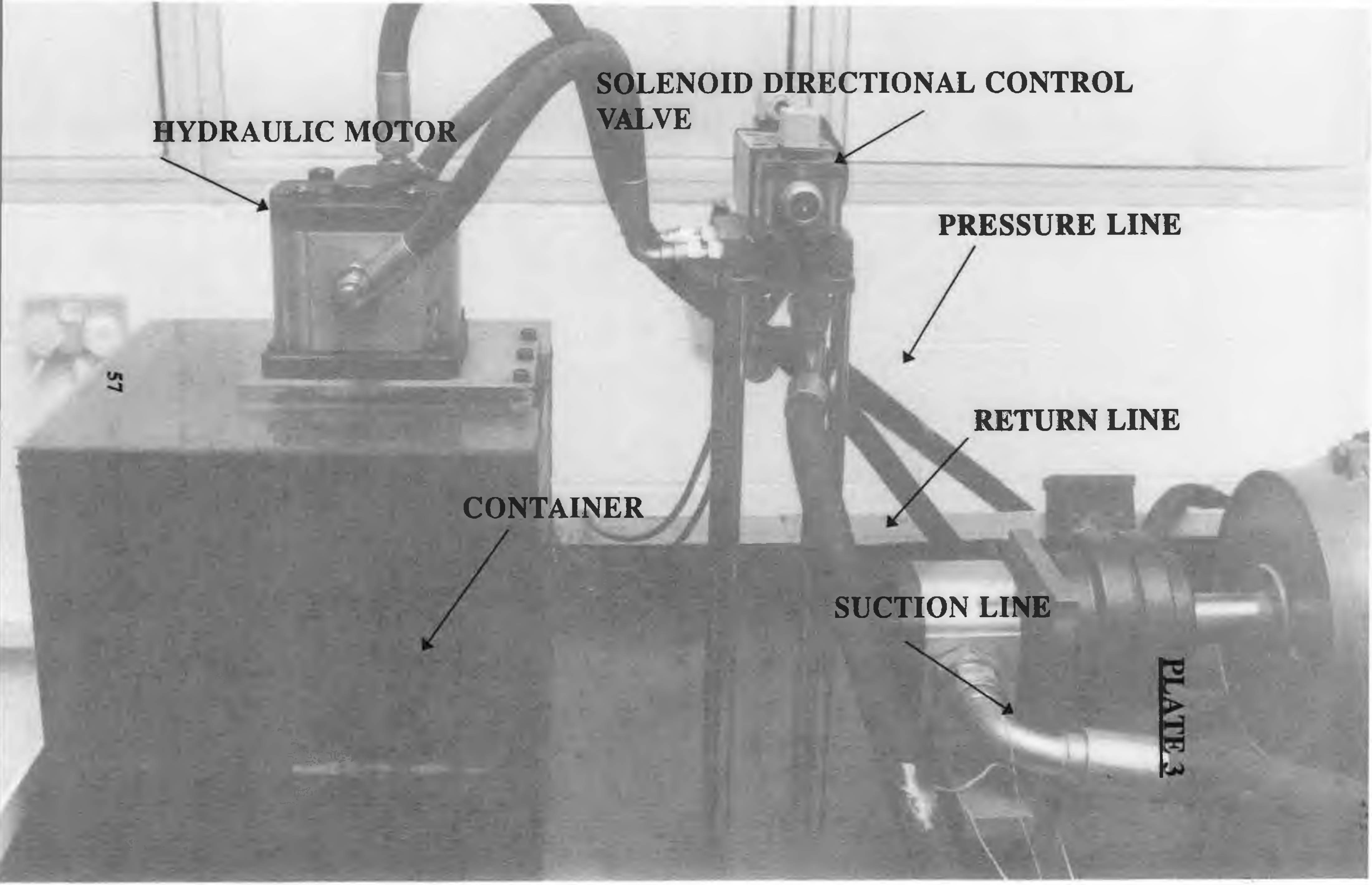

HYDRAULIC MOTOR
SOLENOID DIRECTIONAL CONTROL VALVE
PRESSURE LINE
RETURN LINE
CONTAINER
SUCTION LINE

PLATE 3

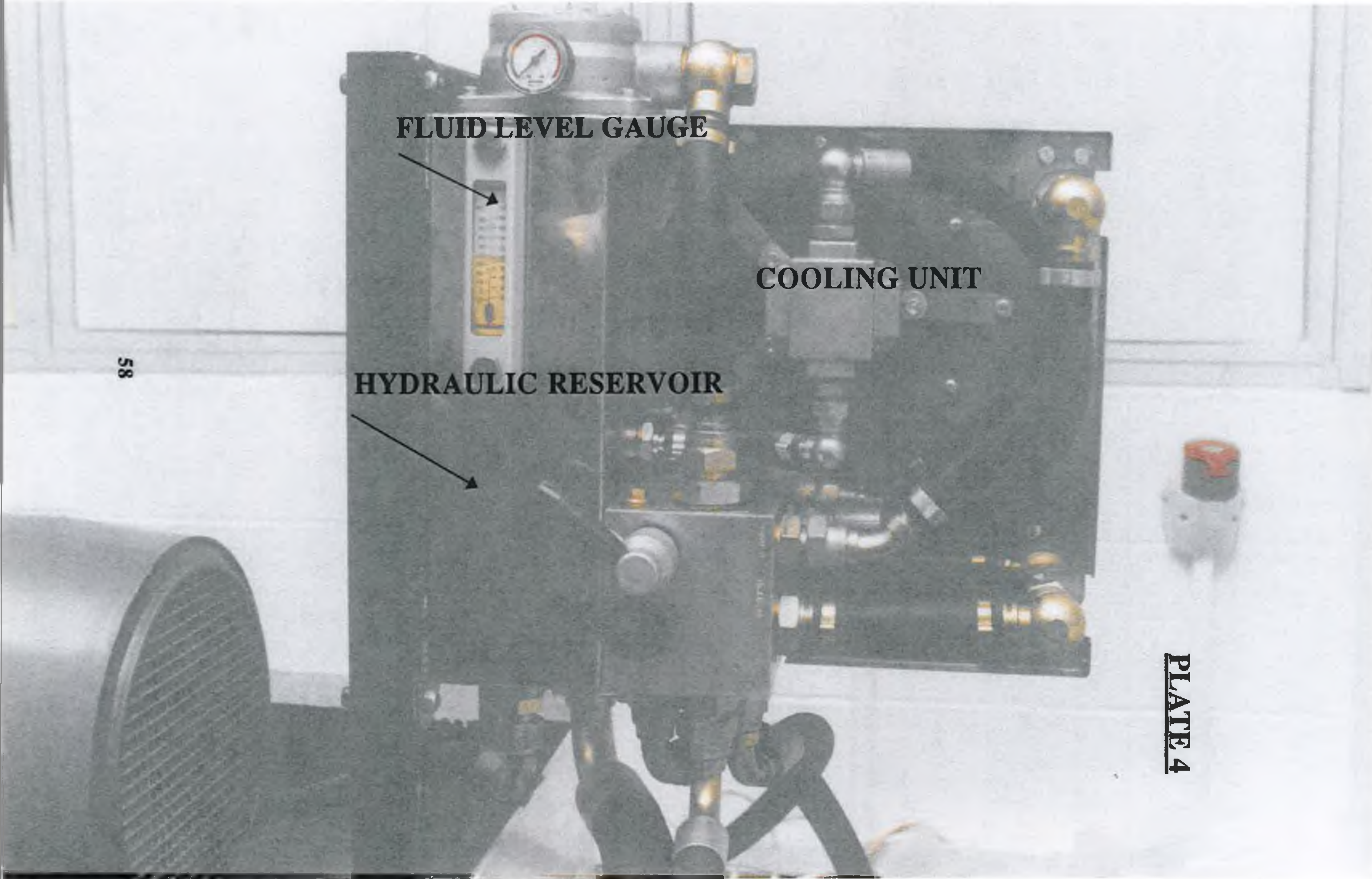
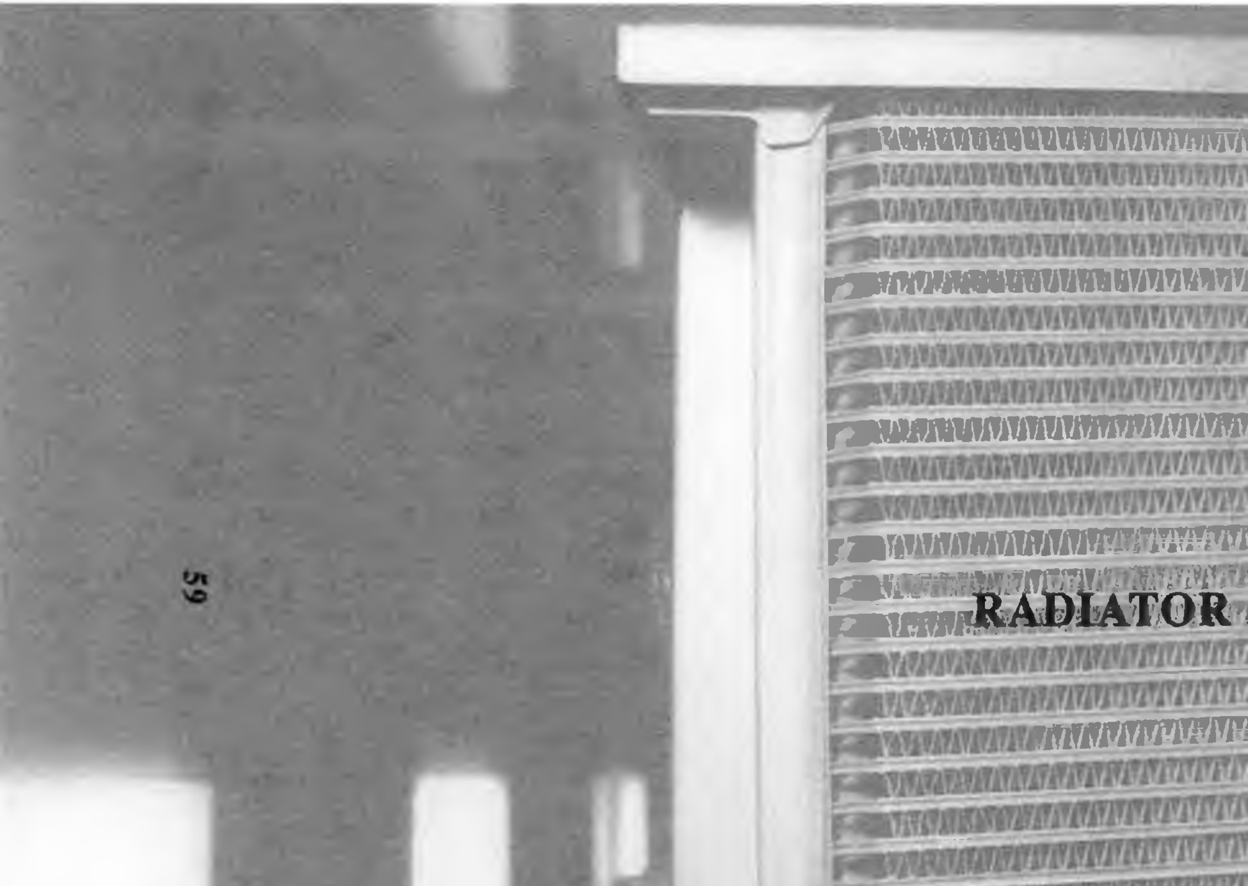


PLATE 4



RADIATOR

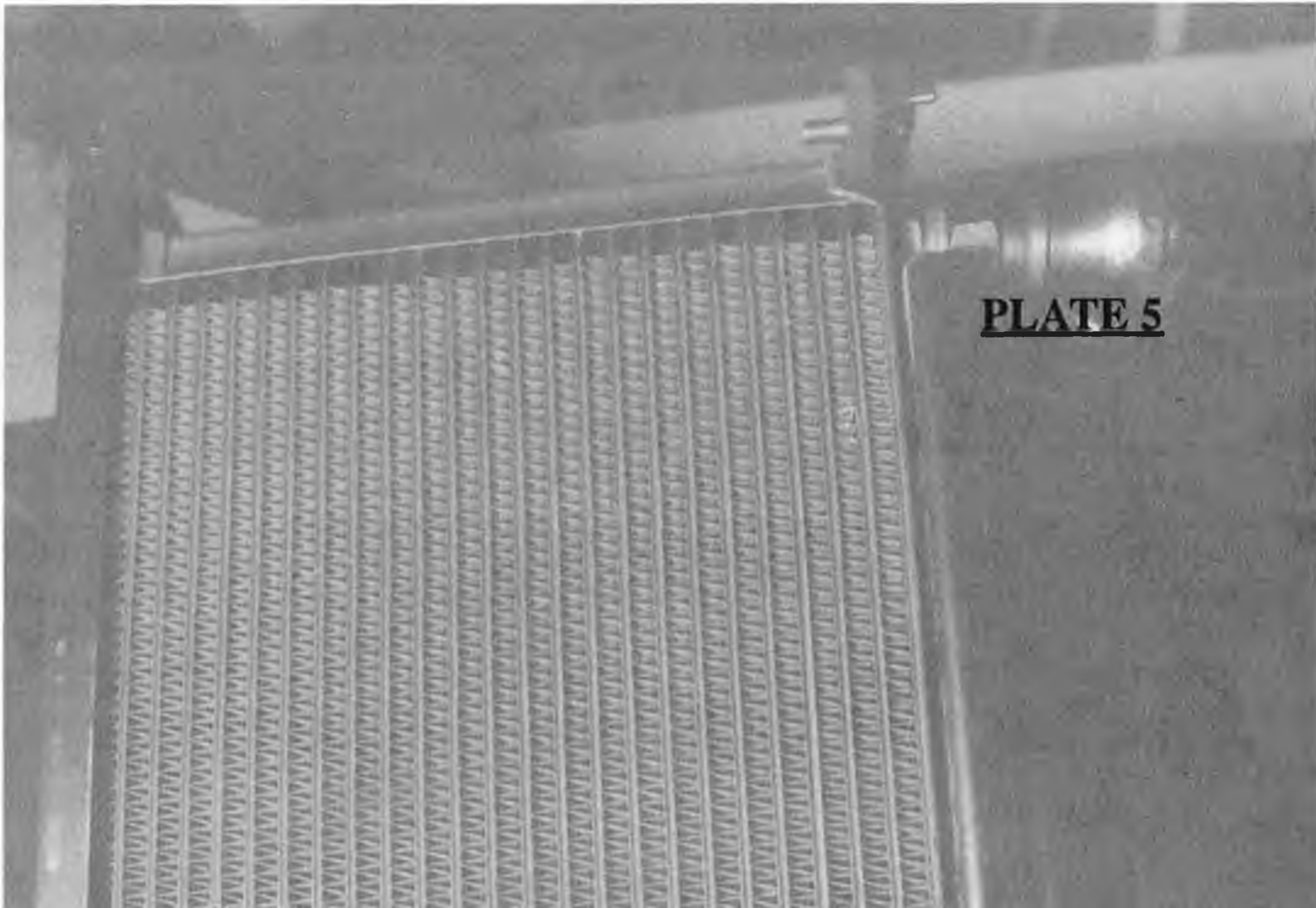


PLATE 5

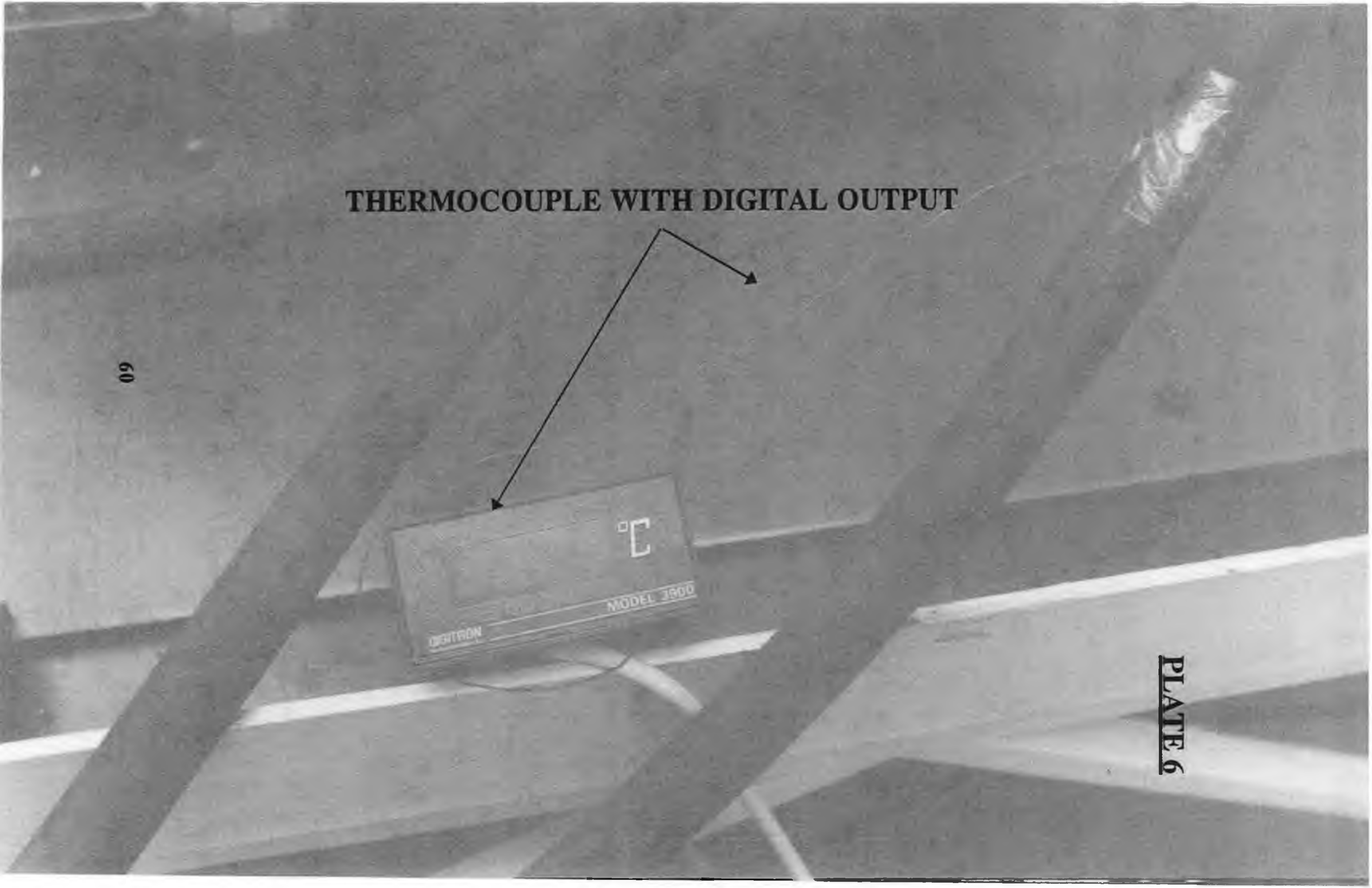
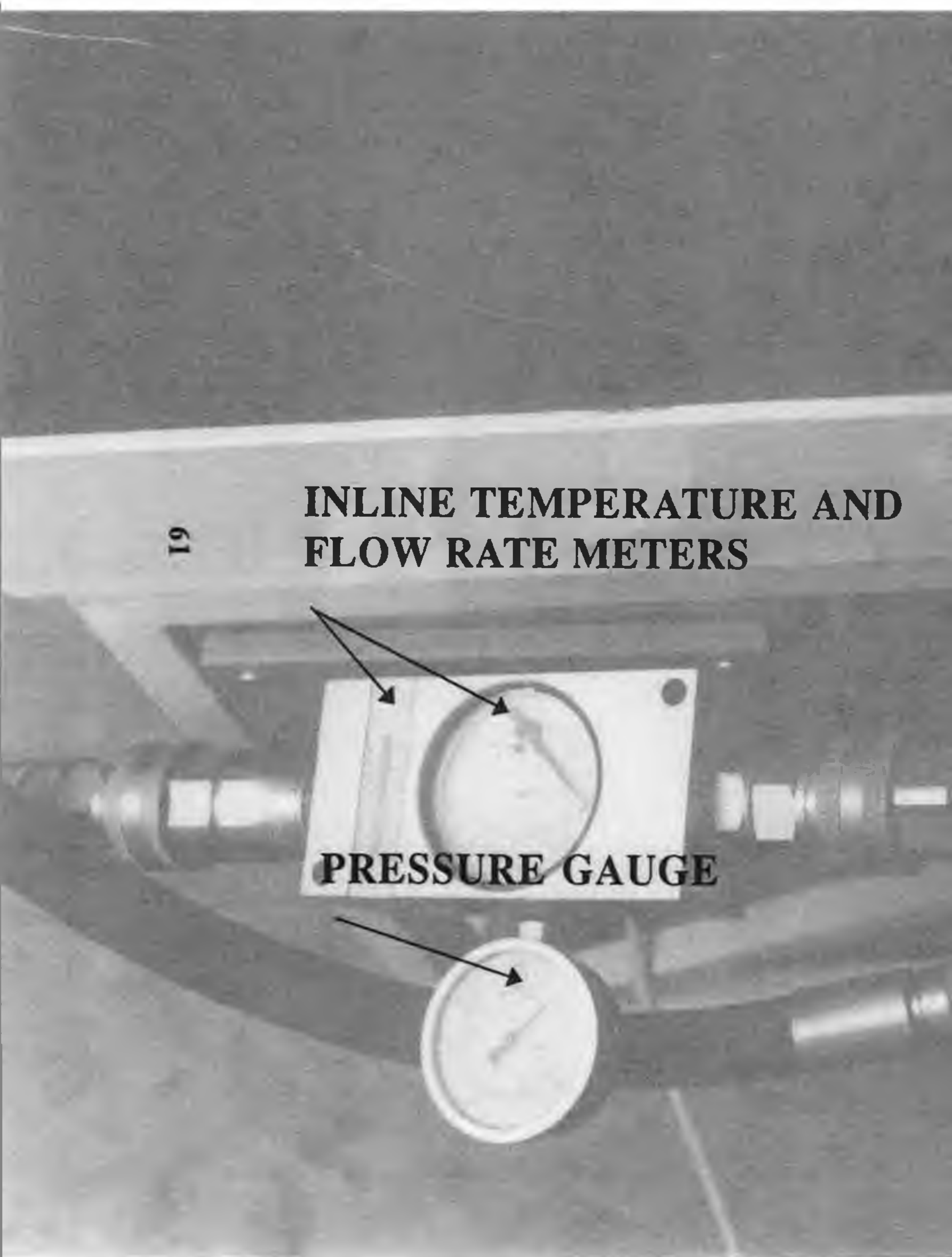


PLATE 6


INLINE TEMPERATURE AND
FLOW RATE METERS
PRESSURE GAUGE

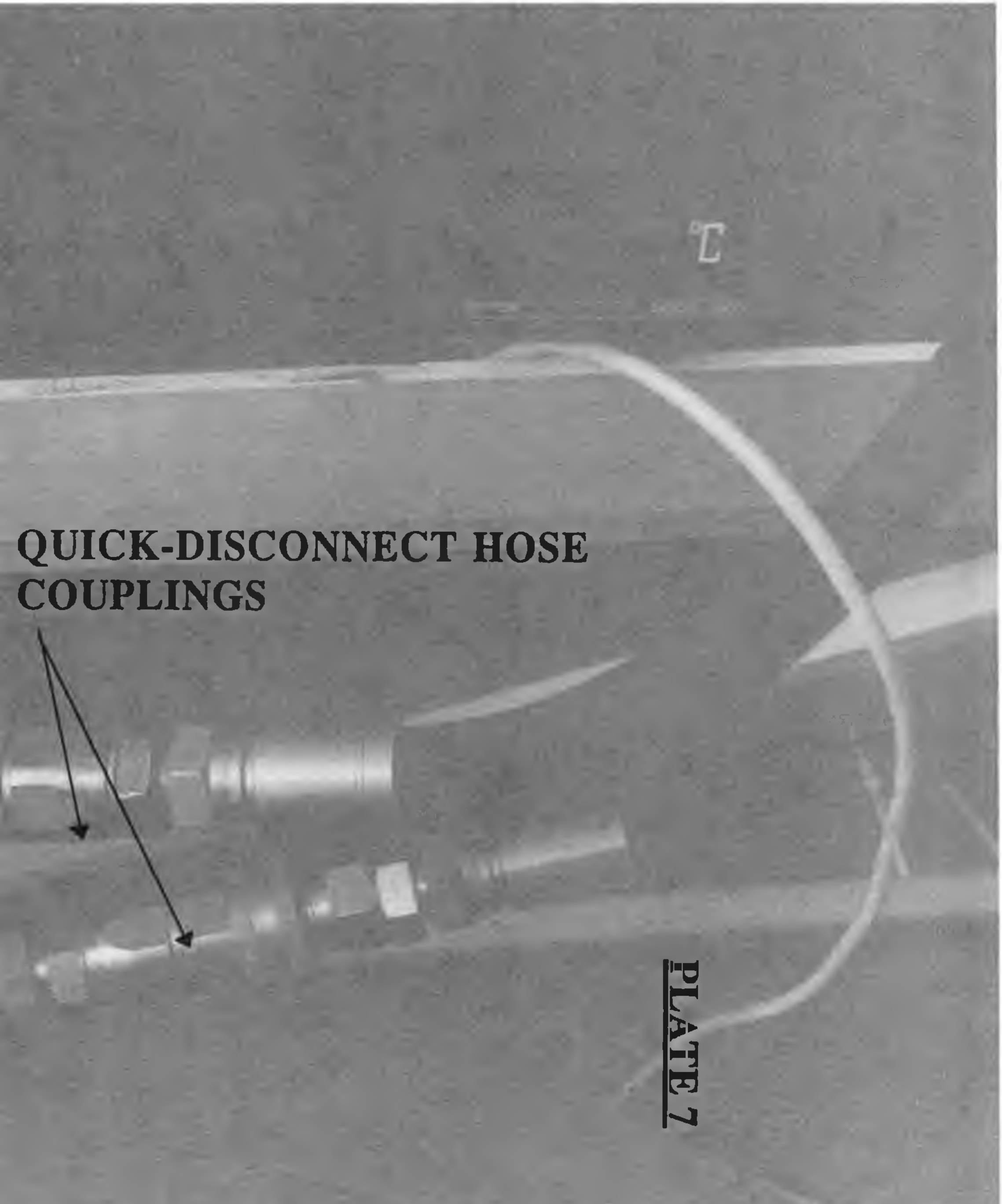


PLATE 7

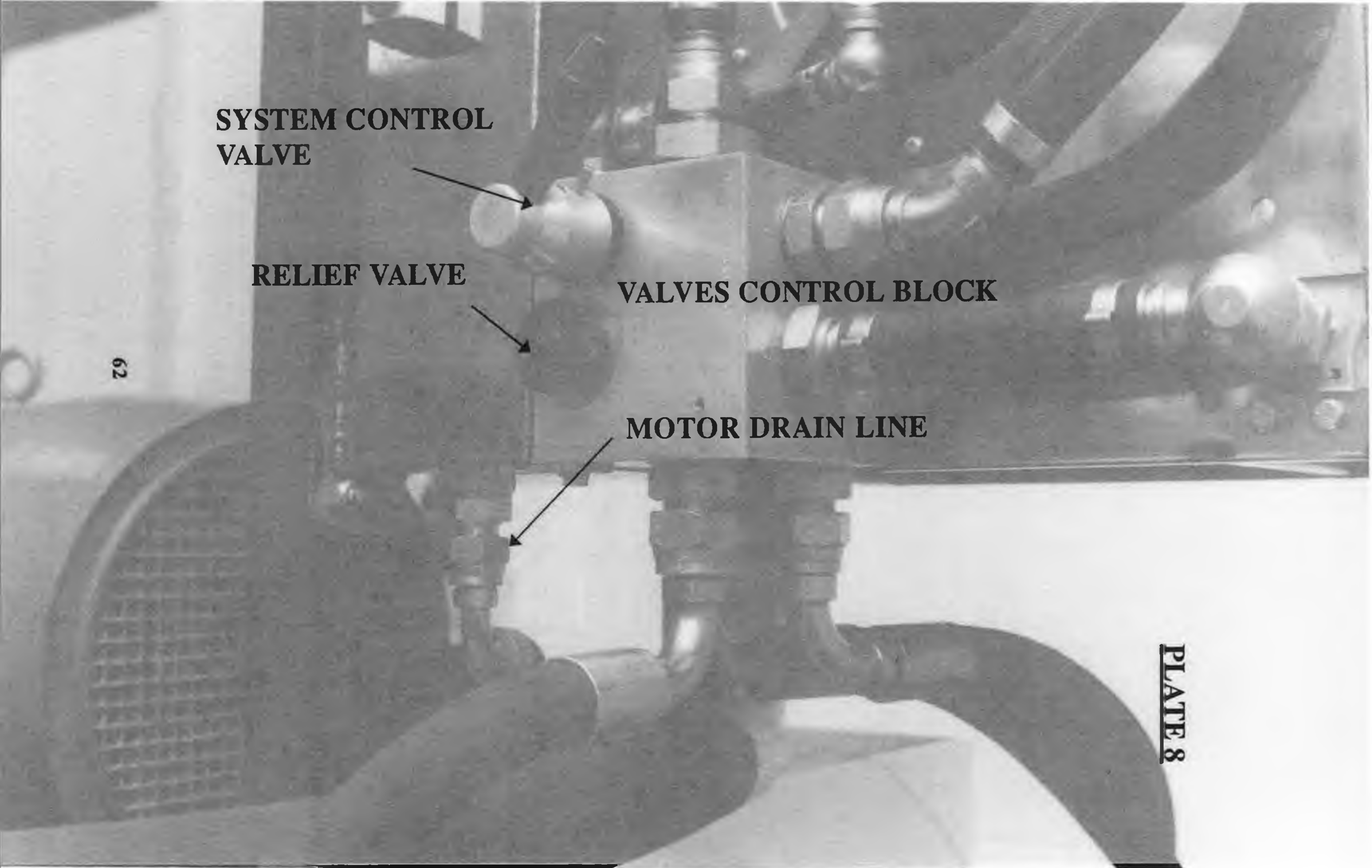
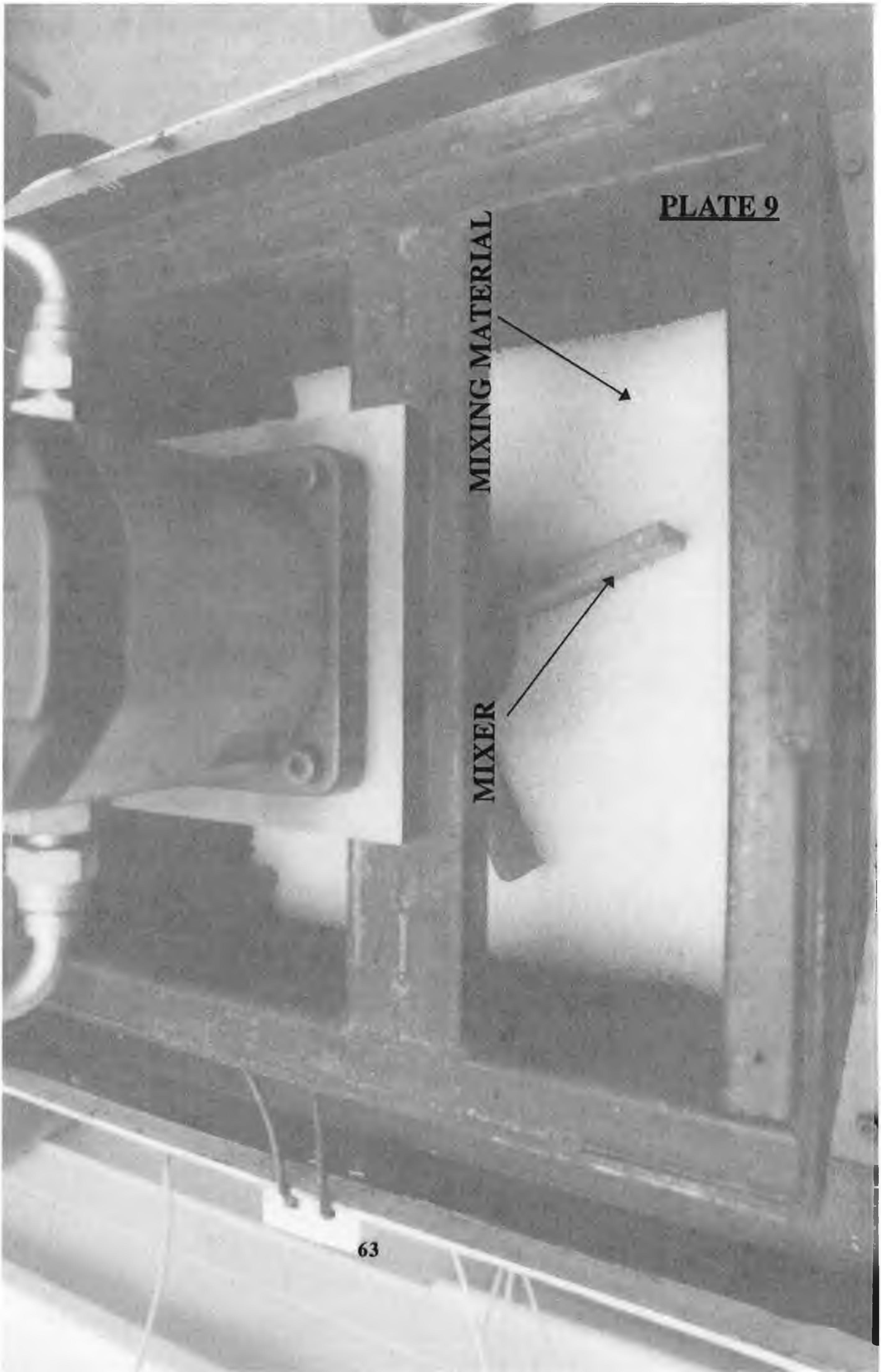

SYSTEM CONTROL
VALVE
RELIEF VALVE
VALVES CONTROL BLOCK
MOTOR DRAIN LINE

PLATE 8

PLATE 9



CHAPTER: 3

MATHEMATICAL MODEL AND SIMULATION

3.1. Introduction: In order to study and predict the temperature behaviour in any hydraulic system, it is important to develop a suitable mathematical model. As a first step, such a model has been developed in this study based on the assumption that the hydraulic system is an open system. The mathematical model is applied to analyze the temperature distribution in an open hydraulic mixer system. The power losses is converted into heat energy which causes the variations in the system temperatures. In a practical hydraulic system, the fluid temperature is different from the wall temperature of the pipes in the loop because of the heat flow transferred to the loop surrounding the operating system.

3.2. The present analysis: The analysis reported in references (13) and (14) was developed based on the assumption that 1) both the inside and outside pipe wall temperatures of the system were the same as the fluid temperature.

2) the heat generated by the pressure losses due to pipe friction were negligible.

Assumption (1) implies that the heat transfer process in which the rate of heat transfer by conduction and convection through the system pipe walls has been ignored. Therefore, this assumption may lead to errors in predicting the accurate fluid temperature in a practical hydraulic system. Assumption (2) is unreasonable since in some systems friction generates heat energy which should be taken into account.

In the present work a mathematical model for temperature analysis is developed to consider a hydraulic system (such as the open hydraulic mixer system) as an open thermal system in which the fluid temperature throughout the circuit and the pipe wall temperature are also uniform throughout the circuit. In addition, in this analysis the differences between the fluid and pipe wall temperatures have been considered so that the transient difference in the heat flows transferred from fluid to the pipe wall and from the pipe wall to the system surrounding atmosphere can be estimated to predict transient behaviour of outside pipe wall temperature. Furthermore, the rate of heat transferred by conduction, convection and radiation and, the heat generated by the pressure losses due to pipe friction have been taken into account. The present theoretical analysis also, has been improved to include the differences in pipe wall temperatures at the different parts of the system pipelines due to different pipe wall materials.

In this work the system main loop is considered as an open thermal system where uniform temperature is assumed in the loop fluid, while the reservoir is another open thermal system. Hence, the fluid energy exchanges between the loop and the reservoir can be estimated. However, in a practical hydraulic system differences exist between fluid temperatures of the system pipelines in the main loop due to the differences in the heat generation in the different parts of the system. So, a further study for analysis of the thermodynamic processes in a hydraulic system has been investigated in this thesis. The aim of that is to consider the hydraulic system loop as a series of open thermal systems with a constant fluid temperature in each part of the system so that the difference between fluid temperatures in the loop of the system can be calculated.

Before outlining the analysis, it was felt necessary to give a brief summary of some relevant fundamental thermodynamic concepts. The theory of the First Law applied to an open thermal system has been used to derive some equations in order to analyze basic thermodynamic and heat transfer processes associated with an open pipe section of hydraulic systems. This set of equations can then be applied for temperature analysis in the hydraulic system.

3.3. Closed and Open Thermal Systems: The concept of a system plays an important part in thermodynamics; it may be defined as a region in space containing a quantity of matter whose behaviour is being investigated. This quantity of matter is separated from its surrounding by a boundary, which may be a physical boundary, such as the wall of pipes, or some imaginary surface enveloping the region. Before any thermodynamics is attempted it is essential to define the boundary of the system, because it is across the boundary that work and heat are said to be transferred. The term surroundings is restricted to those portions of matter external to the system which are affected by changes occurring within the system. When the same matter remains within the region throughout the process under investigation it is called a closed system, and only the work and heat cross the boundary. An open system, on the other hand, is a region in space defined by a boundary across which matter may flow in addition to work and heat.

3.4. The First Law for Open Systems: It is known that the first law of thermodynamics is applied to a closed thermal system where no mass flow crosses the boundary.But, in most engineering applications it is difficult to separate a mass of the working substance and treat it as a closed thermal system. Zeuner (32) and Gillespie and Coe (33) suggested that it is possible to regard the continuous flow process as a series of non flow processes undergone by an imaginary closed system, as shown in Fig (21) which illustrates this concept. The same concept was also suggested by Keenan in Reference (34). However, in this way continuous flow may be regarded as a succession of these non-flow processes, carried out for every element of fluid entering the open system. Therefore, it now becomes possible to apply the energy equation which has been developed from a study of closed systems. The application of First Law of Thermodynamics in a closed system are well referenced by Bejan (35), McConkey (36), Wallace (37), and Look and Sauec (38). The first Law of Thermodynamics for a closed system can be written on a per unit time basis as

$$Q-W=\frac{dE}{dt} \tag{1}$$

The following analysis will later be applied to a length of pipe as a part of a hydraulic system. Fig. (22) shows the main features of an open system, namely heat transfer interactions per unit time, Q; work transfer interactions per unit time W; and portions of the boundary that are crossed by the flow of mass. For simplicity, this Figure shows only one of each type of boundary crossing, one inlet port labelled "in," and one outlet port labelled "out." The open system, or the control volume, is the rectangular region contained between the inlet and outlet ports, in other words, the dashed lines labelled "in" and "out" are part of the boundary of the open system.

Since equation (1) applies strictly to closed system, it must first identify a system with a fixed mass inventory that is unambiguously related to the open system of interest. If M_{open} is the mass inventory of the open system at a certain point in time t, then it can be thought of as the fixed mass inventory M_{closed} that at time t flows through the control volume. According to Fig (22), the relationship between M_{open} and M_{closed} is as follows:

$$M_{closed}(constant) = M_{open,t} + \Delta M_{inlet} = M_{open,(t+\Delta t)} + \Delta M_{outlet} \qquad (2)$$

For the process from state (1) (time t) to state (2) (time t+Δt) executed by the closed system, the First Law of Thermodynamics (1) reads:

$$E_{closed,(t+\Delta t)} - E_{closed,t} = Q\Delta t - W\Delta t + (P\Delta V)_{inlet} - (P\Delta V)_{out} \qquad (3)$$

The last two terms appearing on the right hand side account for the PdV type work transfer associated with the deformation of the closed system from time t to time t +Δt ; P is the local pressure, that is, the pressure in the immediate vicinity of the port. Relations similar to equation (2) express the relative size of the energy inventories of closed and open systems:

$$E_{closed,t}=E_{open,t}+\Delta E_{inlet} \tag{4}$$

$$E_{closed,(t+\Delta t)}=E_{open,(t+\Delta t)}+\Delta E_{outlet} \tag{5}$$

Furthermore, the ΔE and ΔV can be re-written in terms of their per unit mass counterparts e and v as

$$(\Delta E)_{inlet,outlet}=(e\Delta M)_{inlet,outlet} \tag{6}$$

and

$$(\Delta V)_{inlet,outlet}=(e\Delta M)_{inlet,outlet} \tag{7}$$

Like the port pressure P, the specific energy and volume (e and v , respectively) are properties of the intensive state of the fluid that cross the boundary at time t. Combining eqs (2) and (5) for the purpose of eliminating the terms that refer to energy inventory of the closed system (E_{closed}), the following equation is obtained:

$$\frac{1}{\Delta t}(E_{open,(t+\Delta t)} - E_{open,t} = Q - W + [(e+Pv)\frac{\Delta M}{\Delta t}]_{inlet} - [(e+Pv)\frac{\Delta M}{\Delta t} \tag{8}$$

Invoking the limit $\Delta t \rightarrow 0$, writing m for mass flowrate $\Delta M/\Delta t$, dropping the subscript "open" from the energy inventory of the control volume, and allowing for the existence of more than one inlet port and outlet port, the most general statement of the First Law of Thermodynamics for an open system is obtained as:

$$\frac{dE}{dt} = Q - W + \sum_{inlet} m(e+Pv) - \sum_{outlet} m(e+Pv) \tag{9}$$

What makes this statement more general than the per unit time version of the first law for closed systems, is the presence of the terms $m(e+Pv)$. These terms represent the energy transfer associated with the flow of mass across the system boundary. Finally, in the absence of macroscopic forms of energy storage other than kinetic and gravitational ones, the specific energy e can be decomposed into $(u+\frac{1}{2}V^2+gz)$ see Reference 35. The results of this decomposition is that of the *specific enthalpy, h, where*

$$h = u + Pv \qquad (10)$$

shows up explicitly in the terms accounting for energy transfer via mass flow. Therefore,

$$\frac{dE}{dt} = Q - W + \sum_{inlet} m(h + \tfrac{1}{2}V^2 + gz) - \sum_{outlet} m(h + \tfrac{1}{2}V^2 + gz) \qquad (11)$$

or

$$\frac{dE}{dt} = Q - W + m_1(h_1 + \tfrac{1}{2}V_1^2 + gz_1) - m_2(h_2 + \tfrac{1}{2}V_2^2 + gz_2) \qquad (12)$$

where Q is the heat flowrate transferred into the system, W is the rate of work done to the surroundings and m_1 and m_2 are the rates of mass flow.

In the above equation the open system has a quantity of internal energy E which can be defined as:

$$E = M(h + \tfrac{1}{2}V^2 + gz) \qquad (13)$$

In 1966, Kestin proposed an engineering generalization of the enthalpy concept under the name of methalpy [symbol h° in Ref (39), p. 223] $h° = e + Pv = h + 1/2\ V^2 + gz$ which is intended to mean "beyond enthalpy" or "transcending enthalpy"

3.5. Fluid Temperature in a Pipe Section of a Hydraulic System:

Any specific length of pipe in a hydraulic system can be regarded as an open thermal system undergoing a thermodynamic process as shown in Fig. (23). The amount of work W transferred into the pipe section due to the power loss caused by the pressure loss in the pipe at each end of the pipe section will be converted into heat. The energy transfer taking place across the boundaries of the system is the heat energy Q_{fc} which represents heat transferred from fluids to the pipe wall by the forced convection. The mass transfer occurs simultaneously with energy transfer flow with rate of Q_1 and temperature of T_{in} is entering the system in which outlet flow has temperature of T_f and rate of Q_2.

If a uniform fluid temperature is assumed within the open section of the pipe, and the change in fluid kinetic and potential energies for a specific hydraulic pipeline are negligible, the thermodynamic flow process associated with the section of the pipe can be studied by applying Equation (12) for open thermal system, i.e

$$\frac{dE}{dt}=Q_1-W+m_1h_1-m_2h_2 \qquad \textbf{(14)}$$

Substituting Equation (13) into Equation (14) to define the system internal energy by ignoring its kinetic and potential energies, yields:

$$Q_1 - W + m_1 h_1 - m_2 h_2 = \frac{d(M_f h)}{dt} \quad (15)$$

where M_f is the mass of fluid within the pipe section. From the definition of specific heat, it is known that the specific heat at constant volume (C_v) is defined as the heat required to raise the temperature of unit mass by one degree during a reversible constant volume process,and the specific heat at constant pressure (C_p) as the heat required to raise unit mass by one degree during a reversible constant pressure, these may be written as :

$$C_v = \left(\frac{dh}{dt}\right)_v$$
$$C_p = \left(\frac{dh}{dt}\right)_p \quad (16)$$

In the case being investigated, the effect of pressure on the internal energy is small and hence the heat exchange due to compression or expansion of the fluid is ignored. Therefore, a constant process is used to find the fluid enthalpy as

$$h = C_v T = C_p T \quad (17)$$

Using the above equation in equation (15), gives :

$$Q_1 - W + m_1 C_p T_{inlet} - m_2 C_p T_f = \frac{d(M_f C_p T_f)}{dt} \tag{18}$$

where T_f is assumed to be the uniform fluid temperature. The above equation can also be written as

$$Q_1 - W + m_1 C_p T_{inlet} - m_2 C_p T_f = M_f C_p \frac{dT_f}{dt} + m_f C_p T_f \tag{19}$$

In this equation, the mass flowrate at any instant of time is expressed $m_f = m_1 - m_2$. In practice, the mass flowrate through a pipe section is generally assumed to be constant. Therefore, the equation (19) can be re-written as:

$$M_f C_p \frac{dT_f}{dt} = Q_1 - W + m_f C_p (T_{inlet} - T_f) \tag{20}$$

The transient term ($M_f C_{pf}$ dT_f/dt) has always been ignored by others in the previous studies due to the only consideration of the steady state. This term, however, could be more significant for the transient fluid temperature changes in any hydraulic system.

In the above equation, the rate of heat flow Q_1 transferred from fluid to the pipe wall is defined through the following extensive considerations

3.6. Heat Transfer through a Pipe Wall in Hydraulic System:

Basic theoretical studies of heat transfer have been established for years. For instance, Holman (40), Yildiz and Mecati (41), Rogers and Mayhew (42), and Adam (43) have studied extensively in terms of this subject. For the benefit of the reader a brief summary is given for applying the theory to the analysis of the pipe wall temperature in a hydraulic system.

Heat transfer is that science which seeks to predict the energy transfer which may take place between material bodies as a result of a temperature difference. Thermodynamics teaches that this energy transfer is defined as heat. The science of heat transfer seeks not merely to explain how heat energy may be transferred, but also to predict the rate at which the exchange will take place under certain specified conditions.

In studying heat transfer it is customary to consider three distinct mechanisms of heat flow: conduction, convection, and radiation.

1) Conduction: Conduction is the transfer of heat from one part of a body at a higher temperature to another part of the same body at a lower temperature, or from one body at a higher temperature to another body at a lower temperature in physical contact with it.

2) Radiation: Radiation, or more correctly thermal radiation, is electromagnetic radiation emitted by a body by virtue of its temperature and at the expense of its internal energy. Thus, thermal radiation is of the same nature as visible light, x ray, the difference between them being in their wave lengths and the source of generation.

3) Convection: Convection relates to the transfer of heat from a boundary surface to a fluid in motion, or to the heat transfer across a flow plane within the interior of flowing fluid. If the fluid motion is induced by a pump, a blower, a fan, or some similar device, the process is called *forced convection*. If the fluid motion occurs as a result of the density difference produced by the heat transfer itself, the process is called *free or natural convection*.

It is generally possible to calculate the heat transfer by each mode separately and then sum up the results. In this way a complex problem can be resolved, and the relative importance of the various modes of heat transfer can be assessed.

In the present analysis the heat transfer radially through the pipe wall material only is discussed, and the effects of conduction, radiation and convection are investigated.

3.6.1 Forced Convection: The study of heat transfer by forced convection is usually concerned with the calculation of rates of heat exchange between fluid and solid boundaries. Because the fluid is flowing past the inside wall, a condition of forced convection applies to heat transfer across the wall as in Fig. (24) which shows a view of a pipe section containing the heat transfer at any instant of time. Q_1 expresses the rate of heat transferred from moving fluid to the pipe wall due to the forced convection arising from the movement of the fluid. It is known that the transfer of heat by forced convection can be written as :

$$Q_1 = h_f A_1 (T_f - T_1) \qquad \textbf{(21)}$$

where A_1 is the inside surface of the pipe section, h_f is the forced convection heat transfer coefficient which depends on the Nusselt number $N_u = h.L/k$. Here L is the characteristic linear dimension and k is the thermal conductivity of fluid.

The value of the Nusselt number for the forced convection in fully developed laminar flow through a tube is given in Refs (36) and (42) as follows:

(1) Nu=3.65 for laminar flow in a tube at constant wall temperature boundary condition;

(2) Nu= 4.36 for laminar flow in a tube at constant heat flux which is equivalent to constant temperature gradient at the wall

As the case being investigated is a tube which is heated by flow passing through it, case (2) applies in evaluating heat transfer coefficient by forced convection at the inner surface of the pipe wall.

When the flow is turbulent, Reynolds analogy and dimensional analysis can be applied. The conclusions of these analysis may be summarised by introducing the Nusselt number, Nu, where

$$N=KF\{(Pr)(Re)\} \tag{22}$$

In above, the Prandtl number Pr and the Reynolds number Re are defined as $Pr=V\,\mu/k$ and $Re=\rho\ V\ L/\mu$, V is the fluid velocity . Experiments can be performed in order to evaluate K and to determined the function F. From such experiments the Nusselt number for turbulent flow through tube has been given in Refs.

(42) and (44) as Nu= 0.0243 $Re^{0.8}$ $Pr^{0.4}$.

3.6.2. Conduction and Natural convection and Radiation: The heat transfer from pipe wall atmosphere Q_2 involves conduction, natural convection and radiation as shown in Fig. (25)

The heat flow rate over a length L of the pipe through the pipe wall from R_1 to R_2 by conduction can be written as :

$$Q_c=\frac{2\pi kL}{Ln(\frac{R_2}{R_1})}(T_1-T_2) \tag{23}$$

The heat transferred by radiation can be determined by the Stefan-Boltzmann Law. It is found that the emissive power of a black body is directly proportional to the fourth power of its absolute temperature i.e

$$E_B=\sigma T^4 \tag{24}$$

The energy emitted by a non-black body is given as :

$$E=\epsilon\sigma T^4 \tag{25}$$

where the Stefan-Boltzmann constant σ has the value of $56.7x10^{-9}$ in $W/m^2.K^4$ and ϵ is the emissivity of the body.

Similarly, the energy emitted by the black surroundings is given by

$$E_B=\sigma T^4 \tag{26}$$

Also it is noted that the energy emitted from one body, like the wall of a pipe section, is completely absorbed by the surroundings. Thus, the heat transferred from the body to its surroundings is given as

$$Q_r=\epsilon\sigma A_2(T_2^4-T_a^4) \tag{27}$$

where A_2 is the outside surface area of the pipe section. Hence, the flow transferred from the wall to its surrounding atmosphere by conduction,natural convection and radiation, Q_2, can be expressed as:

$$Q_2=\frac{2\pi kL}{Ln(\frac{R_2}{R_1})}(T_1-T_2)=h_aA_2(T_2-T_a)+\epsilon\sigma A_2(T_2^4-T_a^4) \tag{28}$$

where h_a is the heat transfer coefficient of free convection, T_1 and T_2 are the uniform inside and outside pipe wall temperatures respectively, T_a is the atmospheric temperature surrounding the pipe section. The value of ϵ for commonly used pipe wall material in hydraulics is given in Table (15) taken from Ref. (45). If the variation of T_a is assumed to be insignificant compared to other factors, then T_a can be regarded as a constant value during the system operation. The heat transfer coefficient by natural convection h_a can be similarly found from experiment. Experimental values for h_a are to be found in various handbooks. For instance, Ref. (40) gives a relation for a natural convection for a horizontal pipe as

$$h_a=1.32\left(\frac{\theta}{d}\right)^{\frac{1}{4}} \quad when \quad 10^{14}\langle Gr \langle 10^{9}$$
$$h_a=1.25\theta^{\frac{1}{3}} \quad when \quad 10^{9}\langle Gr \langle 10^{12} \qquad \mathbf{(29)}$$

In the above equation, h_a is in W/m.K, θ is the temperature difference in K and d is in m. The Grashof number Gr is defined as $Gr=(gL^3\ \beta\ \theta)\ /\ \nu$, where g is the gravitational acceleration, L is the linear dimension, β is the coefficient of cubical expansion and ν is the kinematic viscosity. Simplified formulae for natural convection for both the laminar and turbulent range in the different configurations are taken from Fig (22.16), Ref. (42) and summarised in Table (16).

3.7. Pipe Wall Temperature in Hydraulic System: In the steady state, the heat flow transferred into the pipe wall Q_1 must be equal to the heat transferred out, Q_2.

Under the unsteady condition, the difference between Q_1 and Q_2 must be equal to the increment of the pipe wall internal energy. In other words, if the pipe wall is regarded as a closed thermal system, the heat flow supplied to the pipe wall must be equal to the gain in internal energy. Hence;

$$Q_1 - Q_2 = \frac{dE}{dt} \tag{30}$$

From equations (13) and (17) :

$$\frac{dE}{dt} = \frac{d(M_p C_p T_2)}{dt} \tag{31}$$

where the inside and outside surface pipe wall temperature T_1 and T_2, are assumed to be uniform and both change at the same rate .Thus, equation (30) becomes:

$$M_p C_p \frac{dT_2}{dt} = Q_1 - Q_2 \tag{32}$$

where the constant mass M_p and the specific heat C_p of the pipe wall material are noted. The values of C_p for usual pipe wall materials and oil used in hydraulics are listed in Table (17) quoted from Ref. (45).

If the rate of heat flow through the pipe wall is assumed to be transferred under one dimensional conditions, then the heat radial flows through the pipe wall can be expressed by Fourier's Law as:

$$Q=\frac{kA_m(T_1-T_2)}{R_2-R_1} \tag{33}$$

where k is the thermal conductivity of the pipe wall in W/m.K. The values of k for often used pipe materials in hydraulics can be also obtained from Ref. (45) and a brief selection is given in Table (18). In the above equation, A_m is the logarithmic mean area and can be calculated from :

$$A_m=\frac{2\pi L(R_2-R_1)}{Ln\frac{R_2}{R_1}}=\frac{A_2-A_1}{Ln\frac{R_2}{R_1}} \tag{34}$$

where L is the length of the pipe section, R_1 and R_2 are the internal and external radiuses of the pipe section. This rate of heat transferred through the pipe wall by conduction must be equal to the rate of heat transferred from fluid to the pipe wall by forced convection at the boundary of the inner surface of the pipe section. From equation (21) and (33) we have

$$Q_1=h_fA_1(T_f-T_1)=\frac{kA_m(T_1-T_2)}{R_2-R_1} \tag{35}$$

Re-arranging the above equation, the inner pipe wall temperature can be written as

$$T_1=\frac{h_f A_1 T_f+\frac{kA_m}{R_2-R_1}T_2}{h_f A_1+\frac{kA_m}{R_2-R_1}} \qquad (36)$$

In summary the fluid temperature and pipe wall temperature in a section of the pipe at any instant of time can be determined by solving the following simultaneous equations:

$$h_a=1.32\left(\frac{T_2-T_1}{d}\right)^{\frac{1}{4}} \qquad (37)$$

$$Q_2=\frac{2\pi kL}{Ln\frac{R_2}{R_1}}(T_1-T_2)=h_a A_2(T_2-T_A)+\epsilon\sigma A_2(T_2^4-T_a^4) \qquad (38)$$

$$T_1=\frac{h_f A_1 T_f+\frac{kA_m}{R_2-R_1}T_2}{h_f A_1+\frac{kA_m}{R_2-R_1}} \qquad (39)$$

$$Q_1=h_f A_1(T_f-T_1) \qquad (40)$$

$$\frac{dT_2}{dt}=\frac{Q_1-Q_2}{M_p C_{pp}} \tag{41}$$

$$\frac{dT_f}{dt}=\frac{W-Q_1+m_f C_{pf}(T_{inlet}-T_f)}{M_f C_{pf}} \tag{42}$$

where C_{pp} is the specific heat of pipe wall. C_{pf} is the specific heat of fluid. ϵ is the emissivity and k is the thermal conductivity of the pipe wall material.

If the initial conditions are given, the equations can be used to calculate the fluid temperature T_f and the pipe wall temperature T_2 by applying the numerical integration techniques which are used in computer simulation.

3.8. A Mathematical Model for Temperature Analysis in an Open Hydraulic System: In an open hydraulic system in which a reservoir is used, there exists flow energy transfer between the fluid in the reservoir and in the loop. The fluid in the loop is in general considered to be at a different temperature than that of the fluid in the reservoir. If the loop and reservoir fluid temperatures are assumed to be uniform, the hydraulic system may be considered as two open thermal systems; the loop and the reservoir.

For the open thermal system of the loop, the transient fluid temperature can be predicted by applying equation (42), so that :

$$\frac{dT_f}{dt}=\frac{\sum W-\sum Q_1+m_f C_{pf}(T_{Res}-T_f)}{M_f C_{pf}} \tag{43}$$

In the above, $\sum W$ is the total losses, $\sum Q_1$ is the total heat flow transferred from fluid to the pipe walls by forced convection. M_f is the total mass of fluid in the system loop, m_f is the mass flowrate exchange between the loop and the reservoir T_{Res} is the uniform reservoir transient fluid temperature and T_f is the uniform loop transient fluid temperature. In order to find an appropriate theoretical value of the loop fluid temperature, a reservoir temperature model is necessary to determine the uniform fluid temperature in the reservoir.

3.9. An Analytical Model for Temperature of Reservoir in Hydraulic Systems:

In practice, the fluid and wall temperatures in a reservoir are different from one specific point to another depending on its position relative to the ports of the inlet and outlet pipelines, but these temperature differences are relatively small. Therefore, the fluid temperature and the reservoir wall temperature may be assumed to be uniform but different in the fluid and pipe wall. In reality there may be same difference in fluid temperature at different points of the circuit. This could be true for the pipe wall temperature. However, such difference will depend on the size and type of hydraulic circuit. If the difference substantial then same error will arise in the predicted results. Fig. (26) shows a reservoir in which the flow is considered to be an open thermal system.

The heat flow transferred through the vertical reservoir by conduction can be expressed as:

$$Q_{vrcond} = \frac{2k}{S}(A_2 + A_3)(T_f - T_{vwr}) \qquad \textbf{(44)}$$

and the heat flow transferred from the outside vertical reservoir wall to its surrounding atmosphere by natural convection is given by :

$$Q_{vrconv}=2h_{arv}(A_2+A_3)(T_{vwr}-T_a) \quad (45)$$

In the above equation, h_{arv} is in W/m.K, T_{wvr} and T_a are the outside vertical reservoir wall and surroundings atmosphere temperatures respectively ,in K.

According to Reference (40)

$$h_{arv}=1.42\left(\frac{T_{wvr}-T_a}{L}\right)^{\frac{1}{4}} \quad When \quad 10^4\langle Gr\langle 10^9$$
$$h_{arv}=1.31(T_{wvr}-Ta)^{\frac{1}{3}} \quad When \quad 10^9\langle Gr\langle 10^{12} \quad (46)$$

where Gr is defined as :

$$Gr=\frac{gL^3\beta\rho^2(T_{wvr}-T_a)}{\mu^2}=\frac{gL^3\beta(T_{wvr}-T_a)}{\nu^2} \quad (47)$$

The heat conducted through the vertical reservoir wall must be equal to the heat convected from outside of the reservoir wall into the ambient at temperature T_a. Therefore, an energy equation can be stated as

$$Q_{1R}=2\frac{k}{S}(A_2+A_3)(T_f-T_{wvr})=2h_{arv}(A_2+A_3)(T_{wvr}-T_a) \quad (48)$$

This equation can be rearranged as

$$Q_{1R}=\frac{(T_f-T_{wvr})}{\frac{S}{2k(A_2+A_3)}}=\frac{(T_{wvr}-T_a)}{\frac{1}{2h_{arv}(A_2+A_3)}} \quad (49)$$

which after eliminating T_{wvr} becomes

$$Q_{1R}=\frac{T_f-T_a}{\frac{S}{2k(A_2+A_3)}+\frac{1}{2h_{arv}(A_2+A_3)}} \quad (50)$$

where Q_{1R} is the heat flow by conduction and convection from the fluid into its surrounding atmosphere at vertical wall of the reservoir.

The heat conducted through the horizontal wall of the reservoir bottom can be expressed as :

$$Q_{hrcond}=\frac{k}{S}A_1(T_f-T_{rwp}) \quad (51)$$

and the heat flow transferred from the outside horizontal wall of the reservoir bottom to the surrounding atmosphere by natural convection can be expressed as

$$Q_{hrconv}=h_{arp}A_1(T_{WRP}-T_a) \quad (52)$$

In the same way, the heat flow by conduction and convection from the fluid into the surrounding atmosphere at the horizontal wall of the reservoir can be given as

$$Q_{2R}=\frac{T_f-T_a}{\frac{S}{kA_1}+\frac{1}{h_{arp}A_1}} \tag{53}$$

where S is the thickness of the reservoir wall in m and h_{arp} is the heat transfer coefficient by natural convection at the bottom of the reservoir wall.

Similarly, the heat flow transferred from the surface of fluid in the reservoir to the atmosphere by natural convection can be expressed as

$$Q_{3R}=h_{arp1}A_1(T_f-T_t) \tag{54}$$

where,

$$\begin{aligned} h_{arp1}&=1.42(T_f-T_t)^{\frac{1}{4}} \quad When \quad 10^4\langle Gr\langle 10^9 \\ h_{arp1}&=1.31(T_f-T_t)^{\frac{1}{3}} \quad When \quad 10^9\langle Gr\langle 10^{12} \end{aligned} \tag{55}$$

where T_t is the temperature of air inside the reservoir above the fluid surface. In the case when the top of the reservoir is open to the surrounding atmosphere, the value of T_t can be assumed to be the same as that of the surrounding atmospheric temperature, T_a . In the case being investigated, the air in the reservoir top is not directly open to its surrounding atmosphere but indirectly exchanges with outside air through an air breather. In this case, the value of T_t would be slightly higher than that of T_a. Although the value of T_t can be precisely determined by considering the air inside the reservoir as an open thermal system, it is rather complex and unnecessary. Assumption of a slightly higher but constant value of T_t usually suffices.

The vertical wall temperature T_{wrv} can be determined by solving equations (44) and (45) so that,

$$T_{wrv}=\frac{h_{arv}T_a+\frac{k}{S}T_f}{h_{arv}+\frac{k}{S}} \tag{56}$$

Similarly, the outside reservoir wall temperature at the bottom can be calculated by solving equations (51) and (52), so that

$$T_{wrp}=\frac{h_{arp}T_a+\frac{k}{S}T_f}{h_{arp}+\frac{k}{S}} \tag{57}$$

Therefore, fluid temperature in the reservoir can be expressed by the following equation which is another way of representing equation (20).

$$\frac{dT_{fr}}{dt}=\frac{W+m_1C_{pf}T_{inlet}-m_2C_{pf}T_f-Q_{1R}-Q_{2R}-Q_{3R}}{M_fC_{pf}} \tag{58}$$

If the initial parameters are given, the temperature transient of fluid in the reservoir can be calculated by numerical techniques.

In the above equation when the temperature in the reservoir is higher than that recommended ,it is not difficult to involve the characteristics of heat exchanger or cooler heat transfer since,there is a large volume of detailed information about the heat exchanger available in the literature. Therefore, it would be relatively easy to include the effect of a heat exchanger in the above equation just by including a substraction term of the amount of heat being dissipated through the heat exchanger during the operation of the system.

As it was mentioned before, some of the previous studies have ignored the effects of heat transfer processes. However, the effects of heat transfer by conduction, free convection and radiation from the vertical reservoir walls can be expressed as:

$$Q_{1RR} = \frac{T_f - T_a}{\frac{S}{2K(A_2 + A_3)} + \frac{1}{2h_{avr}(A_2 + A_3)}} + \epsilon\sigma(A_2 + A_3)(T_{wv}^4 - T_a^4)$$

The heat flow by convection, conduction and radiation from fluid into surrounding atmosphere at horizontal wall of reservoir bottom and from the surface of fluid in the reservoir can be written as follows:

$$Q_2RR = \frac{T_f - T_a}{\frac{S}{KA_1} + \frac{1}{h_{arp}A_1}} + \epsilon\sigma A_1(T_{wrp}^4 - T_a^4) \qquad \textbf{(60)}$$

$$Q_3RR = h_{arp_1}A_1(T_f - T_t) + \epsilon\sigma A_1(T_{wrp}^4 - T_t^4)$$

By replacing Q_{1R}, Q_{2R} and Q_{3R} in Equation 58 by Q_{1RR}, Q_{2RR} and Q_{3RR}, the fluid temperature in the reservoir with the consideration of heat transfer by conduction , convection and radiation can be determined.

3.10. The Calculation of Power Loss in Hydraulic System:

When fluid is pumped through a fluid power system a certain amount of the energy in the fluid is lost due to friction. Major losses occur as the fluid flows through pipes, hoses, and tubing, while minor losses occur in valves, fittings, bends, enlargements, contractions, and orifices. The procedure for the calculation of power losses in hydraulic system has been established for many years. Particularly, Sullivan (46), references (47) and (48), Banks and Banks (49), Frank (50) and Pinches (51) have written extensively on this subject. Major losses are calculated from a given length of pipe, minor losses, on the other hand, first must be converted to losses through an equivalent length of straight pipe using various experimental friction factors. To arrive at the total power loss for a circuit, the major and minor flow losses are combined and substituted in one of the formulae that determines the pressure drop and power losses associated with pumping the fluid.

The basic equation that governs viscous non-compressible flow in pipes is given by,

$$h_f = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \tag{62}$$

where h_f is the head loss required to pump the fluid, f is a dimensionless friction factor, L is the length of the pipe, D is the internal diameter of the pipe, v is the velocity of the fluid and g is the acceleration due to gravity. This is known as the Darcy-Weisbach formula for viscous flow Ref.(46). The value of the friction factor in the Darcy-Weisbach formula is largely determined by whether the flow is laminar or turbulent. for laminar flow, the friction factor has been determined experimentally to be :

$$f=\frac{64}{Re} \tag{63}$$

During laminar flow, the friction is relatively independent of the surface condition of the inside diameter of the pipe. When the flow is completely turbulent, the factor f is read from Fig. (28) taken from Ref. (46) by locating the place of intersection of the Reynolds number and the relative roughness of the tube surface, and then reading the friction value from the left or right margins.

The relative roughness of the pipe wall is computed as the dimensionless ratio of the absolute roughness ε to the inside diameter of the pipe D, that is,

$$Relative\ \ roughness = \frac{\varepsilon}{Di} \tag{64}$$

the value of ε can be read from Table (19) taken from Ref. (46).

Minor losses occur as the fluid flows through pipe fittings, valves, and bends. These losses are associated with the Bernoulli equation and defined as the number of velocity heads lost due to friction. The value of this factor is determined experimentally see Ref. (50), and then assigned a K-value for that fitting or pipe configuration. Head loss is thus computed from

$$h_{ff} = K\left(\frac{V^2}{2\,g}\right) \qquad (65)$$

where K value for pipe fittings can be computed from

$$K = f\left(\frac{L}{Di}\right) \qquad (66)$$

Fig (29) taken from Ref. (46) illustrates the value of K for different fittings.

It is evident that the head or pressure loss through a sudden enlargement, contraction, fittings, or valve is equivalent to the loss through some length of straight pipe. In addition, $h_f = hf_f$ where h_f is the head loss through restriction of some configuration and h_{ff} is the loss through an equivalent length of straight pipe. Expanding these from Eqs (62) and (65) we obtain

$$h_{ff}=K\left(\frac{Lv^2}{2g}\right) = f\left(\frac{L\ v^2}{D_i 2g}\right) \quad \text{(67)}$$
$$which\ L=D_i\left(\frac{K}{f}\right)$$

In summary, calculating the equivalent length and pressure drop for a hydraulic circuit involves determining the flow rate through the circuit plumbing, computing the Reynolds number, determining the friction factors and equivalent lengths for the pipe and fittings, and then combining the series of equivalent lengths into a total equivalent length. A final calculation determines the total pressure drop of fluid and then power losses in the system. Converting the head loss h_f to pressure drop we have

$$\Delta P= 9802\ x\ S_g\ x\ h_f \quad \text{(68)}$$

where ΔP is the pressure drop in N/m^2 and S_g is the specific gravity of the fluid. Finally, the power loss in the system can be calculated from

$$W = \frac{Q\ \Delta P}{600} \quad [kW] \quad \text{(69)}$$

where ΔP is in bar, Q is in l/min. and W is the power loss in the hydraulic system in kW. The change in the fluid density due to the change in pressure and temperature is neglected. However, the change in oil viscosity due to temperature change has been taken into account. Based on an empirical relationship between the kinematic viscosity and fluid temperature as reported by Mccoul et al (52) under the atmospheric pressure it is expressed as:

$$\nu = e^{10e^{10|A-B\log(t+273)|}} + 0.7$$

where constants A and B depend on the type of fluid. The empirical values of constants A and B have been used in the package on the basis of experimental data obtained from Ref. (53)

3.11. Numerical Technique for Temperature Calculation in Hydraulic System:

A first order Euler integration technique previously applied in references (54-56) has been used for calculating transient fluid temperature. This technique for temperature calculations was developed based on the fact that the temperature levels change relatively slowly in any hydraulic system and the relevant fluid properties can be regarded as constant over a period of time without significant loss of accuracy. The temperature transient equations are solved by using first order Euler integration, the temperatures are assumed to be constant for a fixed short period of time and then are updated. If the simulation time exceeds the constant temperature period, the fluid and pipe wall temperatures given by

Equations (41), (42) and (58) are updated by solving the equations as below

$$\Delta T_2 = \frac{dt_2}{dt} = \frac{Q_1 - Q_2}{M_p C_{pp}} \tag{71}$$

$$T_{2(i)} = T_{2(i-1)} + \Delta T_2 \, \delta t \tag{72}$$

$$\Delta T_{f(loop)} = \frac{dT_{f(loop)}}{dt} = \frac{W - Q_1 + m_f C_{pf}(T_{res} - T_f)}{M_f \, C_{pf}} \tag{73}$$

$$T_{f(loop)(i)} = T_{f(i-1)} + \Delta T_f \, \delta t \tag{74}$$

$$\Delta T_{fres} = \frac{dT_{fres}}{dt} = \frac{W + m_1 C_{pf} T_{inlet} - m_2 C_{pf} T_f - Q_{1R} - Q_{2R} - Q_{3R}}{M_f C_{pf}} \tag{75}$$

$$T_{fres(i)} = T_{f(i-1)} + \Delta T_{fres} \delta t \tag{76}$$

where $T_{2(i-1)}$, $T_{f(i-1)}$ and $T_{fres(i-1)}$ are the initial temperatures of pipe wall and fluid temperature in the loop and fluid temperature in the reservoir at t=0 respectively and δt is the time increment.

3.12. Computer Aided Design and Simulation of Fluid Power Systems:

Digital simulation for fluid power engineers is an extremely powerful design tool. A system or component can be tested at a very early stage and a bad solution can be easily abandoned. Furthermore, designers are encouraged to try new and unorthodox solutions since these can be evaluated at a low cost.

Mainly, the mathematical models have been developed to predict the transient fluid and pipe wall temperature of any hydraulic system. This program can be considered as a general simulation program especially suited to simulate fluid power systems. The fundamental aims of this programme are:

1- To provide a computational tool which can be used by an industrial engineer or researcher with little or no knowledge of computer systems.

2- To provide a facility which allows changes in component parameters or in circuit configuration to be carried out without further need for programming.

3- To provide a facility which allows adding new components without any modification in the programme.

4- To allow simple numerical examination of the behaviour of selected circuit parameters.

This program may be used either at the design stage, where the design of a proposed hydraulic system is investigated or else for a system already in operation.

This program consists of the following parts:

1-**The first part** contains detailed specification of pumps and hydraulic actuators including torque, speed, power and efficiency.

2- **The steady state flow analysis program:** This part predicts the steady state flows and pressures in the hydraulic system, determines the flow rate through the circuit and, whether flow is laminar, transitional or turbulent in order to apply appropriate resistance factors. This part can calculate the equivalent length of the system and then determine either total pressure drop or fluid horsepower loss resulting from the friction of the total equivalent length. Other factors such as the velocity at suction line becomes evident, and pipes can be resized as required to limit velocities at critical places in the circuit.

3-**Power losses:** This part calculates the power losses in the system including the power loss in pump, hydraulic motor, fitting and pipes.

4- **The hydraulic transient thermal analysis:** This part predicts the fluid temperature in the system loop and pipe wall and reservoir temperatures under unsteady condition. It also predicts the effects of system heat generation on the performance of an industrial hydraulic system.

3.12.1. Simulation Diagram: Fig (30) shows the complete system simulation diagram for the dynamic and thermal simulation of open hydraulic systems. The block HP represents the subroutine for the pump which determines the flowrate and input torque to the pump and the efficiencies. The hydraulic motor is simulated by the model HM which gives the torque at the output shaft of the motor, the speed, the total fluid flow lost from slippage in the high pressure port and the motor efficiencies. The hydraulic reservoir is modelled by block HR. The hydraulic dynamic performance of the system has been modelled by block DM through subroutines HC (hydraulic circuit model). This subroutine predicts the pressures in the system, the velocity of flow in each different diameter of the system, whether flow is laminar, transitional or turbulent in order to apply appropriate resistance factors. In addition, this subroutine can calculate the equivalent length of the system and then determine either total pressure drop of fluid horsepower resulting from the friction of the total equivalent length. However, the power losses in all components are summed in the model W_{total} . The power losses in the pipes resulting from friction is represented in block W_{pipes}, while the power losses in the pump, hydraulic motor and fittings are represented in blocks W_{pump} W_{motor} and $W_{fittings}$ respectively. The outside wall temperature of the vertical and at the bottom wall of the reservoir are represented in models T_{wrb} and T_{wrv} respectively. The model T_{wall} represents the outside wall temperature of pipes. The temperature of fluid in the reservoir and loop are represented by $T_{fres.}$ and $T_{floop.}$ respectively.

3.12.2. Data Acquisition for Simulation: A number of simulations have been carried out to predict the changes in system temperature due to the thermodynamic and heat transfer processes in the system. On the basis of the experimental test rig described in section (2.3) and (2.5), the user defined parameters for each model in the simulation block diagram in Fig. (30) have been obtained. The details of these parameters are listed in Table (21) to (47) .

Simulations were carried out for analyzing 60 minute duration experimental temperature test during the system loading. In this simplified approach for the temperature simulation the hydraulic system is regarded as an open thermal system. The system is also considered to be constructed by the same size of pipes and pipe wall material with an uniform fluid temperature throughout the system loop and reservoir. Based on the experimental test rig, the high pressure and return lines are taken as 0.75 inch (19 mm) internal bore single wired hydraulic hoses and the external diameter of the hose is measured as 29 mm. The total length of the pipes is taken as 7 m. The suction line has 1.5 inch (38.1 mm) internal diameter and the external diameter of the hose suction line is measured as (47.5 mm). The total length of suction line is taken as 1.5 m. The type of oil used in the simulation is SHELL TELLUS-37.

Using the data provided in Table (17) and (18), the specific heat and thermal conductivity for oil, are taken as 1964 (J/Kg.K) and 0.144 (W/m.°C) respectively. Fig (12) shows a typical single wire braid hydraulic hose constructed by synthetic rubber tube, single braid reinforcement and synthetic rubber cover. The specific heat and thermal conductivity of such a hose has been taken as if it was made from pure rubber, ie 2010 (J/Kg.K) and (0.013 W/m °C). The density of pure rubber 1100 (Kg/m^3) has been obtained from the data taken from Ref (41). The emissivity of the hose is taken as the value of natural black rubber, ie 0.9. The data in Table (34) for initial fluid, pipe wall and surrounding air temperatures before the system loading have been evaluated from consideration of experimental and test conditions.

3.12.3. Simulation Results and Discussion: Using the input data defined above, the simulations using Fig. (30) have been carried out to predict the theoretical fluid and pipe wall temperatures in the system, and also to investigate the effects of various parameters on the system performance indicated by the level of temperature In these simulations, the electric motor was running steadily at 1500 r.p.m.

According to the mathematical model of the system thermodynamic process, the fluid and pipe wall temperatures of the system should be affected by a number of parameters. A clear appreciation can be obtained from a close examination of a range of simulation results with its governing equations. The analysis of fluid and pipe wall temperatures in the open hydraulic drive mixer system is given as an example. Similar analysis can be considered to examine fluid and pipe wall temperature results in the other system simulations.

The power loss of the components during the system loading is the only heat generation source to cause the increase of fluid temperature in the hydraulic systems. Fig (31) shows the effects of power losses on the fluid temperature, this figure shows the increase in the simulated fluid temperature in response to the increase of the total power loss in the system. In this specific case of the system simulation, when the power loss exceeds the value of 160 Watts, the fluid temperature in the loop will rise above the recommended maximum fluid temperature 60 °C. In order to ensure an appropriate performance of the system, a cooler is needed to be incorporated in the system when total system power loss is higher than 160 Watts. These simulated results have shown that fluid temperatures have two distinct rates of increase: under the simulation condition defined, a fast increase is shown within the first half an hour of the system loading, during the second half of the system loading, the rate of increase in fluid temperature is much slower. Similarly, Fig. (32) suggests that the increase in power losses has a significant effect on the pipe wall temperature which increase following the same pattern as for the fluid temperature.

It has been pointed out that the fluid temperature is also dependent on the fluid mass M_f and heat dissipated through the pipeline of the system Q_1. When the total pipe length of the system is changed, Q_1 due to the corresponding change in the pipe internal surface area and M_f will be affected and so the fluid temperature also changes.

The simulation results in Fig. (33) shows the effects of the pipe length on fluid temperature and Fig. (34) shows the effect of the pipe length on the outside hose wall temperature. These figures demonstrate that when the total pipe length is increased, the fluid and outside wall temperatures of the hose are also increased. Similarly, when the inside diameter of pipe is changed, the fluid mass M_f in the pipe line is changed, hence the fluid temperature would be affected as suggested in the simulation results in Fig. (35). Similarly, the change in the internal pipe diameter shows influence on the outside pipe wall temperature as shown in Fig. (36). The size of the pipe outside diameter has more significant influence on the pipe wall temperature as shown in Fig. (37) This is because the outside surface area of the pipe wall is a dominant factor in determining the heat flow transferred from the outside pipe wall to atmosphere. The fluid temperature is not much affected by the variation of the outside pipe diameter, since the outside pipe diameter has little effects on the heat flow transferred from the fluid to the inside pipe wall, Q_1. The simulation results of the effects of the variation of the outside pipe diameter on fluid temperature is show in Fig. (38).

The type of the pipe wall material plays an important role in determining the temperature distribution, since the values of specific heat, density and thermal conductivity are dependant on the pipe wall material. As expressed in Equation (35), the heat transferred within the pipe wall by conduction is affected by the thermal conductivity k of the pipe wall material. Fig. (39) shows the effect of thermal conductivity of the pipe wall material on fluid temperature. The results show that the reduction of the value of k will increase fluid temperature in the loop due to corresponding reduction in Q_1. Similarly, the value of pipe wall k has considerable influence on the outside pipe wall temperature as shown in the simulation results in Fig. (40). The specific heat of pipe wall material affects the rate of increase in the pipe wall temperature as indicated in Equation (42). This effect is well reflected in the simulation results shown in Fig. (41). Since fluid temperature distribution is affected by the pipe wall temperatures, Fig.(42) shows a similar change on fluid temperature with the variation of the pipe wall specific heat. For the same reason, the mass or density of the pipe wall has influence on the pipe wall and fluid temperatures as shown in Fig. (43) and (44) respectively which show similar trends as those in Figs (41) and (42).

The difference between the pipe wall outside and surrounding atmospheric temperatures is greatly influenced by the effects of pipe wall heat radiation as expressed in Equation (38) by the emissivity of the pipe wall material ϵ . This is due to the heat flow transferred from outside pipe wall to its surrounding atmosphere by the natural convection and radiation. The radiation effect can be as important as the natural convection when the heat transfer coefficient by natural convection h_a is relatively small. The effects of the emissivity of the pipe wall material on the pipe wall and fluid temperatures are demonstrated in the simulation results shown in Fig. (45) and Fig.

(46) respectively.

If the hydraulic hose pipeline in the system is replaced by the commonly used mild steel or copper pipes, the input simulation data in Table (34) will be changed to the data in Table (35) or (36) respectively. A large difference between the outside pipe wall temperatures of steel or copper pipes and rubber hose is shown in simulation results in Fig (47) and (48) as it would be expected due to a big difference between the thermal conductivities of metallic and non-metallic materials as suggested in Table (35) (for steel k=60 W/m.K; copper: k=386 W/m.K; rubber: k=0.013 W/m.K). An interesting phenomenon shown in the simulation fluid temperatures in Figs. (47) and(48) is that fluid temperature in the system with the steel or copper pipeline is higher than it in the system with the hydraulic hose. These simulation fluid temperatures have resulted when the effects of radiation and conduction are taken into account. Despite the fact that the thermal conductivities of steel and copper pipes are much higher than that of rubber hose, the emissivity of steel and copper are lower than of rubber hose as data indicated in Table (15) (for mild steel ϵ=0.32; polished copper: ϵ=0.018; rubber: ϵ =0.94). This comparison of simulation fluid temperature results should question the widely accepted assumption in many text books which ignore the radiation effect under 200 °C.

Further more, Figs (49) and (50) show the comparison of the effects of heat transfer processes. In these Figures the effects of convection, conduction and radiation in the loop and the reservoir have been taken into account. These simulated results show that by ignoring the effect of radiation from the reservoir, the transient temperature in the hydraulic system would be 2.5 °C to 3.5 °C higher in the system which could give rise to error in estimating the temperature distribution in hydraulic systems.

3.13. Temperature Simulations Results for the Reservoir in the Open Hydraulic Drive Mixer System:

A reservoir is required in the majority of hydraulic systems. When it is used, the hydraulic system no longer can be considered as a closed thermal system because the exchange of the flow energies of the fluids in the loop and in the reservoir. An analytical model has been developed in Section (3.7) and (3.8). The principle of the model is to consider the hydraulic system as two open thermal systems; the loop and the reservoir. An assumption has been made to regard the temperatures of the fluids in the loop and reservoir as two separate uniform values. The computer program developed for predicting these two fluid temperatures can be used for simulation of the open hydraulic drive mixer system.

A number of simulations have been carried out to investigate the effect of reservoir geometry on the loop fluid temperature.

The fluid height in the reservoir determines the vertical side areas of the reservoir (A_2 and A_3). as indicated in Equations (44) and (50), the amount of heat flow dissipated through the reservoir side walls Q_{1R} is affected by these areas of the reservoir. This value of heat dissipated through the side walls of the reservoir will affect fluid temperature inside the reservoir as expressed by Equation (58). Moreover, the fluid temperature inside the reservoir and in the system loop are dependent on each other, so the change in the fluid height in the reservoir will result in a corresponding change in the fluid temperature in the system loop. A clear appreciation can be obtained from a collection of simulation results in Figs (51) and (53) which show the effects of fluid height in the reservoir on the fluid and outside hose wall temperatures in the system loop. These figures suggest that any change of the fluid height in the reservoir will not only affect the mass of fluid in the reservoir, which is one of the factors to determine the change rate of the fluid temperature in reservoir, but also change the amount of heat dissipated through the vertical side wall of the reservoir. As a consequence, the steady level of the fluid temperature in the reservoir is also affected by this change of the fluid height. these figures show higher value of H_t for lower value of temperature.

For the same reason, the size of the reservoir base has its influence both on the fluid temperature of the reservoir and the system loop. The simulation results in Figs. (53)and (54) and Figs. (55) and (56) predict the effects of the length and width of the reservoir on the fluid and outside hose wall temperatures in the system loop respectively. These figures are different in trend from those in Figs (51) and (52) which indicate that the lower the values of L_1 and S, higher the values of temperatures. It can be seen that the width of the reservoir is less effective than the height and length of the base of the reservoir on the temperature distribution. Moreover, the simulated results have shown that the effect of the thickness and type of material of the reservoir have very little influence on the fluid and outside hose wall temperatures as shown in Figs. (57), (58), (59) and (60). The fluid temperature in the loop is also affected by the value of the flow returning to the reservoir as expressed in Equation (43) in terms of mass flowrate m_f. As already stated in the temperature analysis model for an open hydraulic system (using a reservoir), the whole hydraulic system can be considered as a combination of two open thermal systems: the loop and the reservoir. The energy transfer between the loop and reservoir thermal systems is expressed in Equation (43) in terms of m_f C_{pf} $(T_{res} - T_f)$, where the mass flowrate $m_f = \rho Q$, T_{res} is the uniform fluid temperature in the reservoir, the first term ρ Q C_{pf} T_{res} represents the flow energy received by the loop thermal system while the second term ρ Q C_{pf} T_f indicates the flow energy from the loop to the reservoir. The effect of the rate of the flow between the loop and reservoir on the loop temperature is demonstrated in the simulation results in Fig. (61). As can be seen, when no flow returns to the reservoir, the increase of fluid temperature in the system loop during the system loading is identical to the fluid temperature of the open hydraulic drive mixer system discussed in the last section. As this flowrate rises, the flow energy transferred from the system loop to the reservoir increases and so the loop temperature will reduce. Meanwhile, the loop fluid temperature will be affected by the rise of the reservoir temperature due to this increase of the flow.

Another feature of these temperature results is that when the flowrate returning to the reservoir exceed about 0.5 l/min, the reduction of the loop temperature with the increase of the system flowrate to the reservoir gradually becomes insignificant. This phenomenon may be explained in the following way: when the flowrate circulating in the reservoir exceeds a value at which an equilibrium of heat dissipation process is established the value of the heat dissipated through the pipes of the system loop and the value of flow energy transferred to the reservoir will become constant. In this case, any further increase of the flowrate will not affect the final value of the flow energy in terms of $\rho\ Q\ C_{pf}\ (T_{res} - T_f)$ but introduce a small reduction in the fluid temperature difference between the loop and the reservoir $(T_{res}-T_f)$. Hence, once the equilibrium of the proportion of heat dissipated through the pipes of the loop and the reservoir is established, the loop temperature will not be affected by any further change in the system flowrate returning to the reservoir as indicated in Fig. (61). Furthermore, the rate of the heat dissipated through the walls of the reservoir which is calculated using the reservoir temperature mathematical model with the defined parameters of the reservoir size, together with the power loss and flow energy transferred from the loop $\rho\ Q\ C_{pf}(T_{res}-T_f)$ have resulted the gradual fluid temperature increase as shown in Fig. (61). When all system flow returns to the reservoir as in the open hydraulic system, since the values of the flow energy and the heat dissipation are the same with the above case, the rise of the power led to a small increase in the reservoir fluid temperature.

The proportion of heat dissipation through the loop pipes is expressed by ΣQ_l in Equation (43). For instance, in the case of the open hydraulic drive mixer system, the simulation results of the heat dissipated through the pipes is shown in Fig. (62). The rest of the loop heat dissipated in the reservoir has led to a small difference between the fluid temperature of the loop and that of the reservoir as shown in comparison of these two temperatures in Fig. (62). Fig. (63) shows the comparison of simulated results of loop and reservoir fluid temperature. This figure illustrates that the heat transfer by radiation must be considered to get more accurate results in terms of the temperature distribution in hydraulic systems. A further investigation into the temperature performance during the system loading may be carried out by considering a big increase in the power loss to the value of 1200 Watts. As shown in comparison to the simulation results for the fluid temperature in the loop and the reservoir in Fig. (64), although the increase in the power loss has increased the rate of change and steady value of the fluid temperatures of the loop and the reservoir, nevertheless the difference between these two fluid temperatures is not as big as it otherwise might have been expected.

From the analysis of the above simulation results, it has been found that the heat dissipation through the pipes of a hydraulic system plays an important part in determining the temperature distributions in the loop and the reservoir. The assumption which ignores this vital thermodynamics and heat transfer processes during the loading of a hydraulic system may lead to greater errors in predicting both the fluid temperature in the loop and that in the reservoir than had been previously thought.

In order to investigate these temperature results obtained by simulation, a range of experimental test has been carried out in an open hydraulic drive mixer system to measure the fluid temperature of the loop and the reservoir in addition to measuring temperature of the pipes wall. The experimental temperature recordings and their comparisons with simulation results will be discussed in the following two chapters.

Normal Emissivity of Various Surfaces			
Surface		T [°C]	Emissivity ϵ
Iron	Polished	425-1025	0.14-0.38
	Oxidized	100	0.77
	Cast Iron	22	0.44
Rubber	Hard	23	0.94
	Soft, grey	23	0.86
Steel	Mild Steel	45-1065	0.18-0.32
Copper	Polished	80	0.018
	Polished	100	0.023
	Black Oxidized	37	0.78
Water		0-100	0.95-0.96

Table (15) Normal emissivity for several used material in hydraulics taken from table (C-1) of ref. (41).

Type of shape	h [W/m^2 °C]	
	Laminar or Transition	Turbulent
Horizontal cylinder of Diameter d	$10^4 < Gr < 10^9$	$10^9 < Gr < 10^{12}$
	$h=1.32(\theta/d)^{1/4}$	$h=1.25\ \theta^{1/3}$
Vertical Plate of height L, or Cylinder of Large Diameter, d	$10^4 < Gr < 10^9$	$10^9 < Gr < 10^{12}$
	$h=1.42(\theta/L=d)^{1/4}$	$h=1.31\ \theta^{1/3}$
Square Plate 1 X 1: heated Plate facing up or Cooled Plate facing Down	$10^5 < Gr < 2x10^7$	$2x10^7 < Gr < 3x10^{10}$
	$h=1.32(\theta/L)^{1/4}$	$h=1.52\ \theta^{1/3}$
Square Plate 1 x 1: Cooled Plate Facing up or Heated Plate Facing Down	$3x10^5 < Gr < 3x10^{10}$	
	$h=0.59(\theta/L)^{1/4}$	

Table(16) Simplified equations for free convection heat transfer coefficients in air at atmospheric pressure taken from Fig(22.17) ref. (42)

Specific Heat of Several Material			
Material		T [°C]	Specific Heat [KJ/Kg.°C]
Iron	Pure	20	0.452
	Wrought 5% C	20	0.46
Steel	Carbon C≈1.5%	20	0.486
	Mild	20	0.487
	Chrome	20	0.452
Copper	Pure	20	0.3831
Rubber		30	2.010
Machine Oil		40	1.964
Water		40	4.1784
Sand		30	0.800

Table (17) Specific heat for several used material in hydraulics system taken from tables (B-4) and (B-5) of ref (41)

Thermal Conductivity of Several Surfaces			
Material		T [°C]	Thermal Conductivity k [W/m. °C]
Iron	Pure	20	73
		100	76
		300	55
	Wrought 0.5% C	20	59
Steel	Carbon 1.5 %	20	36
	Chrome	20	61
	Mild	40	60
Copper	Pure	20	386
		100	379
Rubber	Soft	30	0.012
	Hard	30	0.013
Machine Oil		40	0.144
Water		40	0.628

Tale (18) Thermal conductivity for several used materials in hydraulic system taken from tables (B-4) and (B-5) of Ref. (41)

Absolute Roughness of Several Surfaces		
Material	Roughness ε	Unit
Drawn Tubing	0.000 0015	m
Steel Pipe	0.000 046	m
Galvanized Iron	0.000 2	m
Cast Iron	0.000 26	m
Rubber	0.000 30 to 0.0030	m

Table (19) Absolute roughness of commercially pipe and tubing taken from ref. (46)

Density of Several Material			
Material		T [°C]	ρ [Kg/m^3]
Iron	Pure	20	7897
	Wrought 0.5% C	20	7849
Steel	Mild	30	7833
Rubber	Soft	30	1100
	Hard	30	1190
Machine Oil		40	876.05
Water		20	1000

Table (20) Density for several used materials in hydraulic systems taken from table (B-2), (B-3) and (B-4) of ref. (41)

Fluid Properties			
No.	Parameters	Data	Unit
1	Oil Type	Shell TELLUS 37	
2	Oil Temperature	15	°C
3	Density	872	Kg/m^3
4	Kinematic Viscosity	150	cSt
5	Absolute Viscosity	99	cP

Table (21) Parametric data used in simulation for fluid properties

Component No 1: Electric Motor			
No.	Input Parameters	Data	Unit
1	Speed	1500	rpm
2	Power	8.2	Kw
3	Torque	150	Nm

Table (22) Parametric data used in simulation for the electric motor

Component No 2: Hydraulic Pump			
No	Input Parameters	Data	Unit
1	Speed	1500	rpm
2	Type: External Gear		
	Flowrate	36	L/min
	Displacement	24	Cm^3/rev
	Continuous Maximum Pressure	220	Bar

Table (23) Parametric data used in simulation for hydraulic pump

Component No 3: Hydraulic Motor			
No	Input Parameters	Data	Unit
1	type:External Gear with External Drain		
2	Maximum Speed	2800	rpm
3	Flowrate	54	L/min
4	Displacement	36	Cm^3/rev
5	Continuous Maximum Pressure	220	Bar

Table (24) Parametric data used in simulation for hydraulic motor

Component No 4:High Pressure Hose before Directional Control Valve			
No.	Input Parameters	Data	Unit
1	Pipe Internal Diameter	19.05	mm
2	Pipe Length	3.5	m
3	Pipe Volume	0.992	Litres
4	Air Saturation Pressure	0	Bar

Table (25) Parametric data used in simulation for high pressure pipeline before directional control valve

Component No 5:High Pressure Hose After Directional Control Valve			
No.	Input Parameters	Data	Unit
1	Pipe Internal Diameter	12.7	mm
2	Pipe Length	0.3	m
3	Pipe Volume	0.038	Litres
4	Air Saturation Pressure	0	Bar

Table (26) Parametric data used in simulation for high pressure pipeline after directional control valve

Component No 6:Pressure Relief Valve			
No	Input Parameters	Data	Unit
1	Flow	40	L/min
2	Pressure Range	20 - 140	Bar
3	Pressure Drop	3	Bar
4	Operating Temperature	-30 to +90	°C

Table (27) Parametric data used in simulation for pressure relief valve

Component No 7: Flow Control Valve			
No	Input Parameters	Data	Unit
1	Flow	40	L/min
2	Pressure Range	20 - 140	Bar
3	Pressure Drop	1	Bar
4	Operating Temperature	-30 to +90	°C

Table (28) Parametric data used in simulation for flow control valve

Component No 8: Check Valve			
No	Input Parameters	Data	Unit
1	Flow	40	L/min
2	Pressure Range	20 - 140	Bar
3	Pressure Drop	2	Bar
4	Operating Temperature	-30 to + 90	°C

Table (29) Parametric data used in simulation for check valve

Component No 9: Directional Control Valve			
No	Input Parameters	Data	Unit
1	Flow	40	L/min
2	Pressure Range	20 - 140	Bar
3	Pressure Drop	4	Bar
4	Operating Temperature	-30 to +90	°C

Table (30) Parametric data used in simulation for directional control valve

Component No 10: Return Line			
No	Input Parameters	Data	Unit
1	Pipe Internal Diameter	19.05	mm
2	Pipe Length	2	m
3	Air Saturation Pressure	0	Bar

Table (31) Parametric Data Used in Simulation for Pressure Return Line

Component No 11: Suction Line			
No	Input Parameters	Data	Unit
1	Pipe Internal Diameter	38.1	mm
2	Pipe Length	1.5	m
3	Air Saturation Pressure	0	Bar

Table (32) Parametric data used in simulation for pressure suction line

Component No 12: Mixer				
No	Input Parameters	Data		Unit
1	Diameter of Mixer	300		mm
2	Initial Angular Velocity	0		rev/min
3	Number of Blades	4		
4	Density of Substances Surrounding the Mixer	Granular Polymer	800	Kg./m^3
		Sand	1515	
		Sandstone	2300	

Table (33) Parametric Data Used in Simulation for Mixer

Component No: 13 Open System Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	29	mm
3	Length of Pipe	7	m
4	Initial Fluid Temp.	297	°K
5	Initial Outside Pipe	295	°K
6	Initial Surrounding	293	°K
7	Type of Oil Used	Shell TELLUS 37	
8	Conductivity of Pipe Wall Material	0.013	W/m.K
9	Conductivity of Fluid	0.144	W/m.K
10	Fluid Specific Heat	1964	J/(Kg.°K)
11	Pipe Wall Specific Heat	2010	J/(Kg.°K)
12	Emissivity of Pipe Section	0.94	
13	Density of Pipe Wall	1100	Kg./m^3

Table (34) Parametric data used in simulation for temperature and fluid property calculation in the open hydraulic drive mixer system with hydraulic hoses.

Component No: 14 Open System Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	25	mm
3	Length of Pipe Section	7	m
4	Initial Fluid Temperature	297	°K
5	Initial Outside Pipe	295	°K
6	Initial Surrounding Atmosphere Temperature	293	°K
7	Type of Oil Used	TELLUS37	
8	Conductivity of Pipe Wall Material	386	W/m.K
9	Conductivity of Fluid	0.144	W/m.K
10	Fluid Specific Heat	1964	J/(Kg.°K)
11	Pipe Wall Specific Heat	3831	J/(Kg.°K)
12	Emissivity of Pipe Wall	0.018	
13	Density of Pipe Wall	8954	Kg./m^3

Table (35) Parametric data used in simulation for temperature and fluid property calculation in the open hydraulic drive mixer system with copper pipes.

Component No: 15 Open System Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	25	mm
3	Length of Pipe Section	7	m
4	Initial Fluid Temperature	297	°K
5	Initial Outside Pipe Wall Temperature	295	°K
6	Initial Surrounding Atmosphere Temperature	293	°K
7	Type of Oil Used	SHELL	
8	Conductivity of Pipe Wall Material	36	W/m.K
9	Conductivity of Fluid	0.144	W/m.K
10	Fluid Specific Heat	1964	J/(Kg.°K)
11	Pipe Wall Specific Heat	487	J/(Kg.°K)
12	Emissivity of Pipe Section	0.32	
13	Density of Pipe Wall	7833	Kg./m^3

Table (36) Parametric data used in simulation for temperature and fluid property calculation in the open hydraulic drive mixer system with steel pipes.

Component No: 16 Pipe Wall Temperature Model			
No	Input Parameter	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	29	mm
3	Length of Pipe Section	3.5	m
4	Density of Pipe Wall Material	1100	Kg/m^3
5	Initial Outside Pipe Wall Temperature	295	°K
6	Initial Surrounding Atmosphere Temperature	293	°K
7	Pipe Wall Specific Heat	2010	J/(Kg.°K)
8	Conductivity of Pipe Wall Material	0.012	W/m.K
9	Conductivity of Fluid	0.144	W/m.K
10	Emissivity of Pipe Wall Material	0.86	

Table (37) Parametric data used in simulation for the single wired hydraulic hose in the High pressure pipeline before the directional control valve.

Component No: 17 Pipe Wall Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	12.7	mm
2	External Pipe Diameter	22	mm
3	Length of Pipe Section	0.3	m
4	Density of Pipe Wall Material	1100	Kg/m^3
5	Initial Outside Pipe Wall Temperature	295	°K
6	Initial Surrounding Atmosphere Temperature	293	°K
7	Pipe Wall Specific Heat	2010	J/(Kg.°K)
8	Conductivity of Pipe Wall Material	0.012	W/m.K
9	Conductivity of Fluid	0.144	W/m.K
10	Emissivity of Pipe Wall Material	0.86	

Table (38) Parametric data used in simulation for the single wired hydraulic hose in the high pressure pipeline after the directional control valve.

Component No: 18 Pipe Wall Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	29	mm
3	Length of Pipe Section	2	m
4	Density of Pipe Wall Material	1100	Kg/m^3
5	Initial Outside Pipe Wall Temperature	295	°K
6	Initial Surrounding Atmosphere Temperature	293	°K
7	Pipe Wall Specific Heat	2010	J/(Kg.°K)
8	Conductivity of Pipe Wall Material	0.012	W/m.K
9	Conductivity of Fluid	0.144	W/m.K
10	Emissivity of Pipe Wall Material	0.86	

Table (39) Parametric data used in simulation for the single wired hydraulic hose at the inlet of the hydraulic motor.

Component No: 19 Pipe Wall Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	29	mm
3	Length of Pipe Section	2	m
4	Density of Pipe Wall Material	1100	Kg/m^3
5	Initial Outside Pipe Wall Temperature	295	°K
6	Initial Surrounding Atmosphere Temperature	293	°K
7	Pipe Wall Specific Heat	2010	J/(Kg.°K)
8	Conductivity of Pipe Wall Material	0.012	W/m.K
9	Conductivity of Fluid	0.144	W/m.K
10	Emissivity of Pipe Wall Material	0.86	

Table (40) Parametric data used in simulation for the single wired hydraulic hose in the return line before the directional control valve.

Component No: 20 Pipe Wall Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	38.1	mm
2	External Pipe Diameter	47.5	mm
3	Length of Pipe Section	1.5	m
4	Density of Pipe Wall Material	1100	Kg/m^3
5	Initial Outside Pipe Wall Temperature	295	°K
6	Initial Surrounding Atmosphere Temperature	293	°K
7	Pipe Wall Specific Heat	2010	J/(Kg.°K)
8	Conductivity of Pipe Wall Material	0.012	W/m.K
9	Conductivity of Fluid	0.144	W/m.K
10	Emissivity of Pipe Wall Material	0.86	

Table (41) Parametric data used in simulation for the single wired hydraulic hose in the suction line.

Component No: 21 Open System Fluid Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	29	mm
3	Length of Pipe Section	7	m
4	Fluid Specific Heat	1964	J/(Kg.°K)
5	Initial Fluid Temperature	297	°K
6	Type of Oil Used	SHELL	

Table (42) Parametric data used in simulation for fluid temperature and property calculation in the open Hydraulic drive mixer system using the developed mathematical model.

Component No: 22 Reservoir Fluid Temperature Model			
No.	Input Parameters	Data	Unit
1	Length of Reservoir Base	300	mm
2	Width of Reservoir Base	105	mm
3	Thickness of Reservoir Wall	5	mm
4	Fluid Height in Reservoir	290	m
5	Fluid Specific Heat	1964	J/(Kg.°K)
6	Conductivity of Reservoir Wall	36	W/m.°K
7	Initial Surrounding Atmosphere Temperature	293	°K
8	Initial Outside Reservoir Vertical Wall Temperature	295	°K
9	Initial Outside Reservoir Bottom Wall Temperature	295.5	°K
10	Conductivity of Fluid	0.144	W/m.K
11	Type of Oil Used	SHELL	

Table (43) Parametric data used in simulation for the fluid temperature calculation in the reservoir of the open hydraulic drive mixer system.

Component No: 23 Pipe Fluid Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	29	mm
3	Length of Pipe Section	1.5	m
4	Fluid Specific Heat	1964	J/(Kg.°K)
5	Initial Fluid Temperature	297	°K
6	Type of Oil Used	SHELL	

Table (44) Parametric data used in simulation for fluid temperature and property model in the first part of the high pressure line.

Component No: 24 Pipe Fluid Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	29	mm
3	Length of Pipe Section	2	m
4	Fluid Specific Heat	1964	J/(Kg.°K)
5	Initial Fluid Temperature	297	°K
6	Type of Oil Used	SHELL	

Table (45) Parametric data used in simulation for fluid temperature and property model in the second part of the high pressure line.

Component No: 25 Pipe Fluid Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	19.05	mm
2	External Pipe Diameter	29	mm
3	Length of Pipe Section	2	m
4	Fluid Specific Heat	1964	J/(Kg.°K)
5	Initial Fluid Temperature	297	°K
6	Type of Oil Used	SHELL	

Table (46) Parametric data used in simulation for fluid temperature and property model in the return line

Component No: 26 Pipe Fluid Temperature Model			
No.	Input Parameters	Data	Unit
1	Internal Pipe Diameter	38.1	mm
2	External Pipe Diameter	47.5	mm
3	Length of Pipe Section	1.5	m
4	Fluid Specific Heat	1964	J/(Kg.°K)
5	Initial Fluid Temperature	297	°K
6	Type of Oil Used	SHELL	

Table (47) Parametric data used in simulation for fluid temperature and property model in the suction line

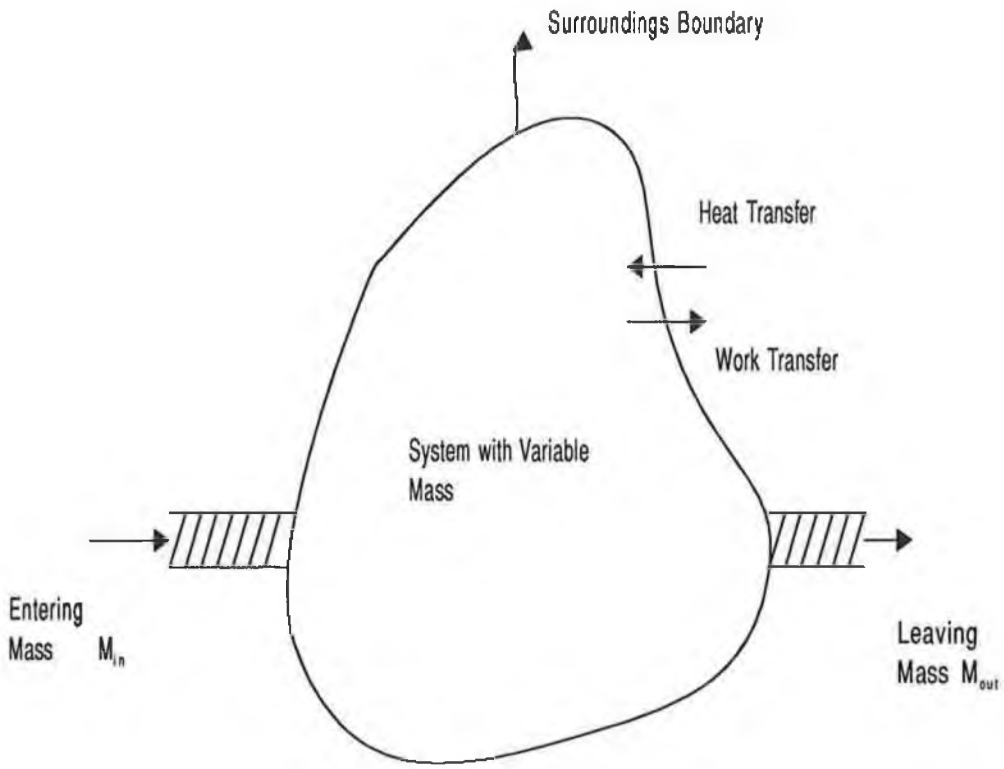


Fig (21) Open thermal system undergoing an imaginary non-flow process

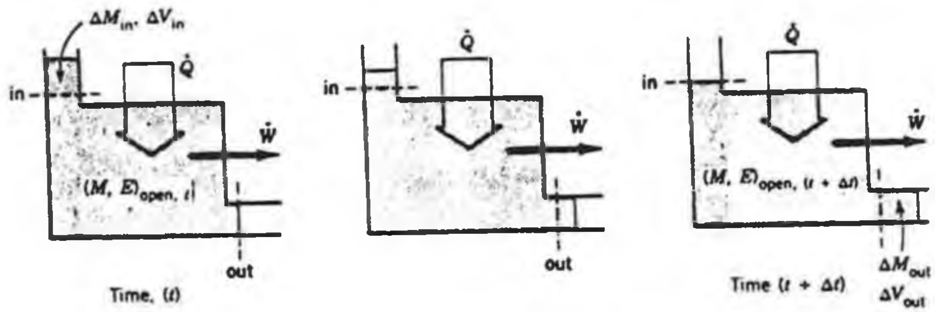


Fig (22) The flow of a closed system (the shaded area) through the space occupied by an open system, conversion of the first-law statement for closed systems into a statement valid for open systems.

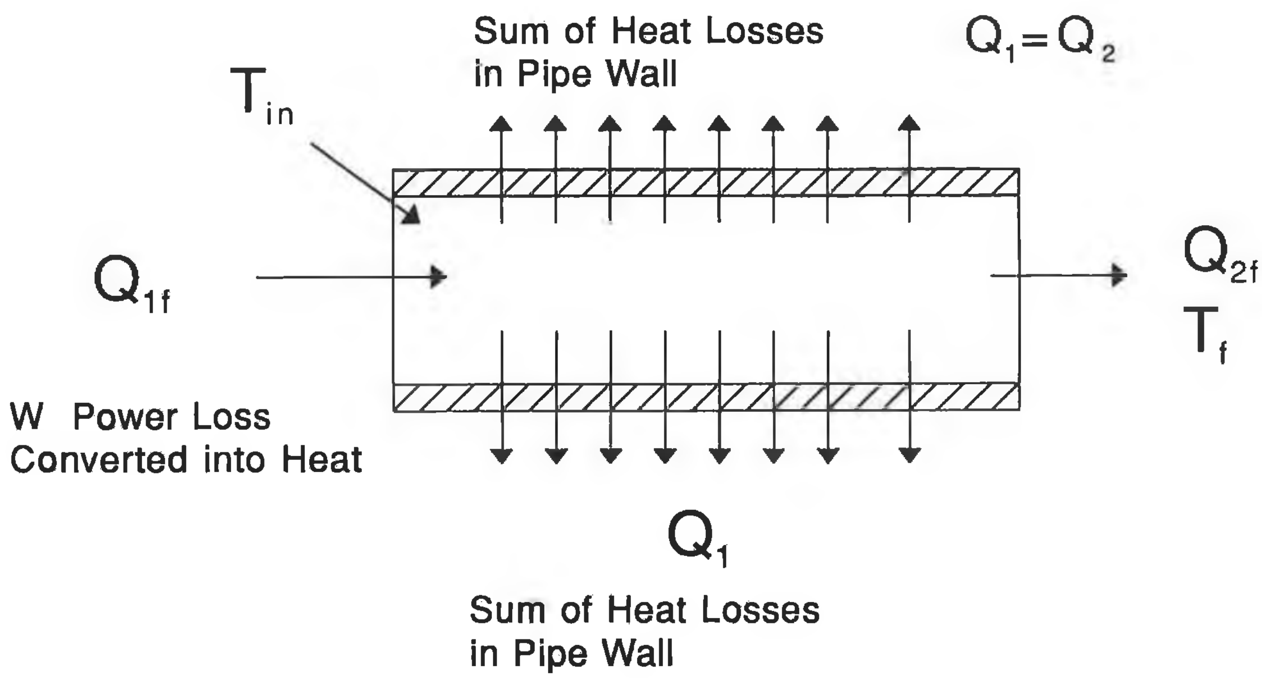


Fig (23) Open thermal system of fluid in a pipe section

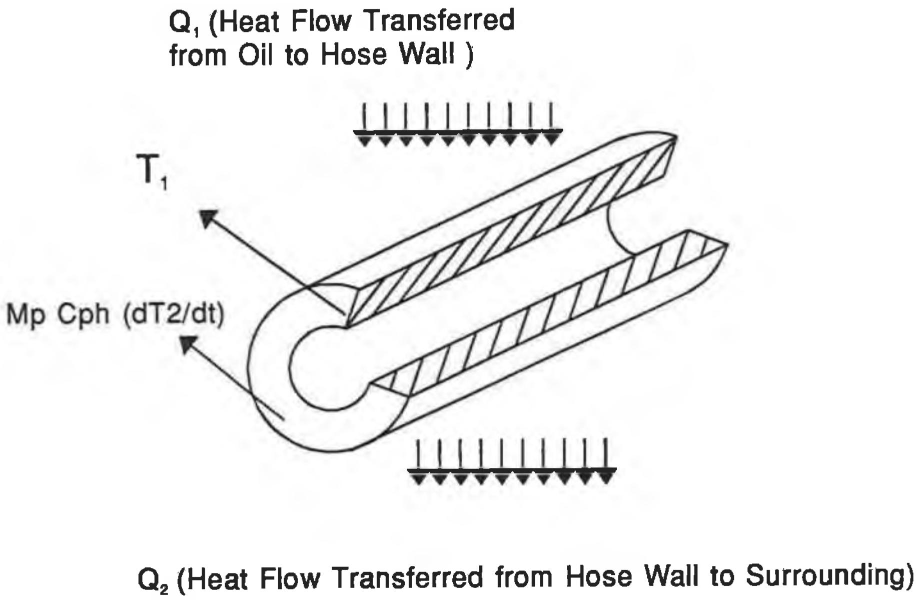


Fig (24) Heat flow losses through the hose in surroundings

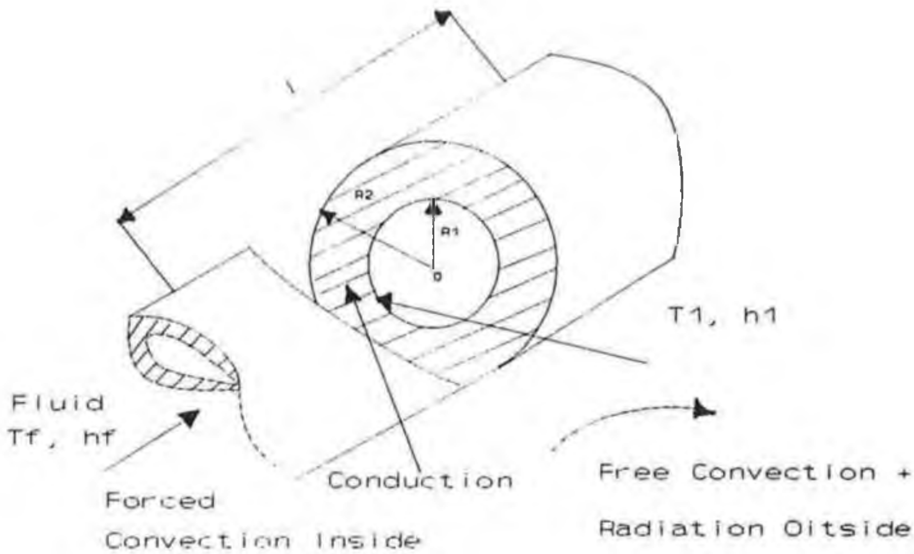
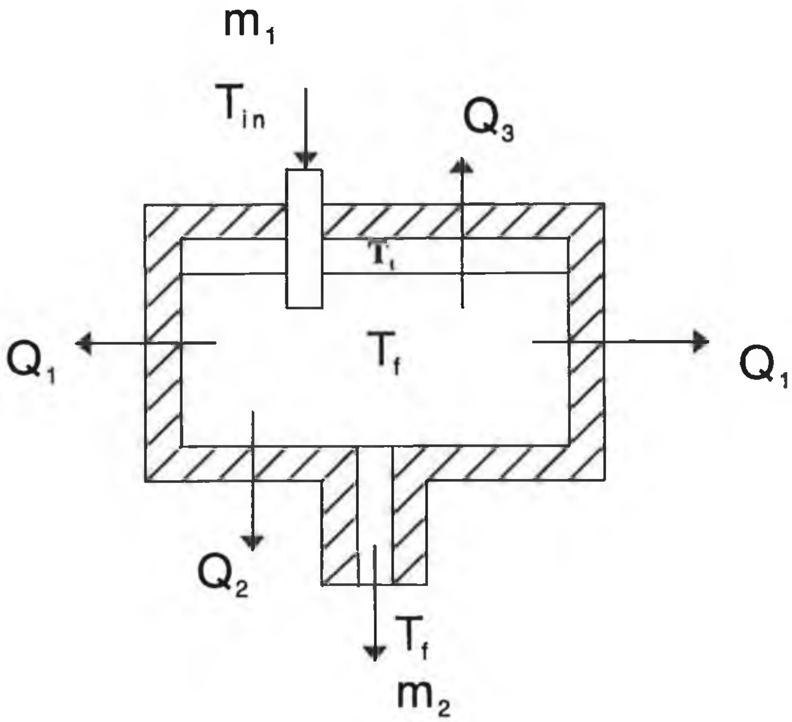


Fig. (25) Heat flow through a hydraulic hose



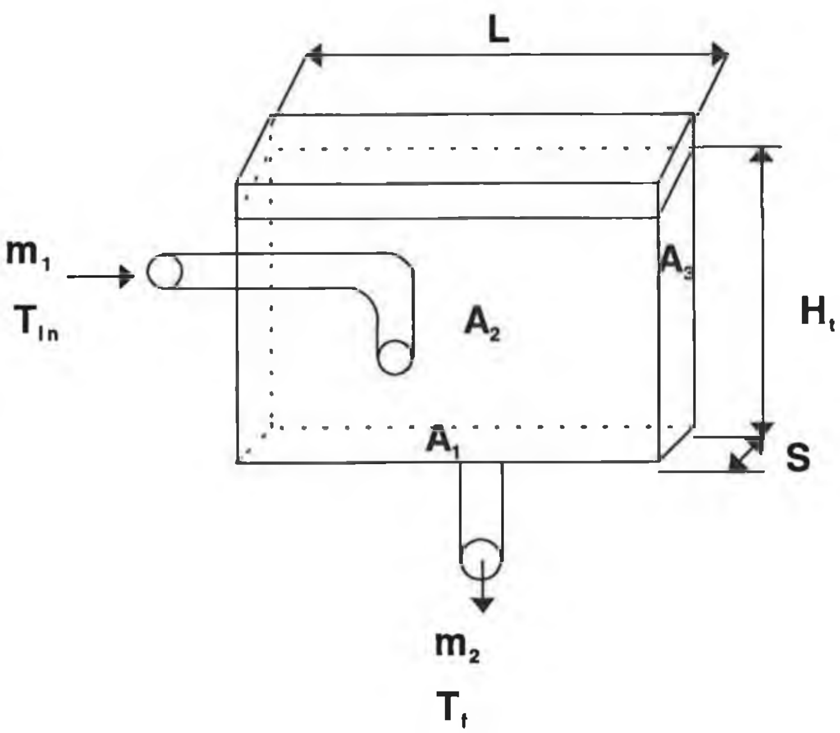


Fig (26) A hydraulic reservoir regarded as an open thermal system.

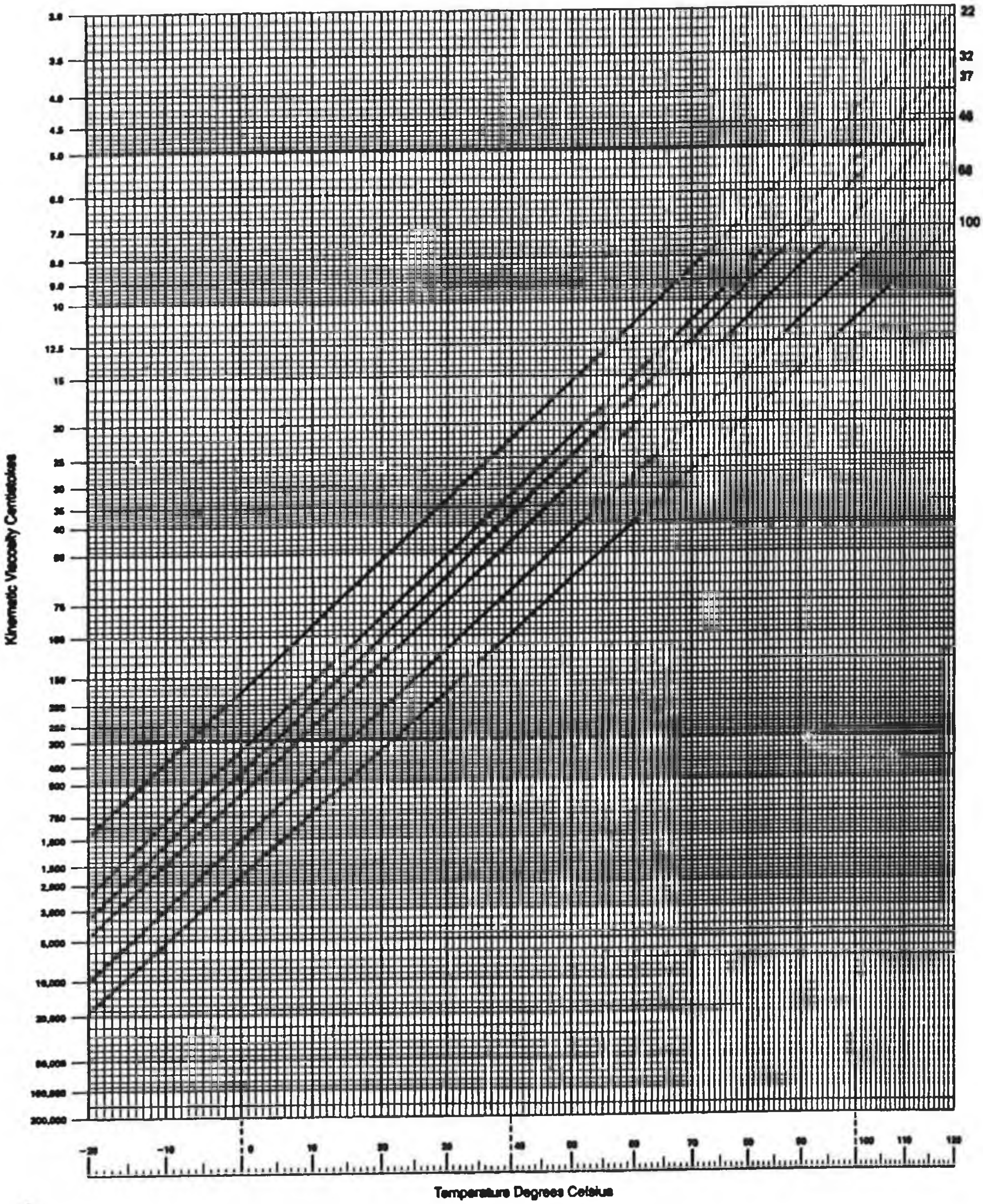


Fig. (27) Viscosity for a number of hydraulic oil SHELL TELLUS used in this research.

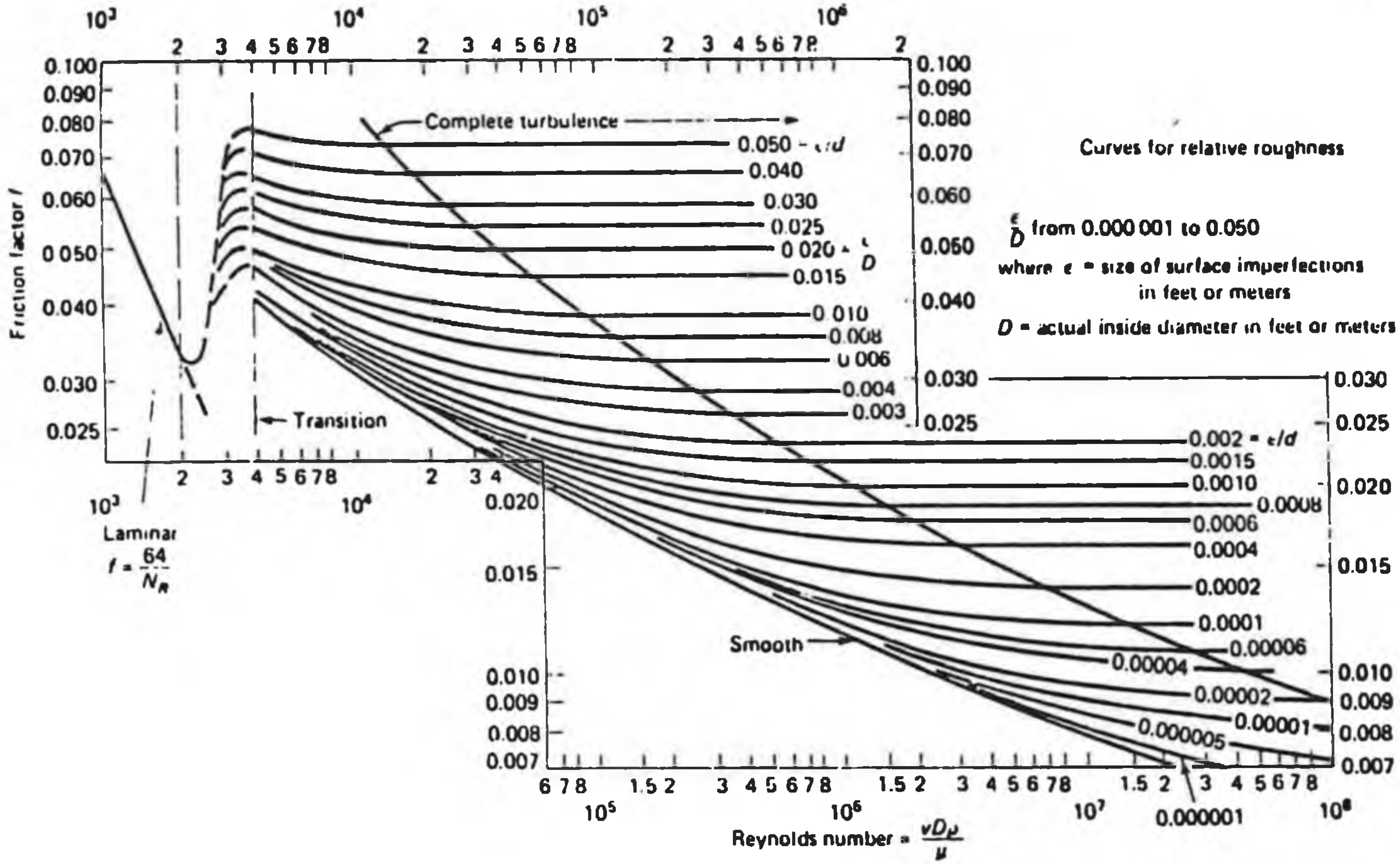


Fig. (28) Friction factor vs. Reynolds number for turbulent flow.

Valves–fittings	$\frac{L}{D}$	*K-value*
swing check valve	135	2.50
globe valve	340	10.00
gate valve		
(full open)	13	0.19
1/4 closed	35	1.15
1/2 closed	160	5.60
3/4 closed	900	24.00
cock valve	18	.26
close pattern return bend	50	2.20
standard tee	60	1.80
standard 90° elbow	30	0.90
standard 45° elbow	16	0.42

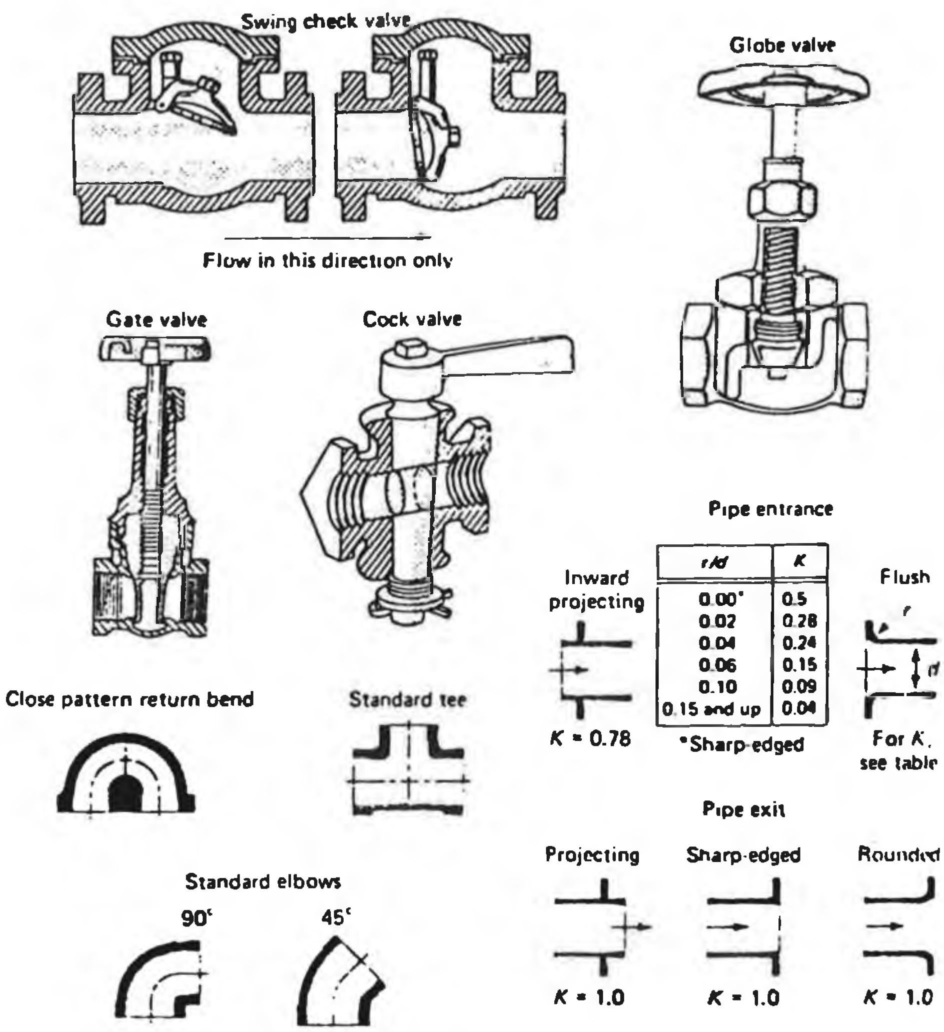


Fig. (29) K-values for several valves and fittings.

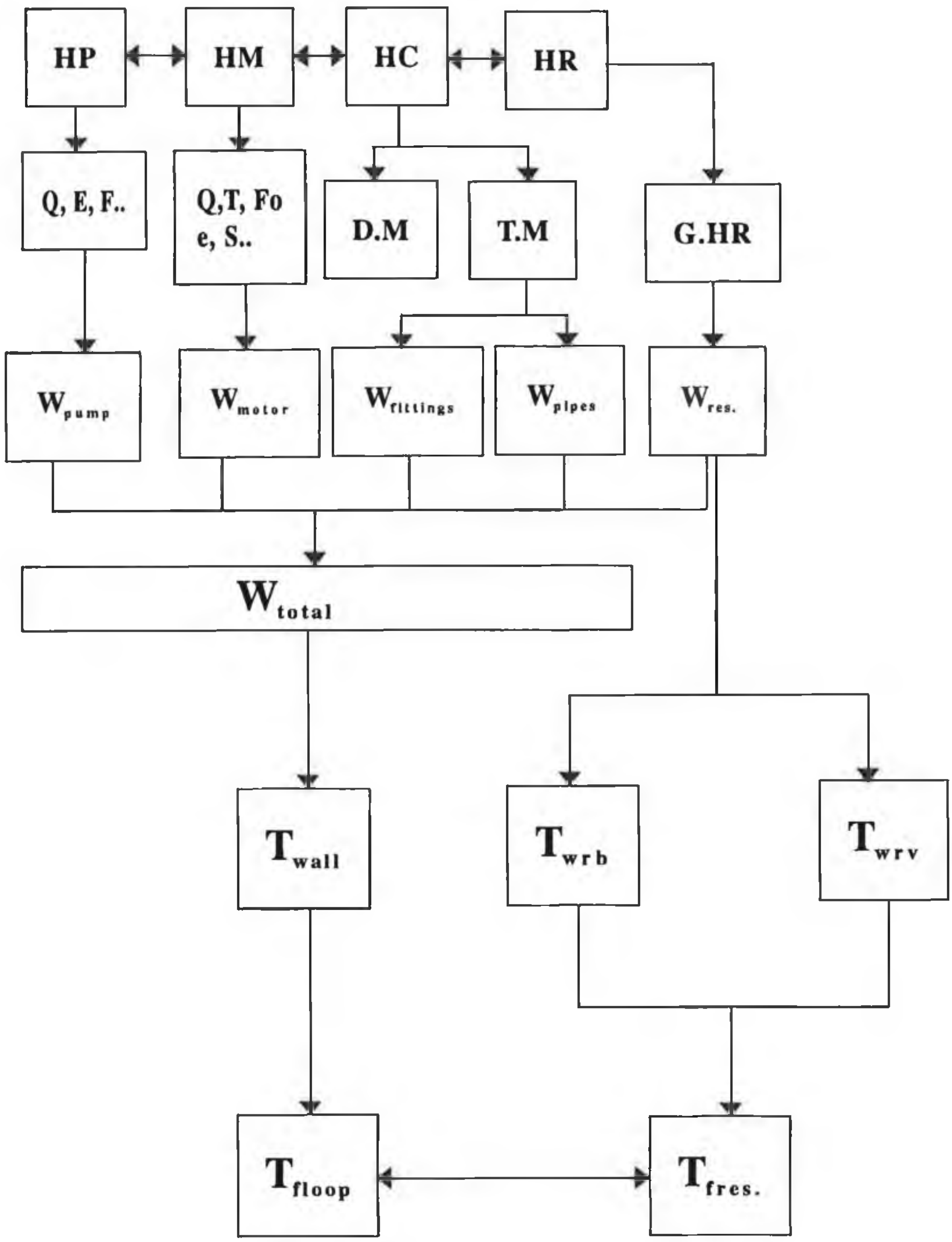


Fig. (30) Block diagram for the dynamic and thermal simulation of the open hydraulic systems using the developed programme.

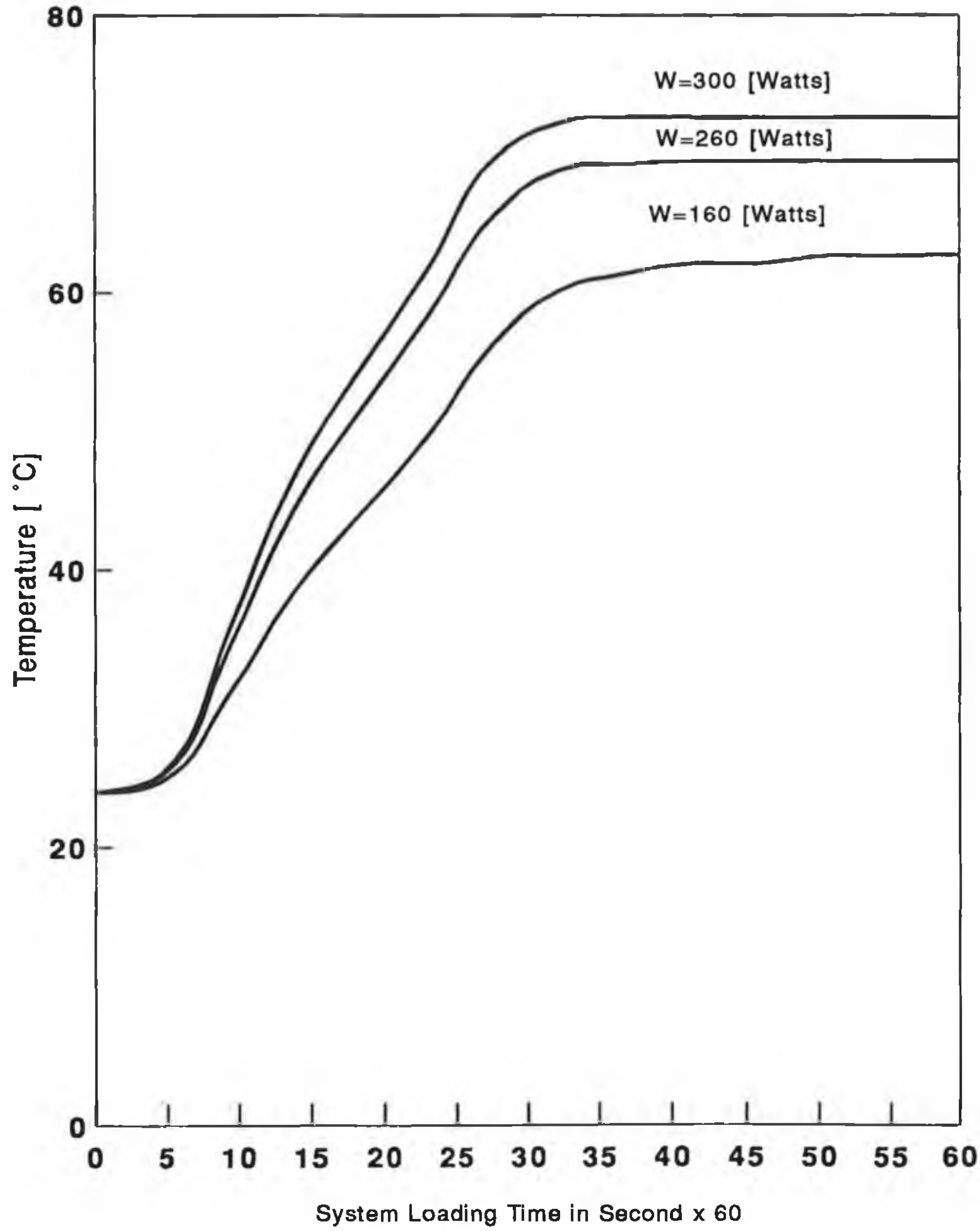


Fig. (31) The effects of power loss on the fluid temperature in the open hydraulic drive mixer system

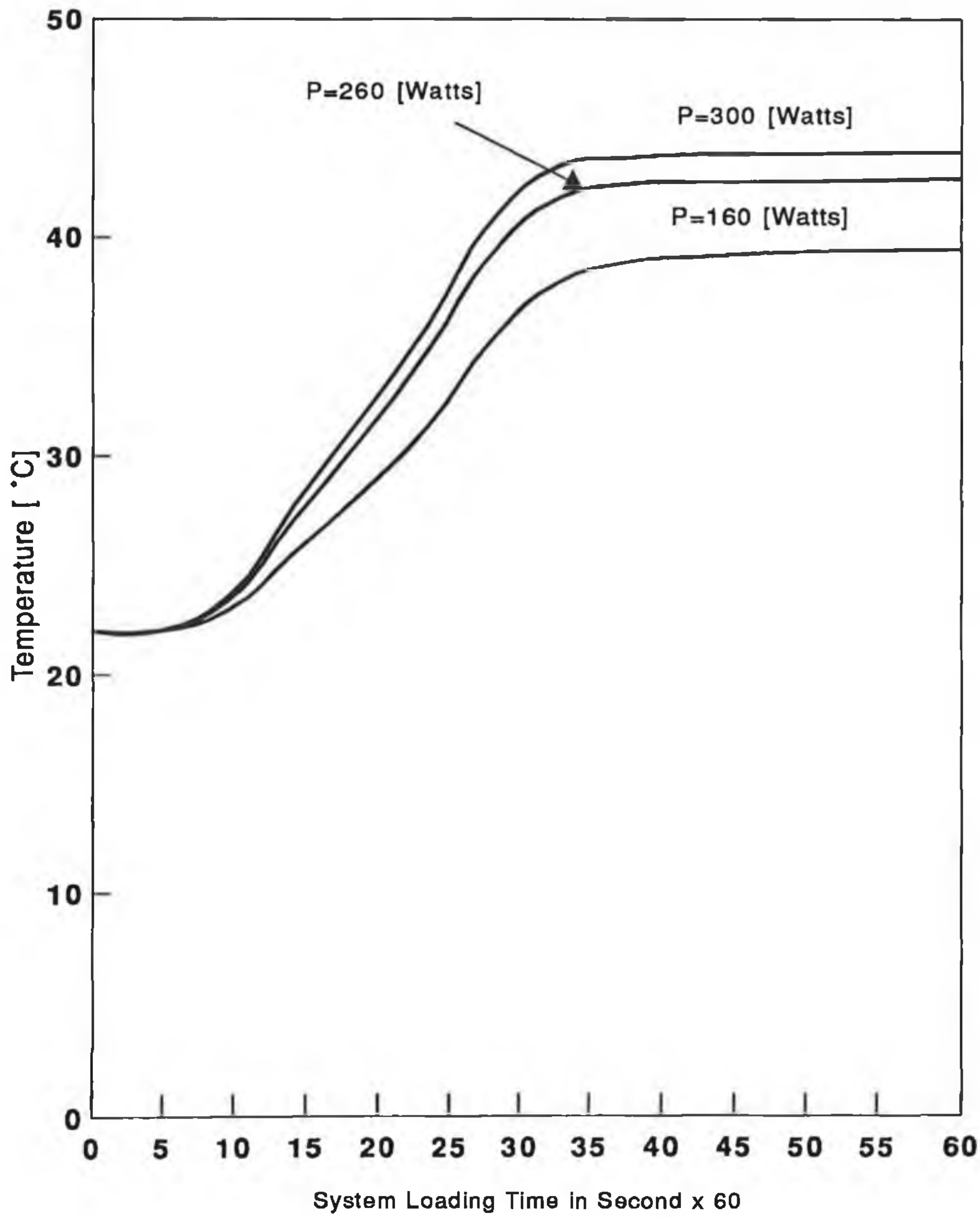


Fig. (32) The effects of power loss on the outside hose wall temperature variation in the open hydraulic drive mixer system.

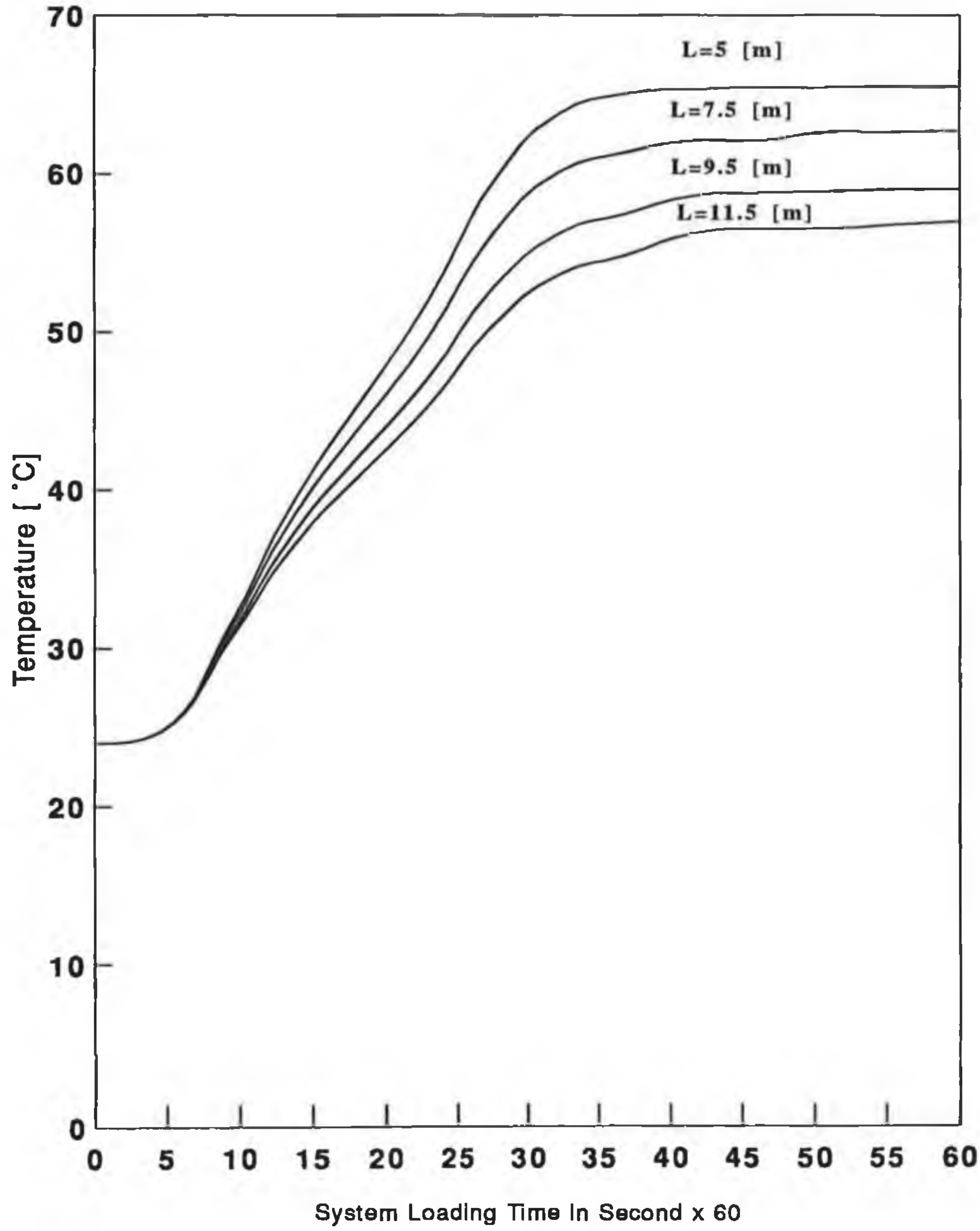


Fig. (33) The effects of pipelength on the fluid temperature variation in the open hydraulic drive mixer system.

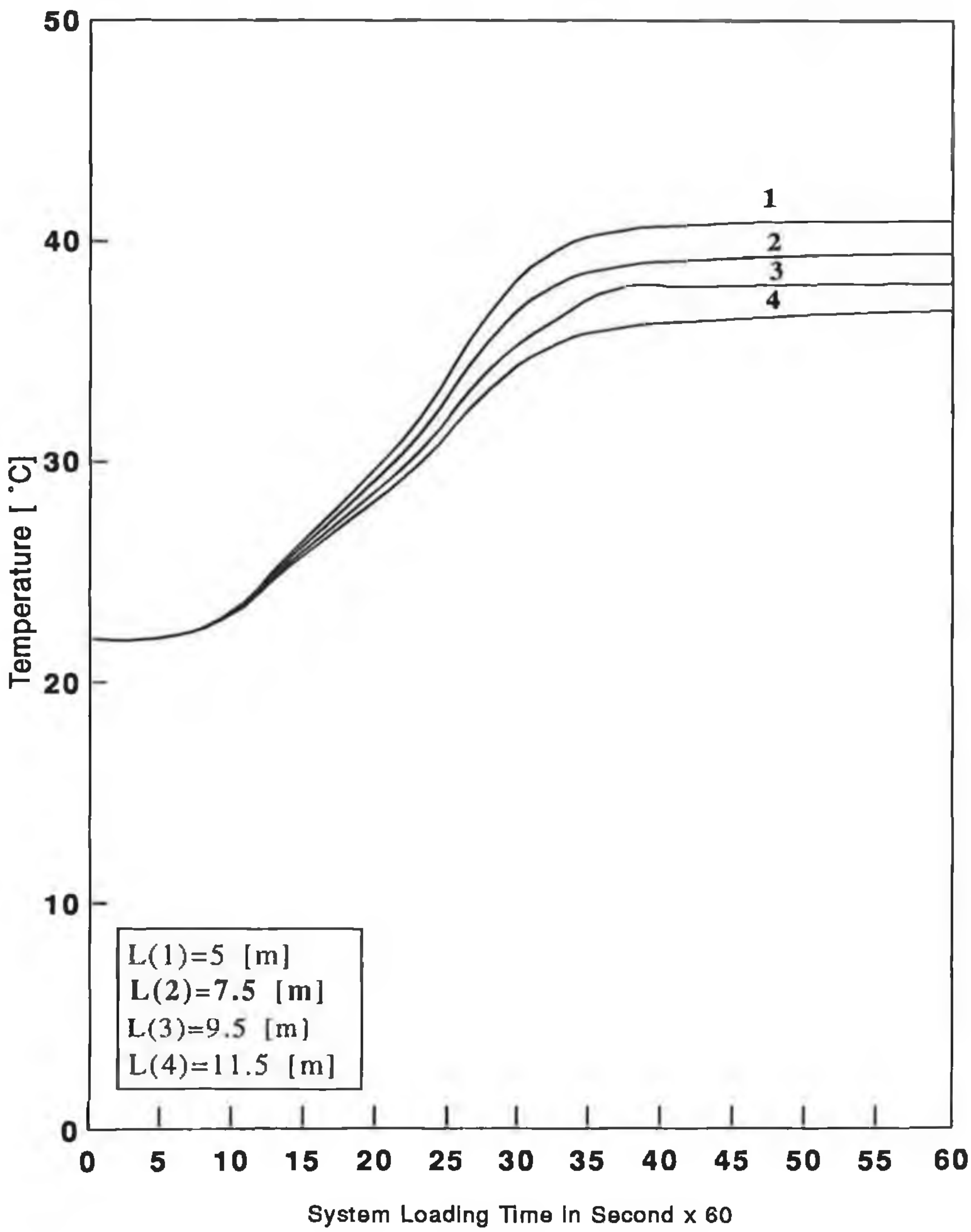


Fig. (34) The effects of pipelength on the outside hose wall temperature variation in the open hydraulic drive mixer system.

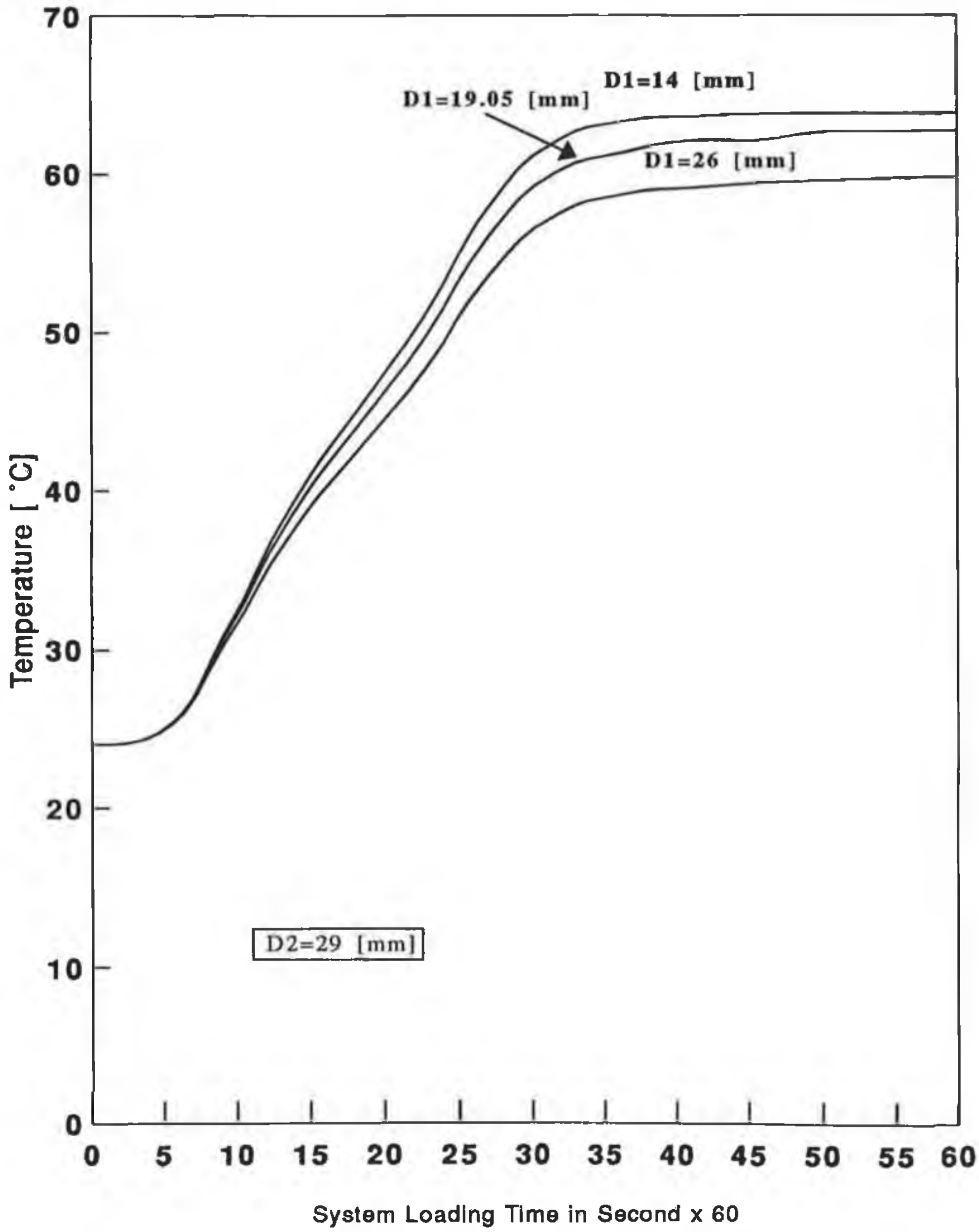


Fig. (35) The effects of internal diameter of hose on the fluid temperature variation in the open hydraulic drive mixer system.

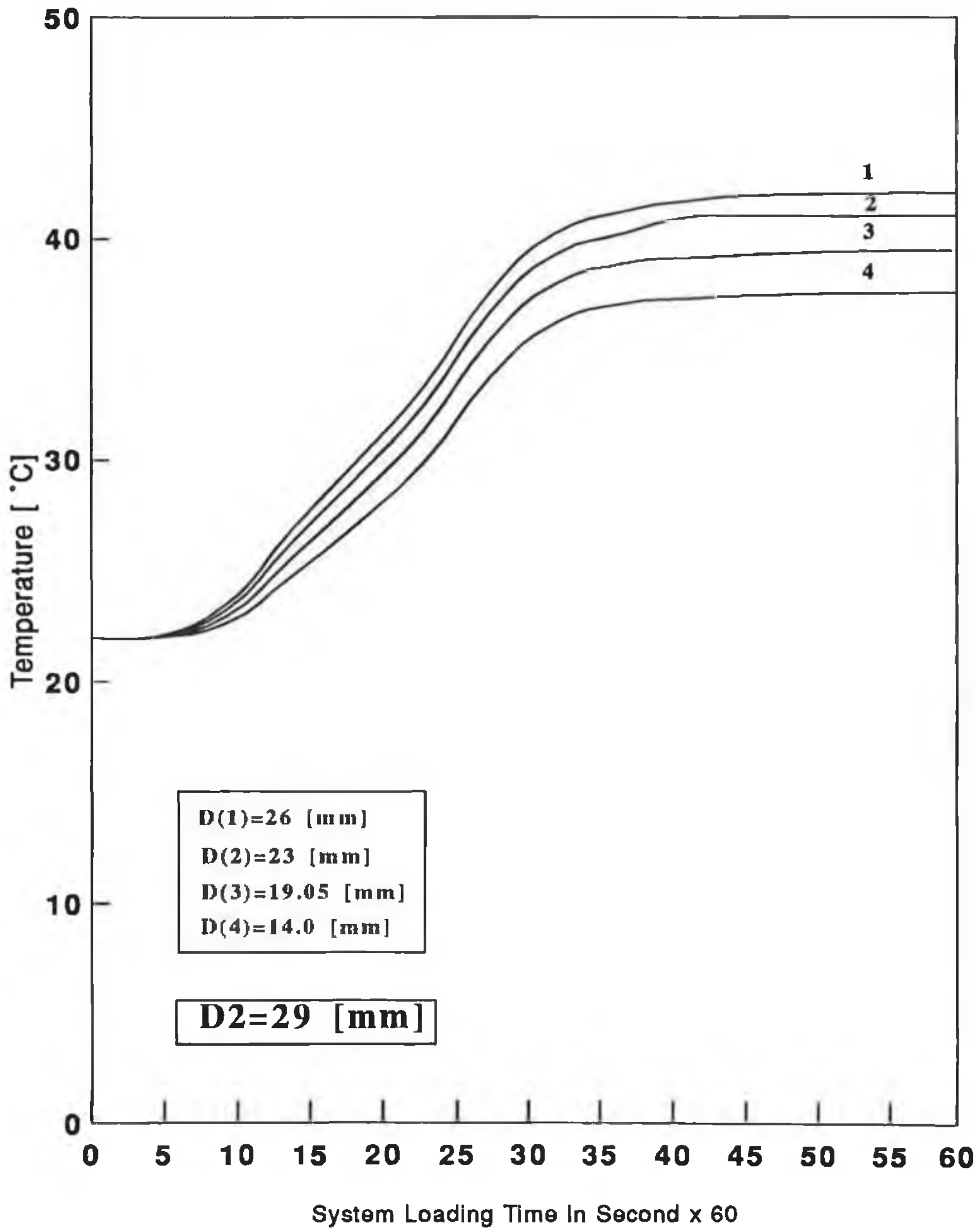


Fig. (36) The effects of internal diameter of hose on the outside hose wall temperature variation in the open hydraulic drive mixer system.

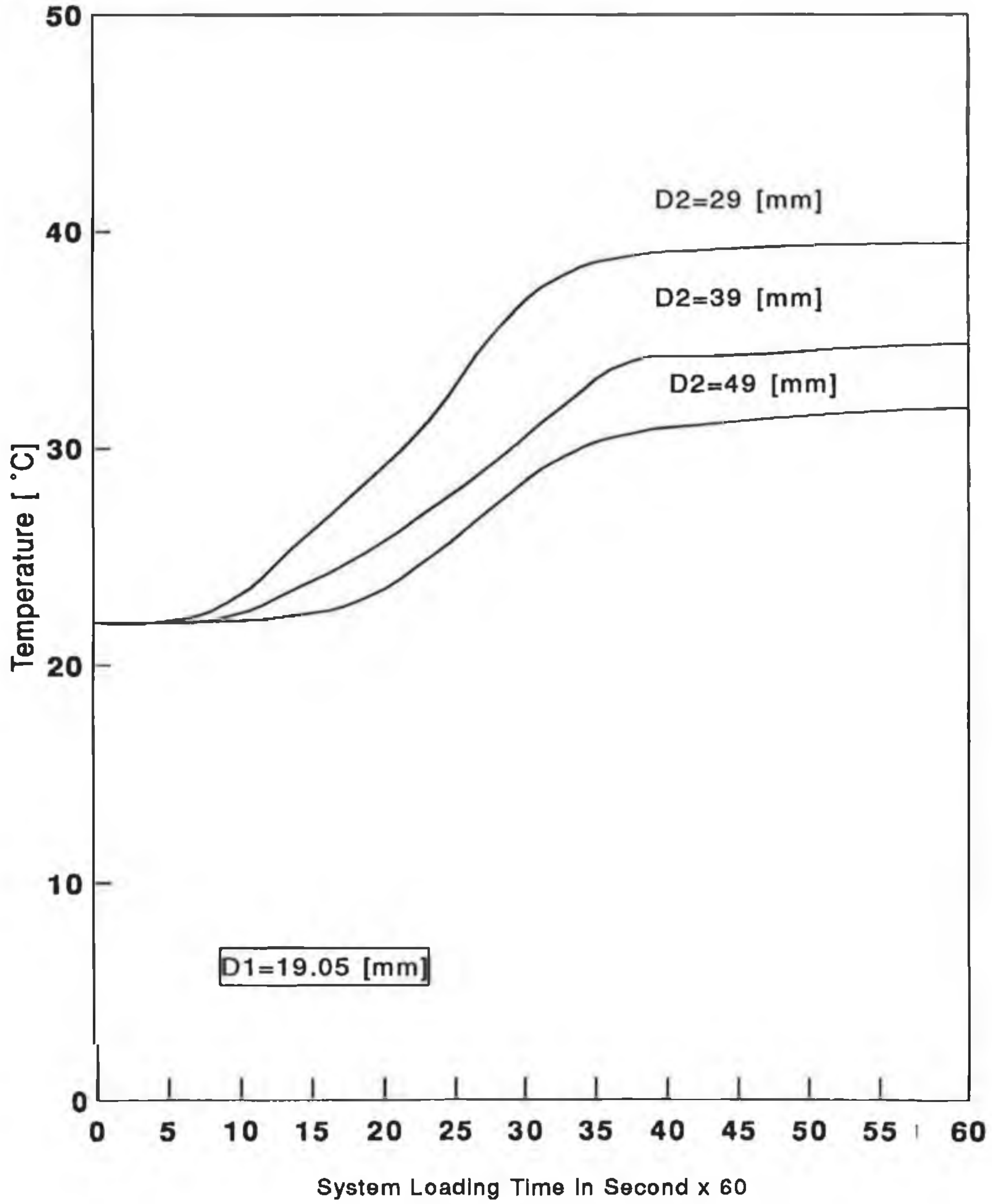


Fig. (37) The effects of external hose diameter on the outside hose wall temperature variation in the open hydraulic drive mixer system.

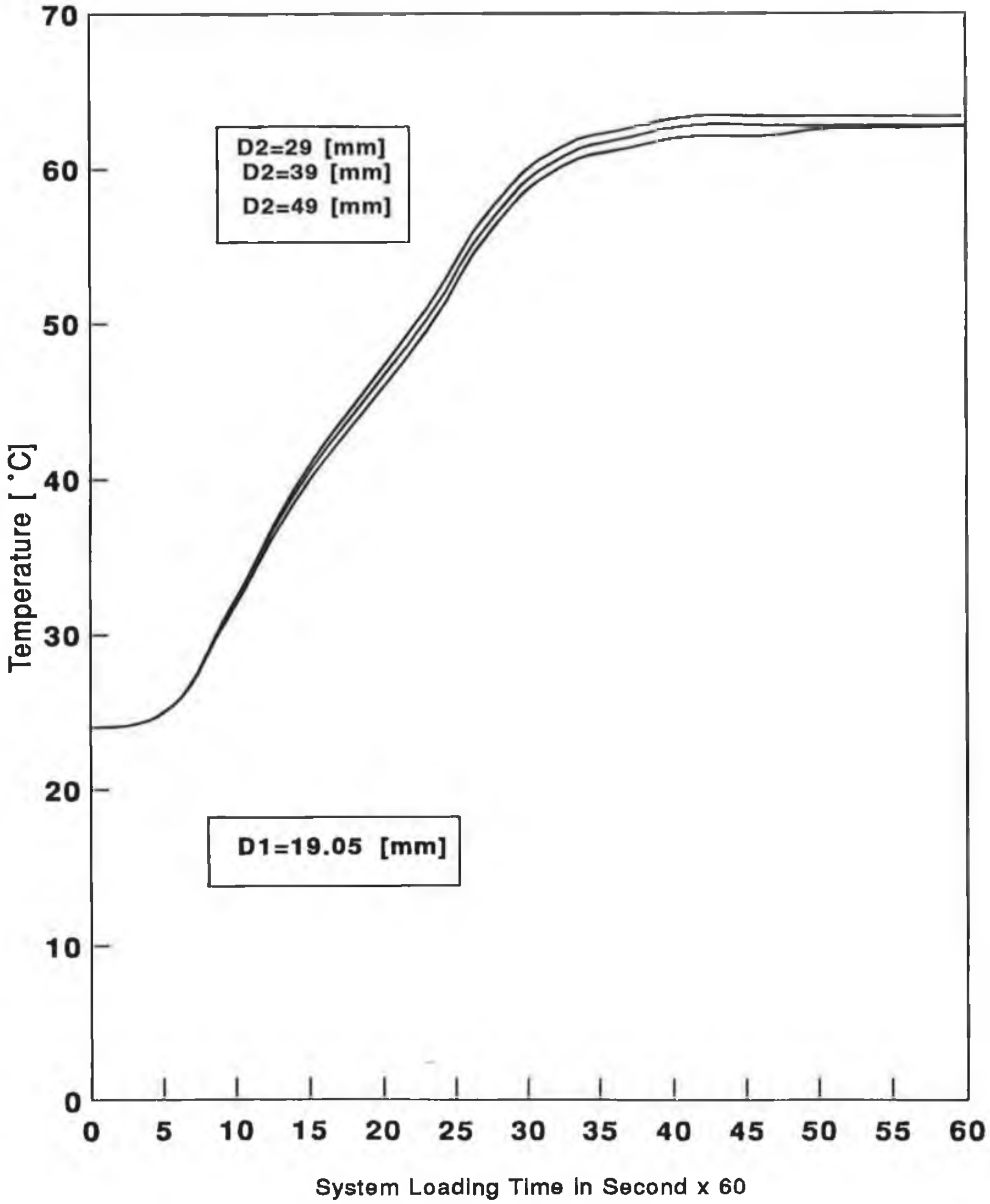


Fig. (38) Simulated results of the effects of external hose diameter on fluid temperature distribution in the open hydraulic drive mixer system.

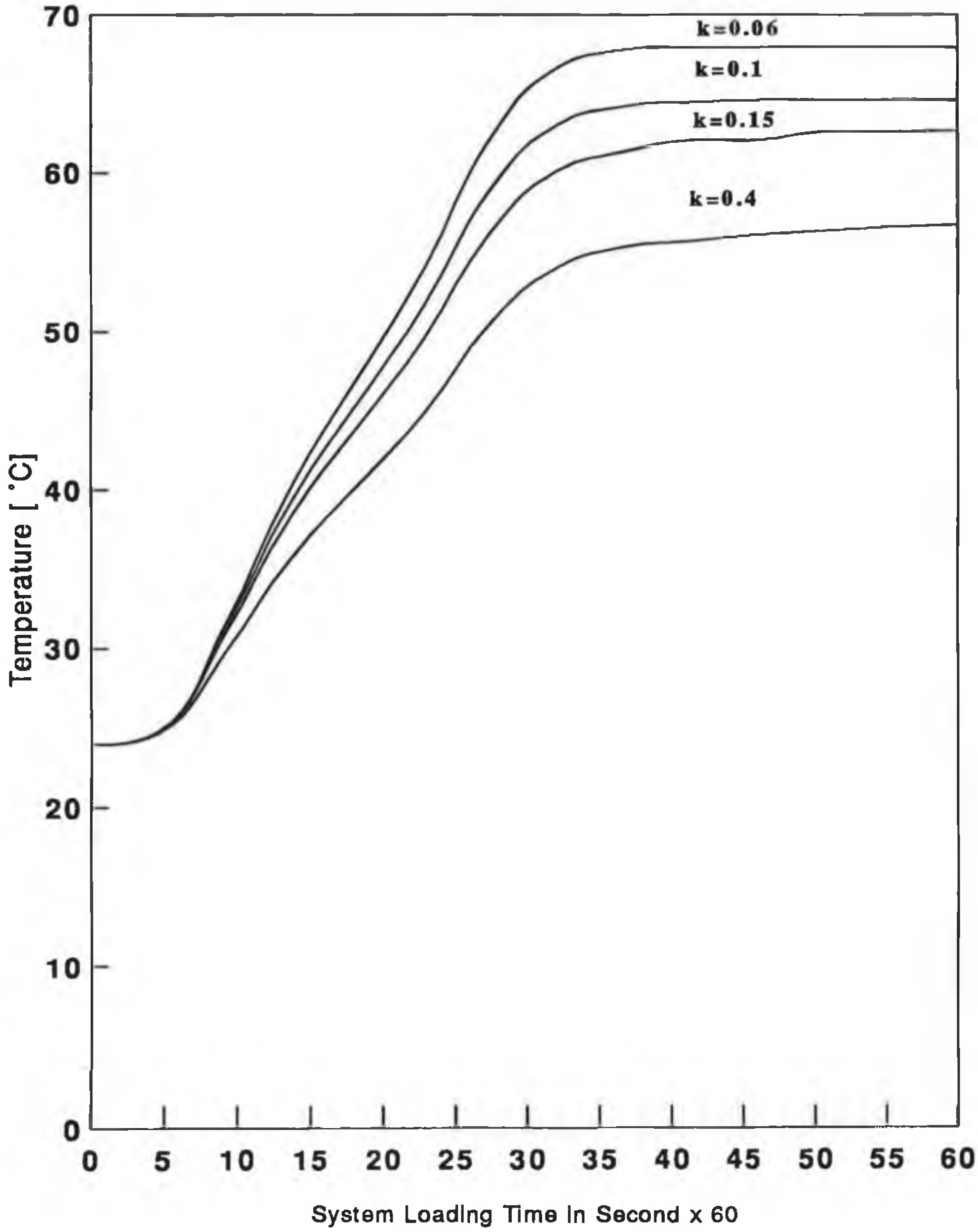


Fig. (39) The effects of conductivity of pipe wall material on fluid temperature variation in the open hydraulic drive mixer system.

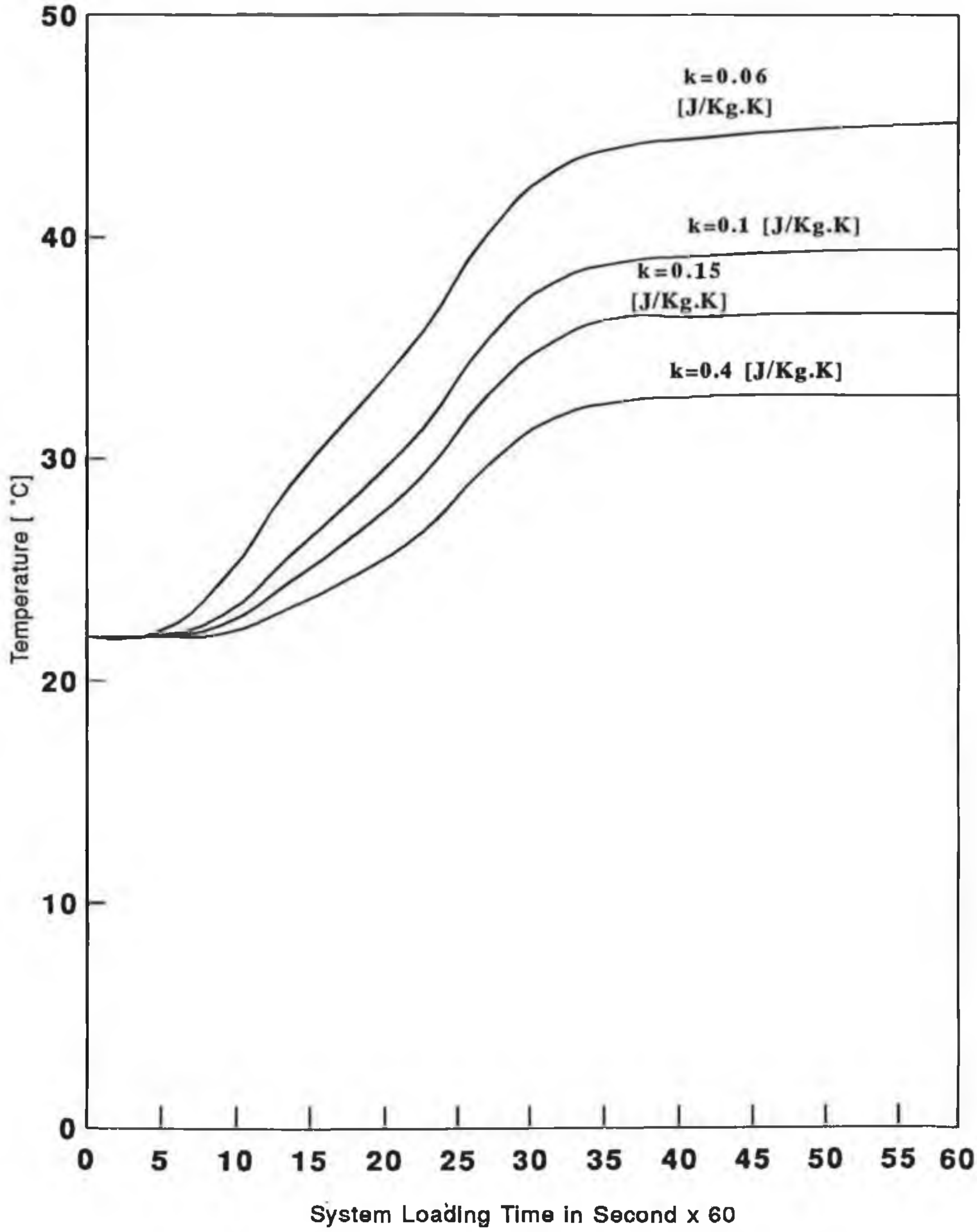


Fig. (40) The effects of conductivity of pipe material on the outside hose wall temperature variation in the open hydraulic drive mixer system.

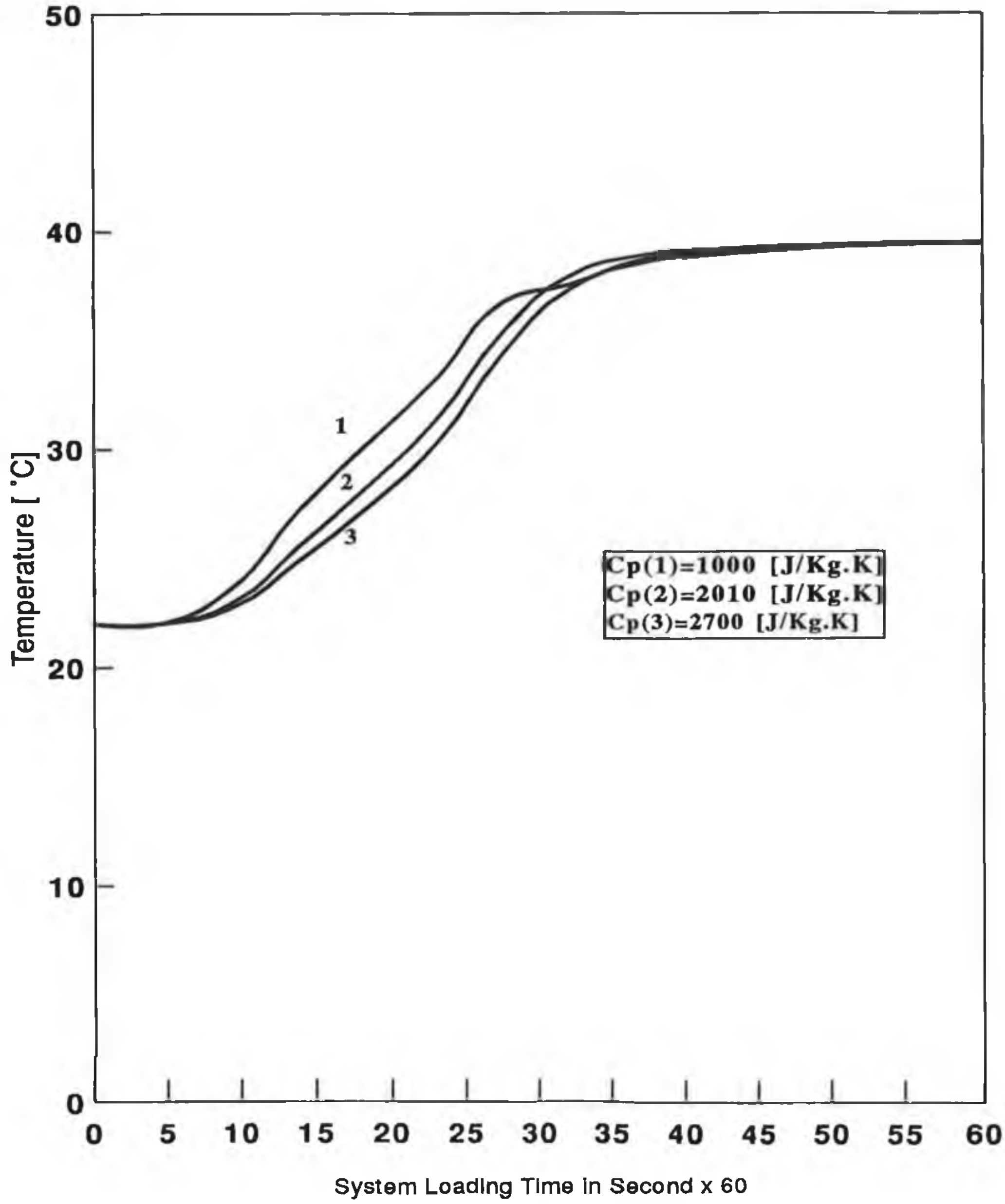


Fig. (41) The effects of specific heat of pipe material on the outside hose wall temperature variation in the Open hydraulic drive mixer system

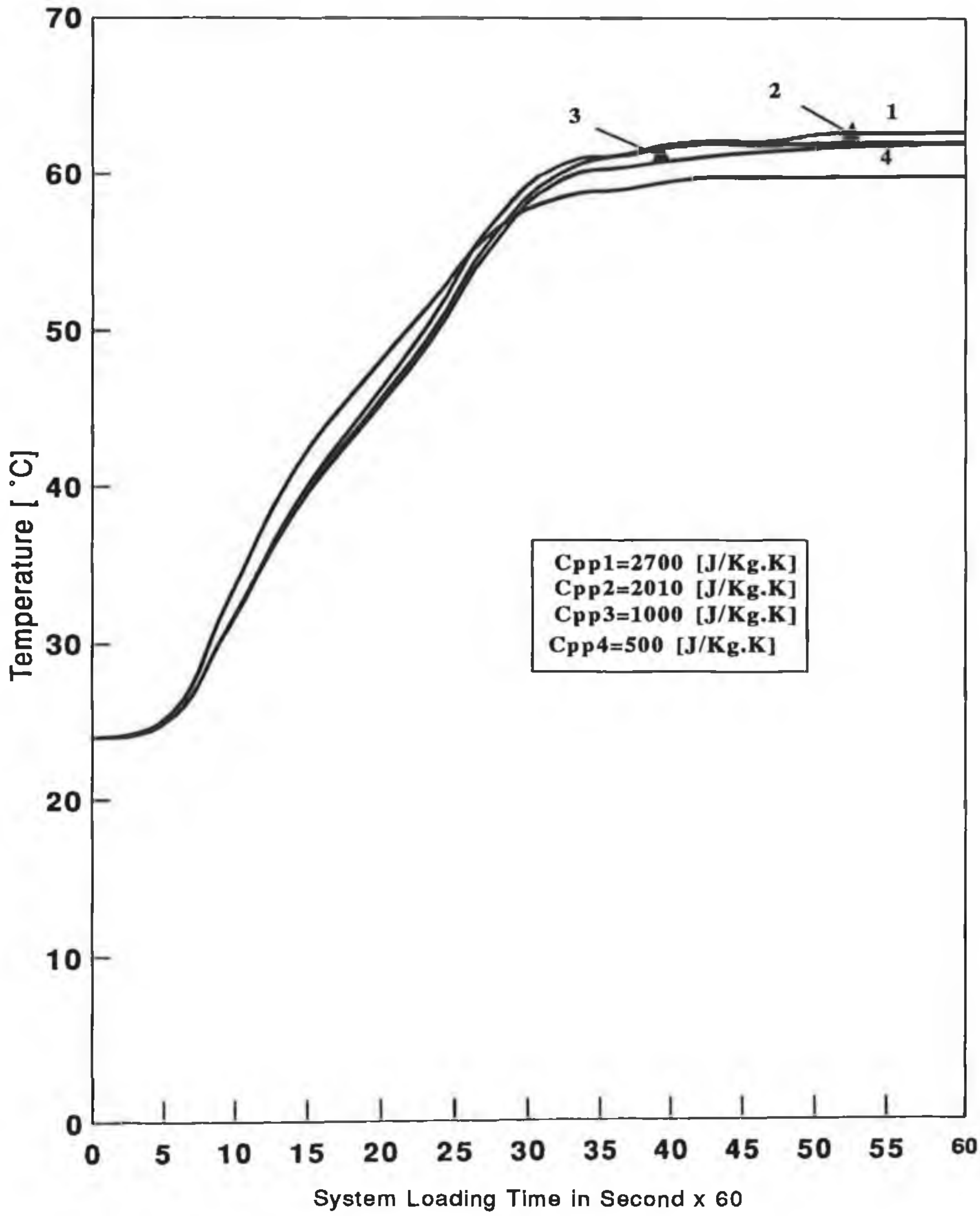


Fig. (42) The effects of specific heat of pipe material on fluid temperature variation in the open hydraulic drive mixer system.

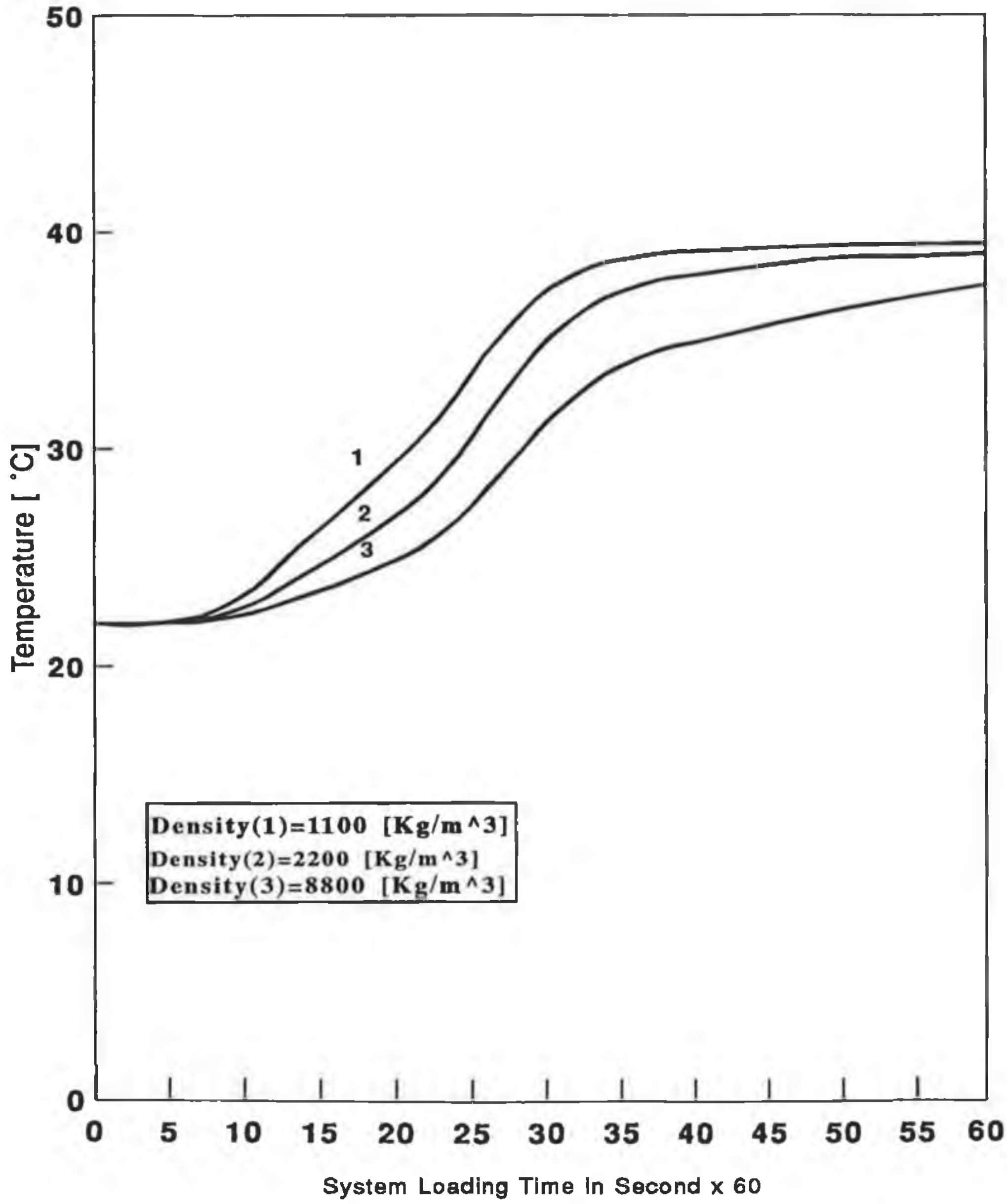


Fig. (43) The effects of density of pipe material on the outside hose wall temperature variation in the open hydraulic drive mixer system

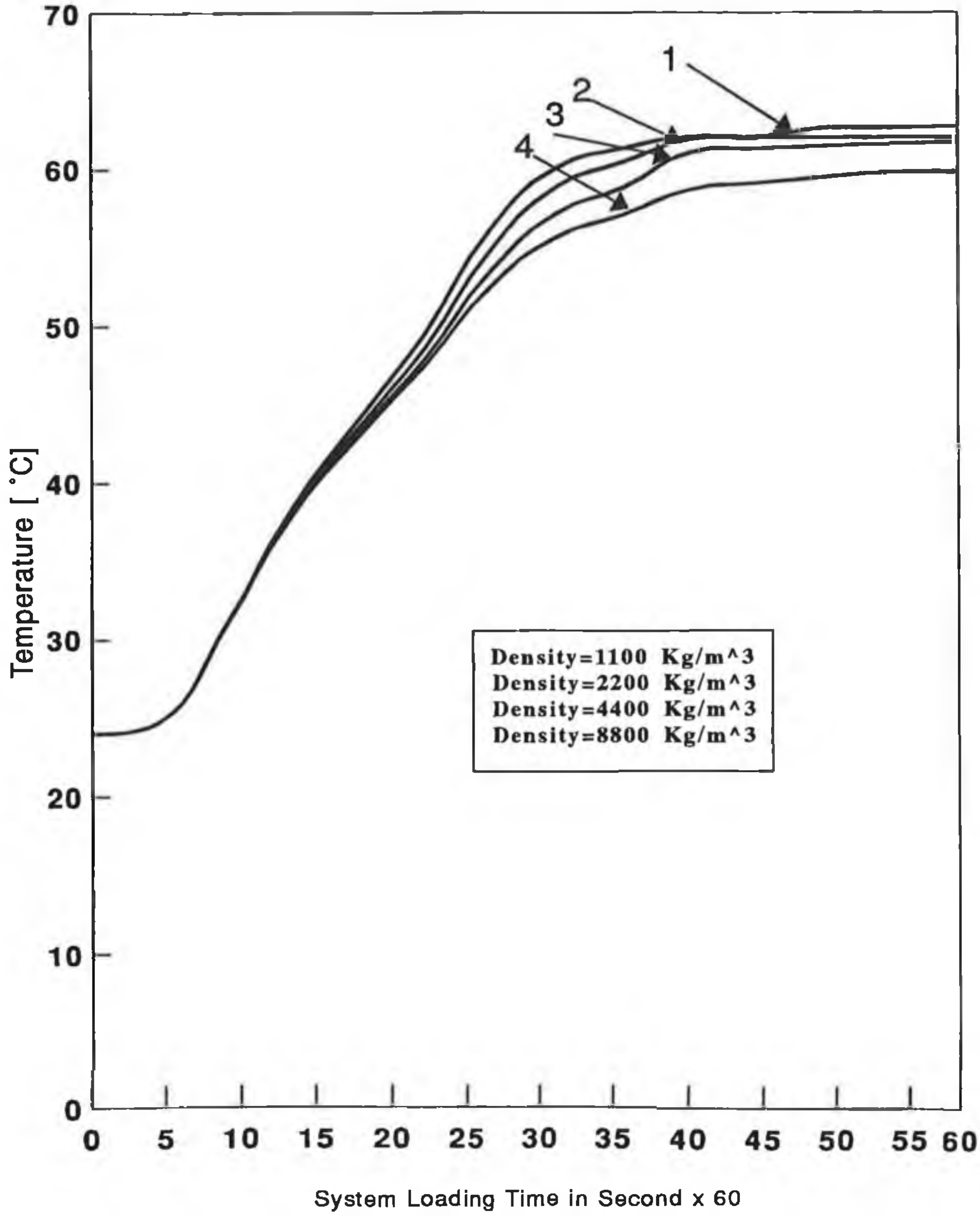


Fig. (44) Simulated results of the effects of density of pipe material on fluid temperature variation in the open hydraulic drive mixer system.

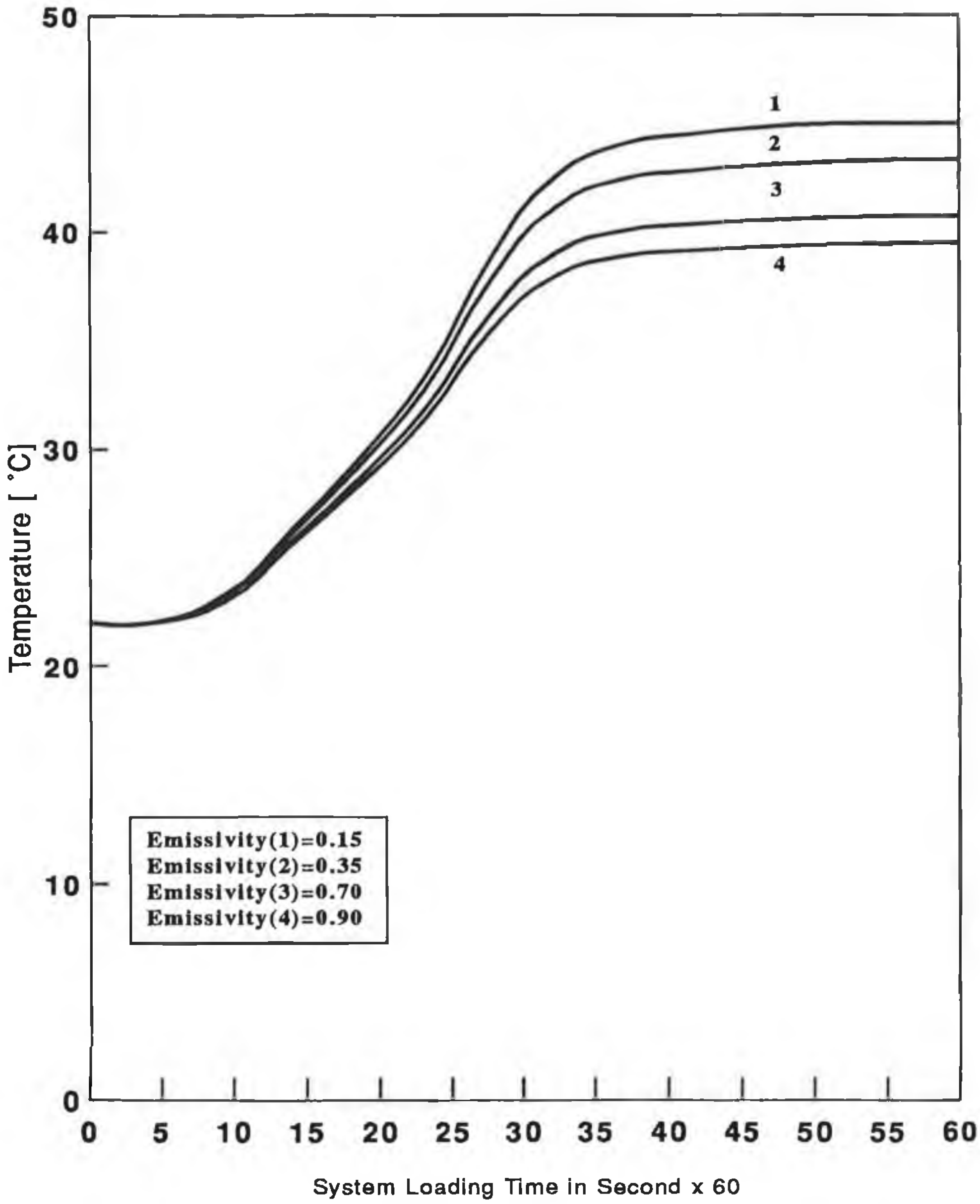


Fig. (45) The effects of emissivity of pipe material on the outside hose wall temperature variation in the open hydraulic drive mixer system.

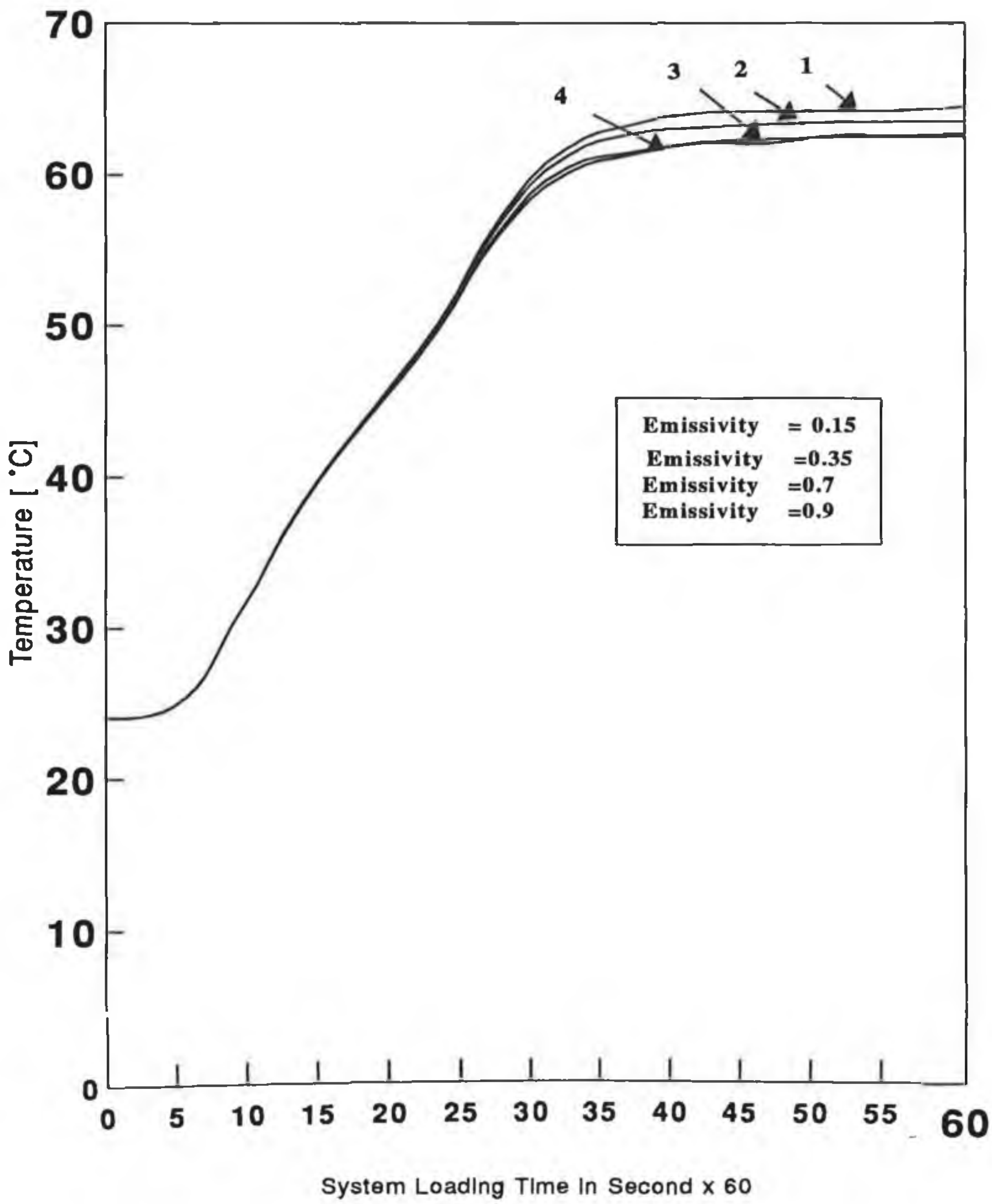


Fig. (46) The effects of emissivity of pipe material on fluid temperature variation in the open hydraulic drive mixer system.

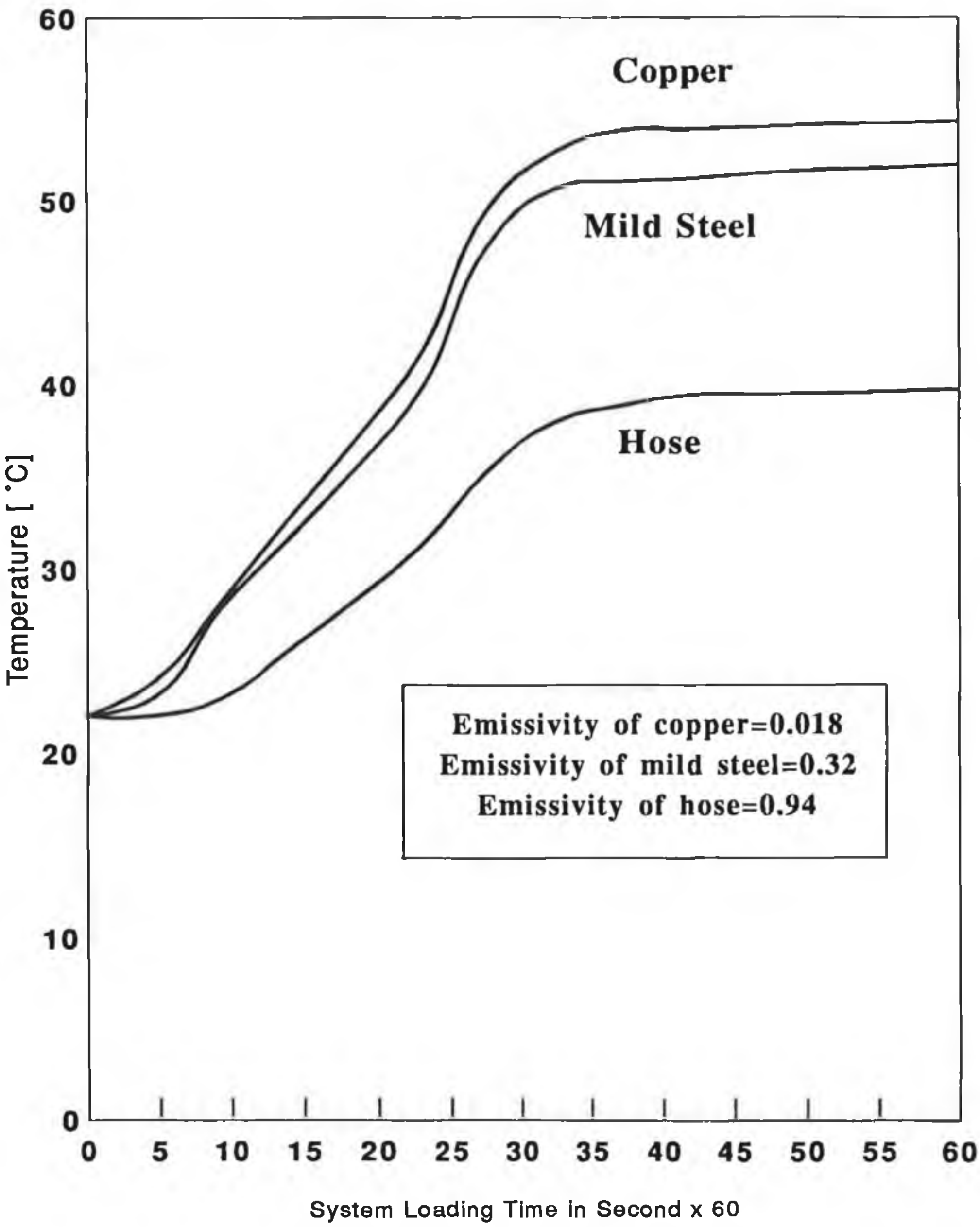


Fig. (47) Simulated results of the effects of type of pipe material on outside pipe wall temperature in the open hydraulic drive mixer system.

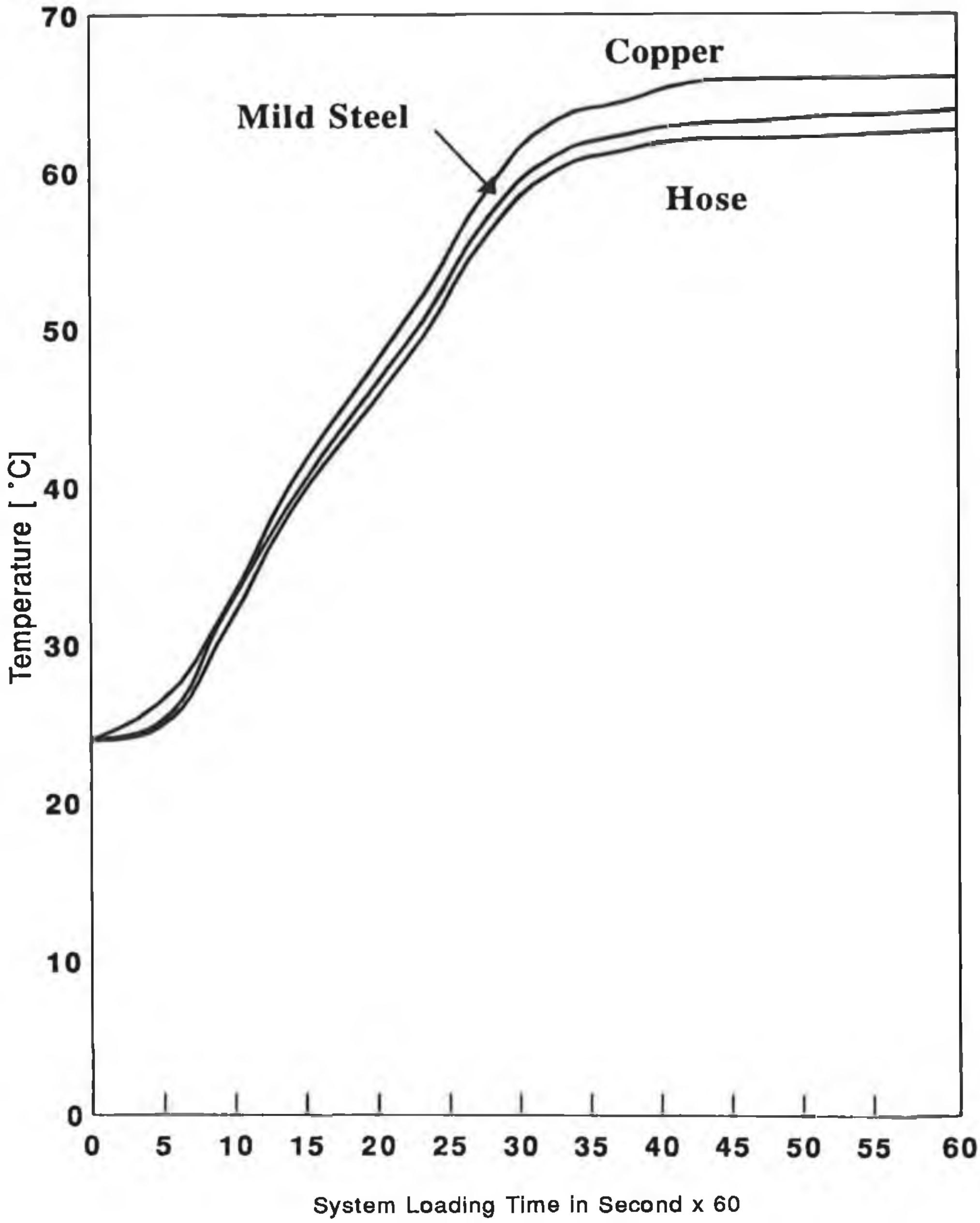


Fig. (48) Simulated results of the effects of type of pipe material on fluid temperature in the open hydaulic drive mixer system.

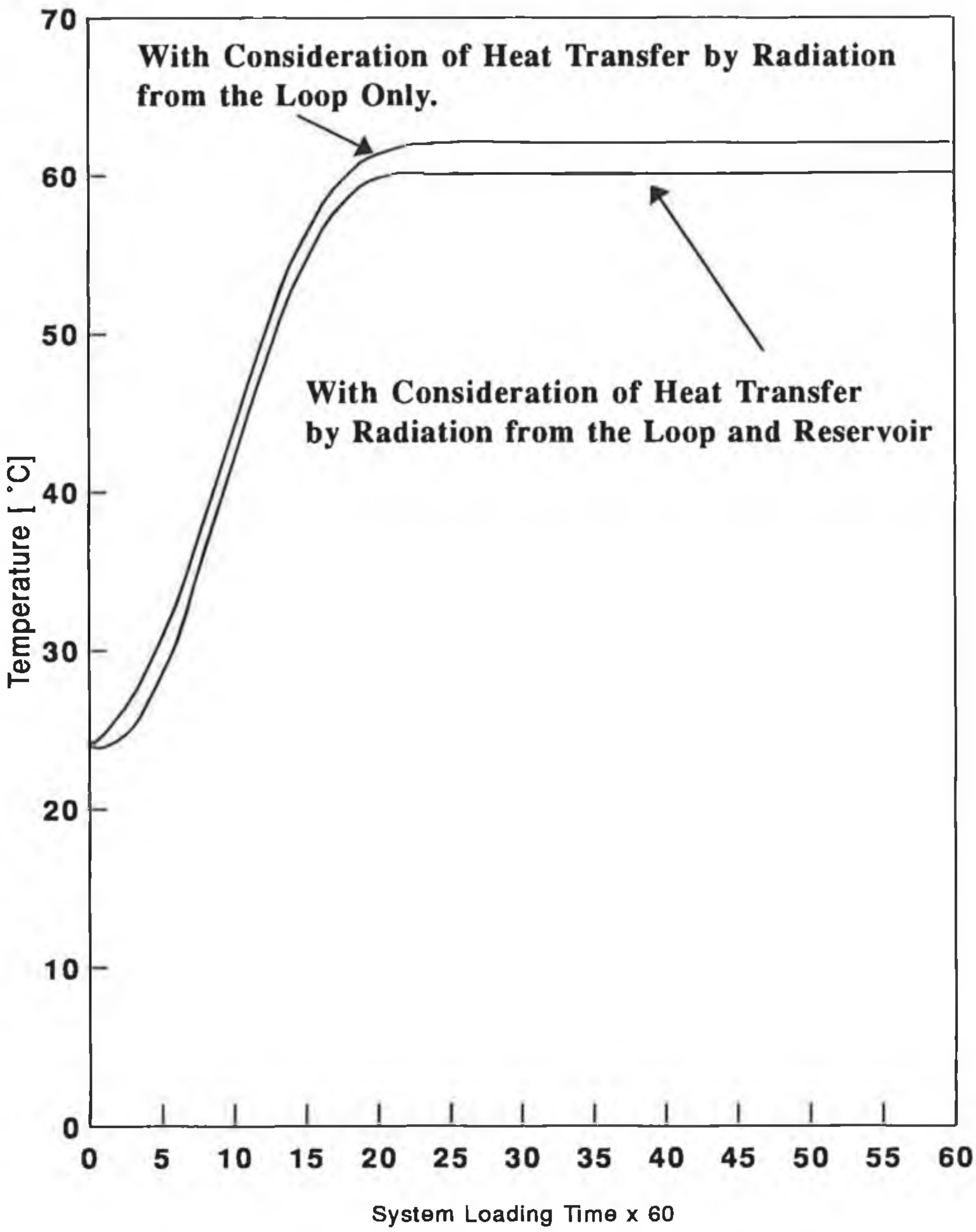


Fig. (49) Comparison of the effects of radiation on the fluid temperature in the open hydraulic drive mixer system.

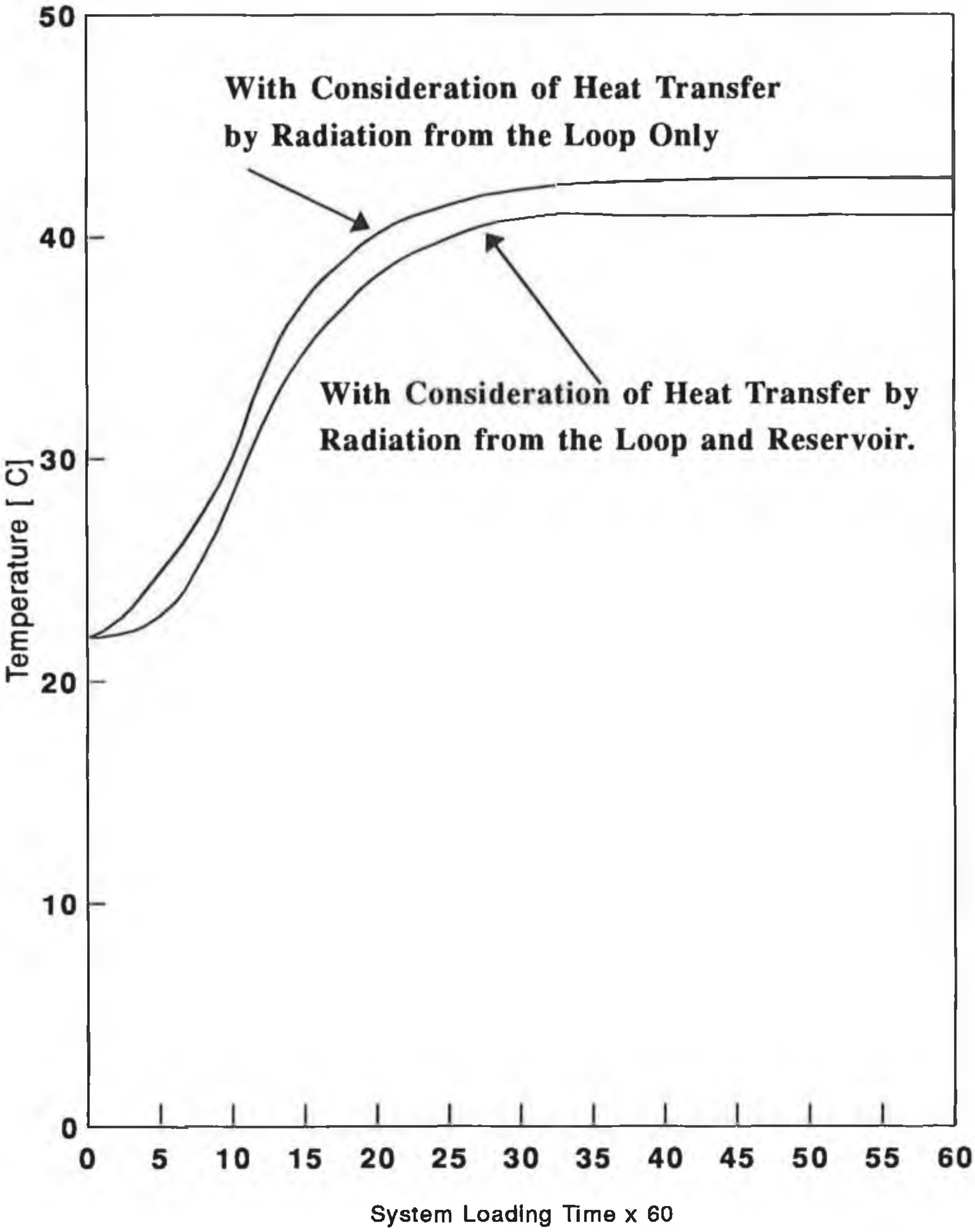


Fig. (50) Comparison of the effects of the radiation on the outside hose wall temperature in the open hydraulic drive mixer system

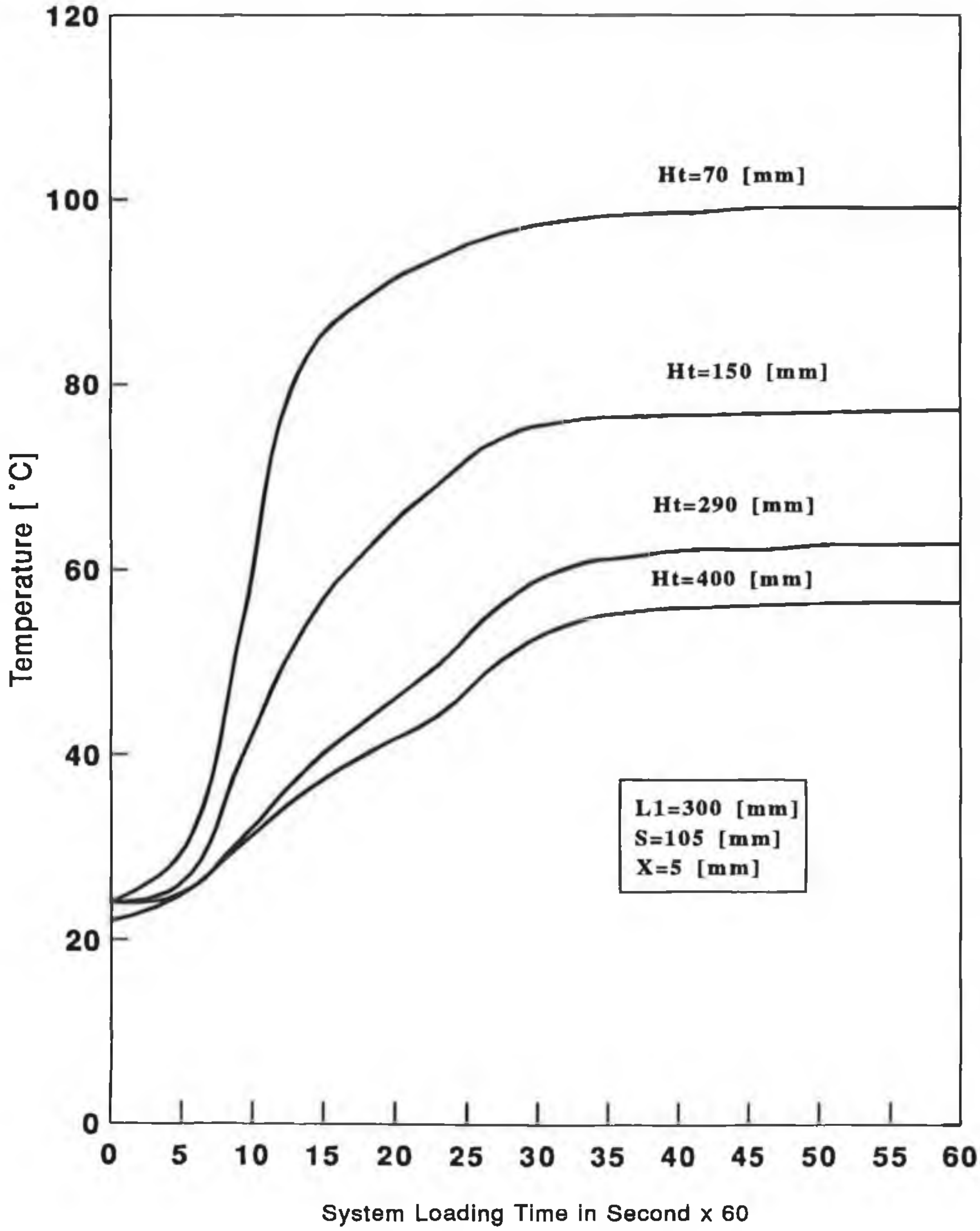


Fig. (51) The effects of fluid height in the reservoir on fluid temperature variation in the open hydraulic drive mixer system.

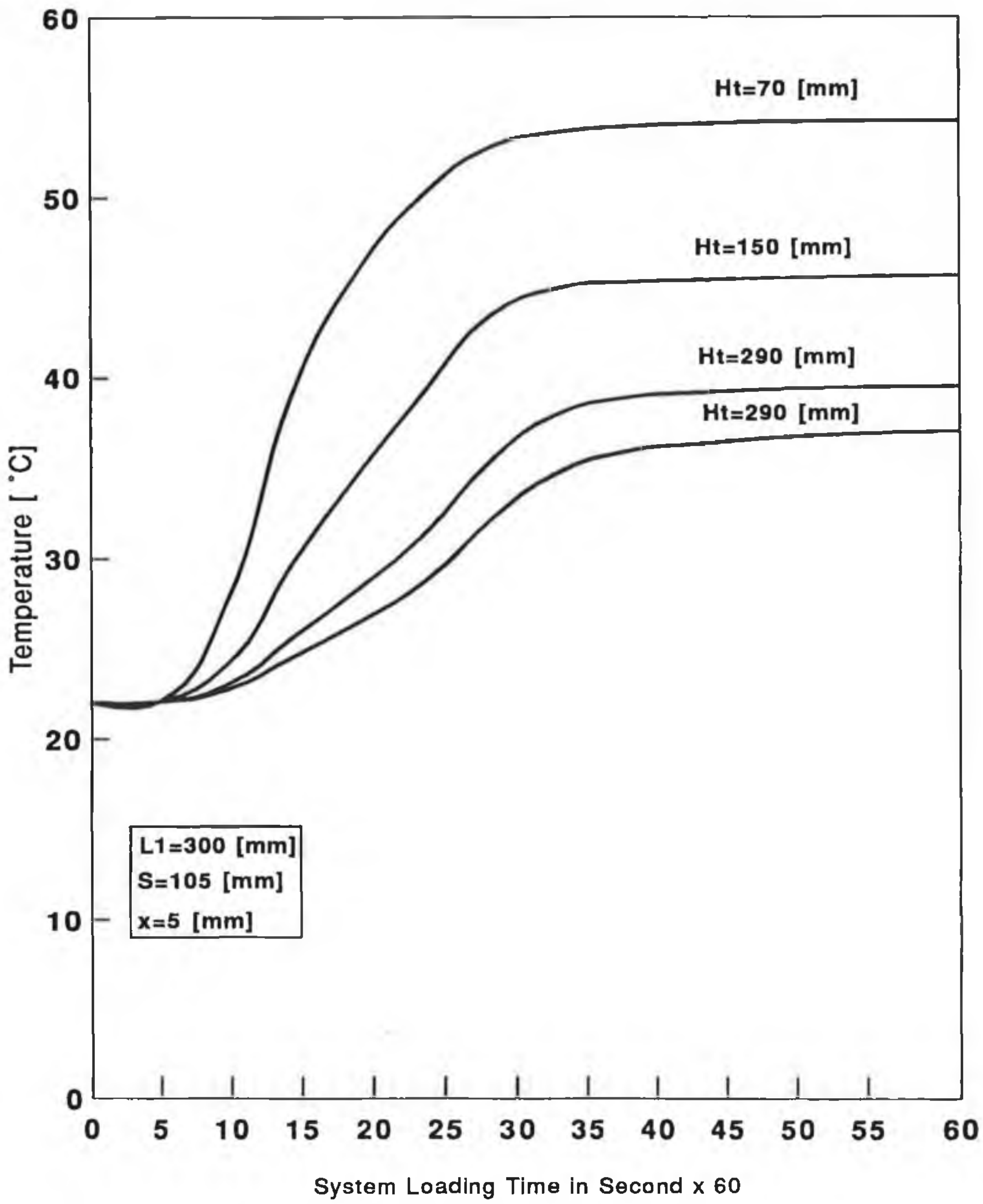


Fig. (52) The effects of fluid height in the reservoir on the outside hose wall temperature variation in the open hydraulic drive mixer system.

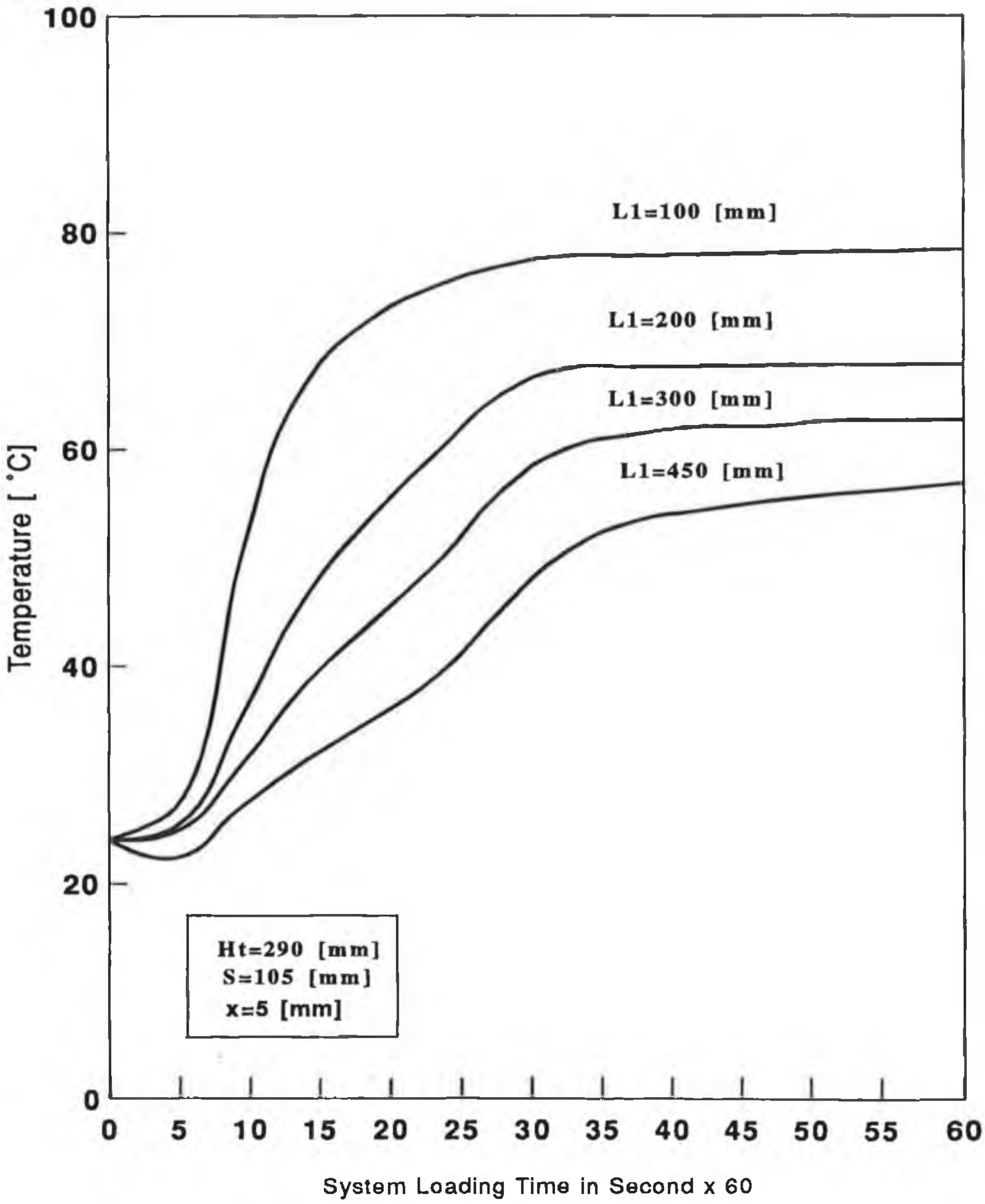


Fig. (53) The effects of length of the Base of the reservoir on fluid temperature variation in the open hydraulic drive mixer system.

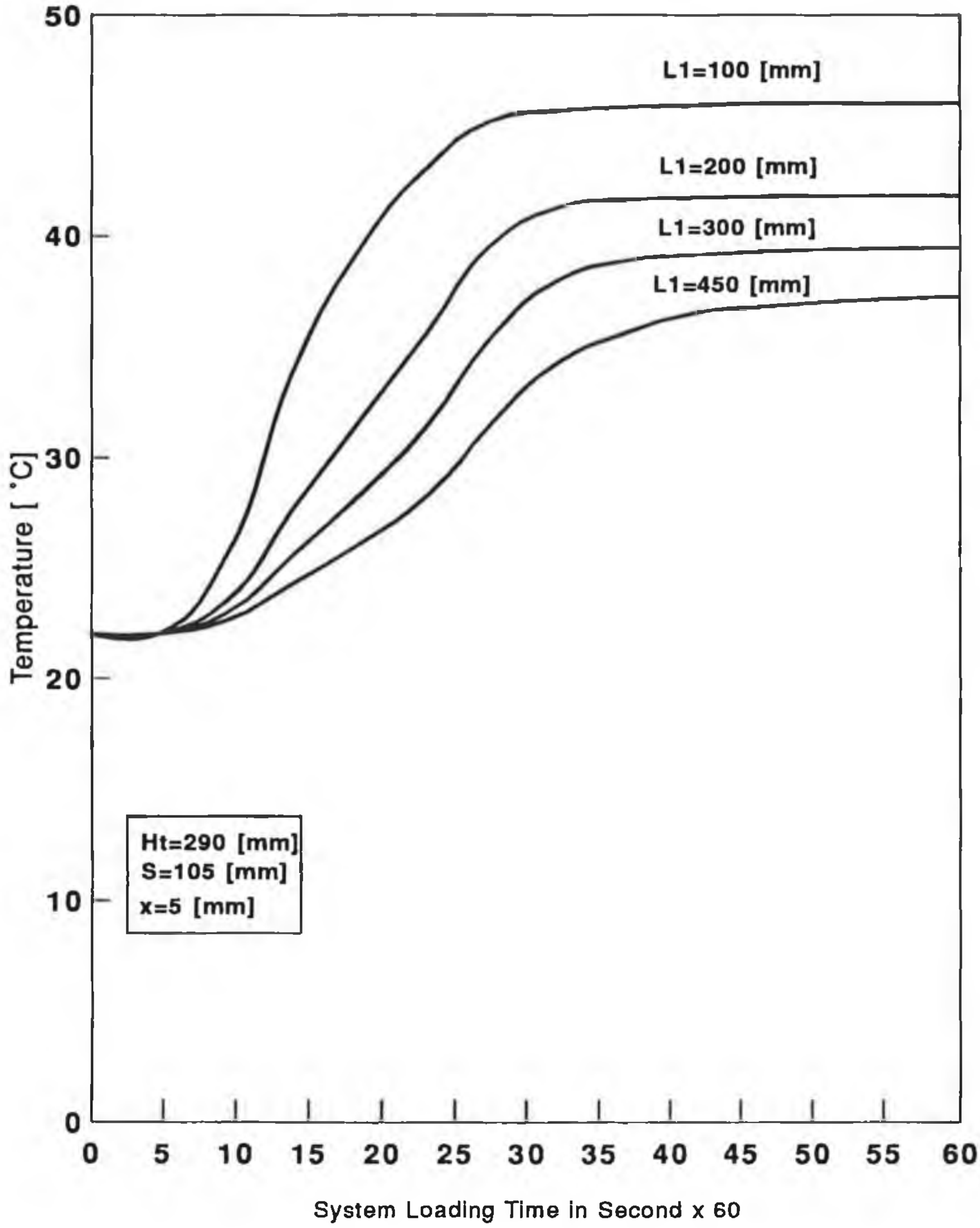


Fig. (54) The effects of the length of the base of the reservoir on the outside hose wall temperature variation in the open hydraulic drive mixer system.

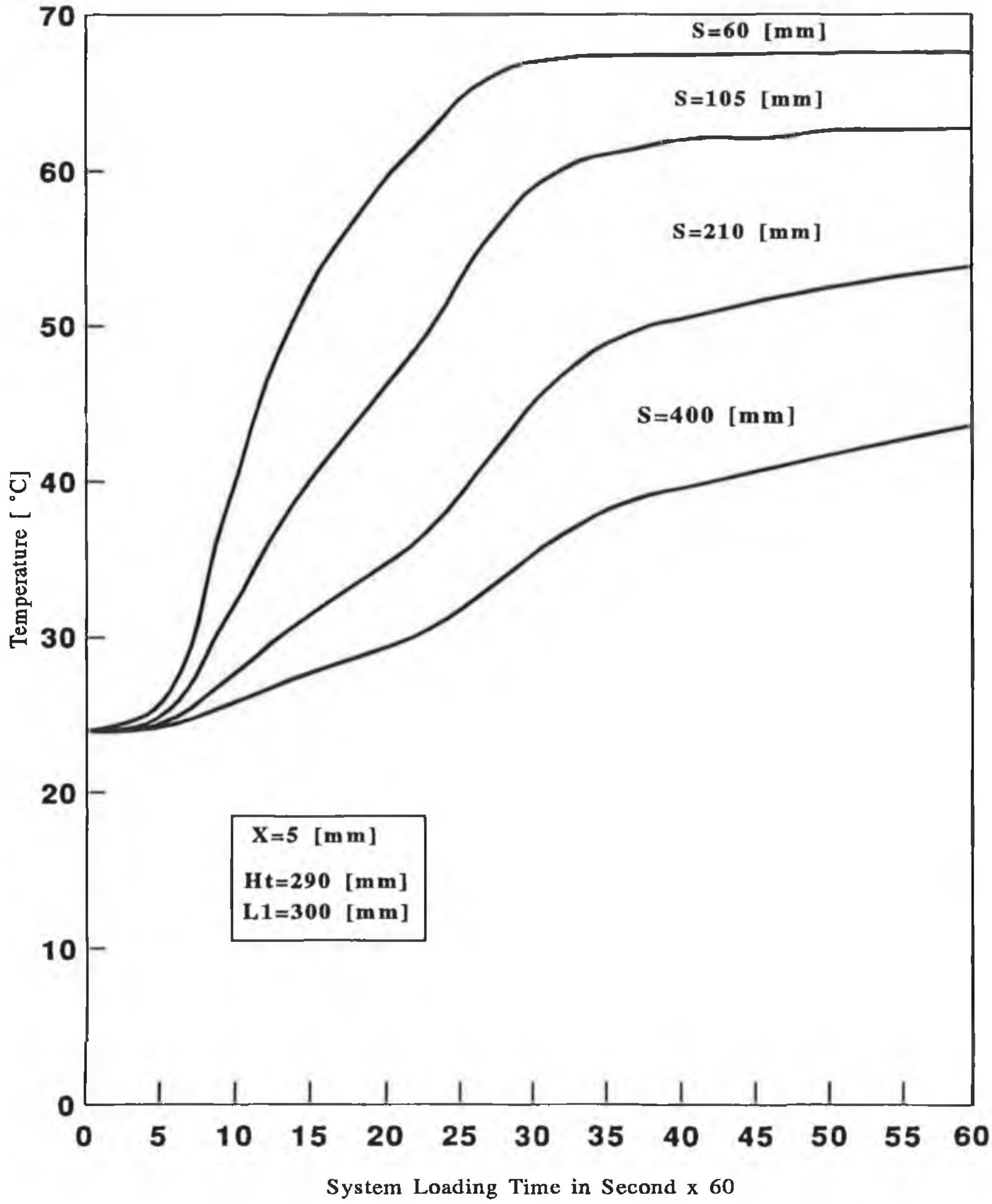


Fig. (55) The effects of the width of the base of the reservoir on fluid temperature variation in the open hydraulic drive mixer system.

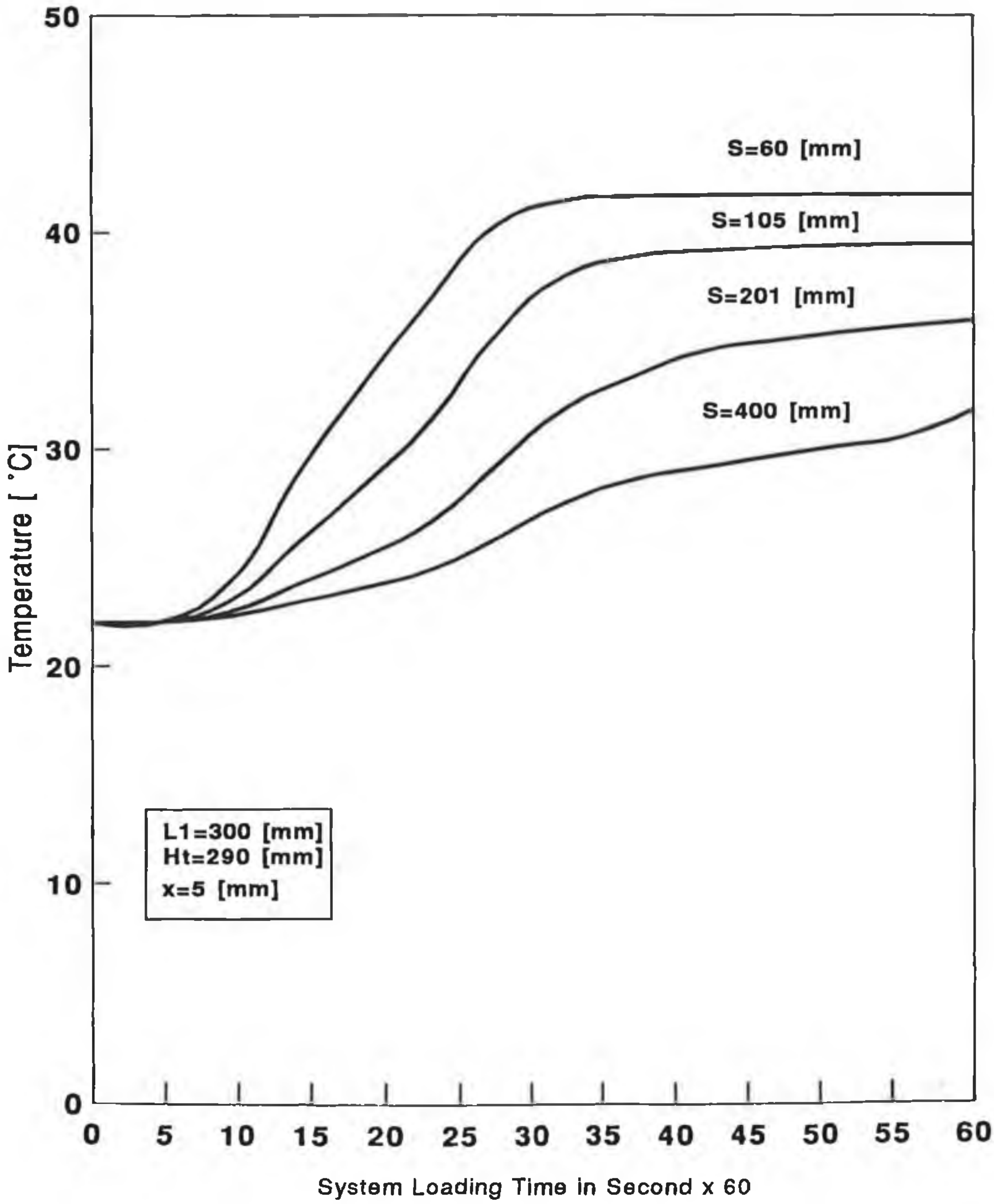


Fig. (56) The effects of the width of the base of reservoir on the outside hose wall temperature in the open hydraulic drive mixer system.

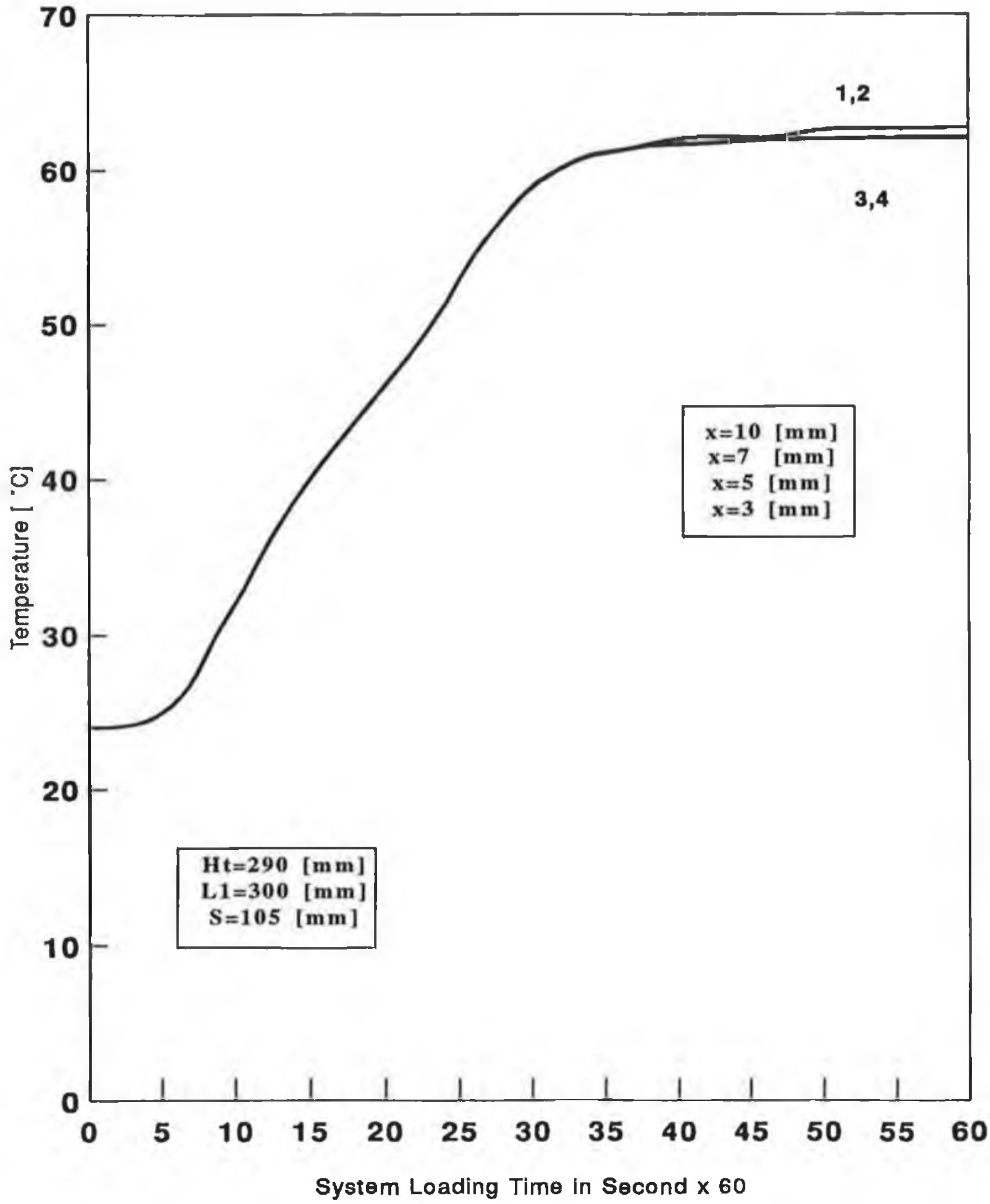


Fig. (57) The effects of thickness of the wall of the reservoir on the fluid temperature variation in the open hydraulic drive mixer system

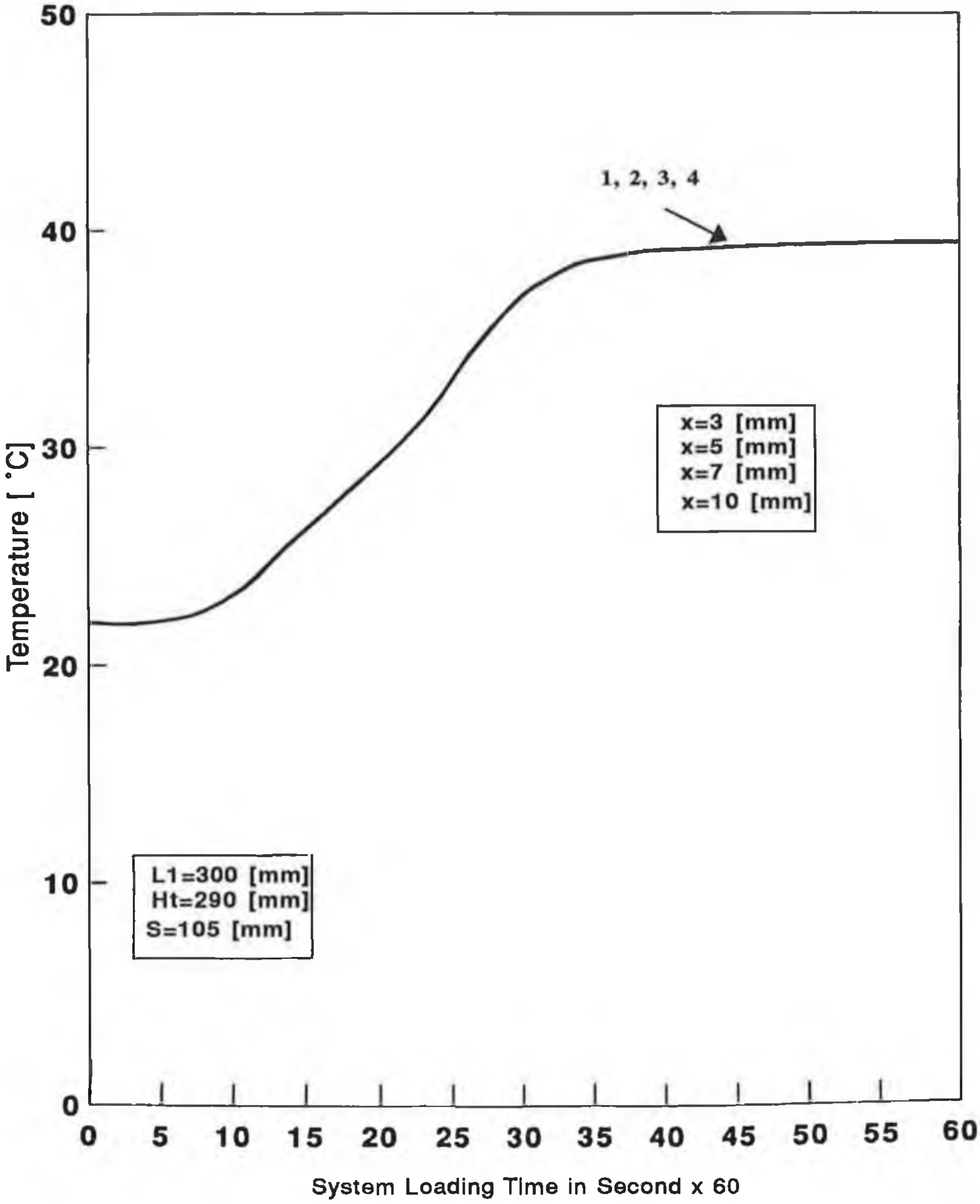


Fig. (58) The effects of the thickness of the wall of the reservoir on the outside hose wall temperature in the open hydraulic drive mixer system.

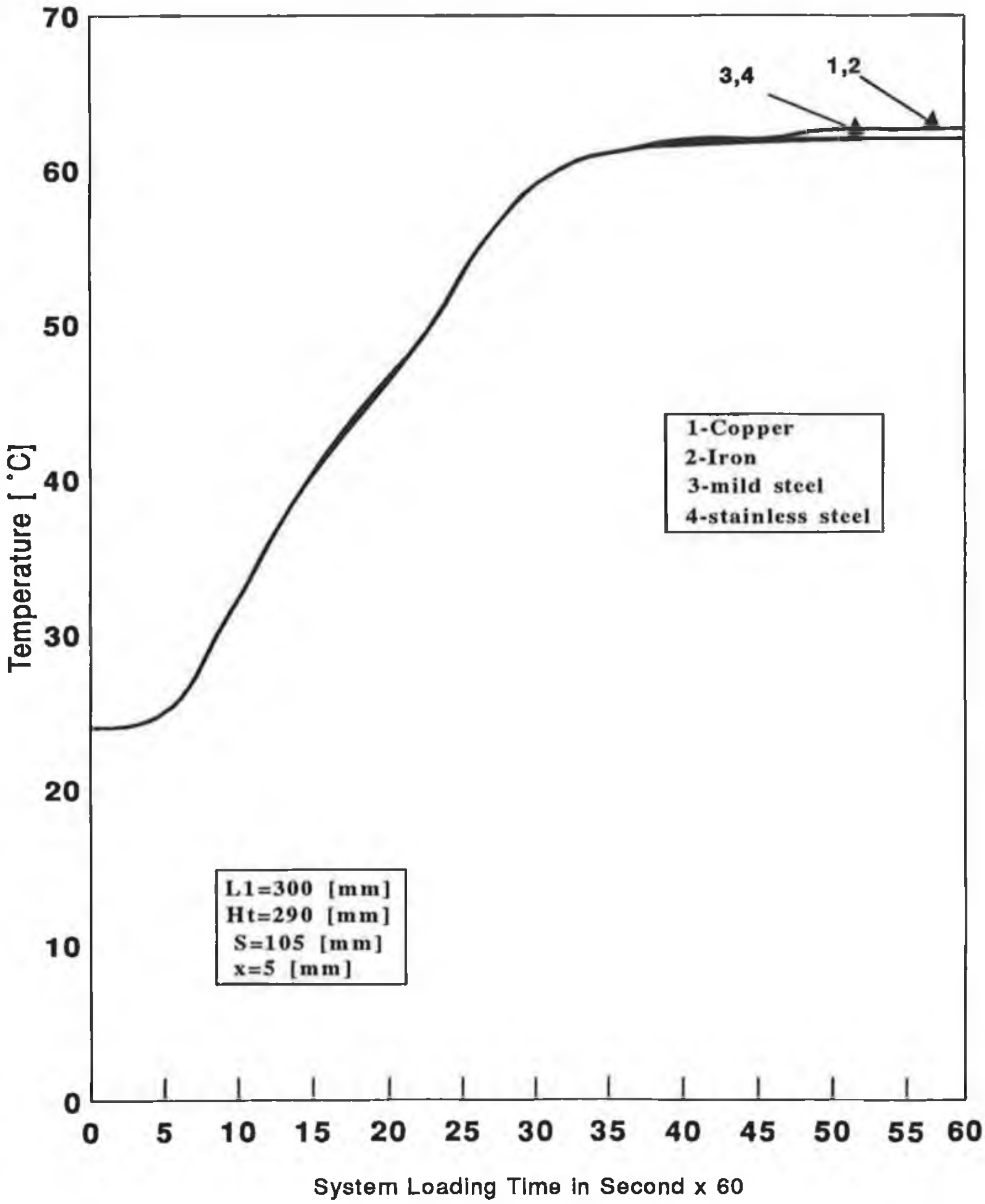


Fig. (59) The effects of Material of the reservoir on fluid temperature variation in the open hydraulic drive mixer system.

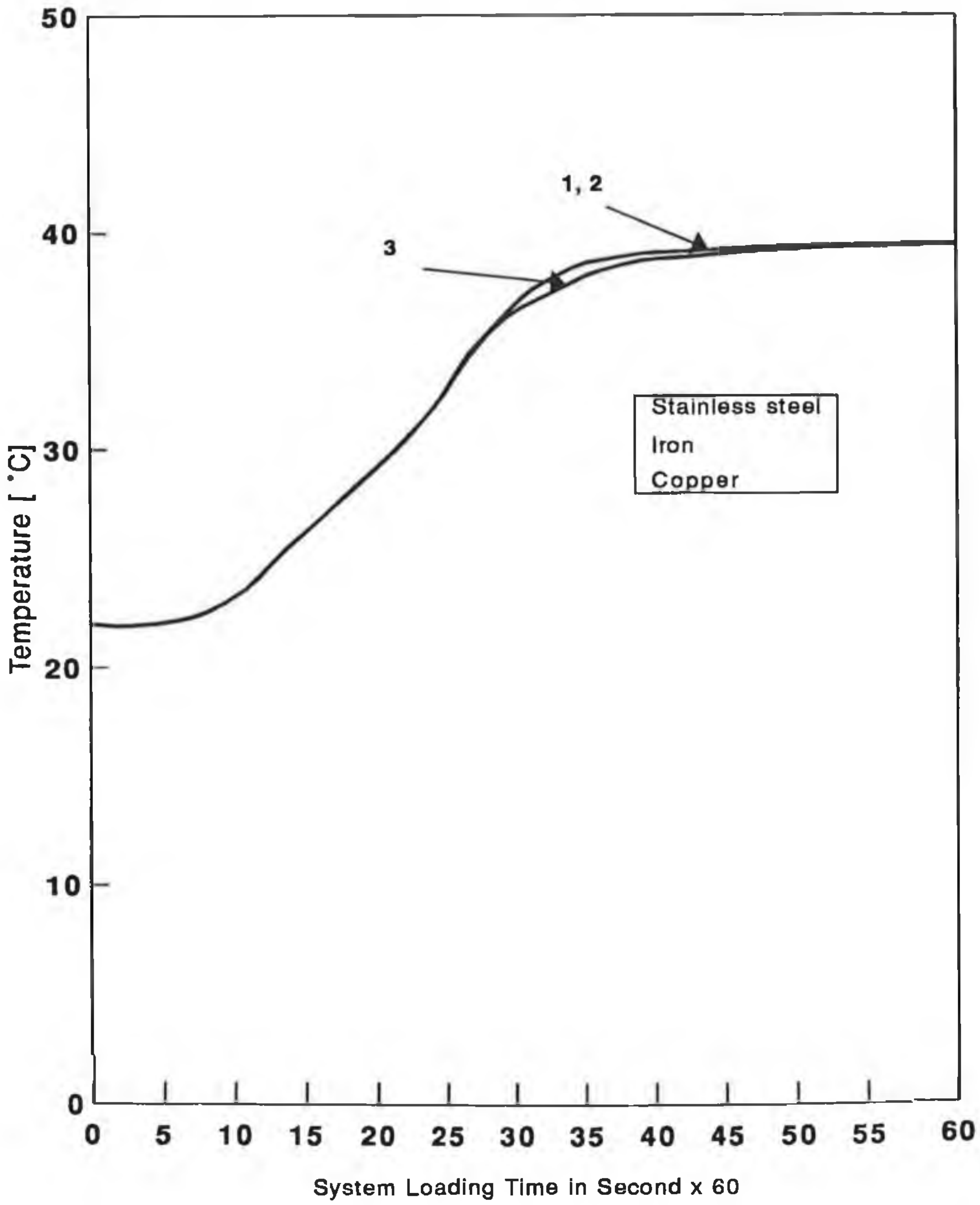


Fig. (60) The effects of material of the reservoir on the outside hose wall temperature variation in the open hydraulic drive mixer system.

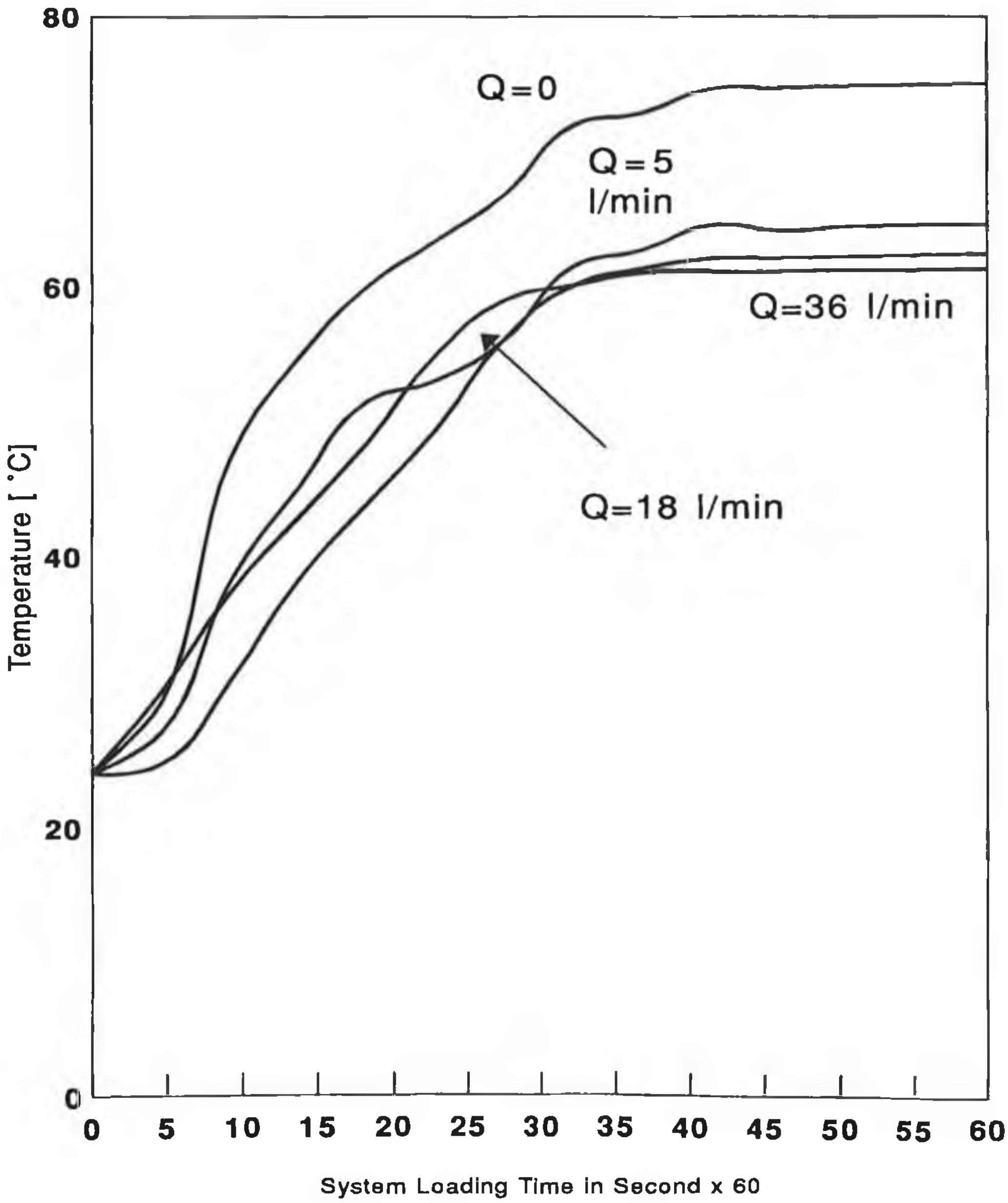


Fig. (61) Simulated results of the effects of exchange flow rate between loop and reservoir on loop fluid temperature in the open hydraulic drive mixer system.

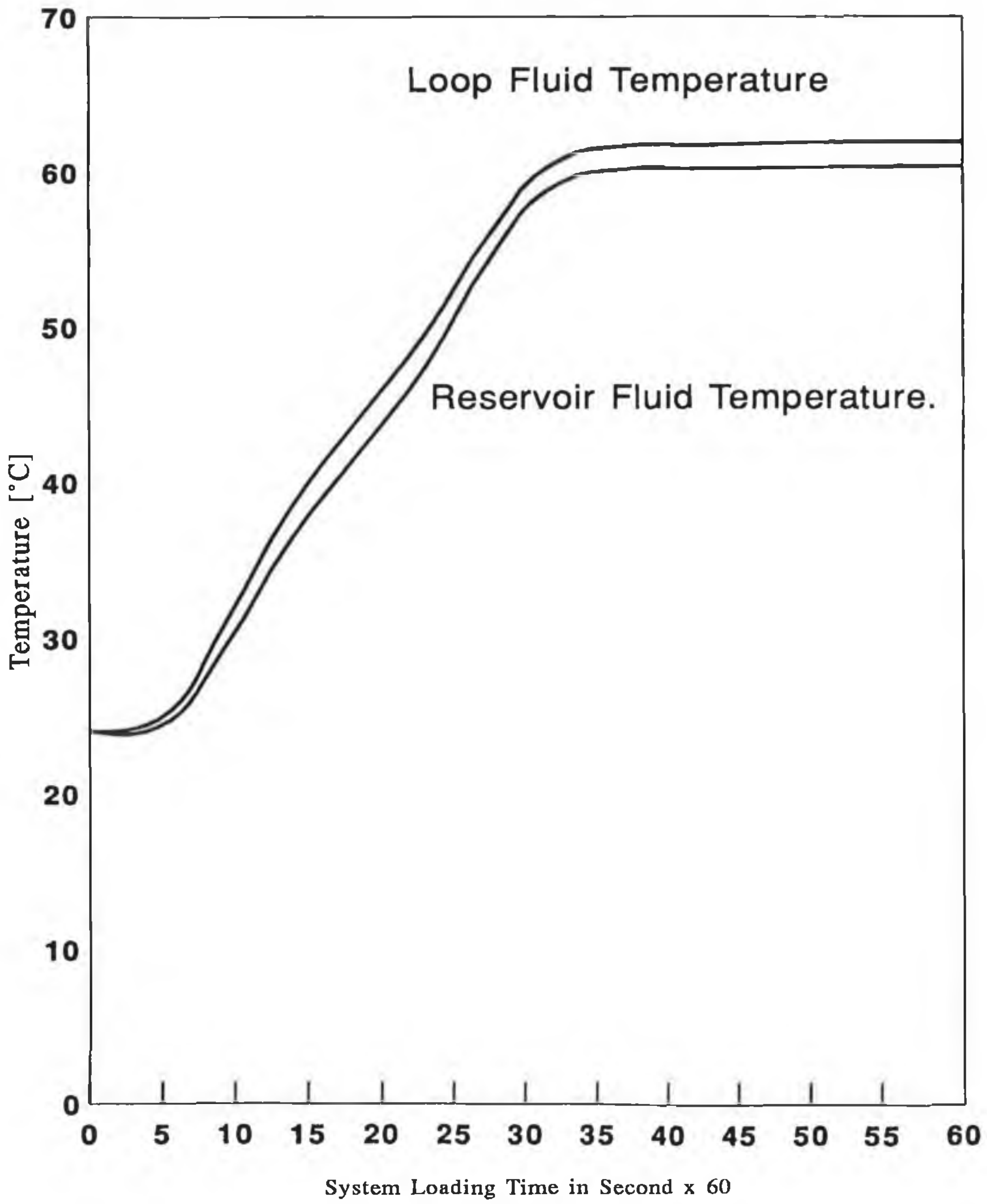


Fig. (62) Comparison of simulated results of loop and reservoir fluid temperature in the open hydraulic drive mixer system.

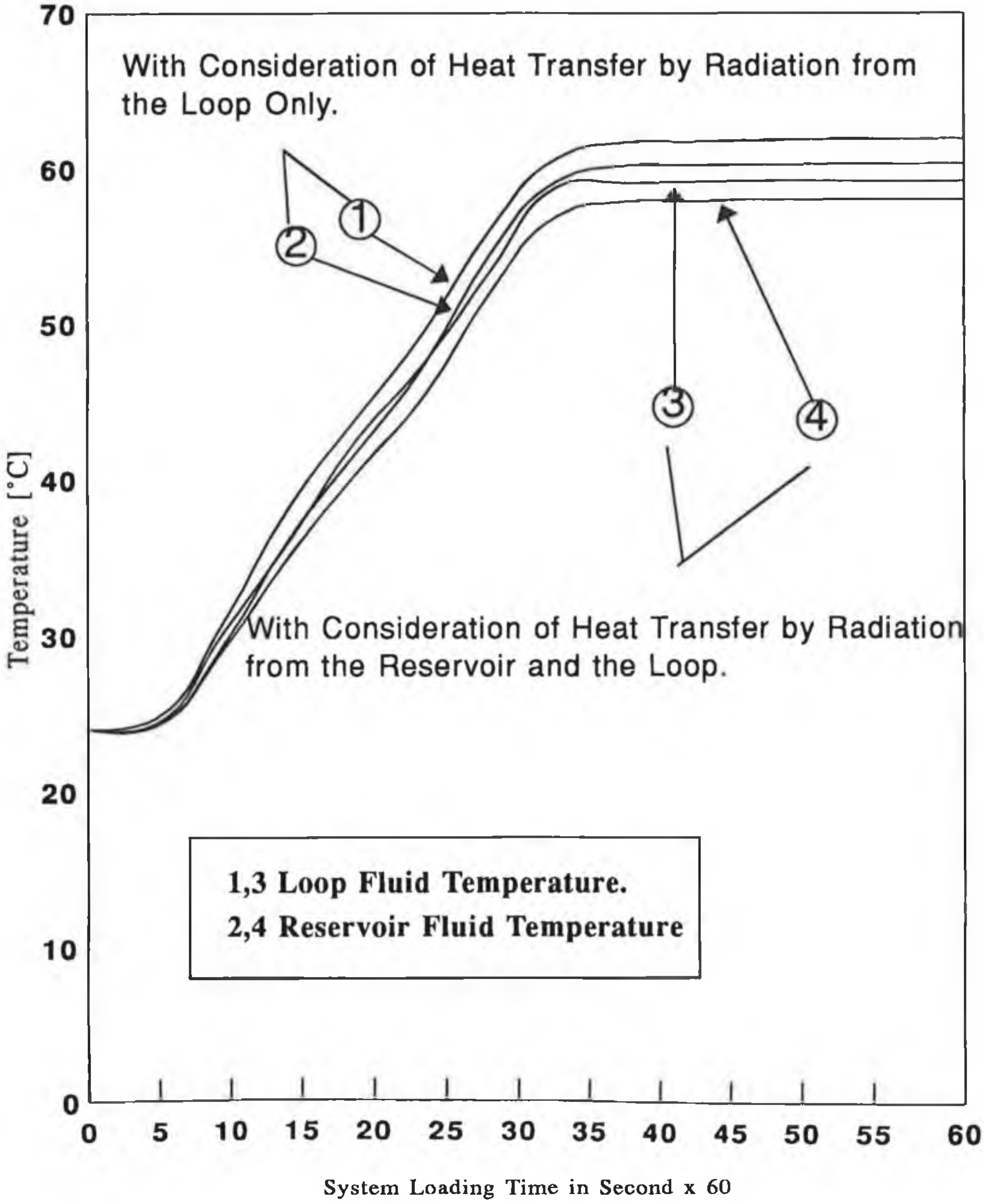


Fig. (63) Comparison of simulated results of loop and reservoir fluid temperature with consideration of heat Transfer by radiation in the open hydraulic drive mixer system.

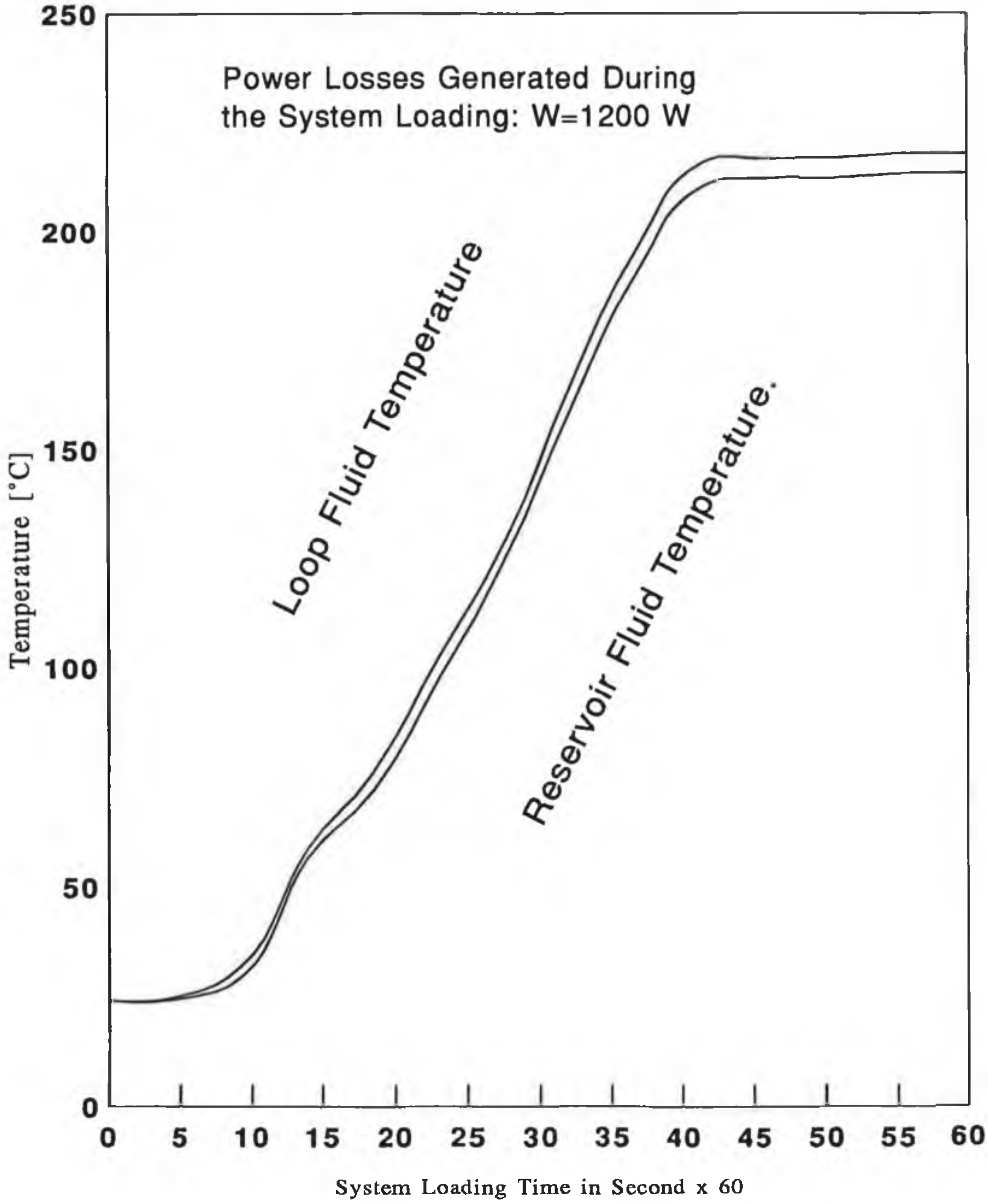


Fig. (64) Comparison of simulated results of loop and reservoir fluid temperatures with a big increase of the system power loss in the open hydraulic drive mixer system.

CHAPTER: 4

Experimental Investigation and Results

4.1. Experimental Procedure: The following procedure was followed before carrying out any test. The hydraulic oil reservoir was filled with Shell TELLUS-37 oil to the level mark on the sight glass. The setting of the relief valve was adjusted on 150 bar as a system working pressure. The container was filled with different material to get different load. The system control valve was in the OFF position. Four thermocouples with digital output were attached firmly on the surface of suction line, pressure line and return line. Three quick-disconnectable hose couplings were coupled with the hoses where frequent connection are made and broken between the hoses.

After the above arrangement, the hydraulic pump was run at 1500 r.p.m and the solenoid directional control valve was switched on. The hydraulic system was run in off load condition for 5 minutes to allow air to be fully expelled from the system. In order to run the drive motor (hydraulic motor), the handle of the control valve was moved to ON position. The entire system installation was checked for oil leaks and rectified when necessary.

The system was run for one hour under load in order to reach the steady thermal condition. Initially, the pipe wall and fluid temperatures changed rapidly and the temperature readings were therefore recorded at one minute interval for the first five minutes, every two minutes for the following 25 minutes and then every five minutes towards the end of the tests.

Following closely the above test procedure, an extensive test programme was undertaken to investigate experimentally the feasibility of providing an efficient oil cooler to small sized hydraulic circuits which will reduce heat problems, reservoir size and will be an integral part of the circuit. In addition, the experimental results were recorded to verify the developed mathematical model.

4.2. Experimental Results: A typical set of temperature recordings from the experimental test is presented in Fig. (65) to Fig (94). These temperature values were obtained from the experimental recordings at each time interval, but for the convenience of temperature analysis, all data of the experimental results have been simply joined together with curves. Some minor scatter was observed as shown in some experimental temperature results. Nevertheless, these experimental curves can indicate the trends of temperature rise with sufficient accuracy.

Fig. (65) shows the measured temperature of the outside hose wall at the entry and exit points of the hydraulic motor for a flow rate of 30 l/min It was observed in this figure that the measured temperatures increased rapidly within the first 30 minutes of the system loading time while in the second 30 minutes of the operation, the change in the temperature was lower than the first 30 minutes. This is due to the achievement of the steady state of the fluid temperature.

Figs (66) to (69) show the measured temperature of the outside hose wall of second part of pressure line and return line of the hydraulic motor for flow rates of 25, 20, 10, and 5 l/min respectively. The general trends of the results were found to be similar to those in fig. (65). Higher temperatures were recorded at higher loads and lower flowrate as shown in figs (68) and (69).

Fig. (70) shows the measured temperature of the outside hose wall of the first part of the pressure line and suction line of hydraulic pump for flow rate of 30 l/min This figure shows higher rate of temperature as recorded during the first 35 minutes of the system loading time. For system loading time in excess of about 50 minutes, the recorded temperatures were observed to be approximately constant due to the stability of steady state condition.

Figs (71) to (74) give the measured temperatures of the outside hose wall of the first part of the pressure line and suction line of the hydraulic pump for flow rates of 25, 20, 10, and 5 l/min respectively. The results of these figures were found to be similar to those in fig. (70). The measured temperatures were greater as the load was increased. For the system loading time in excess of about 55 minutes, the temperature readings were found to be approximately constant.

Fig. (75) shows the experimental results of temperature variation of fluid for flow rate of 30 l/min in the open hydraulic drive mixer system. In this figure, the temperature increases as the system loading time increases for up to 30 minutes; after 30 minutes of the operation under the load, the temperature is changed slightly due to the thermal steady state. Figs. (76) to (78) were found to be similar to those in Fig. (75).

Figs (79) and (80) show the results of the effects of flow rate on the outside hose wall temperature of the return and pressure lines respectively. These figures show that the measured temperature of the outside hose wall is increased as flow rate is decreased therefore pressure is increased. It was observed that the change of the measured temperature at flow rate of 10, 5 l/min (P=110, 140 bar) is much higher compared to those shown in the same figures for flow rate of 30, 25 and 20 l/min

Figs. (81) and (82) demonstrate the experimental results of the effects of the flow rate on fluid temperature of return line and pressure line in the open hydraulic drive mixer system. The general trends of the results were found to be similar to these in Figs. 79 and 80. Higher values of temperature were observed for lower values of flowrate.

The constant temperature values of steady state have not been achieved in some experimental result as shown in Fig. (83) which shows the comparison of hose wall temperature between pressure and return lines in the open hydraulic drive system. A possible reason is the gradual increase in the atmospheric temperature surrounding to the test rig during the tests of system loading. Moreover, fluid temperature at the motor inlet in the second part of the high pressure line is approximately 3 °C higher than of that recorded at the pump outlet in the first part of the high pressure line as shown in Fig. (84). However, on the basis of the well known temperature-pressure drop coefficient 5 °C/100 bar ,Ref. (48), in a pipe line, it can be seen that the 3 °C difference value in the experimentally measured fluid temperatures in the parts of the high pressure line is only partly caused by the pressure losses across the fittings, but mainly caused by the errors in measurements of the experimental fluid temperature.

Similarly, a small temperature difference is shown in the corresponding pipe wall temperatures as shown in Fig. (85). These small temperature differences can be ignored, so that the fluid temperature as well as pipe wall temperature can be regarded as having uniform value over the pipe line. However, the experimentally obtained temperature results from the different parts of the system show a significant difference between the fluid and pipe wall temperature, typical examples are shown in Figs. (86) and (87). These temperature differences between the fluid and pipe wall can not be ignored even if the experimental measurement errors have been taken into account.
Fig. (88) demonstrates the hose wall temperature in different parts of the system. This figure indicates a big difference in value of the experimentally obtained pipe wall temperature. The maximum difference between the measured wall temperatures in the different parts of the pipeline is about 7 °C. This is due to pressure drop through the fittings and the errors in the temperature measurements.

In the experimental tests, the maximum fluid temperature, especially at flowrate of 10 and 5 l/min, was above the usual recommended working value of 50-60 °C. The typical set of temperature recordings from the experimental test is presented in Fig. (89) to Fig. (94). Fig. (89) shows the experimental results of comparison of fluid temperature in the pressure line before and after using the cooling unit as an integral part of the system. As can be seen, the temperature in the system before the cooler is about 10 °C higher than recommended. But, after the oil cooler the temperature is about 45 °C. That means by using the oil cooler the temperature decreases about 25 °C. Fig. (90) gives the experimental results of comparison of fluid temperature in the return line before and after the cooler. This figure is similar in trend as that in fig. 89. This figure suggests that the temperature would be decreased about 35 °C by using the oil cooling unit in the open hydraulic drive mixer system. However, by using the cooling unit in the system the working fluid temperature is maintained below the recommended value because a system that is allowed to overheat can cause costly seal deterioration and fluid oxidation. In addition, oil viscosity will decrease and system operation may be become erratic. Fig. (91) shows the comparison of fluid temperature between the return line, pressure line and the temperature of the fluid returning to the reservoir. This figure suggests that the temperature can be reduced by 40 % by using the cooling unit.

Fig.(92) shows the flowrate versus temperature with the effects of the fan speed in the cooling unit on the performance of the open hydraulic drive mixer system. This figure suggests that as the flowrate decreases, the working temperature in the system increases. As expected, as the flowrate is increased, the fan speed in the cooling unit increases, therefore the temperature of the fluid is decreased. Figs.(93) and (94) give the effect of flowrate on the fan speed and working pressure respectively. these figures demonstrate that as the flowrate increases, the fan speed and working pressure decrease.

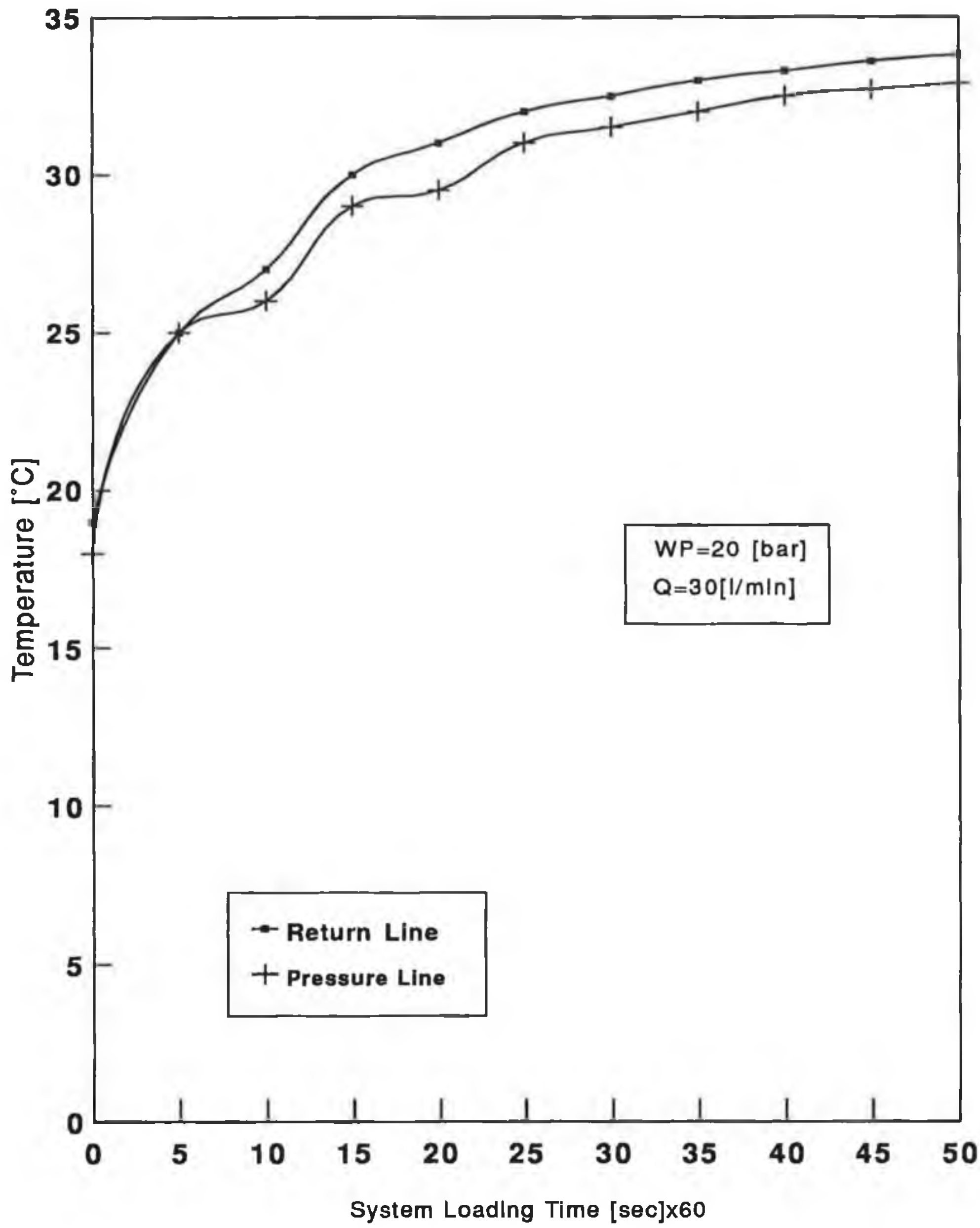


Fig. (65)Experimental results of temperature variation of outaide hose wall of hydraulic motor for Flow rate of 30 l/min in the open hydraulic drive mixer system.

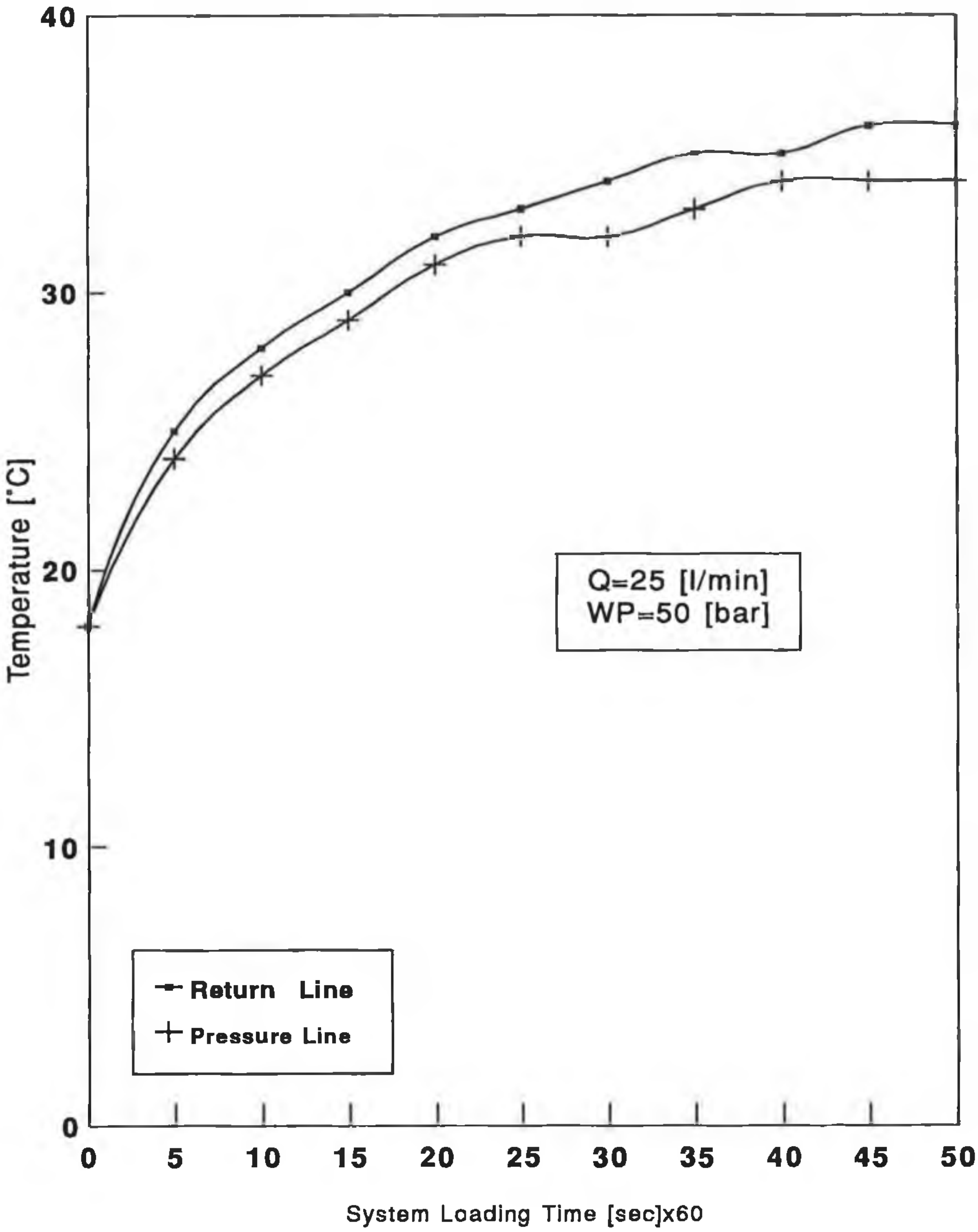


Fig. (66) Experimental results of temperature variation of the outside hose wall of the hydraulic motor for flow rate of 25 l/min. in the open hydraulic drive mixer system.

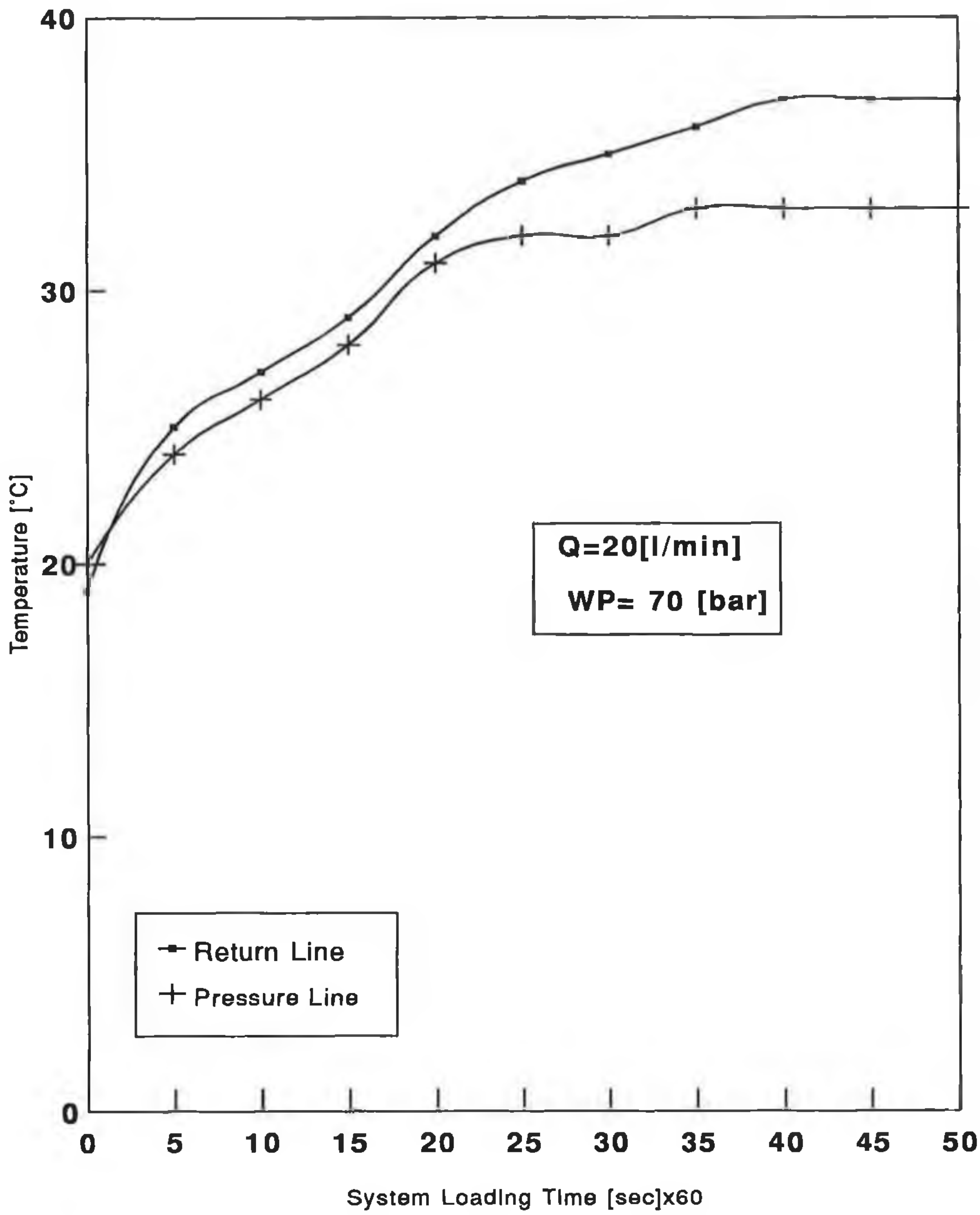


Fig. (67) Experimental results of temperature variation of the outside hose wall of return line of hydraulic motor at flow rate of 20 l/min in the open hydraulic drive mixer system.

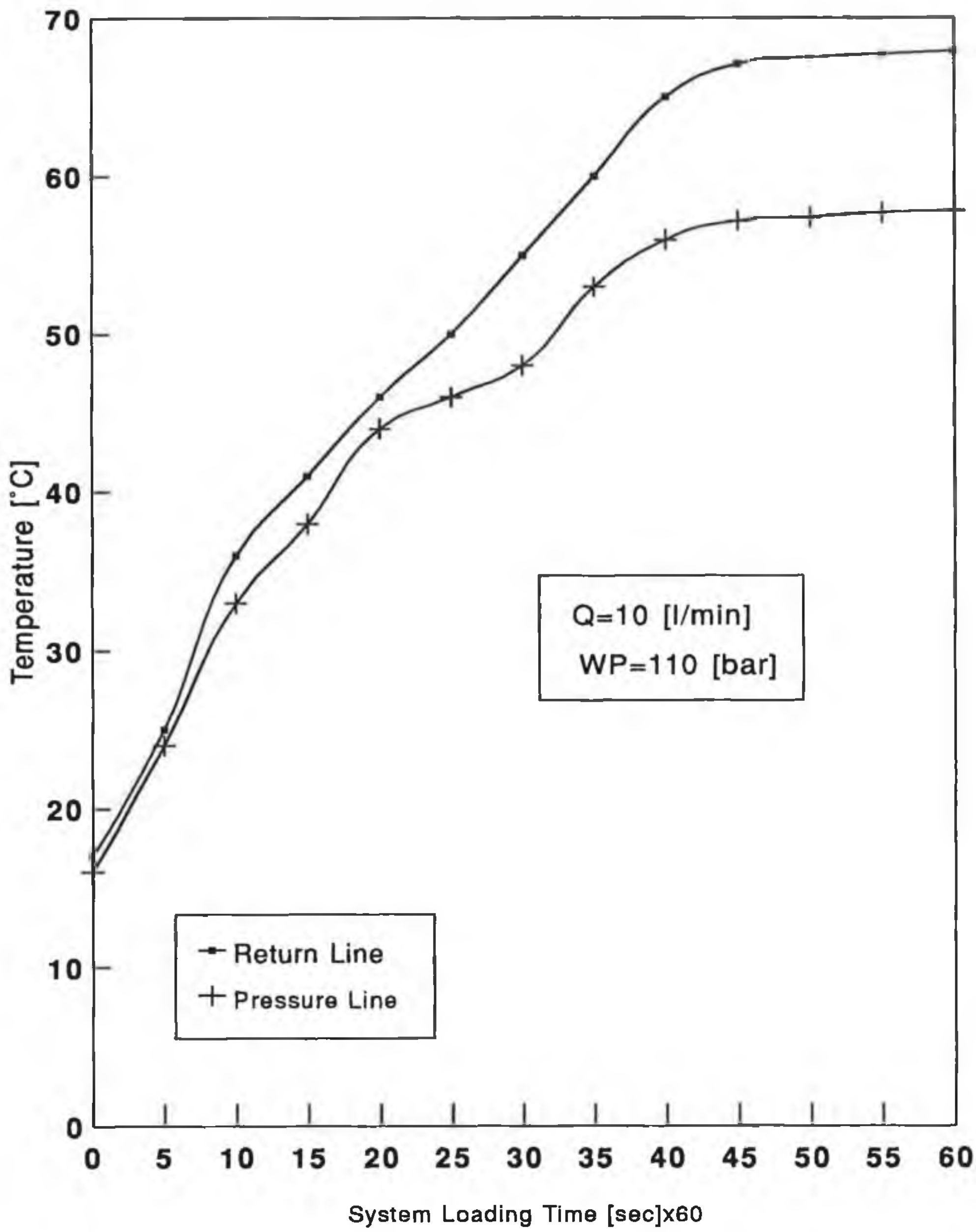


Fig. (68) Experimental results of temperature variation of outside hose wall of hydraulic motor for flow rate of 10 l/min in the open hydraulic drive mixer system.

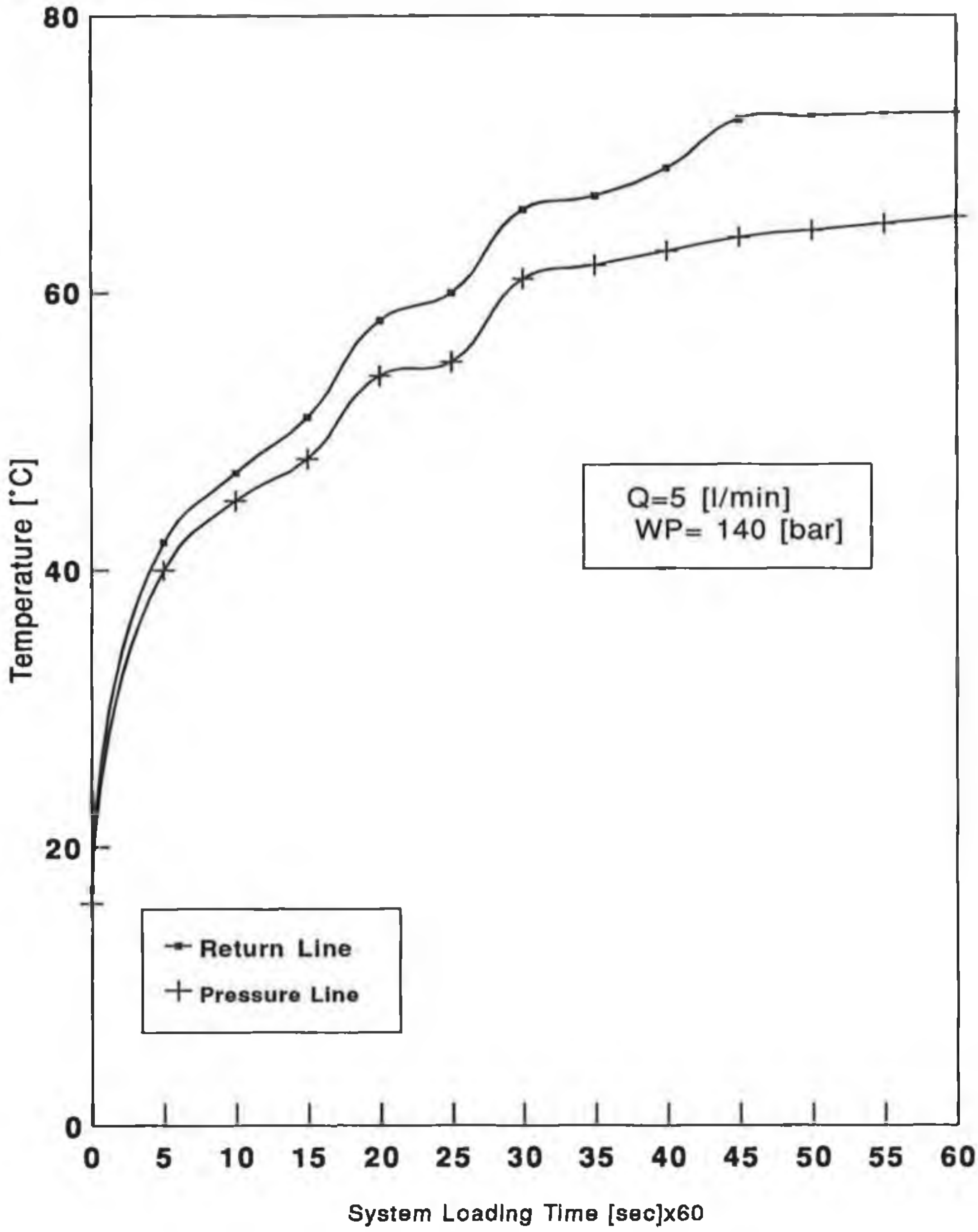


Fig. (69) Experimental results of temperature variation of outside hose wall of hydraulic motor for flow rate of 5 l/min in the open hydraulic drive mixer system.

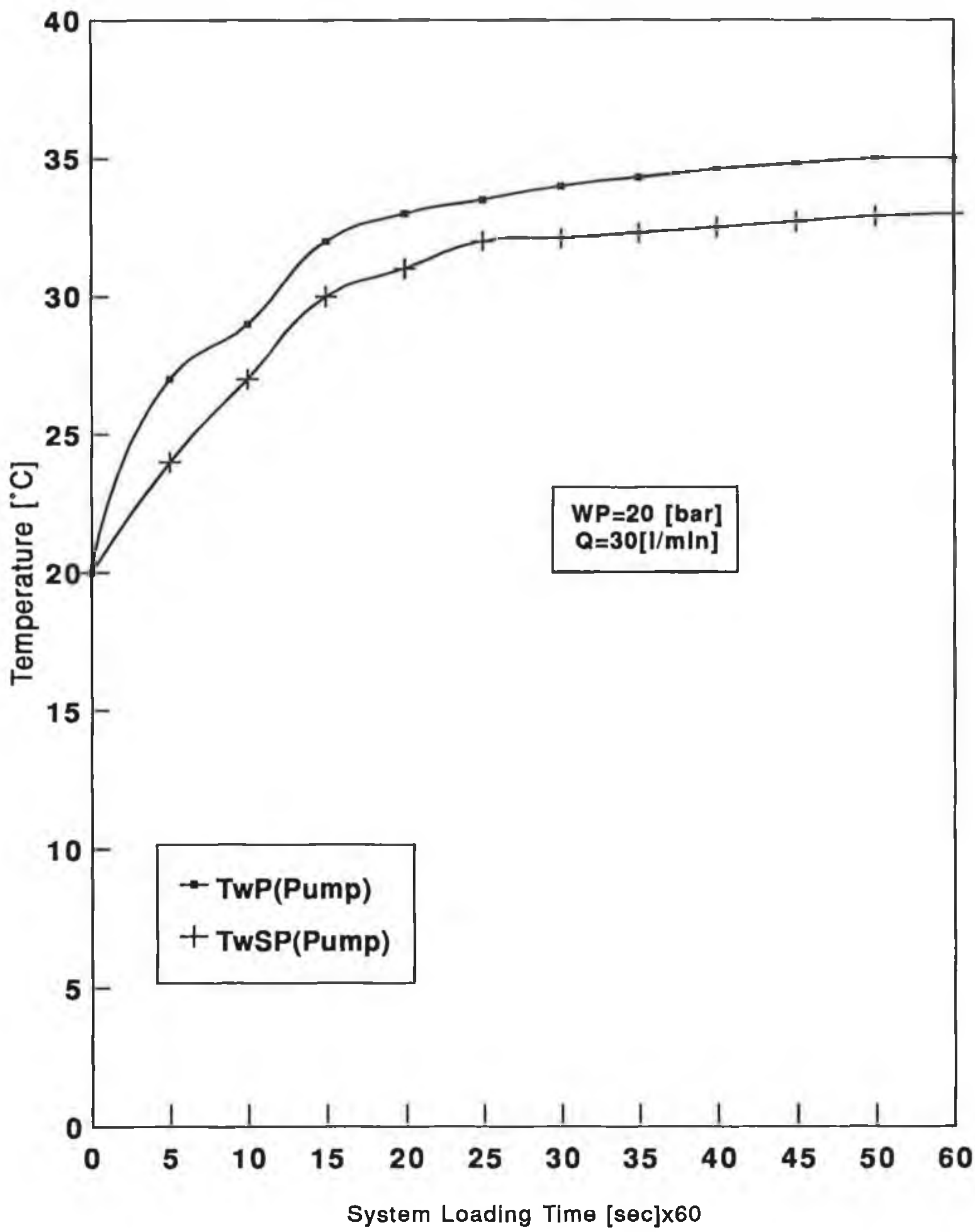


Fig. (70) Experimental results of temperature variation of outside hose wall of hydraulic pump for flow of 30 l/min in the open hydraulic drive mixer system.

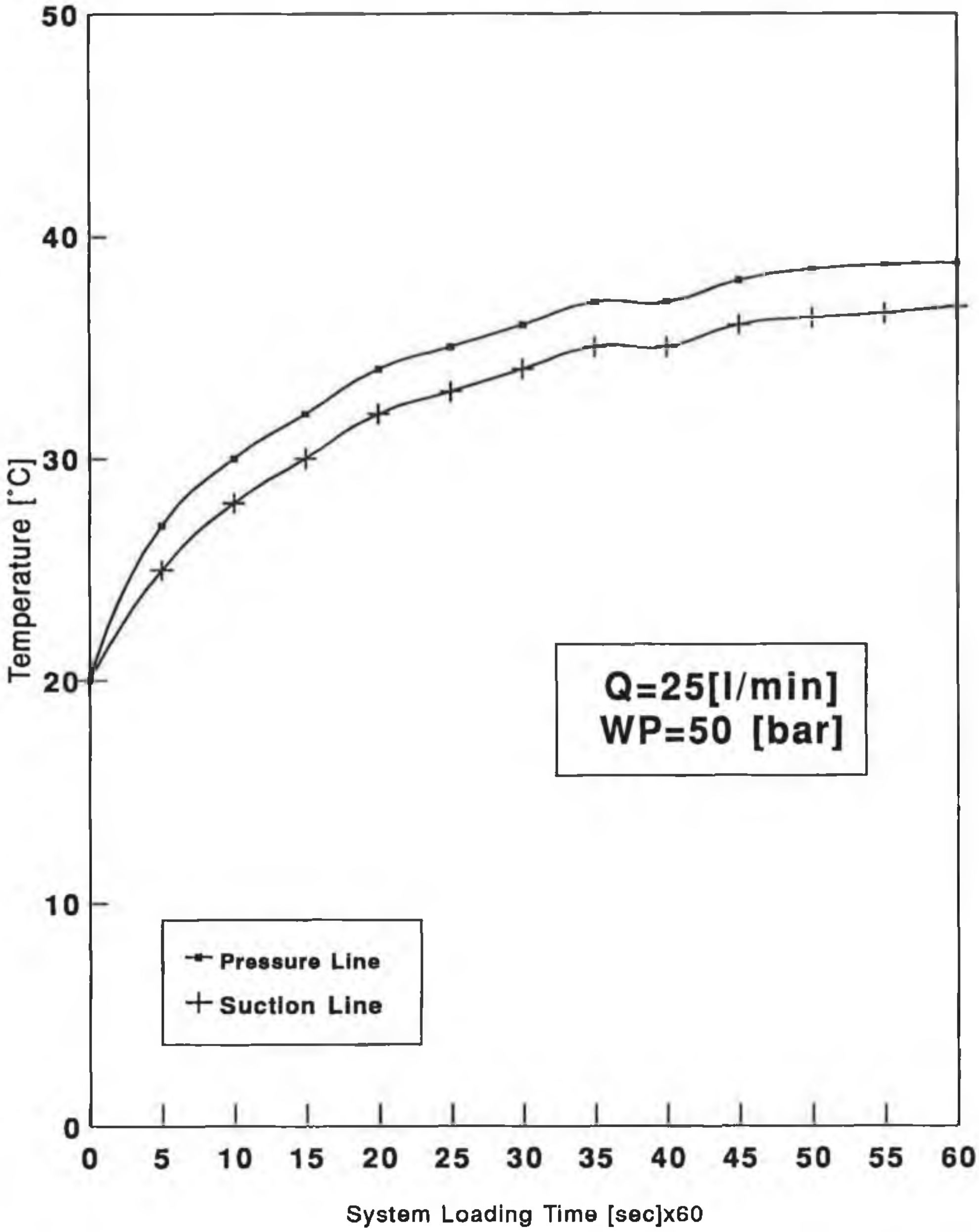


Fig. (71) Experimental results of temperature variation of outside hose wall of hydraulic pump for flow rate of 25 l/min in the open hydraulic drive mixer system.

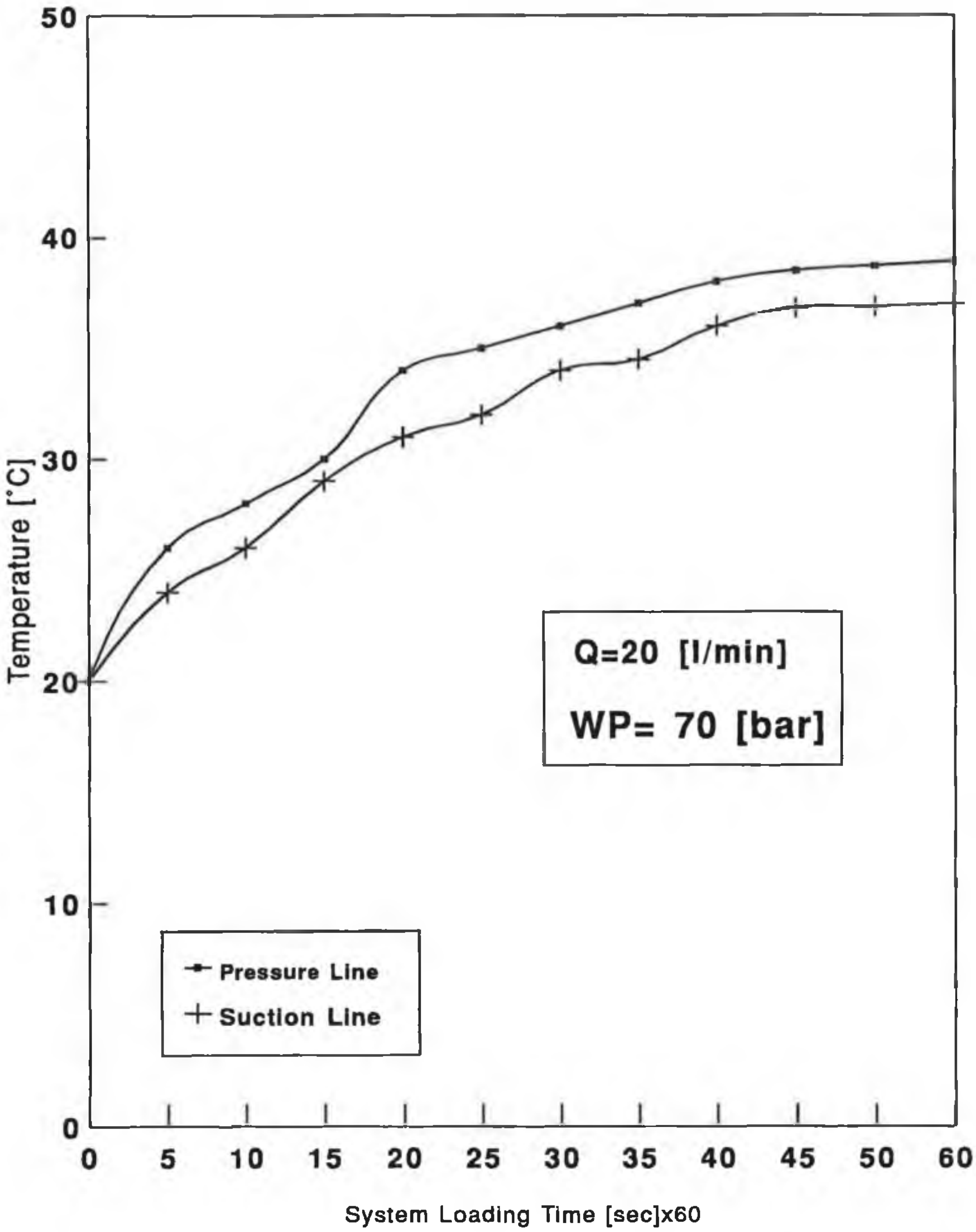


Fig. (72) Experimental results of temperature variation of outside hose wall of hydraulic pump for flow rate of 20 l/min in the open hydraulic drive mixer system.

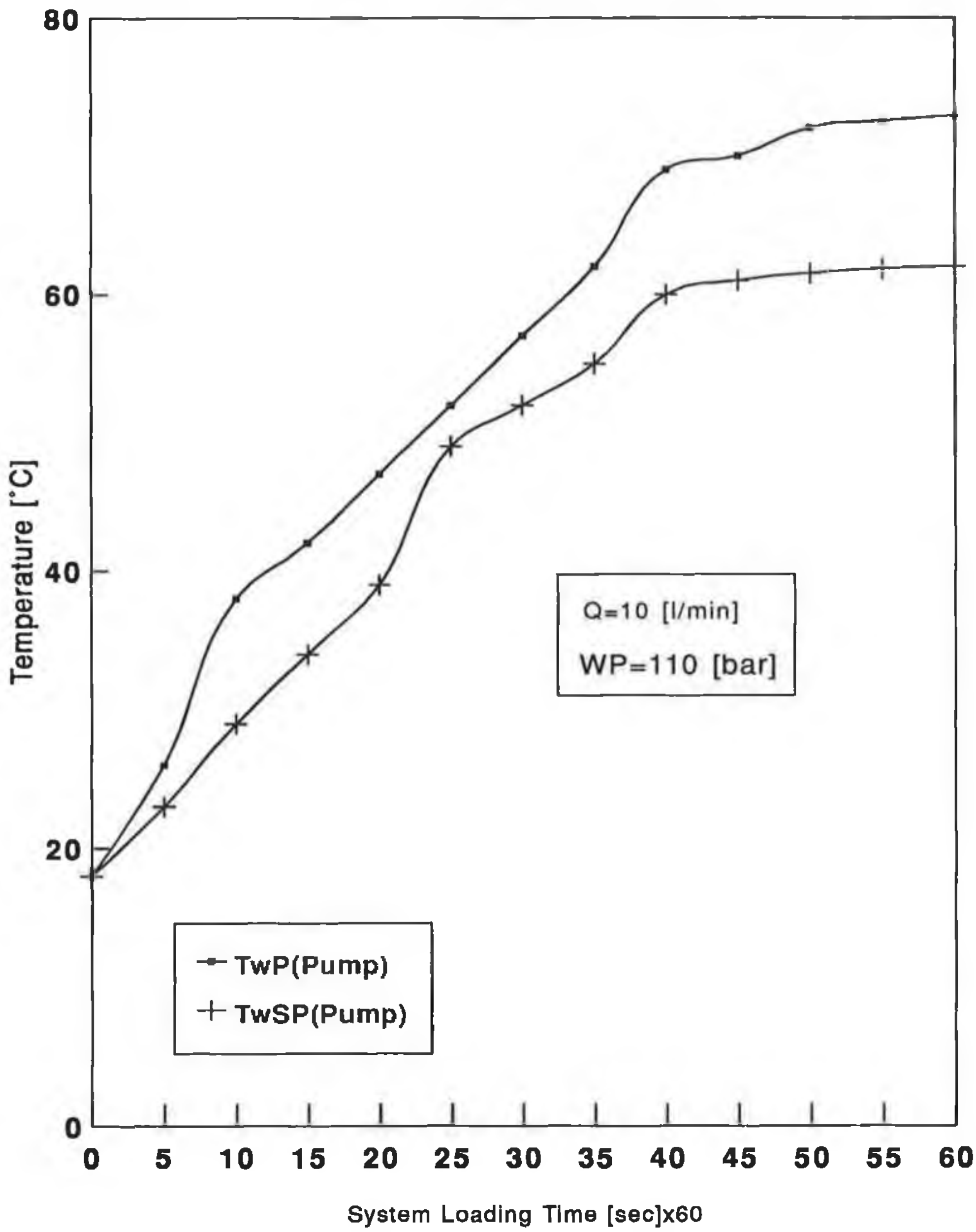


Fig. (73) Experimental results of temperature variation of outside hose wall of hydraulic pump for flow rate of 10 l/min in the open hydraulic drive mixer system.

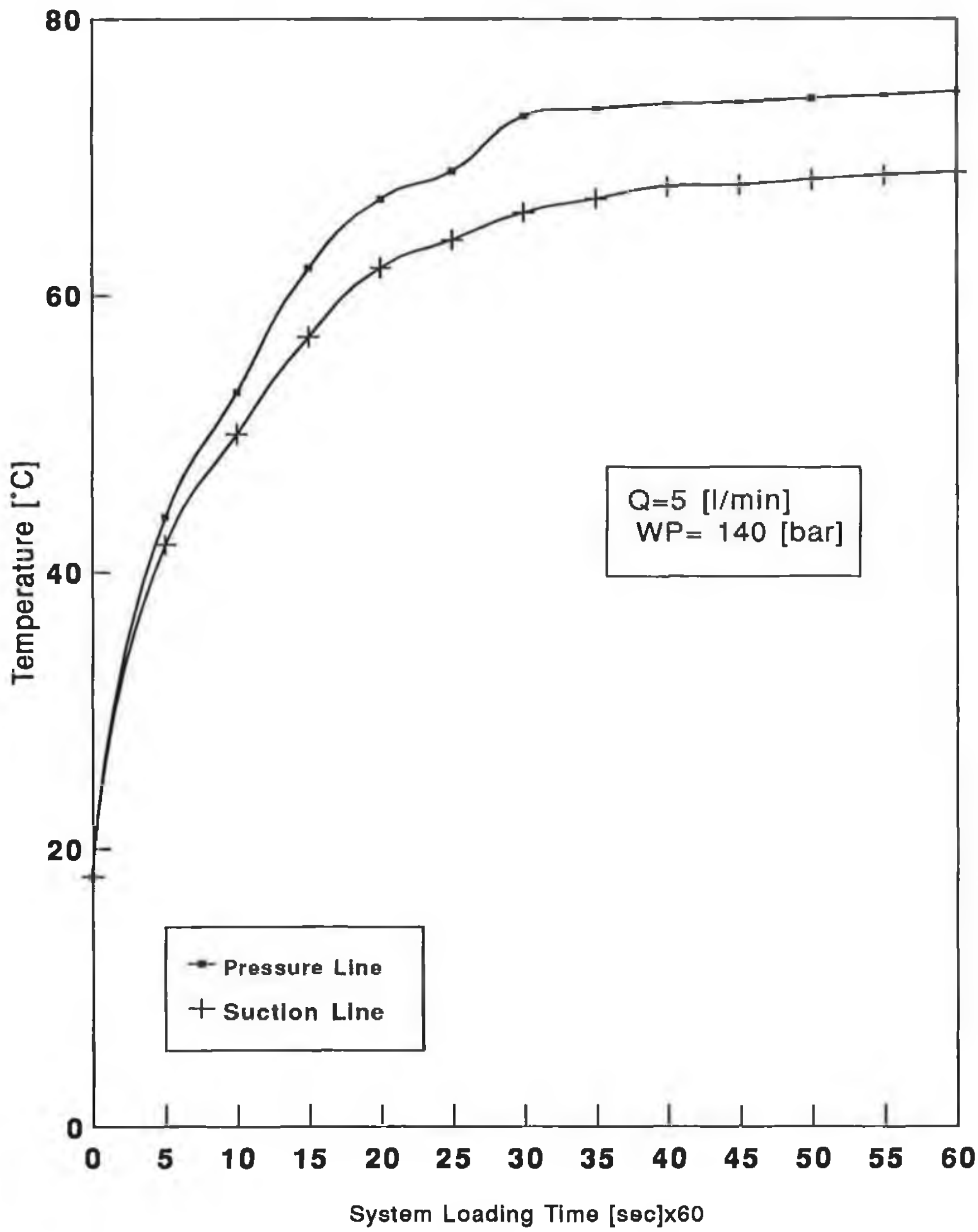


Fig. (74) Experimental results of temperature variation of outside hose wall of hydraulic pump for flow rate of 5 l/min in the open hydraulic drive mixer system.

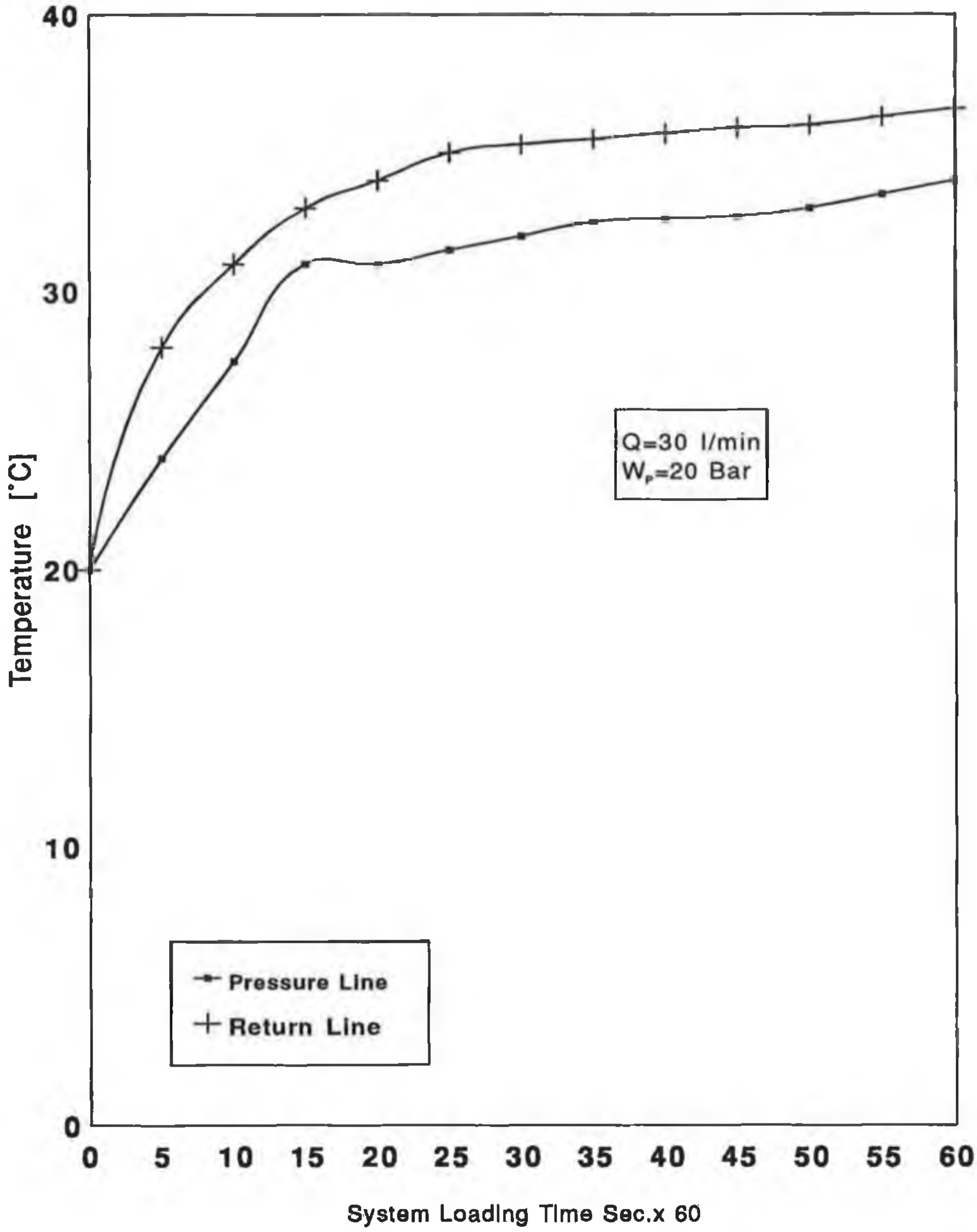


Fig. (75) Experimental results of temperature variation of fluid for flow rate of 30 l/min in the open hydraulic drive mixer system

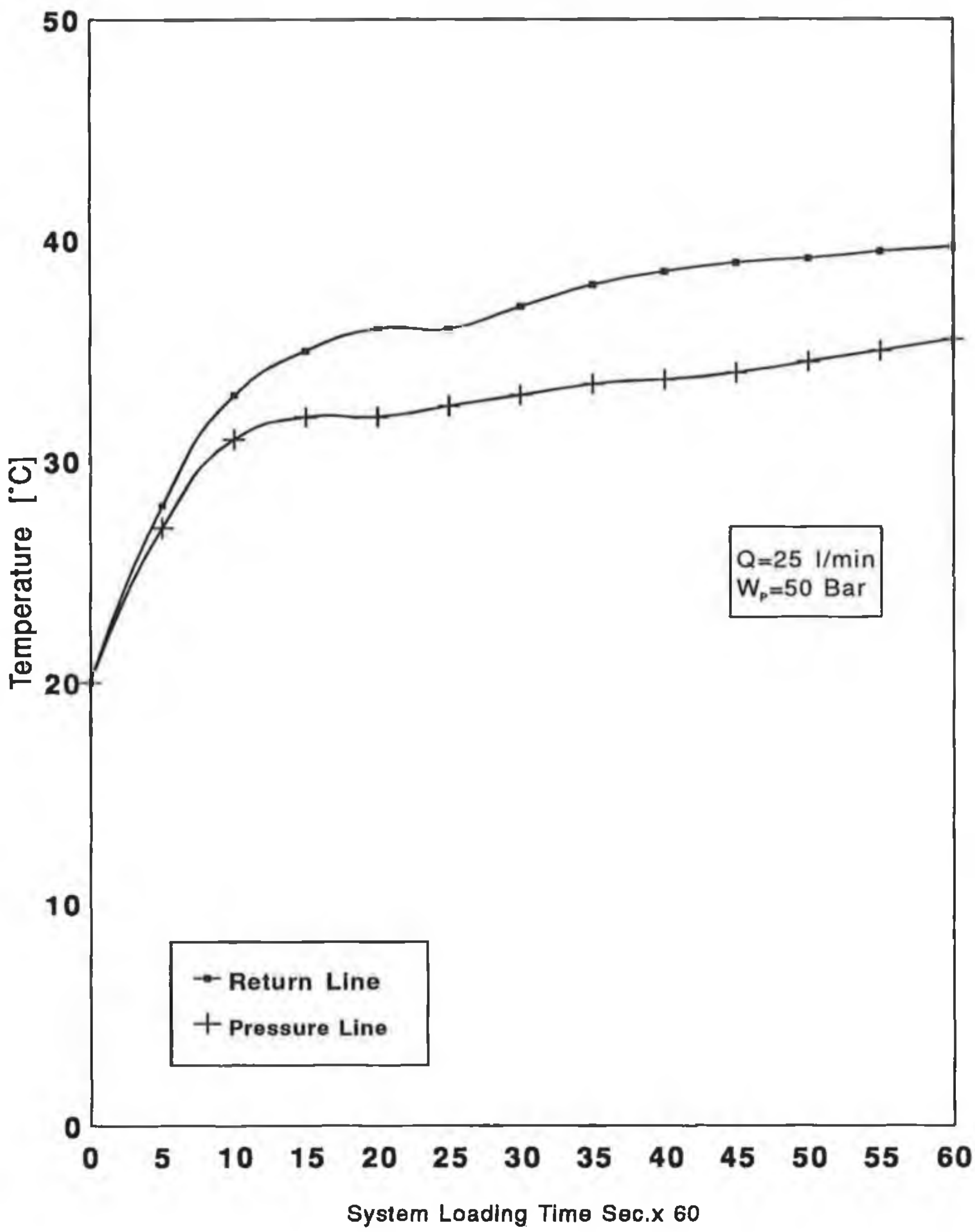


Fig. (76) Experimental results of temperature variation of fluid for flow rate of 25 l/min in the open hydraulic drive mixer system

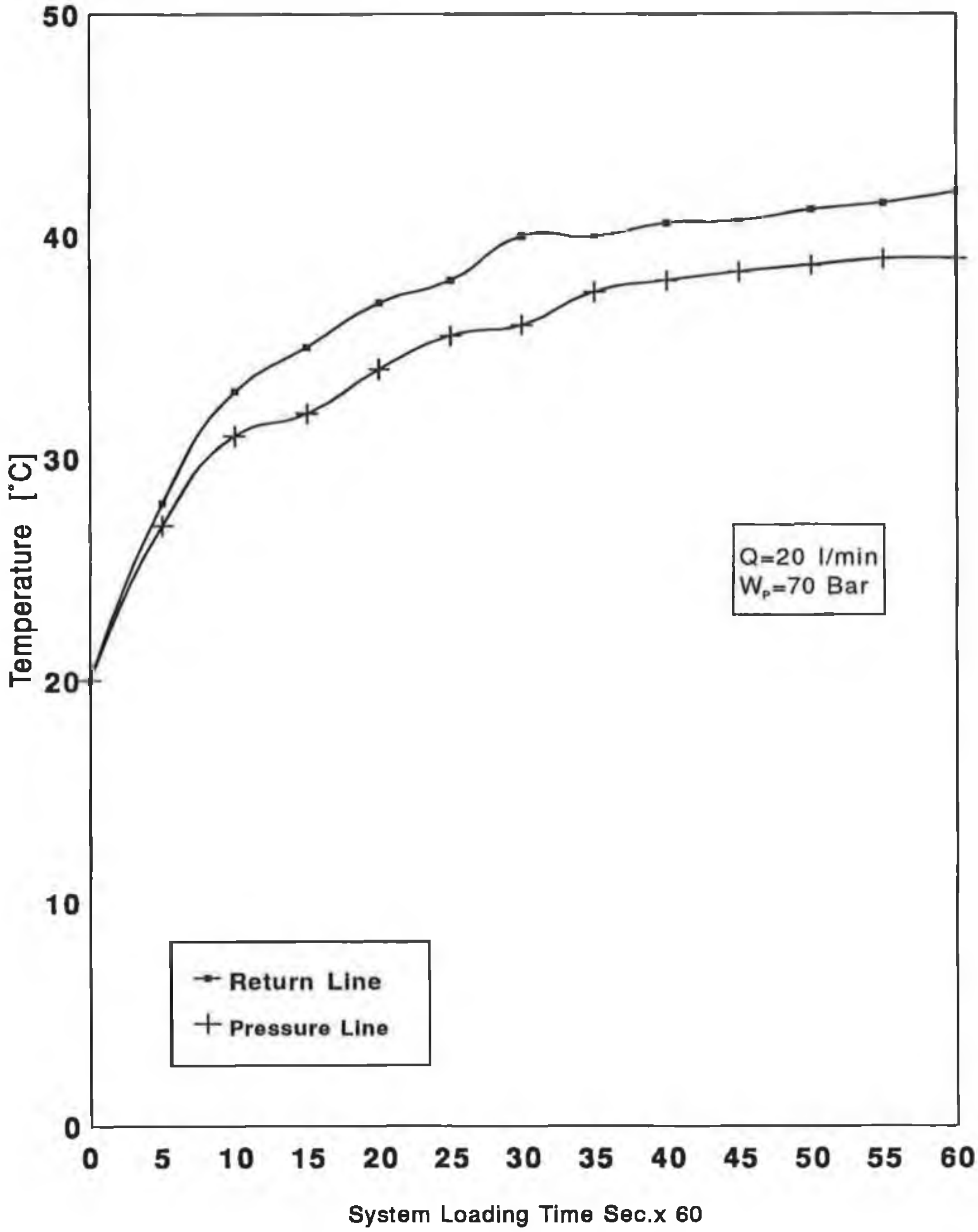


Fig. (77) Experimental results of temperature variation of fluid for flow rate of 20 l/min in the open hydraulic drive mixer system

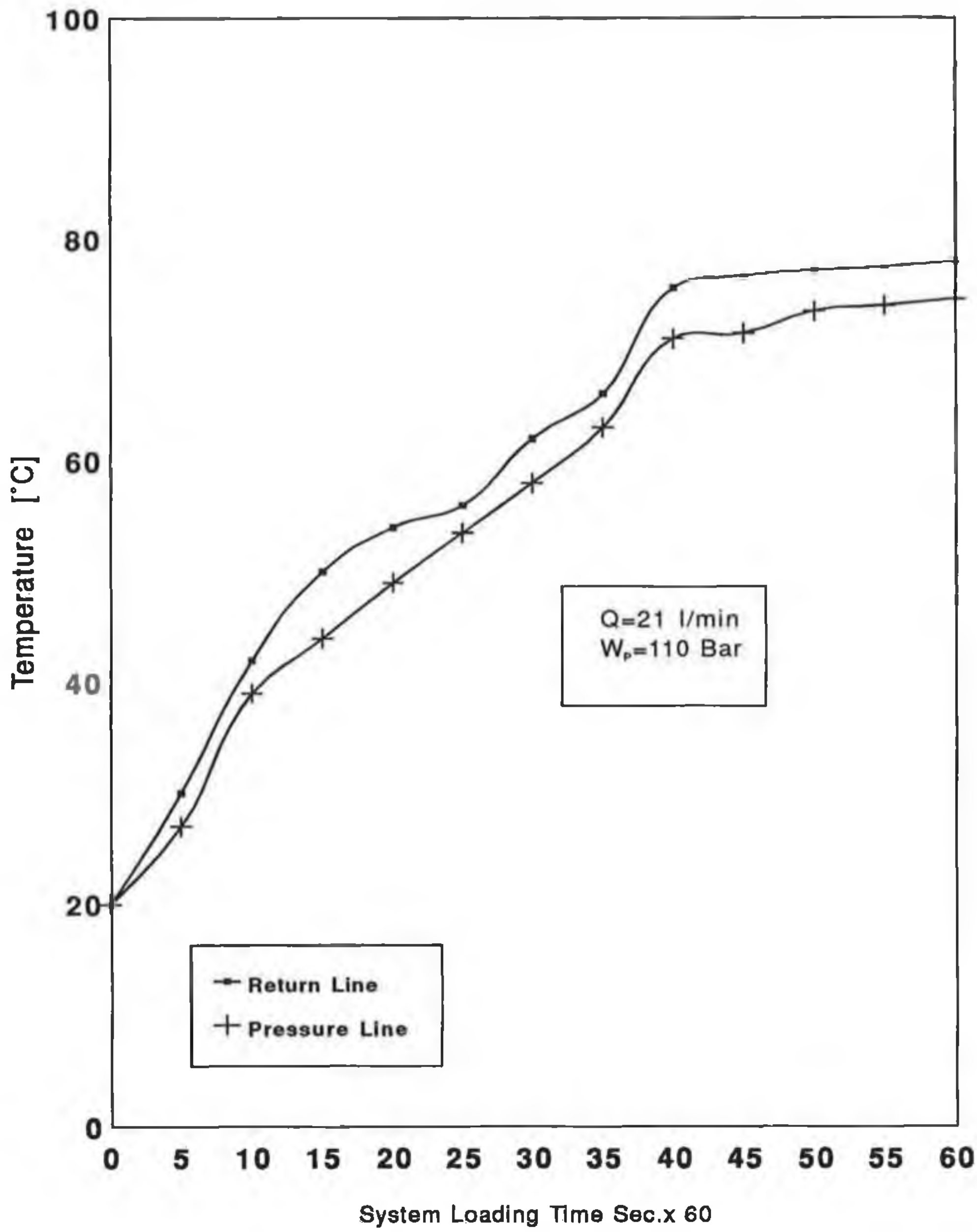


Fig. (78) Experimental results of temperature variation of fluid for flow rate of 10 l/min in the open hydraulic drive mixer system

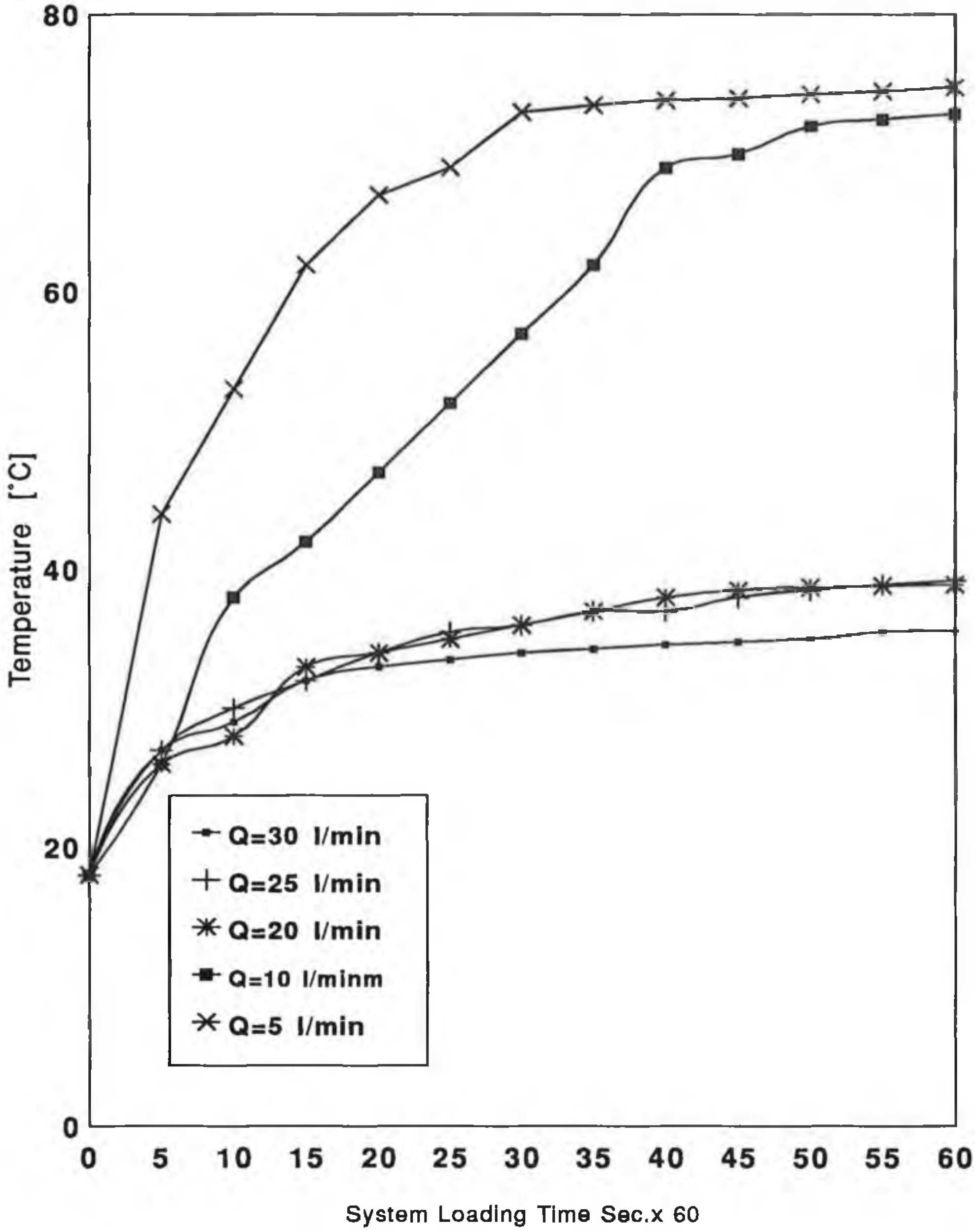


Fig. (79) Experimental results of the effects of flow rate on the outside Hose wall temperature variation of the return line in the open hydraulic drive mixer system.

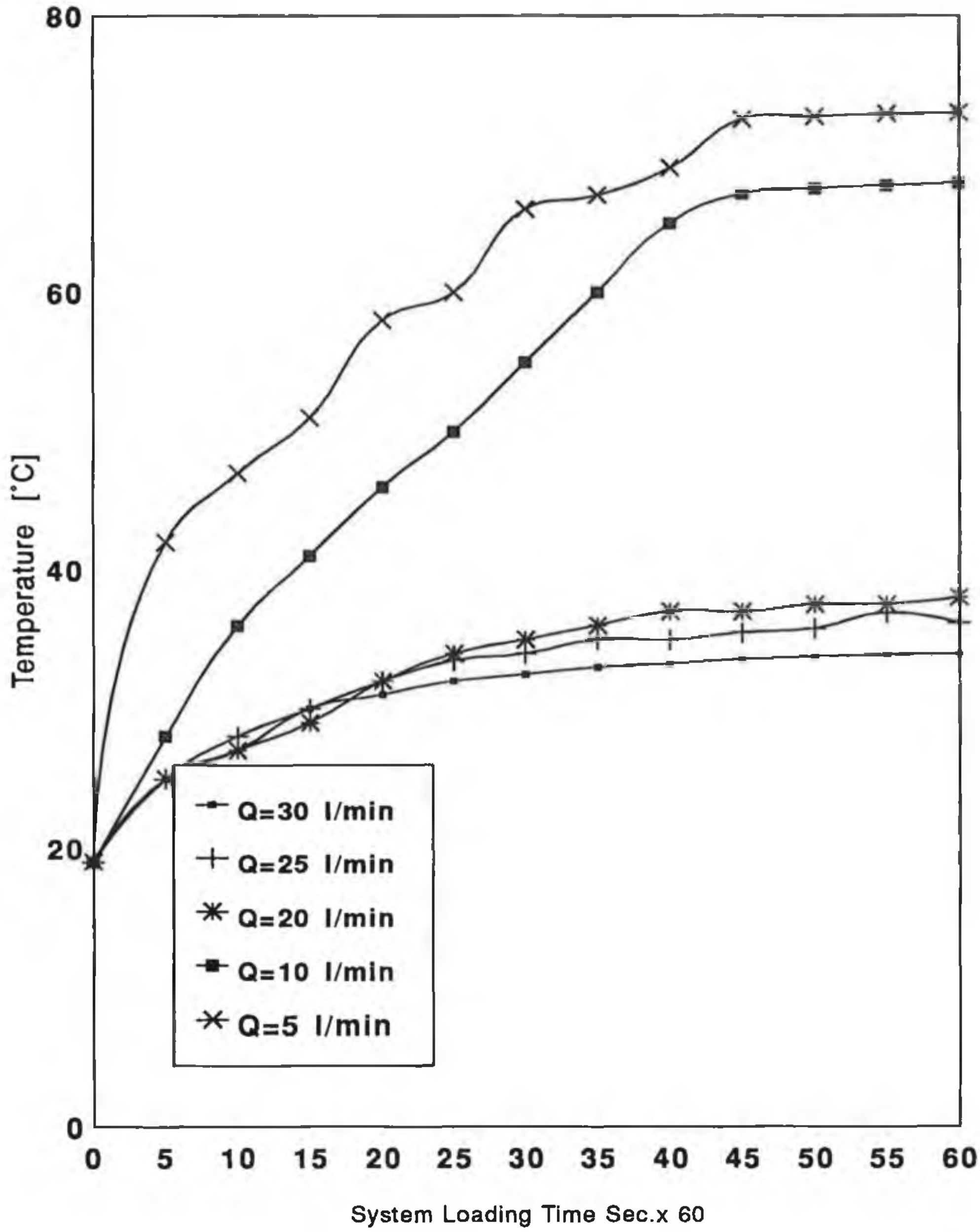


Fig. (80) Experimental results of the effects of flow rate on the outside hose wall temperature variation of the pressure line in the open hydraulic drive mixer System.

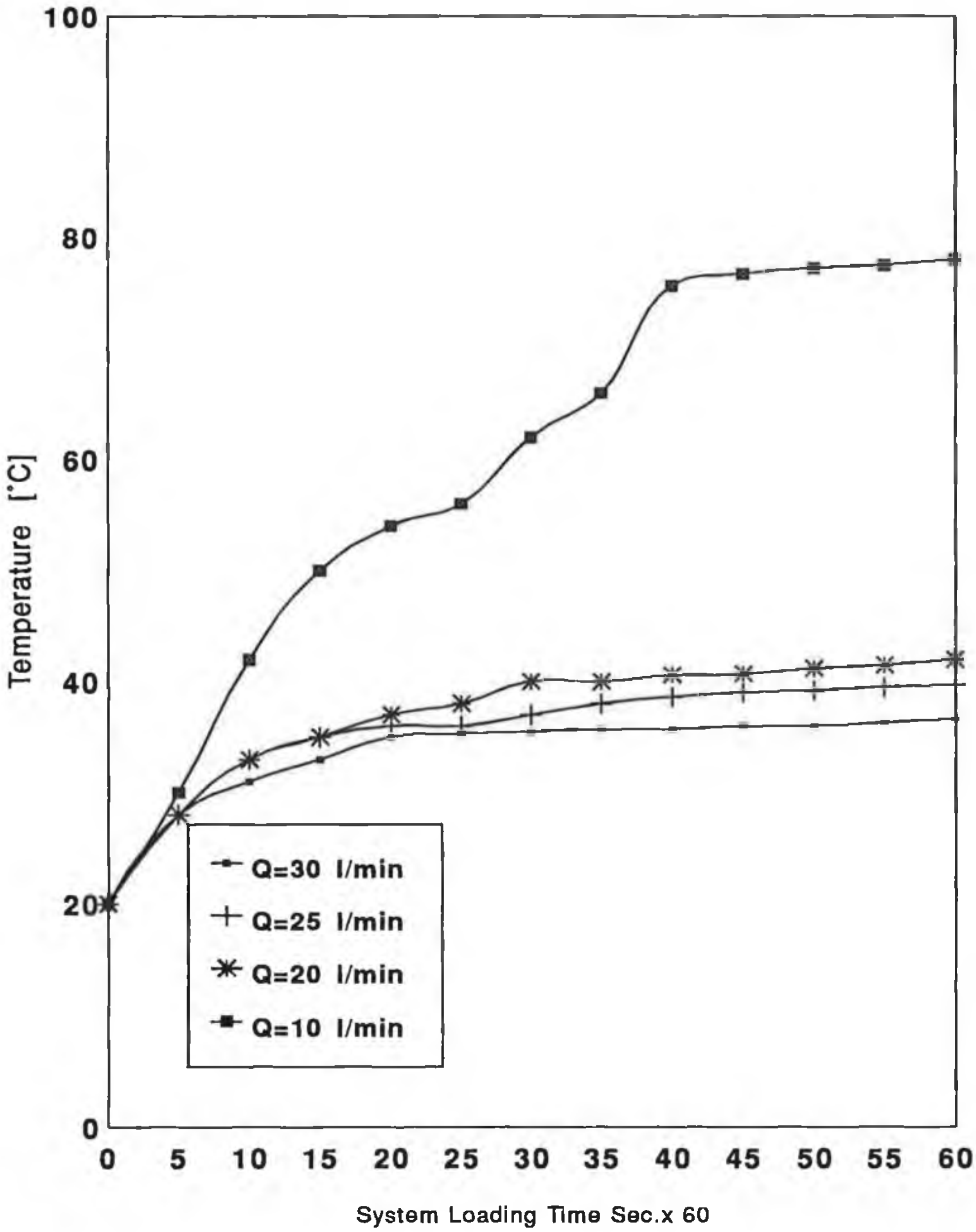


Fig. (81) Experimental results of the effects of flow rate on fluid temperature variation of the return line in the open hydraulic drive mixer system.

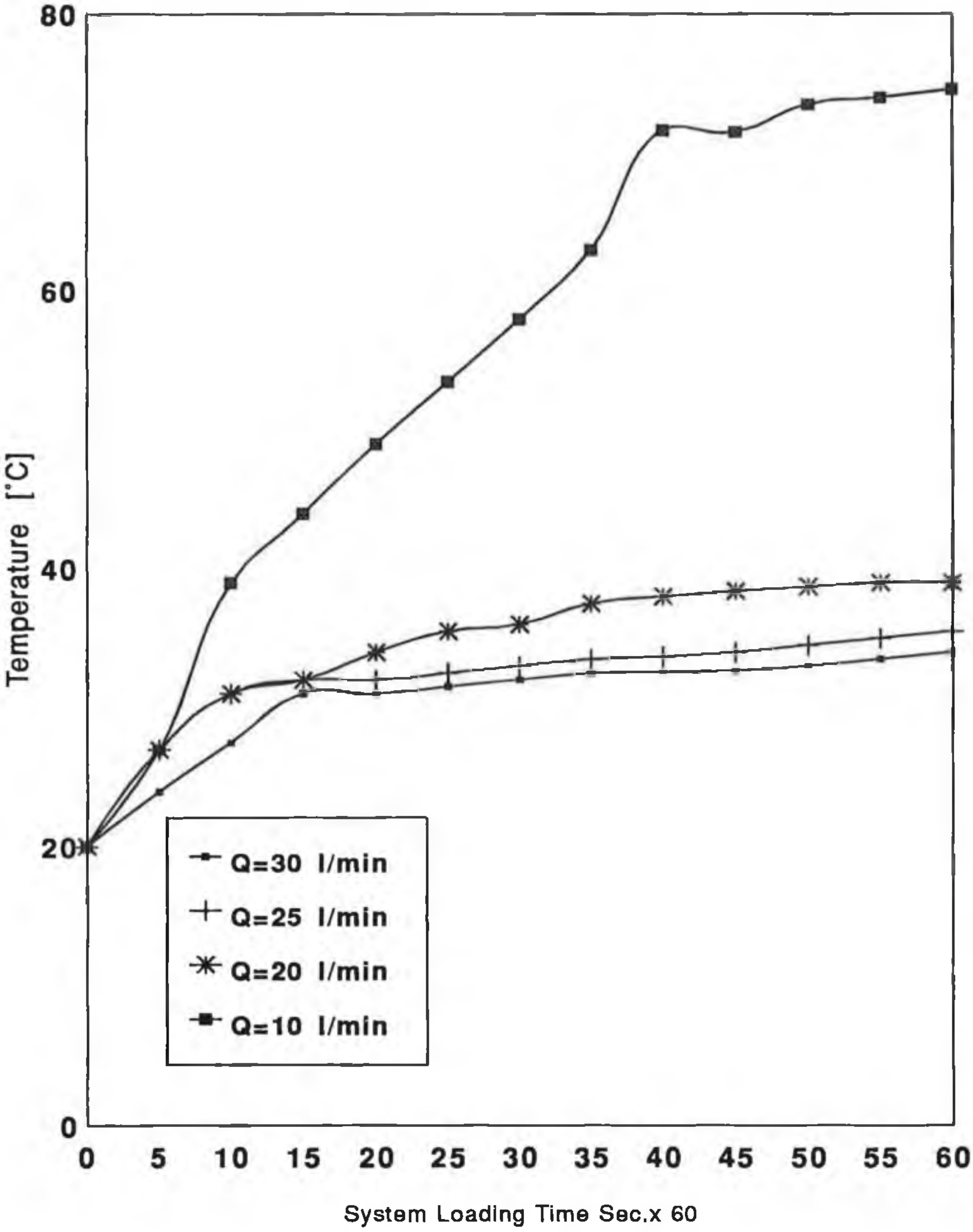


Fig. (82) Experimental results of the effects of flow rate on fluid temperature variation of the pressure line in the open hydraulic drive mixer system.

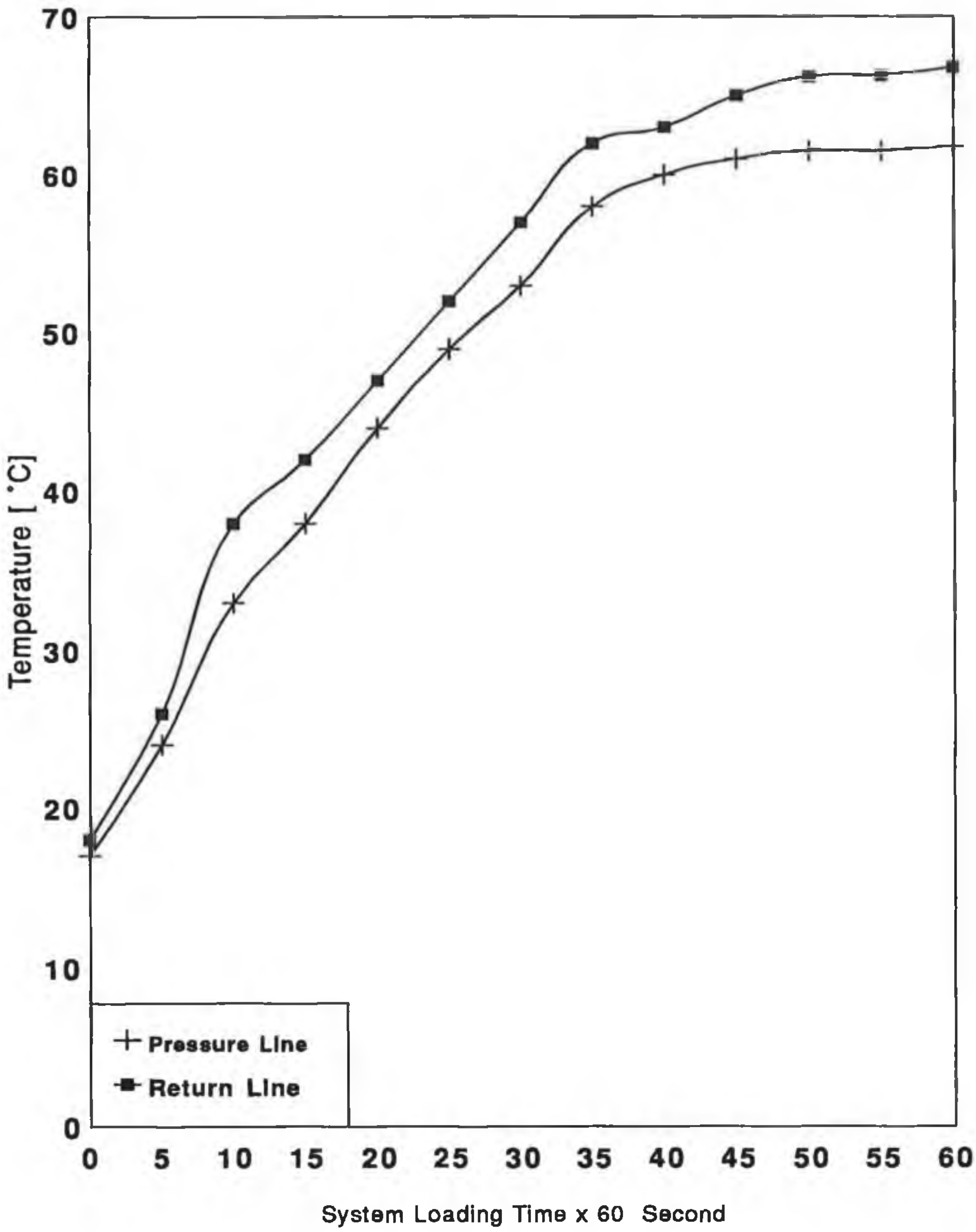


Fig.(83) Experimental results of the comparison of hose wall temperature between pressure and return lines in the open hydraulic drive mixer system.

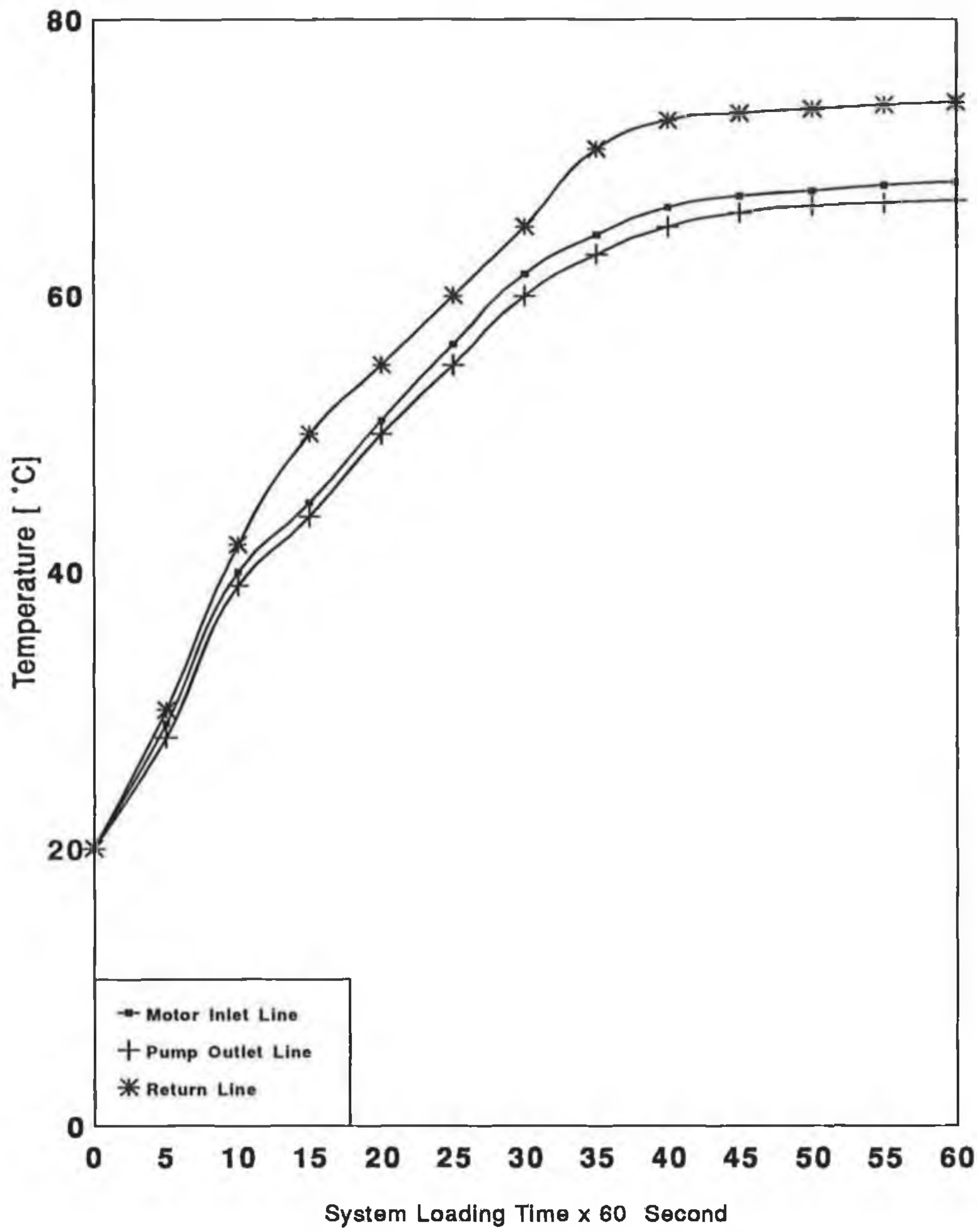


Fig.(84) Experimental results of fluid temperatures variation in the open hydraulic drive mixer system.

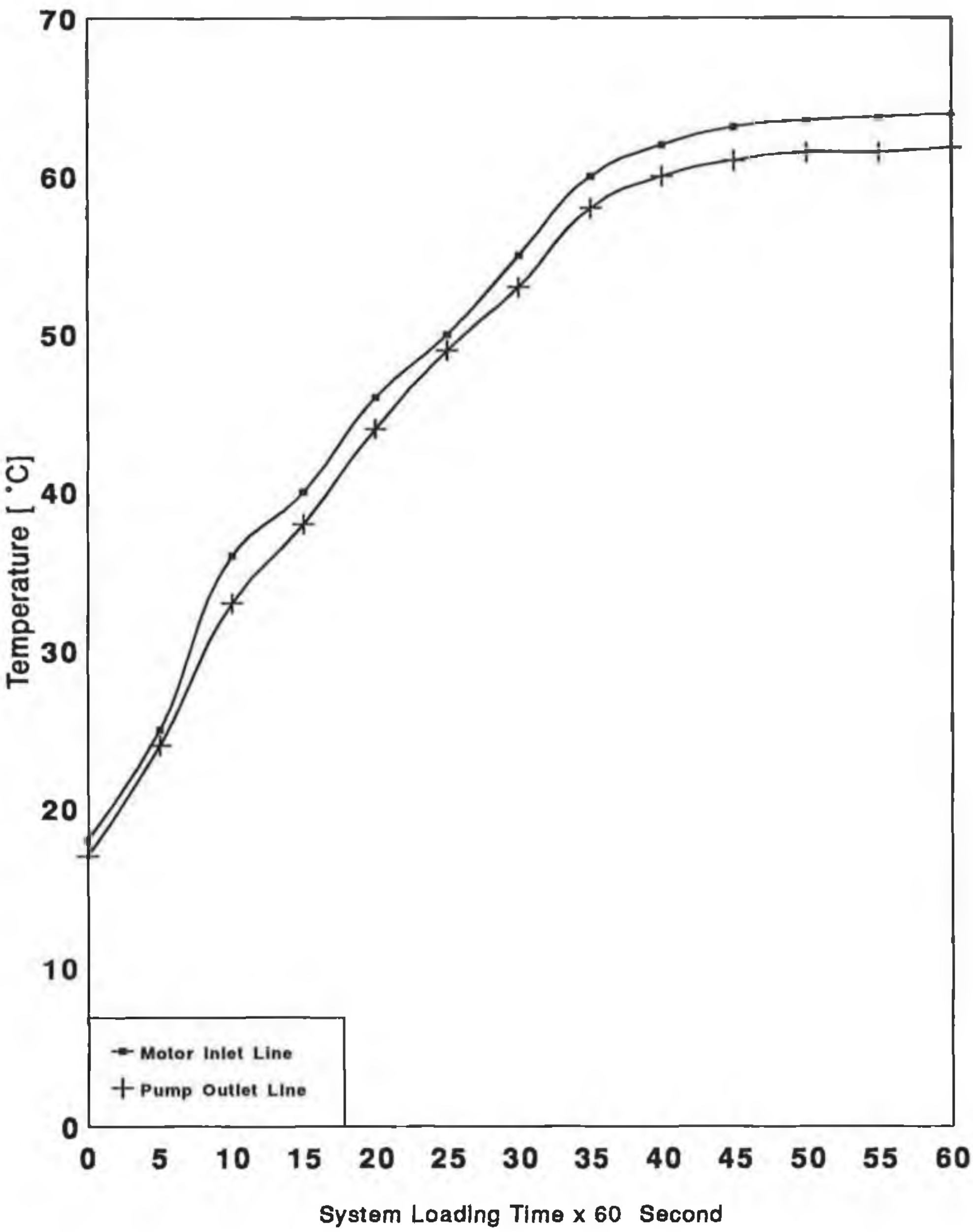


Fig.(85) Experimental results of the comparison of hose wall temperatures in the high pressure line of the open hydraulic drive mixer system.

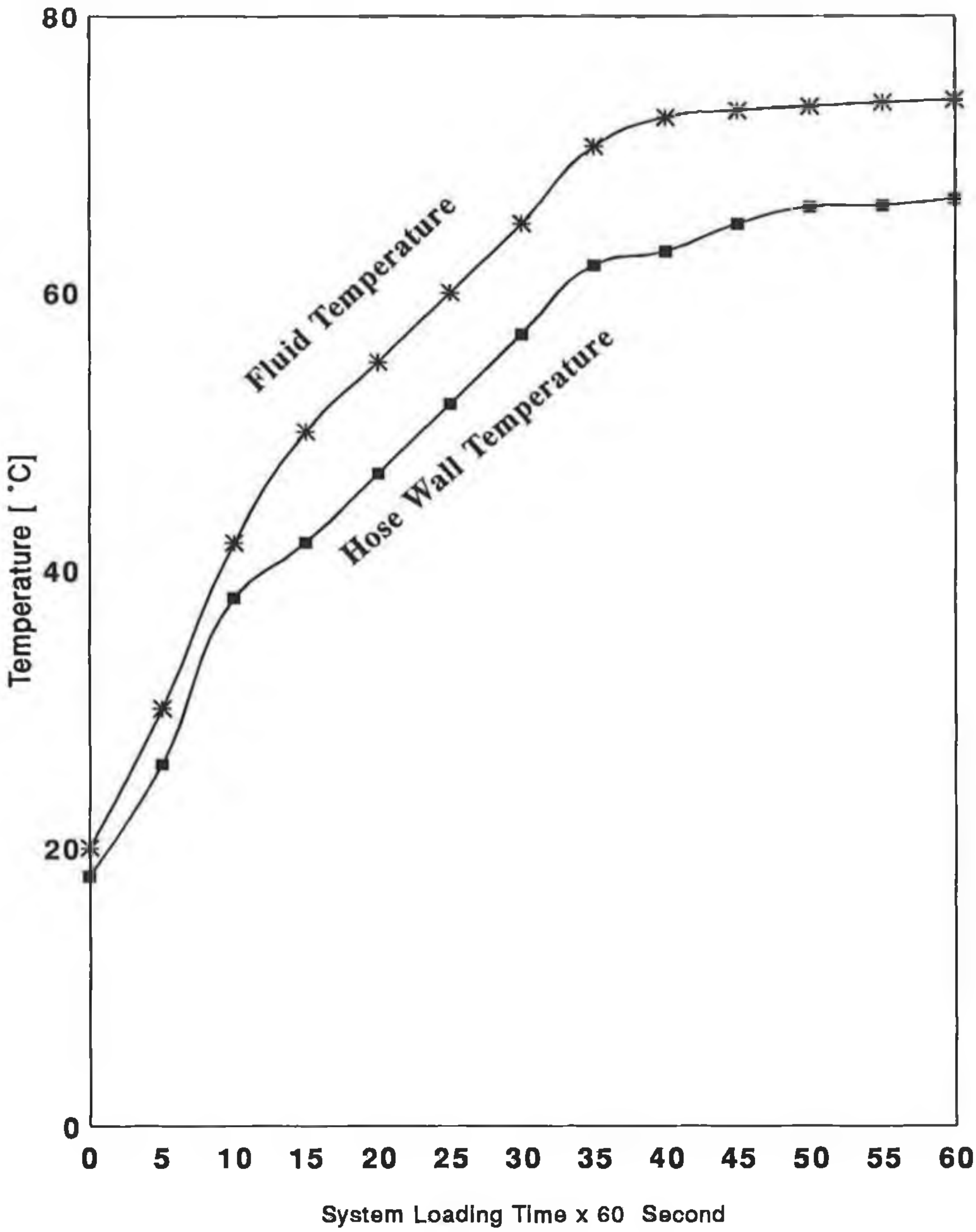


Fig.(86) Experimental results of the comparison of fluid and hose wall temperature at return line in the open hydraulic drive mixer system

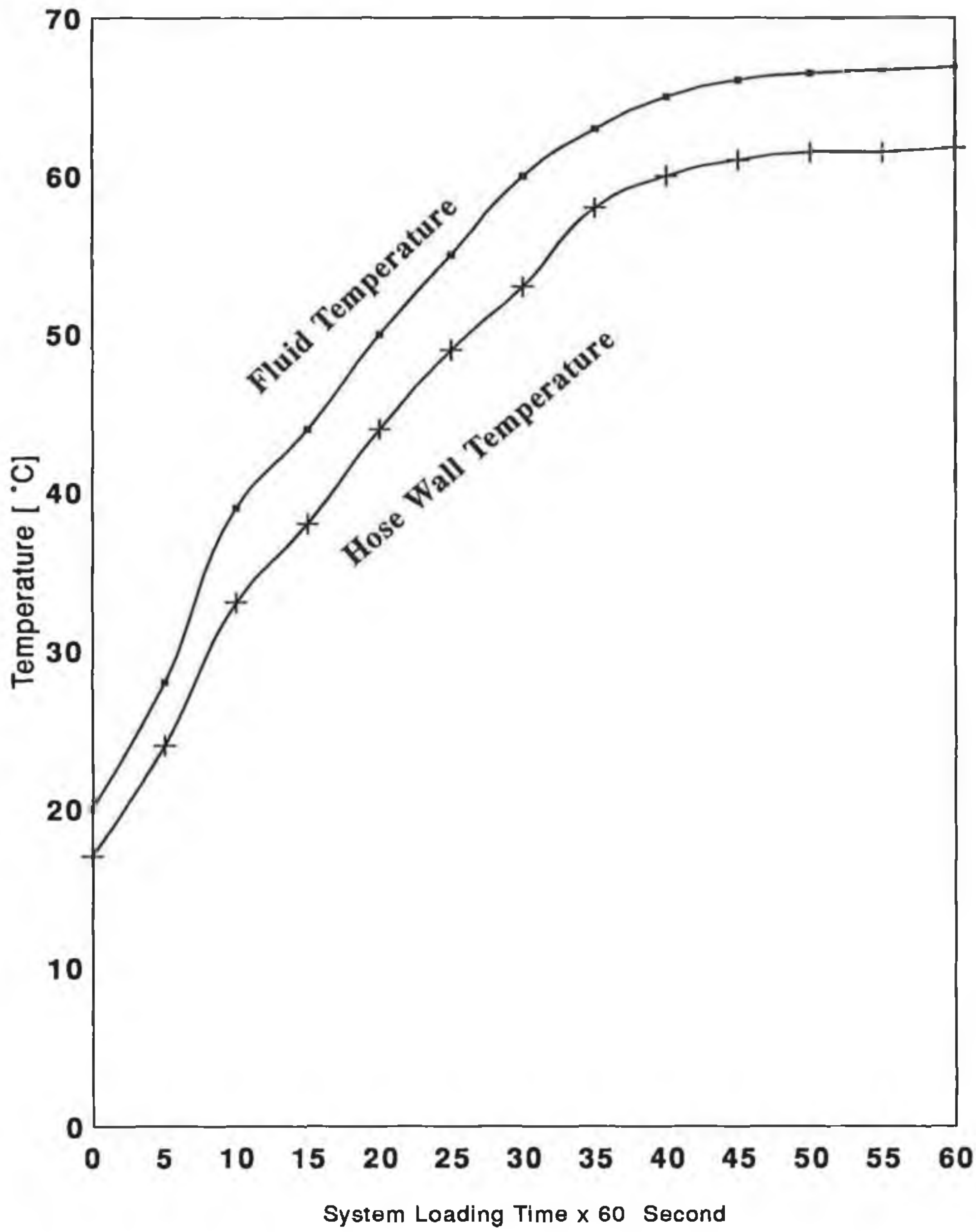


Fig.(87) Experimental results of the comparison of fluid and hose wall temperature at high pressure pipe line in the open hydraulic drive mixer system

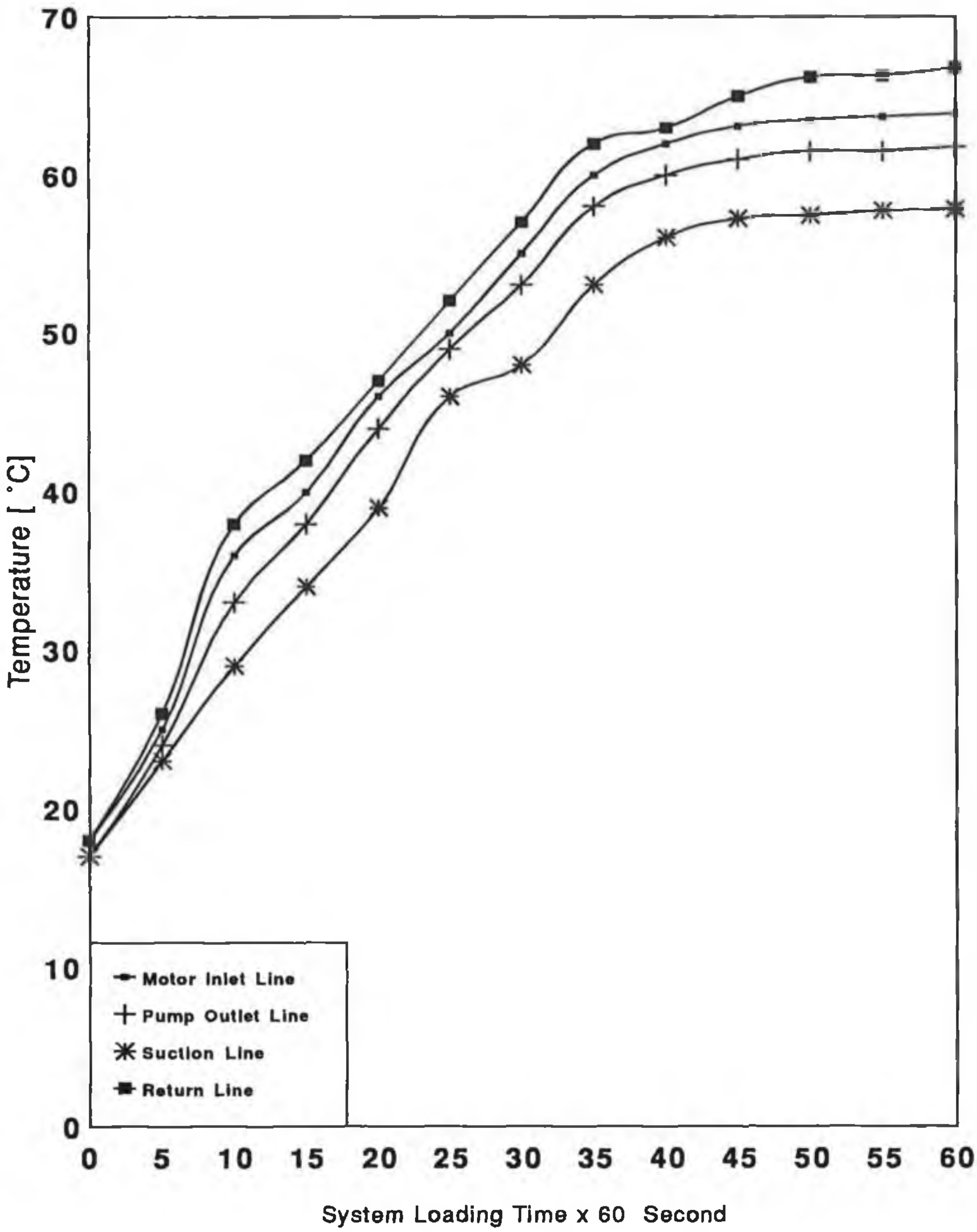


Fig.(88) Experimental results of hoses wall temperatures variation in the open hydraulic drive mixer system.

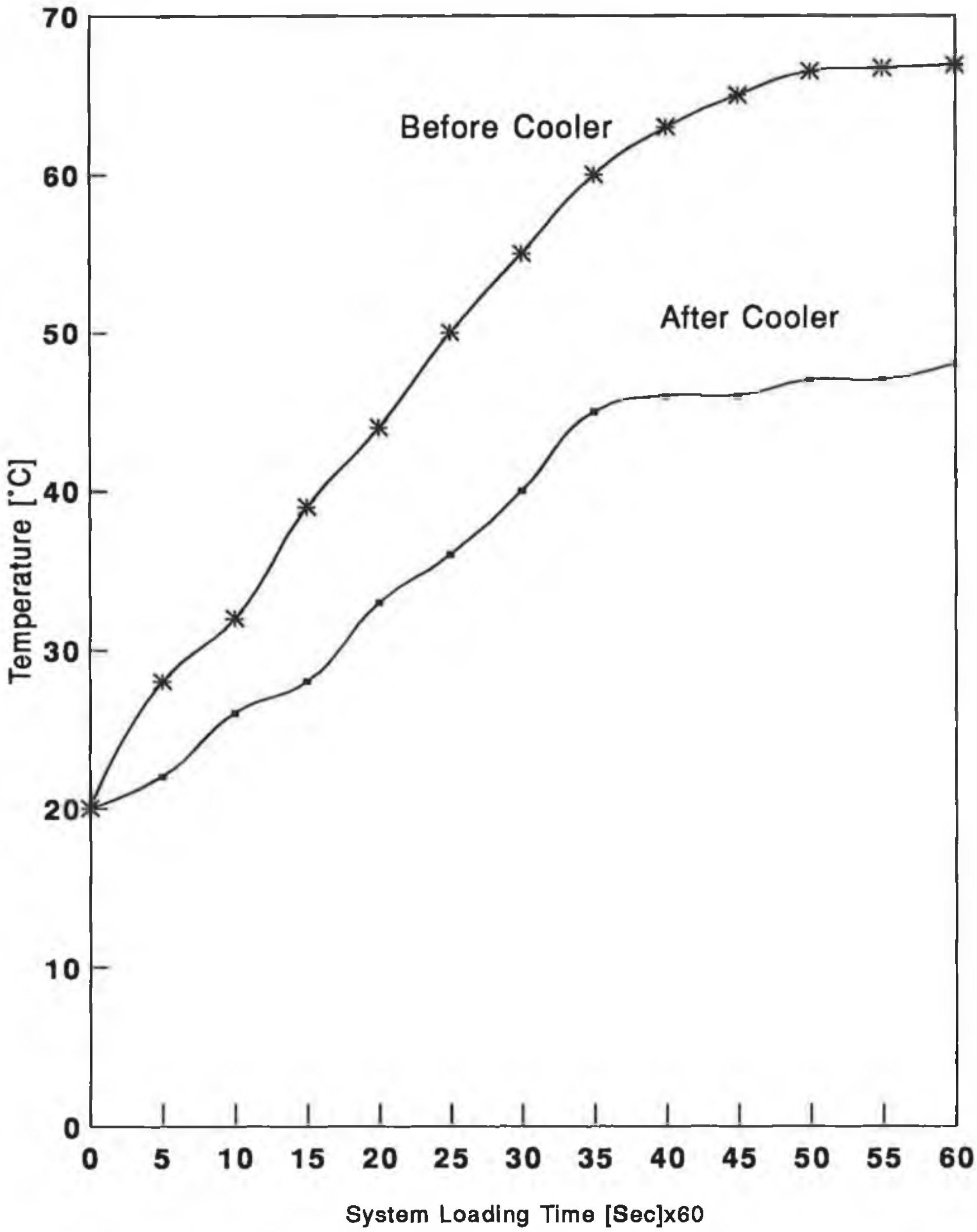


Fig. (89) Experimental results of comparison of fluid temperature in the pressure line before and after cooler in the open hydraulic drive mixer system.

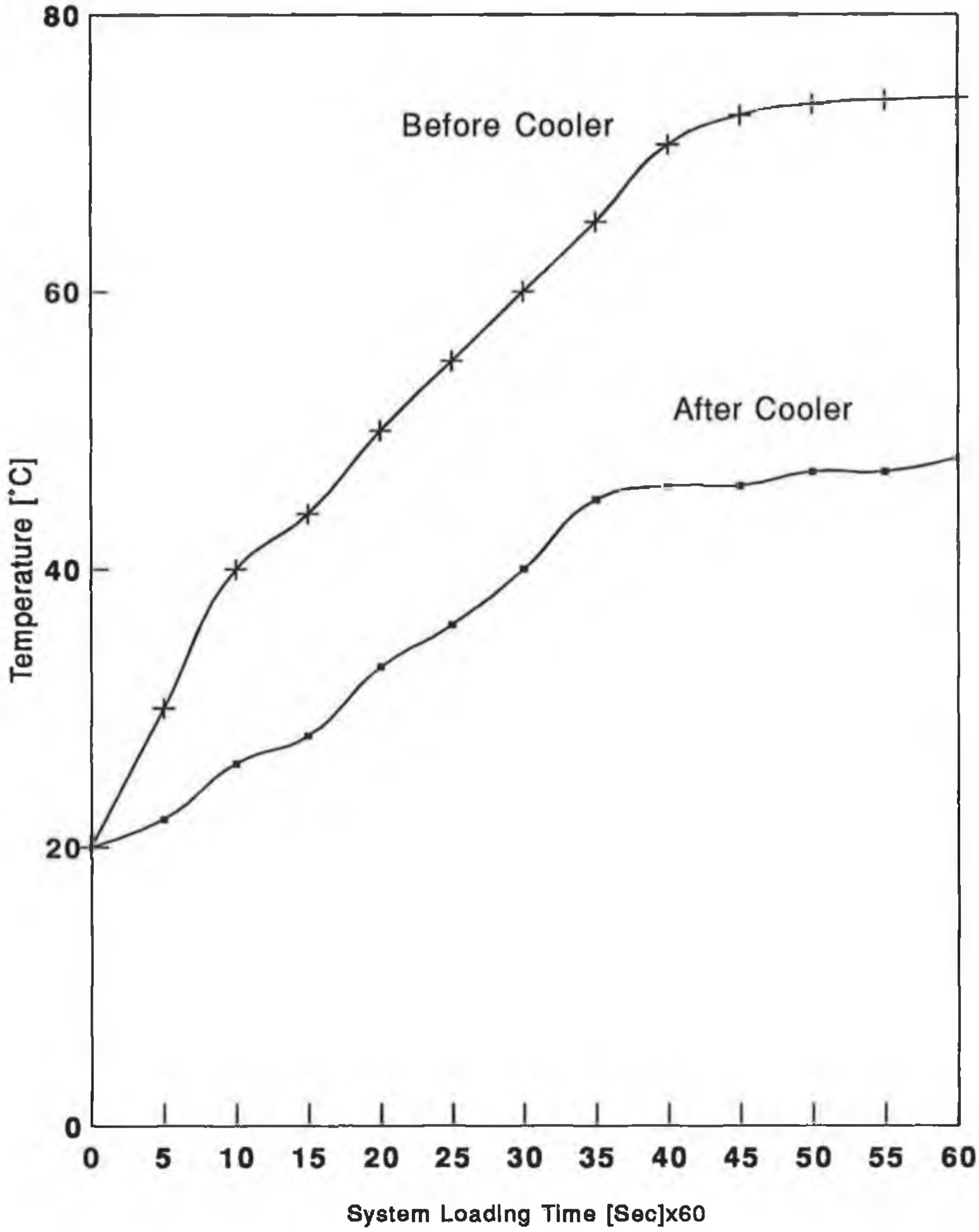


Fig. (90) Experimental results of comparison of fluid temperature in the return line before and after cooler in the open hydraulic drive mixer system.

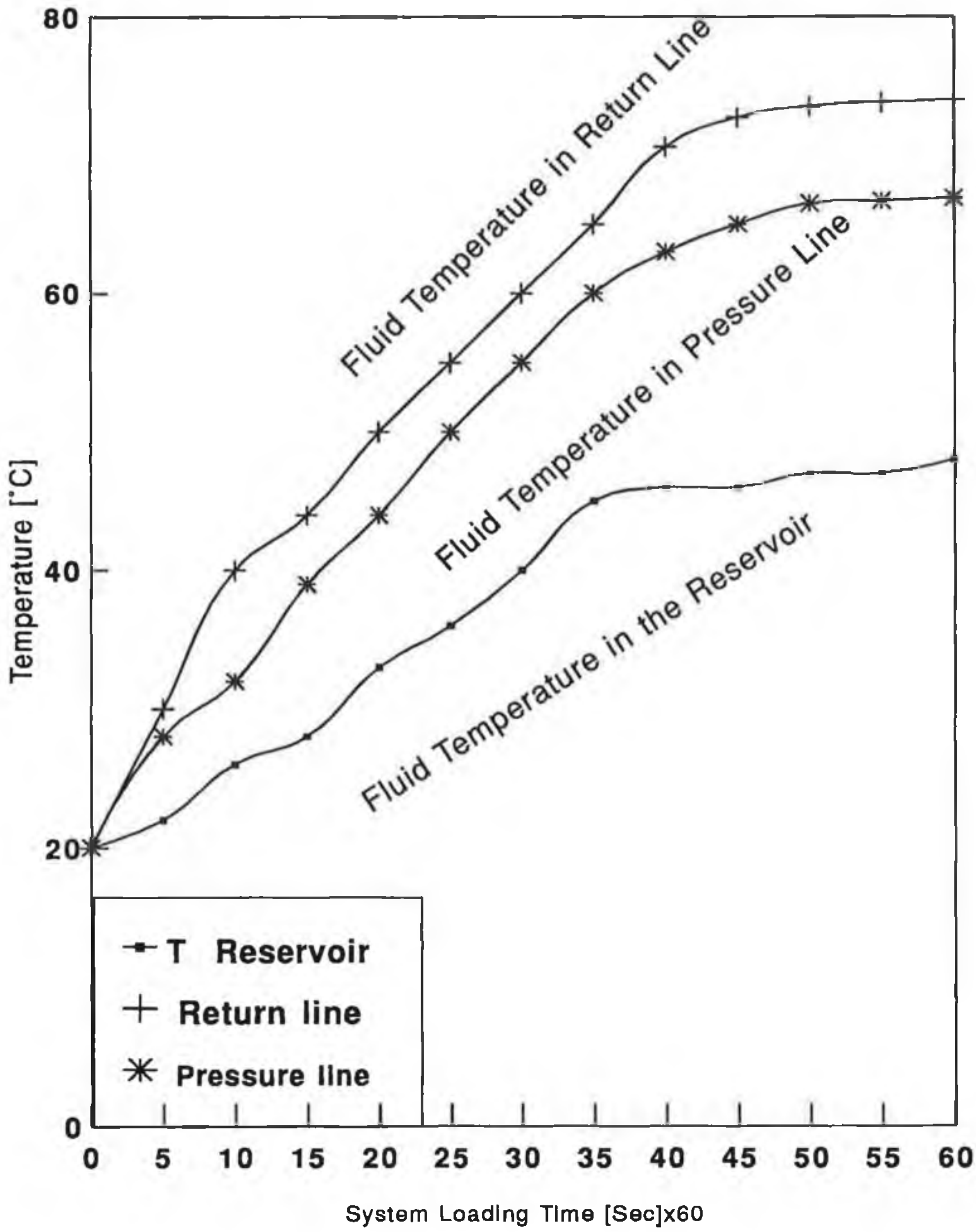


Fig. (91) Experimental results of comparison of fluid temperature between the return line and the Pressure Line Before and After cooler in the open hydraulic drive mixer system.

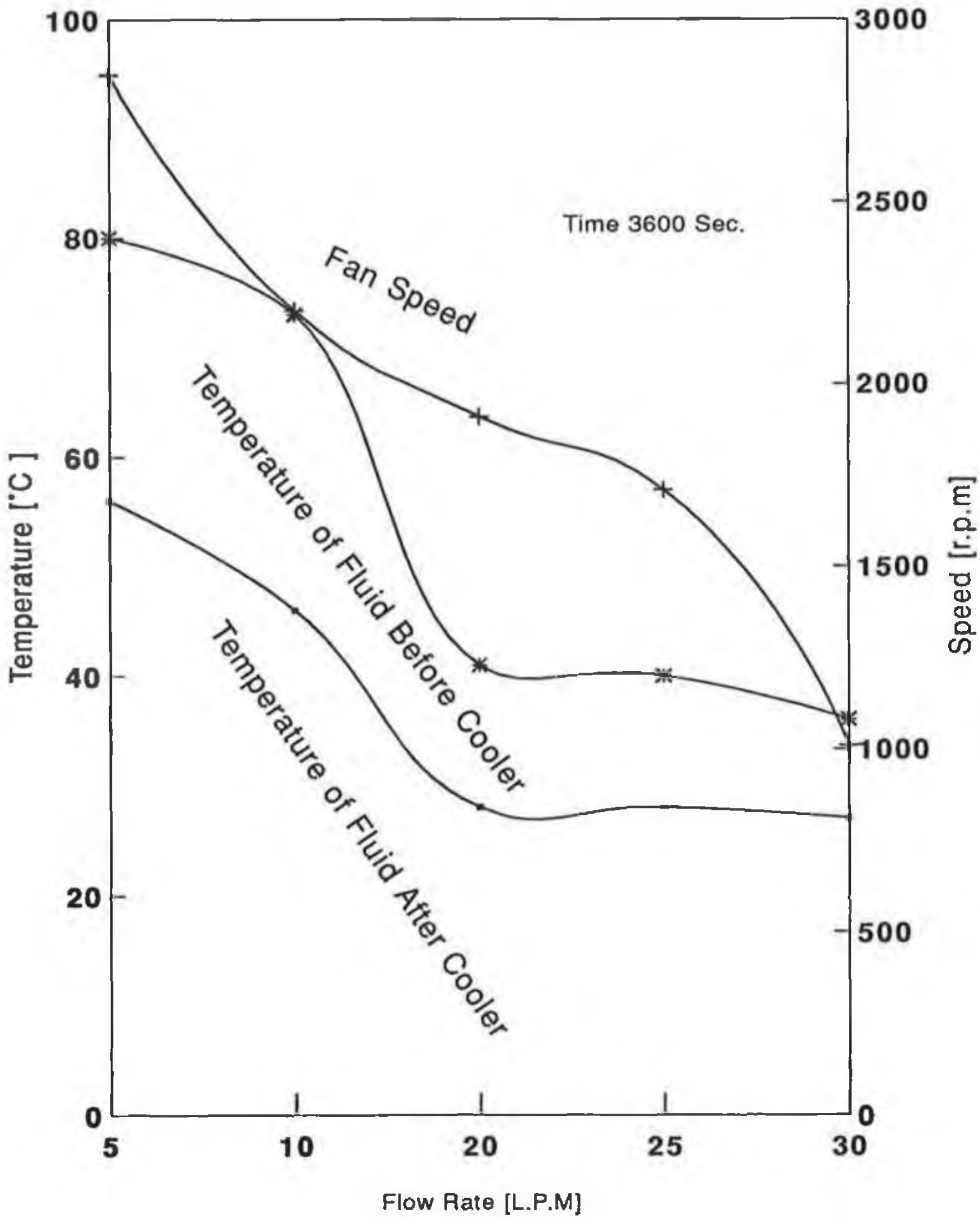


Fig. (92) Experimental results of the effects of fan speed in the cooling unit on performance of the open hydraulic drive mixer system.

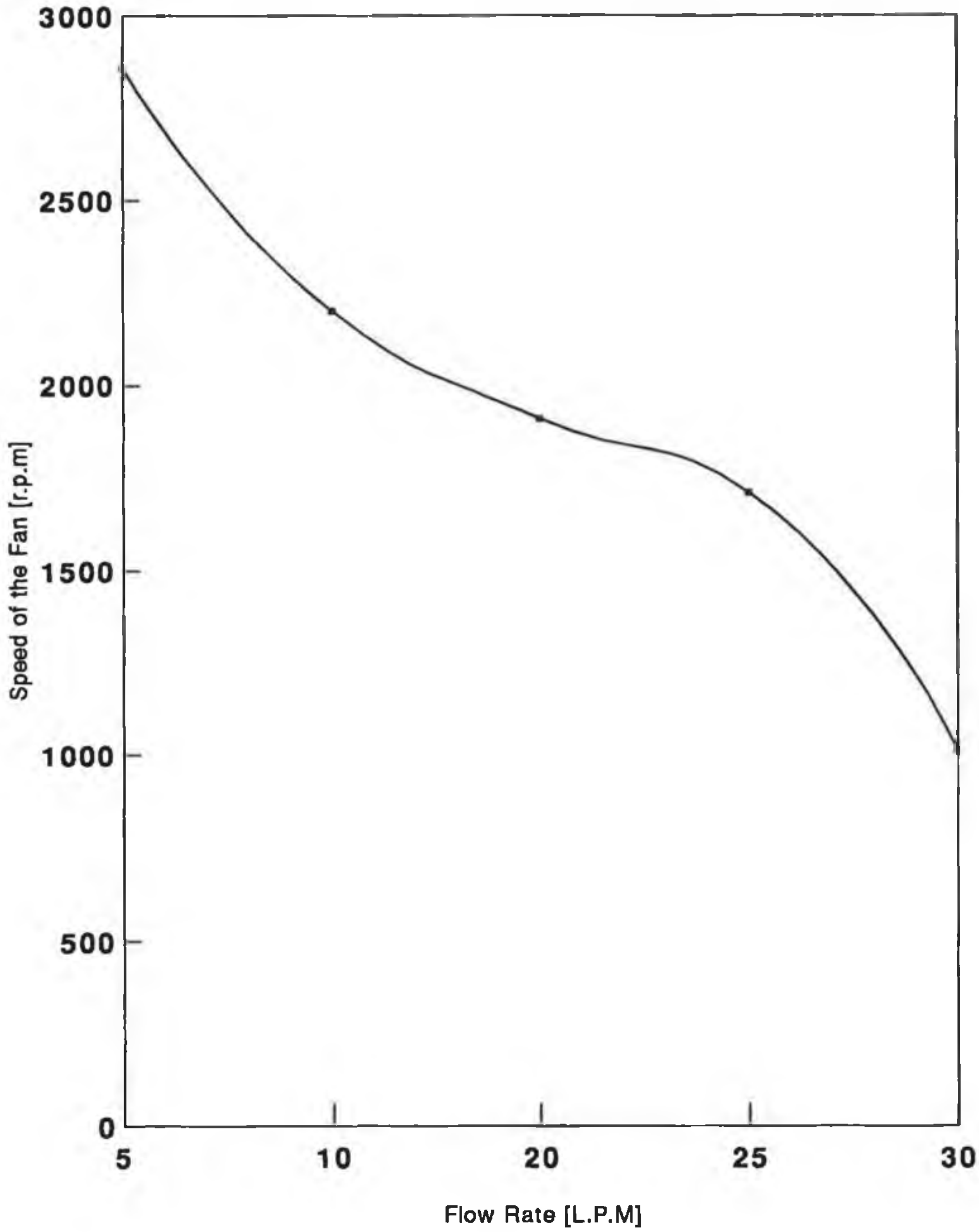


Fig. (93) Experimental results of the effects of flow rate on fan speed in the cooling unit in the open hydraulic drive mixer system.

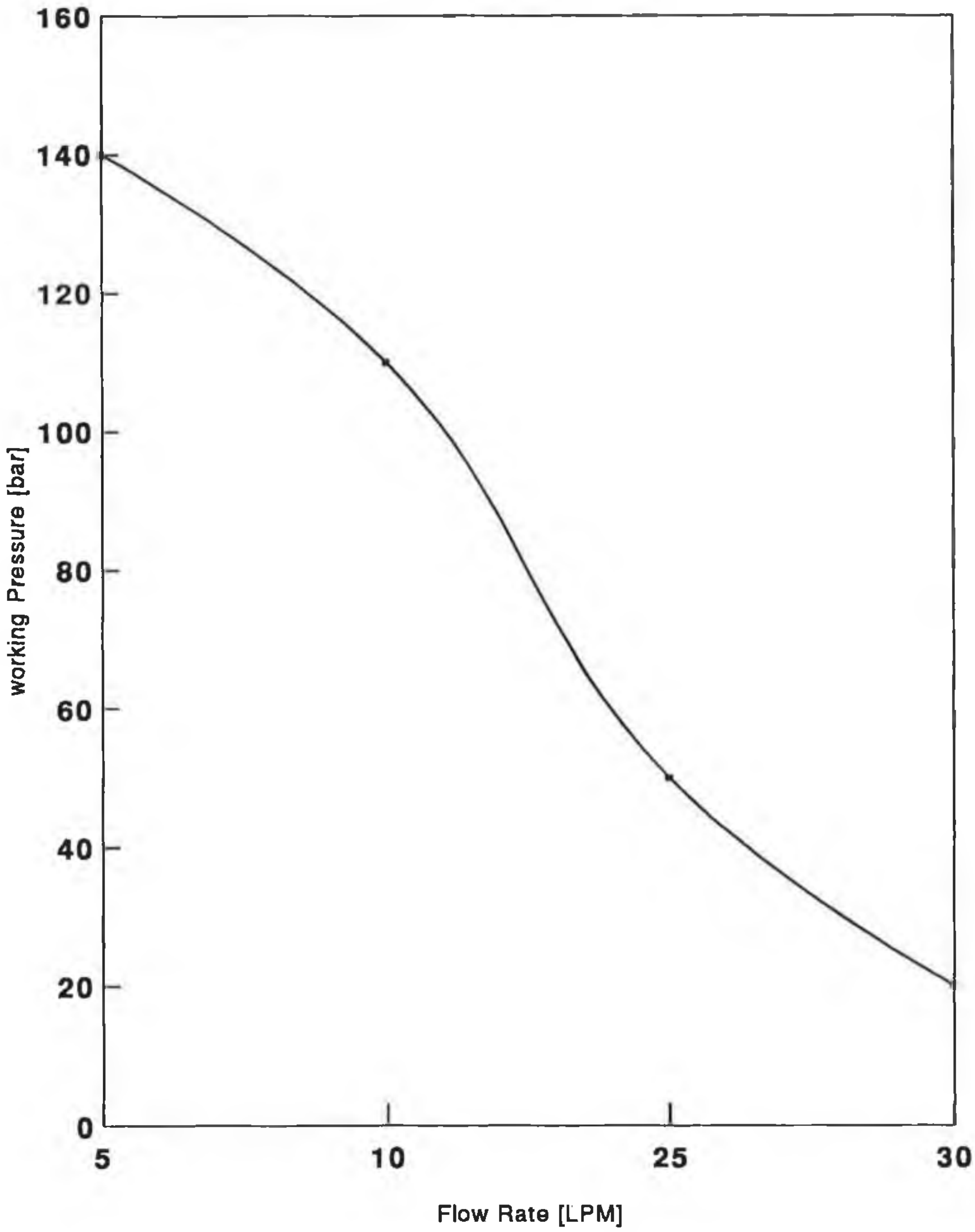


Fig. (94) Experimental results of the effects of flow rate on working pressure in the open hydraulic drive mixer system.

CHAPTER: 5

DISCUSSION

5.1 Discussion on the Theoretical and Experimental Results:

A mathematical model for predicting temperature distribution in hydraulic systems has been developed, and the feasibility of providing an inexpensive but efficient oil cooler to small sized hydraulic circuit which will reduce reservoir size, heat problem and will be an integral part of the circuit has been investigated. To study the performance of the system, an extensive experimental and theoretical investigation has been undertaken during which a considerable amount of data were obtained.In this chapter, it is aimed to highlight some of the important results obtained experimentally and theoretically and to carry out a comparison of typical results.

5.1.1. Discussion on the Procedure and Experimental Results

A number of interesting results have been observed while carrying out the experimental tests, using the hydraulic drive mixer system. During the course of the experimental programme, parameters such as length, diameter and material of the pipes, geometry of the reservoir and the cooling unit were varied in order to investigate their effects on the performance of the system.

Four thermocouples with a digital output were used in these experiments to measure the hose wall temperatures. Fluid temperature in the reservoir was measured with a dial mounted gauge which uses a temperature sensitive fluid that is metal encased and immersed in the hydraulic fluid. Two inline temperature meters were used to measure the temperature of the fluid flowing through the pipes. Actual motor, pump and fan speeds were obtained using a remote hand-held digital reading tachometer which measures the speed of a mark rotating with the output shaft.

The hydraulic reservoir was filled with SHELL TELLUS-37 hydraulic oil to the level mark on the sight glass. The setting of relief valve was adjusted on 150 bar. As mentioned in chapter two, the mixer mechanism was filled with different material to get different loads.

The system was run for one hour under load in leading towards the steady thermal condition. Changes of fluid and pipe wall temperatures were rapid initially and the temperature readings were therefore recorded every one minute for the first five minutes, every two minutes for the following 25 minutes and then every five minutes

towards the end of tests.
Following closely the above discussed test procedure, an extensive test programme was undertaken to verify the developed mathematical model and to investigate experimentally the possibility of providing an efficient oil cooler to small sized hydraulic circuit which will reduce heat problems, reservoir size and will be an integral part of the circuit. The typical set of temperature recordings from the experimental test is presented in Fig. (65) through Fig. (94).

Figs.(65) to (74) show the measured temperature of the outside hose wall for different part of the pipeline in the circuit at different flowrates. These results show that the recorded measured rate of temperature rise at the outside hose wall were greater during the first 35 minutes of the system loading time. For system loading time in excess of 50 minutes, the recorded temperatures were observed to be approximately constant due to the achievement of steady state condition as shown in Fig. (70).

It has been observed that the constant temperature values of steady state condition have not been achieved in some experimental results as shown in Fig. (83). A possible reason is the gradual increase in the atmospheric temperature surrounding to the test rig over the duration of the tests of system loading. For example, the room temperature in the laboratory could be different in a single day or different on different days of the week. Moreover, the experimental results have shown that the fluid temperature in the second part of the pressure line is approximately 3 °C higher than that recorded at the first part of the high pressure line as shown in Fig. (84). The reason could be the well known temperature-pressure drop coefficient 5 °C/100 bar, see Ref. 48, in a pipe line. Similarly, Fig. (88) shows that the maximum difference between the measured wall temperatures in the different parts of the pipeline is about 7 °C. This is due to pressure drop thorough the fittings and the errors in the temperature measurements.

The temperature results obtained experimentally from different parts of the system show a significant difference between fluid and pipe wall temperature. These temperature differences can not be ignored even if the experimental measurement errors have been taken into account, typical examples are shown in Figs (86) and (87).

Since the hydraulic reservoir in the conventional open hydraulic system requires a large size and space and the energy consuming boost pump in the closed hydraulic system requires an additional driving power, neither the traditional open hydraulic system nor the closed hydraulic system may be the best design choice under such special working conditions. In this research an extensive programme of experimental tests have been carried out to investigate the possibility of using an efficient oil cooler as an integral part of the hydraulic circuit to small sized hydraulic systems which will reduce heat problems and reservoir size in the conventional open hydraulic systems. The typical sets of temperature recordings from the experimental test is presented in Fig. (89) to Fig. (94).

These experimental results have shown that the temperature would be decreased by between 25 to 35 °C through using the oil cooling unit as an integral part of the hydraulic circuit. Typical examples are given in Figs 89, 90 and 91. As can be seen in Fig. (89) the temperature in the system before the cooler is about 70 °C which is about 10 °C higher than that recommended, but after the oil cooler the temperature is about 45 °C. That means by using the oil cooler the temperature decreases by about 25 °C while Fig. (91) suggests that by using the cooling unit, the temperature could be reduced by 40 %. As a comparison, under the same condition where the electric motor is operated at a nominal speed of 1500 r.p.m., the reservoir capacity in a conventional open system is normally chosen as 6 to 7 times of the system flowrate. This means that a 48 to 56 capacity litres reservoir has to be used.

5.1.2. Discussion on the Analysis, Simulation Package and Theoretical Results

A mathematical model has been developed in order to study the temperature behaviour in any hydraulic system. The mathematical model is developed to consider a hydraulic system (such as the open hydraulic drive mixer system) as an open thermal system with the assumption that the temperatures of fluid and pipe wall are uniform. In addition, the system main loop is considered as an open thermal system where uniform temperature is assumed to represent the loop fluid temperature, while the reservoir is another open thermal system. However, if the temperature in one part of the loop significantly differs from another part in a hydraulic system, the assumption of uniform loop fluid temperature in the present analysis can no longer be applied. The results may improved by regarding the system loop as a series of open thermal systems with a constant fluid temperature in each section of the system. In this way, an open thermal system of a pipe section can be applied for theoretical solution of the fluid temperatures at the different part of the system. The principle of this suggestion is to develop a series of fluid temperature models for the different sections of pipe in the system, then fluid temperature in each particular section of the pipe can be estimated separately.

Software has been developed incorporating the above analysis in order to predict the transient fluid and pipe wall temperatures of any hydraulic system under unsteady state conditions. This program is especially suitable to simulate fluid power system, the main aims of this program are:
(i) To provide a computer aided design tool which can be used by an industrial engineer or researcher with very little knowledge of computer systems.

(ii) To provide a facility which allows changes in component parameters or in circuit configuration to be carried out without further need for programming.

(iii) To provide a facility which allows adding new components without any modification in the programme.
Fig. (30) explains in details the complete system simulation diagram.

In this research the simulations were carried out for analyzing a large number of 60 minutes duration experimental temperature tests during the system loading. As in the experiments the system is considered to be constructed by the same size of pipes and pipe wall material. However, a uniform fluid temperature throughout the system loop and reservoir was assumed in the simulation. The simulation results are shown from Figs. (31) to Fig. (63) and discussed previously, but here are the main points.

The magnitude of the inside diameter of pipe is more influential on fluid temperature than outside pipe wall temperature (See Figs 35 and 36). This is because the reduction in the internal pipe diameter reduces the pipe internal surface area and at the same time increases the heat coefficient by forced convection. Hence, the combined effects cancel the effect of the change in diameter. However, the size of the pipe outside diameter has considerable influence on the pipe wall temperature as shown in Fig. (37).

The simulation results have shown that the type of pipe wall material plays an important role in determining the temperature distribution since the magnitude of pipe wall specific heat, density and thermal conductivity are dependant on the pipe wall material (See Figs. 39 to 44). The simulation results have also shown a large difference between the outside pipe wall temperature of steel or copper pipes and rubber hose as shown in Figs 47 and 48 as it would be expected due to a big difference which exists between the thermal conductivities of metallic and non-metallic material. An interesting phenomenon shown in the simulation fluid temperatures and illustrated in Figs. (47) and (48), is that fluid temperature in the system with the steel or copper pipeline is higher than that in the system with the hydraulic hose. These simulation fluid temperatures have resulted when the effects of radiation and conduction were taken into account.

Despite the fact that the thermal conductivities of steel and copper pipes are much higher than that of rubber or hose, the emissivity of steel and copper are lower than that of rubber hose as data indicated in Table (15). This comparison of simulation fluid temperature results should question the widely accepted assumption in many literature in which the radiation effect under 200 °C is ignored.

A number of simulations have been carried out to investigate the effect of reservoir geometry on the temperature distribution. The results have shown that the fluid height in the reservoir, and the length and width of the reservoir have influence on the fluid temperature of the reservoir and the system loop as indicated in Figs. (49) to (54).

5.2. Comparative Performance of the Hydraulic Systems: Fig. (95) shows the comparison between the fluid and hose wall temperature obtained experimentally. This figure shows considerable difference between the fluid and hose wall temperatures where the hose wall temperature is predicted to be about 6 °C lower than the fluid temperature. This difference between the fluid and hose wall temperature is beyond the level of experimental error. The results obtained experimentally, show good agreement with the theoretical results.

Fig. (96) shows the comparison between the experimental and theoretical fluid temperature in the pressure pipe line. In this figure a close agreement between theory and experiment is observed. Fig. (97) shows the comparison between experimental and theoretical results of the hose wall temperature. This figure also shows similar trend as that in Figure (96).

Comparison has shown a significant difference between the results predicted according to reference (15) and those predicted using the present mathematical model. According to Reference (15) in which the effect of heat transferred by conduction in the hydraulic system and by radiation from the surfaces of the reservoir were ignored, a considerable difference in the estimate of the temperature distribution in the hydraulic systems is suggested. Typical examples are given in Figs (98) and (99). Fig. (98) shows the comparison of the fluid temperature predicted according to the present analysis and the analysis in Reference (15) with that obtained experimentally. This figure shows a closer agreement between the experimental results and theoretical results predicted according to the present analysis which slightly overestimates the temperature rise. Fig. (99) shows the same trend as that in Fig. (98). in respect of the pipe wall temperature.

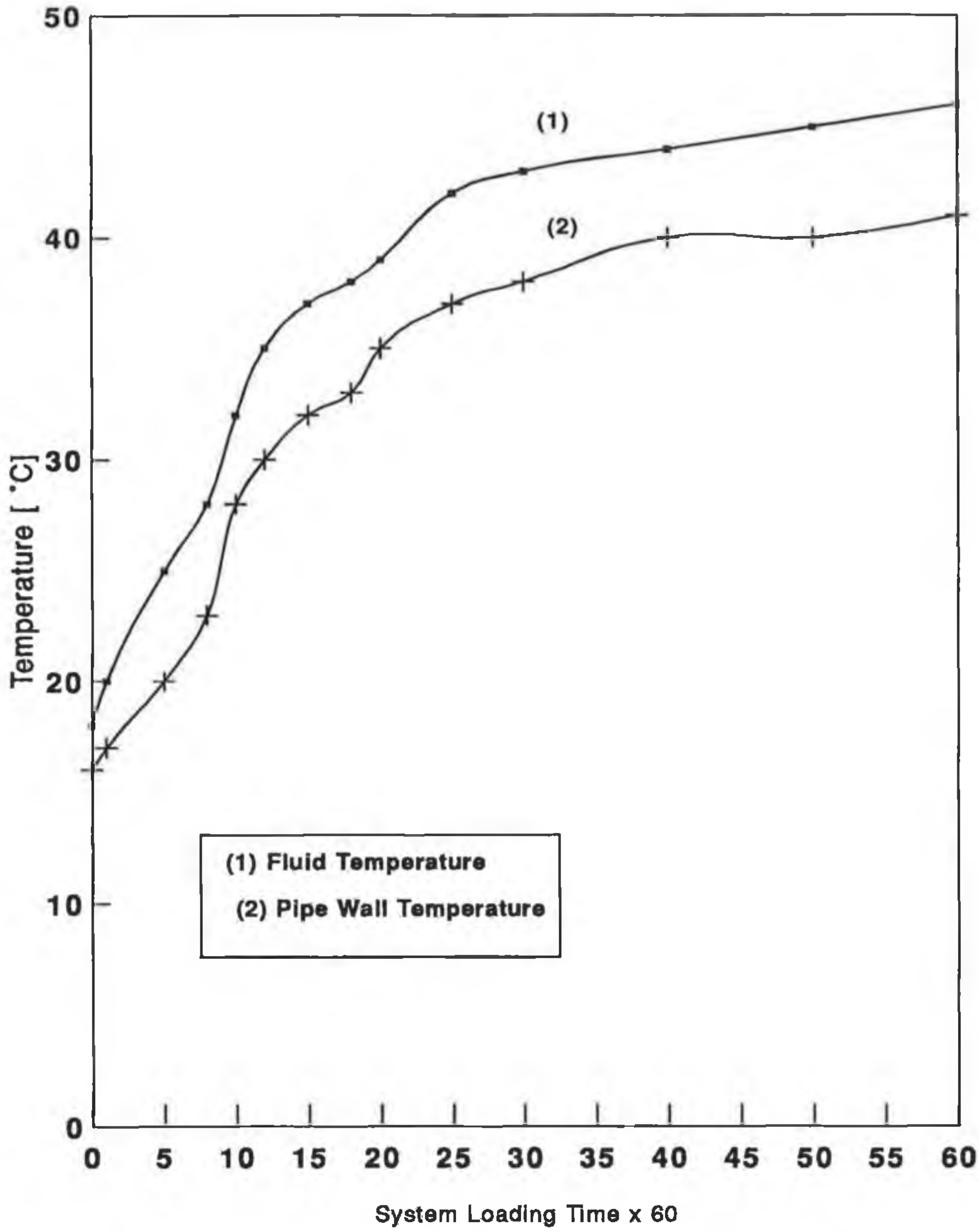


Fig. (95) Comparison between fluid and pipe wall temperature in the open hydraulic drive mixer system.

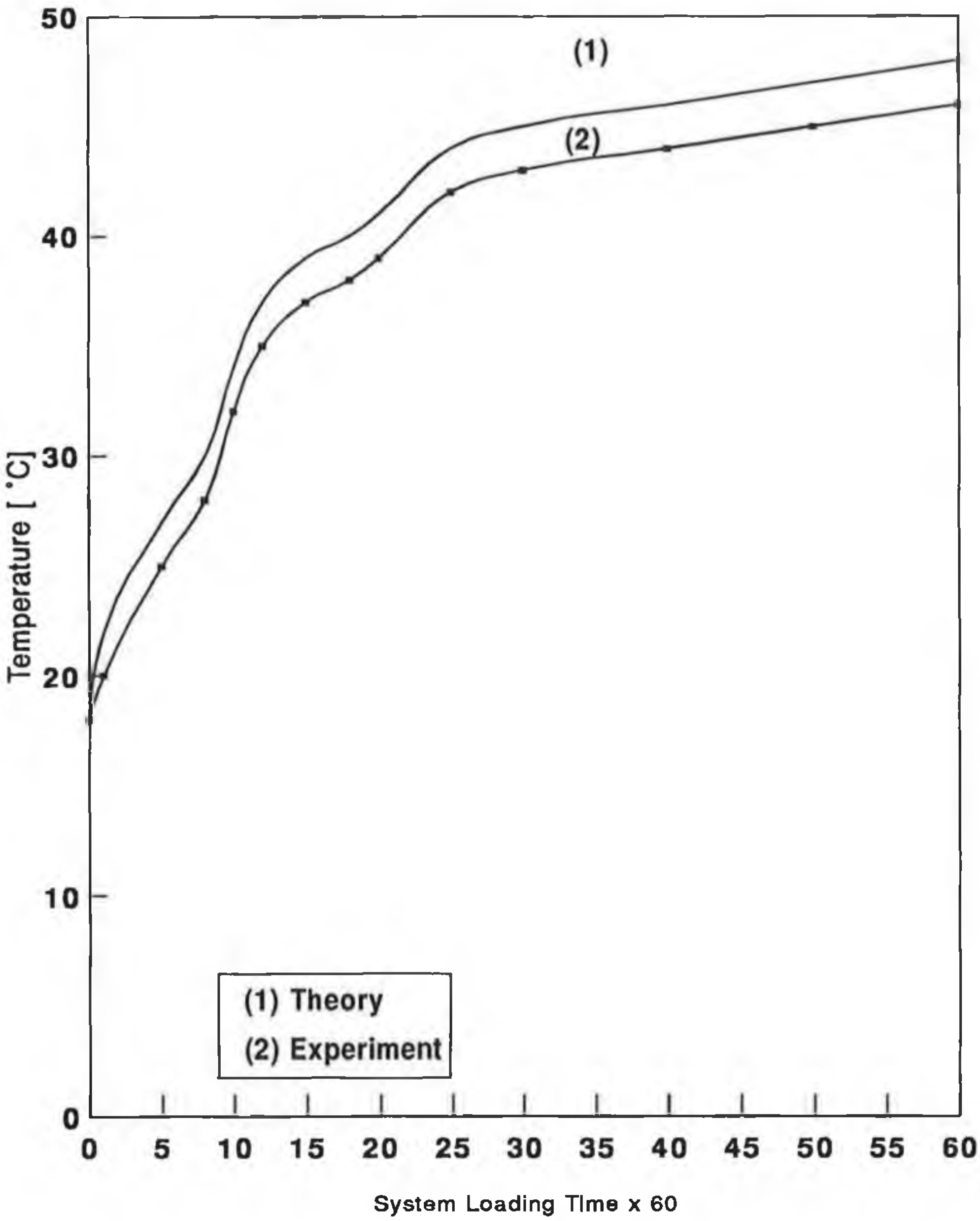


Fig. (96) Comparison between experimental and theoretical results of fluid temperature in the pressure line in the open hydraulic drive mixer system.

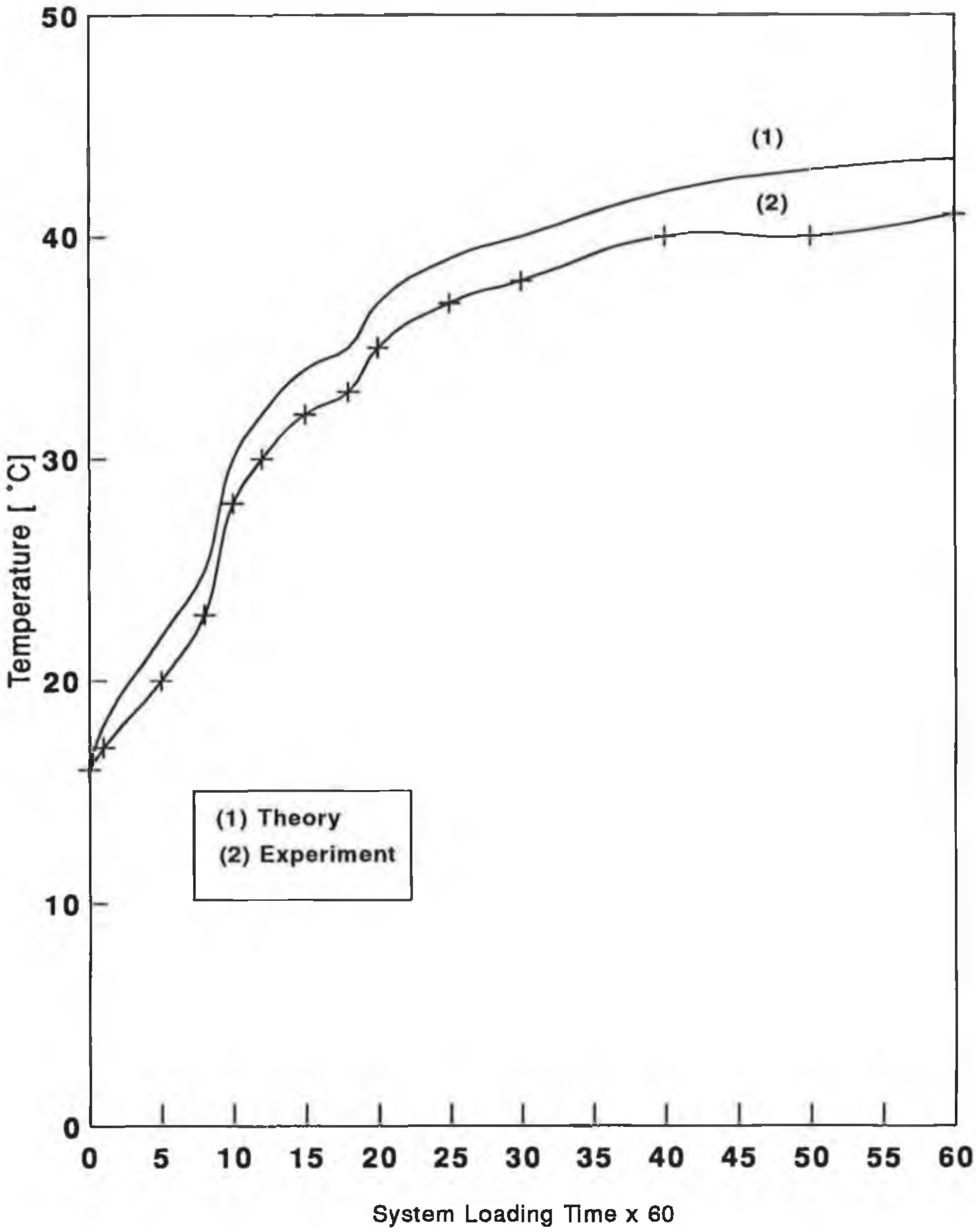


Fig. (97) Comparison between experimental and theoretical results of hose wall temperature in the pressure line in the open hydraulic drive mixer system.

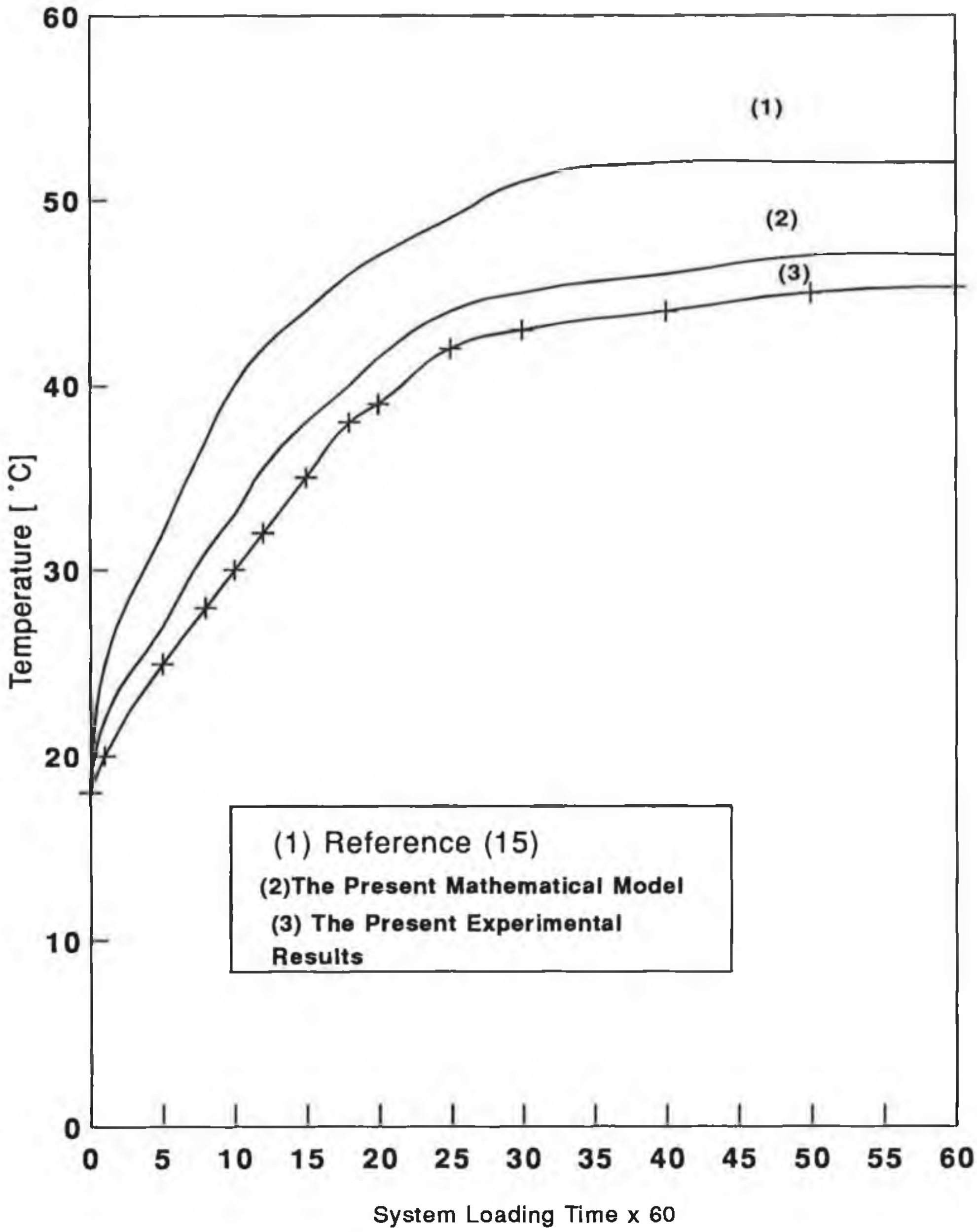


Fig. (98) Comparison of the fluid temperature between the present analysis and the analysis in Reference (15) in the open hydraulic drive mixer system.

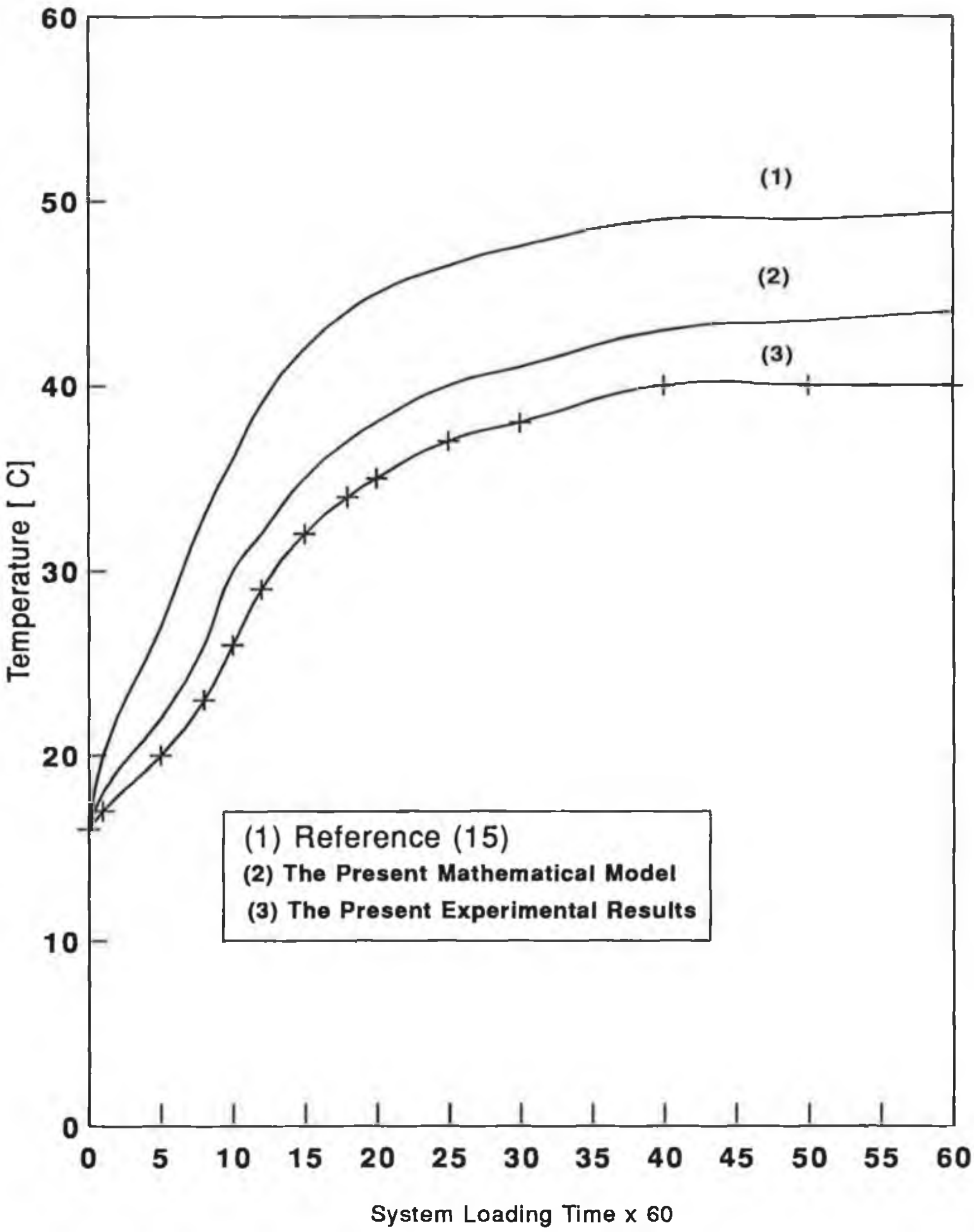


Fig. (99) Comparison of the outside wall temperature of the hose between the present analysis and the analysis in Reference (15) in the open hydraulic drive mixer system.

CHAPTER: 6

Conclusion and Suggestion for Future Work

6.1 Conclusion

In this thesis a mathematical model for temperature distribution in hydraulic systems has been developed. The heat transferred by conduction, convection and radiation have been taken into account. A study of thermodynamic behaviour of the hydraulic drive mixer system has led the way to develop this mathematical model for temperature prediction in hydraulic systems. In this analysis the heat transfer between the fluid and pipe walls have been taken into account. This analysis can be applied to simulate thermodynamic behaviour of other hydraulic systems. The simulation results have shown that a considerable difference exists between the fluid and pipe wall temperatures in the hydraulic system. A particular interesting development of the work has been the simulation investigations of radiation effects under room temperature condition. On the basis of the work it has been concluded that the radiation effects in the region considered are far more significant than previously thought. This has questioned the widely accepted assumption in such many previous literature which ignores the radiation effects under 200 °C.

Numerical integration technique have been used in this work to solve the various equations formulated in the mathematical model to predict the performance of hydraulic systems in terms of the temperature distribution in hydraulic systems under unsteady state conditions.
The study of the temperature behaviour of a hydraulic system is of particular importance when savings in space and energy are being considered. A digital computer simulation package has been developed to be used in hydraulic system design. The main advantage of this package is that it is user friendly and very little prior programming skill is needed. This Package enables the hydraulic engineer to alter system parameters and determine the resulting system response in a relatively cheaper, quicker and safer way than practical testing would allow.

In this research an extensive programme of experimental tests has been done to investigate the validity of the mathematical model and to investigate the feasibility of providing a cooling unit as an integral part in the circuit of the hydraulic systems. The experimental results have shown that the temperature could be reduced by about 40 % of the working temperature and, that the temperature can always be kept below the recommended level which is between 50 - 60 °C. In addition, the main result of the experiments has shown that the hydraulic reservoir is reduced in the size by 6 to 8 times that of the conventional reservoir normally needed in the open hydraulic systems.

The simulations described in this thesis have shown sufficiently close agreement with the experiments to validate the mathematical models. They have been used to give full explanation of the behaviour of the system considered. They can be also used to be an effective design tool for the system components including cooler, filter and reservoir sizing.

It can be seen from the examination of temperature simulations and corresponding experimental results that the theoretical analysis have been implemented into the developed software package and hence they can be improved to cover temperature modelling in other more complex hydraulic systems.

6.2. Suggestions for Future Work

The performance of an open hydraulic system with compact size of the reservoir has been tested experimentally. In the analysis it was attempted to incorporate as many factors as possible. However, further work could usefully be conducted in the following areas.

6.2.1. Theoretically

The mathematical models have been developed based on the assumption that uniform temperature is represented in the loop fluid and wall temperatures. But, when the fluid temperature in one part of the loop significantly differs from another part in a hydraulic system, the assumption of uniform loop fluid temperature in the present analysis can no longer be applied. In this case it is suggested that more accurate theoretical investigation should be carried out to predict the temperature distribution in different parts of the loop during the system loading.

In the developed software package, the user is required to enter specific data. Such data should include pipe and reservoir thermal properties, valves factors, roughness of pipes..etc. It is suggested that data should be collected which more adequately suit the needs of simulation and all the required data should be included in the package itself as data base or expert system and the user can easily pick up the required information.

6.2.2. Experimentally

In the experimental work all pipes in the hydraulic system were rubber hose, but the theoretical results have shown that if the pipelines of hydraulic hose in the system are replaced by the commonly used mild steel or copper pipes, a large difference between the outside pipe wall temperatures of steel or copper pipe and rubber hose will exist. Experimental investigations should be carried out to verify these theoretical results.

It is recommended that a most promising development would be to carry out a test programme on a practical hydraulic system such as liquid or powder tanker or practical hydraulic drive mixer system. This would be essential before it can be suggested that the simulation package would enable savings in size, cost and energy supply of the hydraulic systems.

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APPENDICES

Published Papers

1-Al Natour, I. and Hashmi, M.S.J. "A Mathematical Model and Computer Aided Design for the Temperature Distribution in an Open Hydraulic System"

Has been accepted for publication in the **International journal of NUMERICAL METHODS FOR HEAT & FLUID FLOW, U.K**

2- Al Natour, I. and Hashmi, M.S.J. " Development of a High Performance Integral Oil Cooler and Compact Sized Reservoir Unit for Hydraulic Systems"

The **Transaction of ASME in the Journal of Engineering for Industry, USA.**

3- Al Natour, I. and Hashmi, M.S.J. "Numerical Solution of a Mathematical Model for Temperature Analysis in Hydraulic Systems under Unsteady Conditions"

International Congress on Numerical Methods in Engineering and Applied Sciences, Concepcion, Chile on November 16-20, 1992.

4- Al Natour, I. and Hashmi, M.S.J. "Mathematical Model in Temperature Analysis in Open Power Hydraulic Systems with Consideration of Heat Transfer and Thermodynamics Process in the Hydraulic Reservoir."

COMPUTER PROGRAM

MAIN PROGRAM

```
DECLARE SUB Main ()
DECLARE SUB thermal ()
DECLARE SUB CIRCUIT ()
DECLARE SUB MOTOR ()
DECLARE SUB Pump ()
DECLARE SUB YESORNO ()
DECLARE SUB TURBULANT ()
' Color constants
CONST BLACK = 0
CONST BLUE = 1
CONST GREEN = 2
CONST CYAN = 3
CONST RED = 4
CONST MAGENTA = 5
CONST BROWN = 6
CONST WHITE = 7
CONST BRIGHT = 8
CONST YELLOW = BRIGHT + BROWN
CONST BLINK = 16

' General constants
CONST FALSE = 0, True = NOT FALSE
CONST maxEntries = 13

' SHARED variables (keep these to an absolute minimum)
COMMON SHARED kolor%, Filename$

' DECLARE statements for the QBSCR Screen Routines

DECLARE FUNCTION BlockSize% (L%, r%, t%, B%)
DECLARE FUNCTION ColorChk ()
DECLARE FUNCTION GetBackground% (row%, col%)
DECLARE FUNCTION GetForeground% (row%, col%)
DECLARE FUNCTION GetString$ (leftCol!, row%, strLen%, foreColor%,
        backColor%)
DECLARE FUNCTION GetVideoSegment! ()
DECLARE FUNCTION MakeMenu% (choice$(), numOfChoices%, justify$,
        leftColumn!, rightColumn!, row%, marker$, fg%, bg%, hfg%,
        hbg%, qfg%, qbg%)
DECLARE FUNCTION SubMenu% (choice$(), currentMenu%,
        numOfChoices%, justify$, leftColumn!, rightColumn!, row%,
        marker$, fg%, bg%, hfg%, hbg%, qfg%, qbg%)
DECLARE FUNCTION ScreenBlank$ (delay)
DECLARE SUB Banner (st$, row%)
DECLARE SUB BlockRestore (L%, r%, t%, B%, scrArray%(), segment!)
DECLARE SUB BlockSave (L%, r%, t%, B%, scrArray%(), segment!)
DECLARE SUB BuildScreen (file$, mode%)
DECLARE SUB Center (st$, row%)
DECLARE SUB ClrScr (mode%, fillChar$)
DECLARE SUB DisplayEntry (entry$, qfg%, qbg%, hfg%, hbg%, fg%,
        bg%, marker$, actionCode%)
DECLARE SUB GetScreen (file$)
DECLARE SUB PutScreen (file$)
DECLARE SUB MakeWindow (topRow!, leftCol!, botRow!, rightCol!,
        foreColor%, backColor%, windowType%, frameType%,
```

```
        shadowColor%, explodeType%, label$)
DECLARE SUB MultiMenu (menusArray$(), numEntries%(),
        menuTitles$(), justify$, marker$, shadowCode%, fg%, bg%,
        hfg%, hbg%, qfg%, qbg%, menuSelected%,
        menuEntrySelected%)
DECLARE SUB OffCenter (st$, row%, leftCol%, rightCol%)
DECLARE SUB QBPrint (st$, row%, col%, fore%, back%)
DECLARE SUB ScrnRestore (firstLine%, lastLine%, scrArray%(),
        segment)
DECLARE SUB ScrnSave (firstLine%, lastLine%, scrArray%(),
        segment)
DECLARE SUB Wipe (top%, bottom%, lft%, rght%, back%)
DECLARE FUNCTION BlockSize% (L%, r%, t%, B%)
DECLARE FUNCTION ColorChk ()
DECLARE FUNCTION GetBackground% (row%, col%)
DECLARE FUNCTION GetForeground% (row%, col%)
DECLARE FUNCTION GetString$ (leftCol!, row%, strLen%, foreColor%,
        backColor%)
DECLARE FUNCTION GetVideoSegment! ()
DECLARE FUNCTION MakeMenu% (choice$(), numOfChoices%, justify$,
        leftColumn!, rightColumn!, row%, marker$, fg%, bg%, hfg%,
        hbg%, qfg%, qbg%)
DECLARE FUNCTION SubMenu% (choice$(), currentMenu%,
        numOfChoices%, justify$, leftColumn!, rightColumn!, row%,
        marker$, fg%, bg%, hfg%, hbg%, qfg%, qbg%)
DECLARE FUNCTION ScreenBlank$ (delay)
DECLARE SUB Banner (st$, row%)
DECLARE SUB BlockRestore (L%, r%, t%, B%, scrArray%(), segment!)
DECLARE SUB BlockSave (L%, r%, t%, B%, scrArray%(), segment!)
DECLARE SUB BuildScreen (file$, mode%)
DECLARE SUB Center (st$, row%)
DECLARE SUB ClrScr (mode%, fillChar$)
DECLARE SUB DisplayEntry (entry$, qfg%, qbg%, hfg%, hbg%, fg%,
        bg%, marker$, actionCode%)
DECLARE SUB GetScreen (file$)
DECLARE SUB PutScreen (file$)
DECLARE SUB MakeWindow (topRow!, leftCol!, botRow!, rightCol!,
        foreColor%, backColor%, windowType%, frameType%,
       shadowColor%, explodeType%, label$)
DECLARE SUB MultiMenu (menusArray$(), numEntries%(),
        menuTitles$(), justify$, marker$, shadowCode%, fg%, bg%,
        hfg%, hbg%, qfg%, qbg%, menuSelected%,
        menuEntrySelected%)
DECLARE SUB OffCenter (st$, row%, leftCol%, rightCol%)
DECLARE SUB ScrnRestore (firstLine%, lastLine%, scrArray%(),
        segment)
DECLARE SUB ScrnSave (firstLine%, lastLine%, scrArray%(),
        segment)
DECLARE SUB Wipe (top%, bottom%, lft%, rght%, back%)

' DECLARE statements for routines local to this program

DECLARE FUNCTION Pause% (delay!)
DECLARE SUB ClosingScreen ()
DECLARE SUB Initialize ()
DECLARE SUB KeyPause ()
DECLARE SUB MenuScreen ()
DECLARE SUB MovingMessage ()
```

DECLARE SUB OpenScreen ()

```
ClrScr 5, " "
MakeWindow 16, 15, 22, 70, 15, 1, 0, 3, 16, 1, ""
  FOR i% = 1 TO 5
  READ column%, word$
  FOR row% = 2 TO 18
  LOCATE row%, column%
  PRINT word$
  LOCATE row% - 1, column%
  PRINT SPACE$(LEN(word$))
  SOUND (2400 / row%), 1
  NEXT row%
  NEXT i%
  LOCATE 1, 1
  DATA 25, "welcome", 33, "to", 36, "hydraulic", 48, "system",
  "design"
  PressKey$ = INPUT$(1)

    Initialize
    OpenScreen
    MenuScreen
     bannerString$ = " The Hydraulic System Design by Ibrahim Al
Natour "
done% = FALSE
DO
K$ = INKEY$
IF K$ <> "" THEN
done% = True
ELSE
Banner bannerString$, 3
END IF
LOOP UNTIL done%

      ClosingScreen
```

SUB MenuScreen

```
' Define menu screen file name and set colors for menu
IF kolor% THEN
 scrFile$ = "MENU.CLR"
 fg% = CYAN: bg% = BLUE
 hfg% = YELLOW: hbg% = CYAN
 qfg% = BRIGHT + WHITE: qbg% = BLACK
 COLOR BLUE, BLACK
ELSE
 scrFile$ = "MENU.MON"
 fg% = WHITE: bg% = BLACK
 hfg% = BLACK: hbg% = WHITE
 qfg% = BRIGHT + WHITE: qbg% = BLACK
 COLOR WHITE, BLACK
END IF

' Define menu array for MakeMenu call
DIM menu$(maxEntries)
   menu$(1) = "^Hydraulic Pump"
 menu$(2) = "Hydraulic ^Motor"
 menu$(3) = "^Circuit Design"
 menu$(4) = "^Power Losses"
 menu$(5) = "^Thermal Analysis"
 menu$(6) = "^Miscellaneous"

' Place the menu screen on the display, in a very nifty
 fashion
ClrScr 3, CHR$(176)
BuildScreen scrFile$, 2

' Save the screen for fast restore later
DIM scrArray%(4000)
ScrnSave 1, 25, scrArray%(), GetVideoSegment

' We'll sit in a loop until the user is done pushing keys
done% = FALSE
DO
 ' Make the menu first of all
 choice% = MakeMenu%(menu$(), maxEntries, "L", 29, 53, 7,
 "^", fg%, bg%, hfg%, hbg%, qfg%, qbg%)

 ' Decide what to do based on the user's selection
 SELECT CASE choice%
 CASE 0     ' The ESC key was hit - MakeMenu exits with 0
     done% = True
 CASE 1     ' Pump
     Pump
 CASE 2     ' Motor
     MOTOR
 CASE 3     ' cIRCUIT
     CIRCUIT
 CASE 4     ' ClrScr

 CASE 5     ' thermal analysis
     thermal
```

```
     CASE ELSE
     END SELECT

     ' Restore the screen after selected demo returns
     IF NOT (done%) THEN
         ScrnRestore 1, 25, scrArray%(), GetVideoSegment
     END IF

    LOOP UNTIL done%

END SUB
```

SUB MovingMessage

```
' A few local vars
bigPause = 750
littlePause = 35

' Setup our scrnsave arrays
DIM under%(BlockSize%(34, 46, 19, 19))
DIM over%(BlockSize%(34, 46, 19, 19))
segment = GetVideoSegment

' Set colors
IF kolor% THEN
 COLOR 15, 4
ELSE
 COLOR 0, 7
END IF

' Save portion of screen and make the initial message
BlockSave 34, 46, 19, 19, under%(), segment
LOCATE 19, 34, 0: PRINT " Hit any key ";
BlockSave 34, 46, 19, 19, over%(), segment
IF Pause%(bigPause) = FALSE THEN
 EXIT SUB
END IF

' Move message to left side of screen
FOR x% = 34 TO 6 STEP -2
 BlockRestore x%, x% + 12, 19, 19, over%(), segment
 IF Pause%(littlePause) = FALSE THEN
     EXIT SUB
 END IF
 BlockRestore x%, x% + 12, 19, 19, under%(), segment
NEXT x%
BlockRestore x%, x% + 12, 19, 19, over%(), segment
IF Pause%(bigPause) = FALSE THEN
 EXIT SUB
END IF

' Sit in a loop that moves the message while waiting for a
 keypress
done% = FALSE
DO
 ' Move up
 FOR y% = 18 TO 4 STEP -1
     BlockRestore 4, 16, y% + 1, y% + 1, under%(), segment
     BlockSave 4, 16, y%, y%, under%(), segment
     BlockRestore 4, 16, y%, y%, over%(), segment
     IF Pause%(littlePause) = FALSE THEN
      EXIT SUB
     END IF
 NEXT y%
 IF Pause%(bigPause) = FALSE THEN
     EXIT SUB
 END IF
```

```
    'Move right
    FOR x% = 4 TO 64 STEP 2
        BlockRestore x%, x% + 12, 4, 4, over%(), segment
        IF Pause%(littlePause) = FALSE THEN
         EXIT SUB
        END IF
        BlockRestore x%, x% + 12, 4, 4, under%(), segment
    NEXT x%
    BlockRestore x%, x% + 12, 4, 4, over%(), segment
    IF Pause%(bigPause) = FALSE THEN
        EXIT SUB
    END IF

    ' Move down
    FOR y% = 5 TO 19
        BlockRestore 66, 78, y% - 1, y% - 1, under%(), segment
        BlockSave 66, 78, y%, y%, under%(), segment
        BlockRestore 66, 78, y%, y%, over%(), segment
        IF Pause%(littlePause) = FALSE THEN
         EXIT SUB
        END IF
    NEXT y%
    IF Pause%(bigPause) = FALSE THEN
        EXIT SUB
    END IF

    ' Move left
    FOR x% = 66 TO 6 STEP -2
        BlockRestore x%, x% + 12, 19, 19, over%(), segment
        IF Pause%(littlePause) = FALSE THEN
         EXIT SUB
        END IF
        BlockRestore x%, x% + 12, 19, 19, under%(), segment
    NEXT x%
    BlockRestore x%, x% + 12, 19, 19, over%(), segment
    IF Pause%(bigPause) = FALSE THEN
        EXIT SUB
    END IF

   LOOP UNTIL done%

END SUB
```

SUB YESORNO

```
' ****************SUBROUTINE FOR YES/NOCHECK******************

WHILE ANS$ <> "YES" AND ANS$ <> "yes" AND ANS$ <> "NO" AND ANS$
<> "no"
CLS : LOCATE 15, 15: INPUT "THAT'S A 'YES' or 'NO' I'M LOOKING
FOR "; ANS$
WEND

END SUB
```

SUB thermal

```
ClrScr 5, " "
MakeWindow 8, 12, 11, 68, 14, 1, 0, 2, 16, 1, "Open a File"
LOCATE 9, 15: INPUT "What file name would you like to open";
Filename$
OPEN Filename$ FOR OUTPUT AS #1
OPTION BASE 1
PRINT #1, "GENERAL CHART"
DIM Q1(60), Q2(60), Q3(70), HAV(60), HAP(60), HAP1(60), Twv(60),
Twp(60), DTf(60), Tf(60)
DIM HA(70), H1(60), H2(60), GR(60), T1(60), T2(60), DT2(60),
AA(60), DTFR(60), TFR(60)
i = 1
ClrScr 6, " "
MakeWindow 10, 6, 15, 75, 14, 1, 0, 3, 16, 1, "Input Data for
Pipes"
LOCATE 12, 15: INPUT "Enter the atmospheric temperature, Ta in
K"; Ta
ClrScr 12, " "
MakeWindow 10, 6, 15, 75, 14, 1, 0, 5, 16, 2, " Input Data for
Pipes"
LOCATE 12, 15: INPUT "Enter the kinimatic viscosity MU in
m^2/sec."; MU
ClrScr 3, " "
MakeWindow 10, 6, 15, 75, 14, 5, 0, 2, 16, 2, " Input Data for
Pipes"
LOCATE 12, 15: INPUT "Enter the initial fluid temperature Tf(0)
in K"; Tf(i)
ClrScr 4, " "
MakeWindow 10, 6, 15, 78, 10, 4, 0, 5, 16, 2, "Input Data for
Pipes "
LOCATE 12, 12: INPUT "Enter the temperature of outside wall ofthe
pipe T2(0) in K"; T2(i)
ClrScr 8, " "
MakeWindow 10, 6, 15, 77, 13, 1, 0, 5, 16, 1, "Input Data for
Pipes"
LOCATE 12, 12: INPUT "Enter the inside diameter of pipe ,d1 in
[m]"; d1
ClrScr 9, " "
MakeWindow 10, 6, 15, 75, 13, 14, 0, 6, 16, 1, "Input Data for
Pipes"
LOCATE 12, 15: INPUT "Enter the outside diameter of pipe, d2 in
[m]"; d2
ClrScr 10, " "
MakeWindow 10, 6, 15, 75, 16, 6, 0, 5, 16, 1, "Input Data for
Pipes"
LOCATE 12, 15: INPUT "Enter the length of pipe, L in [m]"; L
ClrScr 1, " "
MakeWindow 10, 6, 15, 75, 14, 1, 0, 2, 16, 2, "Input Data for
Pipe"
LOCATE 12, 15: INPUT "Enter the density of pipe material Rp in
kg/m^3"; Rp
ClrScr 2, " "
MakeWindow 10, 6, 15, 75, 14, 1, 0, 2, 16, 1, "Input Data for
Fluid"
LOCATE 12, 15: INPUT "Enter the density of fluid Rf in kg/m^3";
```

```
Rf
ClrScr 3, " "
MakeWindow 10, 6, 15, 75, 14, 4, 0, 5, 16, 1, "Input Data for
Fluid"
LOCATE 12, 15: INPUT "Enter flow rate of fluid Qf in m^3/s"; Qf
ClrScr 4, " "
MakeWindow 10, 6, 15, 77, 10, 4, 0, 5, 16, 1, "Input Data for
Pipes"
INPUT "Enter the volume of fluid Vf in m^3"; Vf
LOCATE 12, 12: INPUT "Enter the thermal conductivity ,k of pipe
wall material "; K
ClrScr 5, " "
MakeWindow 10, 6, 15, 75, 7, 1, 0, 6, 16, 2, "Input Data for
Pipes"
LOCATE 12, 15: INPUT "Enter the emissivity of pipe wall material
E "; E
ClrScr 6, " "
MakeWindow 10, 6, 15, 75, 14, 4, 0, 2, 16, 1, "Input Data for
Fluid"
LOCATE 12, 15: INPUT "Enter fluid specific heat Cpf"; Cpf
ClrScr 7, " "
MakeWindow 10, 6, 15, 75, 15, 1, 0, 2, 16, 1, "Input Data for
Pipes"
LOCATE 12, 15: INPUT "Enter pipe wall specific heat Cpp"; Cpp
ClrScr 8, " "
MakeWindow 10, 6, 15, 75, 14, 4, 0, 5, 16, 2, "Input Data for
Power Losses"
LOCATE 12, 15: INPUT "Enter total power losses W in Watts"; W
ClrScr 9, " "
MakeWindow 10, 6, 15, 75, 15, 1, 0, 6, 16, 2, "Input Data"
LOCATE 12, 15: INPUT "Enter initial outside vertical tank
temperature in °K Twv"; Twv(i)
ClrScr 10, " "
MakeWindow 10, 6, 15, 79, 15, 1, 0, 6, 16, 2, "Input Data"
LOCATE 12, 15: INPUT "Enter initial tank bottom wall temperature
in °K, Twp "; Twp(i)
ClrScr 11, " "
MakeWindow 10, 6, 15, 79, 14, 4, 0, 6, 16, 2, "Input Data "
LOCATE 12, 15: INPUT "Enter thickness of tank wall in [m], x";
x
ClrScr 12, " "
MakeWindow 10, 6, 15, 79, 14, 4, 0, 6, 16, 2, "Input Data for the
Reservoir"
LOCATE 12, 15: INPUT "Enter density of tank wall material Rt in
[kg/m^3]"; Rt
ClrScr 13, " "
MakeWindow 10, 6, 15, 79, 14, 4, 0, 6, 16, 2, "Input Data for the
Reservoir"
LOCATE 12, 15: INPUT "Enter ht, fluid height in [m]"; ht
ClrScr 14, " "
MakeWindow 10, 6, 15, 80, 14, 1, 0, 6, 16, 2, "Input Data for the
Reservoir"
LOCATE 12, 10: INPUT "Enter Tt, temperature of air inside the
reservoir above the fluid surface in °k"; Tt
ClrScr 1, " "
MakeWindow 10, 6, 15, 75, 14, 1, 0, 6, 16, 2, "Input Data for the
Reservoir"
LOCATE 12, 15: INPUT "Enter L1, length of tank in [m]"; L1
```

```
ClrScr 2, " "
MakeWindow 10, 6, 15, 75, 14, 1, 0, 6, 16, 2, "Input Data for the
Reservoir"
LOCATE 12, 15: INPUT "Enter S, width of tank in [m] "; S
ClrScr 3, " "
MakeWindow 10, 6, 15, 79, 14, 1, 0, 2, 16, 2, "Input Data for the
Reservoir"
LOCATE 12, 15: INPUT "Enter thermal conductivity, k1 of tank wall
material"; k1
ClrScr 4, " "
MakeWindow 10, 6, 15, 79, 14, 1, 0, 2, 16, 2, "Input Data for the
Reservoir"
LOCATE 12, 15: INPUT "Enter specific heat Cpt of tank wall
material"; Cpt
ClrScr 5, " "
MakeWindow 10, 6, 15, 75, 14, 4, 0, 2, 16, 2, "Input Data for the
Reservoir"
LOCATE 12, 15: INPUT "Enter W1 in Watts "; W1
ClrScr 6, " "
FMT1$ = "  ##  ###.##   ###.##  ###.## .######   ###.##
###.##    "
PRINT #1, " i   T1(i)     H1(i)  H2(i)  DT2(i+1)  T2(i+1)
TfR(i+1) "
A11 = (3.14 * d1 ^ 2) / 4
A21 = (3.14 * d2 ^ 2) / 4
A1 = 3.14 * d1 * L
A2 = 3.14 * d2 * L
SEQ = 5.67E-08
VF = (S * L1 * ht) * 1000
HF = (3.65 * K) / d1
G = 9.810001
MF = (Rf * Qf) / 60
AA61 = S * L1
AA62 = ht * L1
AA63 = ht * S
VP = (A21 * L)
MF1 = Rf * VF
MP = Rp * VP
BIT = 1 / Ta
FOR DT = 60 TO 3000 STEP 25
HA(i) = 1.32 * ((T2(i) - Ta) / d2) ^ .25
H2(i) = (HA(i) * A2 * (T2(i) - Ta)) + (E * SEQ * A2 * (T2(i) ^
4 - Ta ^ 4))
R1 = d1 / 2
R2 = d2 / 2
AM = (A2 - A1) / (LOG(R2 / R1))
AA(i) = HF * A1 * Tf(i)
B = (K * AM) / (R2 - R1)
C = HF * A1
T1(i) = (AA(i) + (B * T2(i))) / (C + B)
H1(i) = HF * A1 * (Tf(i) - T1(i))
DT2(i + 1) = (H1(i) - H2(i)) / (MP * Cpp)
IF DT2(I + 1) = 0 THEN END
T2(i + 1) = T2(i) + DT2(i + 1) * DT
DTf1(i + 1) = (W - H1(i)) / (Mf1 * Cpf)
Tf1(i + 1) = Tf(i) + DTf1(i + 1) * Dt
HAV(i) = 1.42 * ((Twv(i) - Ta) / L1) ^ .25
Q1(i) = 2 * HAV(i) * (AA62 + AA63) * (Twv(i) - Ta) + (E * SEQ *
```

```
(A2 + A3) * (Twv(i) ^ 4 - Ta ^ 4))
HAP(i) = 1.42 * ((Twp(i) - Ta) / L1) ^ .25
HAP1(i) = 1.42 * ((Twp(i) - Tt) / L1) ^ .25
Q2(i) = HAP(i) * AA61 * (Twp(i) - Ta) + (E * SEQ * A1 * (Twrp(i)
^ 4 - Ta ^ 4))
Q3(i) = HAP1(i) * AA61 * (Tf(i) - Tt) + (E * SEQ * A1 * (Twrp(i)
^ 4 - Ta ^ 4))
DTFR(i + 1) = (W1 - Q1(i) - Q2(i) - Q3(i)) / (MF1 * Cpf)
TFR(i + 1) = Tf(i) + DTFR(i + 1) * DT
DTf(i + 1) = ((W - H1(i)) + (MF * Cpf * ((TFR(i + 1) - Tf(i)))))
/ (MF1 * Cpf)
IF DTf(i + 1) < 0 THEN GOTO 930222
Tf(i + 1) = Tf(i) + (DTf(i + 1) * DT)
PRINT USING FMT1$; i; T1(i); H1(i); H2(i); DT2(i + 1); T2(i + 1);
TFR(i + 1)
PRINT #1, USING FMT1$; i; T1(i); H1(i); H2(i); DT2(i + 1); T2(i
+ 1); TFR(i + 1)
i = i + 1
Twp(i) = ((HAP(i - 1) * Ta) + ((k1 / x) * Tf(i))) / (HAP(i - 1)
+ (k1 / x))
Twv(i) = ((HAV(i - 1) * Ta) + ((k1 / x) * Tf(i))) / (HAV(i - 1)
+ (k1 / x))
NEXT DT
FMT2$ = "  ##     ###.##  ##.##    ###.##  ##.##  ##.##    .#####
 ###.##"
PRINT #1, " i          Twv       Q1       Twp        Q2      Q3
DTf(i+1)  Tf(i+1)"
FOR i = 1 TO 60
PRINT
PRINT "*********************************************************"
RINT USING FMT2$; i; Twv(i); Q1(i); Twp(i); Q2(i); Q3(i); DTf(i
+ 1); Tf(i + 1)
PRINT #1, USING FMT2$; i; Twv(i); Q1(i); Twp(i); Q2(i); Q3(i);
DTf(i + 1); Tf(i + 1)
IF DTf(i + 1) < 0 THEN END
NEXT i
CLOSE #1
PressKey$ = INPUT$(1)
MakeWindow 20, 6, 12, 70, 14, 4, 0, 2, 16, 2, "Save the Results"
Center " The Results Have been Save in fille General.dat", 20
PressKey$ = INPUT$(1)
ClrScr 15, " "
END
END SUB
```

SUB CIRCUIT

```
DIM H(15), N.OFRE$(15), DP(15), HH(15), HG(15)
ClrScr 3, " "
MakeWindow 10, 5, 15, 70, 14, 1, 0, 1, 16, 2, ""
LOCATE 12, 10
INPUT "DO YOU HAVE MORE THAN ONE DIAMETER "; ANS$
 YESORNO                               'YES OR NO CHECK
WHILE ANS$ = "YES" OR ANS$ = "yes"
ClrScr 3, " "
MakeWindow 10, 8, 15, 77, 17, 6, 0, 1, 16, 1, " Input Data for
Pipes"
LOCATE 12, 10: INPUT "WHAT IS THE DIAMETER OF THE PIPE OF THE
SYSTEM"; d
WHILE d < 0
MakeWindow 18, 9, 22, 55, 14, 4, 0, 1, 16, 1, "Warning"
LOCATE 20, 10: INPUT "THAT MUST BE A NUMBER >  0 "; d
WEND
MakeWindow 4, 6, 11, 75, 14, 1, 0, 1, 16, 2, "Input Data for a
Pump"
LOCATE 7, 10: INPUT "Enter the displacement of a pump
[Cm^3/rev.]"; VP
ClrScr 3, " "
MakeWindow 4, 6, 11, 75, 15, 1, 3, 1, 16, 3, "Input Data for the
Pump"
LOCATE 7, 10: INPUT "Enter the Speed of the Pump [r.p.m] "; NP
ClrScr 15, " "
MakeWindow 6, 6, 11, 75, 14, 3, 0, 3, 16, 1, "Input Data for the
Pump"
LOCATE 8, 10: INPUT " Enter the volumetric efficiency of the
pump"; PVE
QPA = (PVE * VP * .001) * NP
590 a = (3.14 * (d ^ 2)) / 4
ClrScr 15, " "
MakeWindow 13, 6, 18, 77, 14, 1, 2, 3, 16, 1, "Result"
LOCATE 15, 15: PRINT "The area of ("; d; " m) pipe A= "; a;
"[m^2]"
V = (10 ^ -3 * QPA) / (60 * a)
LOCATE 16, 15: PRINT "The speed of the flow in ("; d; " m)  V=";
V; "[m/sec.]"
MakeWindow 20, 6, 23, 70, 14, 4, 0, 3, 16, 1, "Input Data for
Fluid"
LOCATE 21, 15: INPUT "what is kenimatic viscocity MU [m^2/sec]";
MU
ClrScr 12, " "
RN = (V * d) / MU
MakeWindow 6, 6, 14, 70, 14, 1, 0, 3, 16, 1, " Result"
LOCATE 8, 8: PRINT "Reynolds Number Rn= "; RN

IF RN <= 2000 THEN
MakeWindow 11, 30, 14, 55, 14, 4, 0, 1, 0, 1, ""
LOCATE 12, 31: PRINT "This Flow is Laminar"
f = 64 / RN
Center "Press Any Key to Continue ", 20
PressKey$ = INPUT$(1)
ClrScr 3, " "
MakeWindow 12, 6, 16, 75, 15, 2, 2, 3, 16, 1, " Result"
```

```
LOCATE 14, 10: PRINT "The friction factor f for laminar flow is
f="; f
ELSE
TURBULANT
END IF
MakeWindow 12, 6, 16, 75, 14, 4, 0, 3, 16, 1, "Input Data "
LOCATE 14, 10: INPUT "Enter the value of g [m/sec^2] "; G
MakeWindow 18, 6, 22, 77, 14, 1, 0, 3, 16, 2, "Input Data for the
Pipes"
LOCATE 20, 6: PRINT " what is the total length of the diameter
("; d; ") ";
INPUT LL
HLL = f * (LL / d) * (V ^ 2 / 2 * G)
LOCATE 8, 12: PRINT "HLL="; HLL; "[m]"
ClrScr 15, " "
MakeWindow 6, 6, 9, 75, 15, 6, 0, 3, 16, 1, "Input Data "
LOCATE 8, 8: INPUT "How many restrictor are there"; TOT
WHILE TOT <= 0 OR TOT > 15
MakeWindow 18, 6, 22, 60, 14, 4, 0, 3, 16, 1, "Warning"
LOCATE 20, 8: INPUT "Sorry must be > 0 and < 15"; TOT
WEND
ClrScr 8, " "
H(0) = 0
MakeWindow 6, 6, 12, 77, 14, 1, 0, 3, 16, 1, "Input Data "
LOCATE 8, 10: INPUT "Enter the value of specific gravity of
hydraulic oil Sg"; SG
ClrScr 1, " "
FOR L = 1 TO TOT
MakeWindow 6, 6, 16, 77, 17, 6, 0, 3, 16, 1, " Input Data "
LOCATE 8, 10: PRINT "Enter the name of restrictor ("; L; ")"
LOCATE 10, 10: INPUT N.OFRE$(L)
MakeWindow 12, 12, 16, 70, 14, 1, 0, 1, 16, 1, ""
LOCATE 12, 15: PRINT "What is the value of pressure drop("; L;
") [N/m^2]";
INPUT DP(L)
H(L) = (10 * DP(L)) / SG: HH(L) = H(L)
H(L) = H(L - 1) + H(L)
HG(L) = DP(L) * QPA
OMFR = QPA * SG * 1
OSH = 1.75
TI(l) = HG(l) / OSH * OMFR
NEXT L
ClrScr 13, " "
MakeWindow 6, 6, 15, 77, 14, 1, 0, 3, 16, 1, "Results"
LOCATE 7, 7
FMT1$ = " ##  \            \  #####.##   ######.##   #####.##
######.##  "
PRINT "   L       Name        DP [bar]  Head loss[m]  HH [m]
Heat.G [KJ/min] "
FOR L = 1 TO TOT
PRINT USING FMT1$; TAB(8); L; N.OFRE$(L); DP(L); H(L); HH(L);
HG(L)
IF L = TOT THEN PRINT H(L): HTOT = H(L)
NEXT L
LOCATE 24, 15: PRINT "Press any key to continue:";
INPUT a$
ClrScr 15, " "
MakeWindow 6, 6, 16, 75, 14, 1, 0, 1, 16, 1, " Input Data "
```

```
LOCATE 8, 10: INPUT "How many standard 90 elbow is there"; EL1
MakeWindow 13, 8, 16, 60, 15, 4, 0, 1, 16, 1, "Input Data "
HE1 = EL1 * (.9) * (V ^ 2) / (2 * G)
LOCATE 14, 10: INPUT "How many standard 45 elbow is there EL2";
EL2
ClrScr 10, " "
HE2 = EL2 * (.42) * (V ^ 2 / 2 * G)
MakeWindow 12, 6, 17, 77, 14, 3, 0, 3, 16, 1, " Result"
LOCATE 15, 10: PRINT "HE1="; HE1; "[m]"; "HE2="; HE2; "[m]"
HTOTAL = HLL + HE1 + HE2 + HTOT
LOCATE 22, 10
PRINT "Press Any Key to Continue"
PressKey$ = INPUT$(1)
ClrScr 1, " "
MakeWindow 6, 6, 16, 77, 15, 1, 0, 3, 16, 1, "Results"
LOCATE 7, 7
PRINT "The total head loss in ("; d; " m) diameter of the pipe
="; HTOTAL; "[m]"
DP = (SG * HTOTAL)
LOCATE 19, 10
MakeWindow 10, 6, 16, 77, 14, 4, 0, 2, 16, 2, ""
LOCATE 11, 9
PRINT "Total pressure drop in ("; d; "m) diameter of the pipe
dP="; DP; "[bar]"
LOCATE 23, 10
PRINT " Press Any Key to Continue"
PressKey$ = INPUT$(1)
ClrScr 14, " "
MakeWindow 6, 6, 10, 77, 14, 1, 0, 1, 16, 1, ""
LOCATE 7, 8
INPUT "DO YOU WANT TO GO ON FOR NEXT DIAMETER "; ANS$
YESORNO                      'YES OR NO CHECK'
WEND
END SUB
```

SUB MOTOR

```
ClrScr 2, " "
MakeWindow 8, 10, 13, 75, 15, 1, 0, 2, 16, 2, "Input Data for the
Motor"
LOCATE 10, 14: INPUT "Enter the displacement of the hydraulic
motor Vm"; VM
ClrScr 3, " "
MakeWindow 8, 10, 13, 75, 15, 1, 0, 2, 16, 2, "Input Data for the
Pump"
LOCATE 10, 14: INPUT "Enter the displacement of the hydraulic
pump Vp"; VP
ClrScr 3, " "
MakeWindow 8, 10, 13, 75, 15, 1, 0, 3, 16, 2, "Input Data for the
Pump"
LOCATE 10, 14: INPUT "Enter the speed of the pump [rpm]"; NP
380 NM = NP * VP / VM
ClrScr 4, " "
MakeWindow 8, 10, 13, 70, 4, 7, 0, 3, 16, 2, "Input Data for the
Pressure"
LOCATE 10, 13: INPUT "Enter the operating pressure P[bar]"; P
ClrScr 4, " "
MakeWindow 8, 10, 14, 75, 15, 4, 0, 4, 16, 2, "Results"
LOCATE 10, 13: PRINT "The speed of the hydraulic motor is Nm= ";
NM; "[rpm]"
QMT = NM * VM * .001
LOCATE 11, 13: PRINT "Theoretical flowrate of the motor is Qmt=";
QMT; "[L/MIN]"
Center "Press Any Key to Continue", 20
PressKey$ = INPUT$(1)
ClrScr 4, " "
MakeWindow 8, 10, 12, 75, 14, 1, 0, 3, 16, 2, "Input Data for the
Motor"
LOCATE 10, 12: INPUT "Enter the volumetric efficiency of fluid
power motpr mVE "; MVE
QMA = (VM * .001 * NM) / MVE
MakeWindow 6, 8, 12, 75, 14, 1, 0, 3, 16, 2, "Results"
LOCATE 8, 10: PRINT "Actual flowrate of the hydraulic motor is
Qma= "; QMA; "[L/min]"
TM = (VM * .000001) * (P * 100000!) / (2 * 3.14)
LOCATE 9, 10: PRINT "Theoretical output torque of the motor is
Tm="; TM; "[N.m]"
FHPM = QMA * P / 600
LOCATE 10, 10: PRINT "Actual output power of the motor is Fhpm=";
FHPM; "[KW]"
Center "Press Any Key to Continue", 20
PressKey$ = INPUT$(1)
END SUB
```

SUB Pump

```
ClrScr 5, " "
MakeWindow 8, 12, 11, 60, 14, 1, 0, 2, 16, 1, "Open a File"
LOCATE 9, 15: INPUT " What file name would you like to open";
Filename$
OPEN Filename$ FOR OUTPUT AS #1
ClrScr 6, " "
MakeWindow 4, 6, 9, 70, 14, 1, 0, 1, 16, 2, "Input Data for Pump"
160 LOCATE 7, 10: INPUT "Enter the displacement of the pump
[cm^3/rev]"; VP
ClrScr 15, " "
MakeWindow 4, 6, 9, 70, 15, 3, 0, 1, 16, 2, " Input Data for
Pump"
LOCATE 7, 10: INPUT "Enter the operating pressure P [bar]"; P
ClrScr 3, " "
MakeWindow 4, 6, 11, 75, 15, 1, 3, 1, 16, 3, " Input Data for
Pump"
LOCATE 7, 10: INPUT "Enter the speed of the pump Np [rpm]"; NP
QPT = (VP * NP) * .001
ClrScr 15, " "
MakeWindow 6, 6, 11, 70, 14, 3, 0, 3, 16, 1, " Input Data for
Pump "
LOCATE 9, 10: INPUT "Enter the volumetric efficiency of the pump
pVE"; PVE
QPA = (PVE * VP * .001) * NP
FHPP = (QPA * P) / 600
MakeWindow 5, 6, 11, 75, 14, 4, 0, 5, 16, 1, "Results "
LOCATE 7, 10: PRINT "Theoretical flow rate of the pump Qpt= ";
QPT; "[L/min]"
LOCATE 8, 10: PRINT "Actual pump flow rate is Qpa = "; QPA;
"[L/min]"
Center "Press Any Key to Continue", 20
KeyPress$ = INPUT$(1)
ClrScr 15, " "
MakeWindow 5, 6, 11, 75, 14, 1, 0, 2, 16, 2, "Input Data for Pump
"
LOCATE 7, 10: INPUT "what is the input power Bhp1 [KW] "; BHPI
POE = FHPP / BHPI
ClrScr 4, " "
MakeWindow 10, 6, 18, 75, 14, 1, 0, 2, 16, 1, " Results "
LOCATE 12, 10: PRINT "The pump overall efficiency pOE ="; POE
PTE = POE / PVE
LOCATE 13, 10: PRINT "The pump torque efficiency pTE= "; PTE
TP = (VP * .000001) * (P * 100000!) / (2 * 3.14 * PTE)
LOCATE 14, 10: PRINT "The torque at the pump shaft, Tp="; TP;
"[N.M]"
FHP = BHPI * (1 - PTE)
LOCATE 16, 10: PRINT "Pump  frictional horsepower Fhp="; FHP;
"[KW]"
CLOSE #1
Center "Press Any Key to Continue ", 20
KeyPress$ = INPUT$(1)
ClrScr 3, " "
END SUB
```

SUB TURBULANT

```
' *************SUBROUTINE FORTURBULENTFLOW*******************
LOCATE 24, 15: PRINT "Press any key to continue:";
INPUT a$
ClrScr 8, " "
MakeWindow 6, 4, 16, 77, 14, 1, 0, 3, 16, 1, ""
LOCATE 11, 6: PRINT "The friction factor, f, must be obtained
from Darcy-Weisbach diagram "
LOCATE 13, 6: PRINT "as follows:  Relative Roughness=e/D, Where
"
LOCATE 15, 6: PRINT "e is absolute roughness depends on matrial
of the pipe"
MakeWindow 6, 25, 9, 50, 14, 4, 0, 3, 16, 1, ""
LOCATE 7, 27: PRINT "The Flow is Turbulent"
LOCATE 22, 10: PRINT "Press Any Key to Continue"
PressKey$ = INPUT$(1)
MakeWindow 16, 10, 19, 60, 14, 1, 0, 2, 16, 1, " Input Data "
LOCATE 17, 12: INPUT "Enter the value of e "; E
RR = E / d
ClrScr 9, " "
MakeWindow 6, 10, 18, 78, 15, 4, 0, 3, 16, 1, ""
LOCATE 7, 11
PRINT "After computing Relative Roughness, the f-factor can be
read from"
LOCATE 8, 12: PRINT "Darcy-Weisbach by locating the place of
intersection of the"
LOCATE 10, 12: PRINT "Reynolds number and relative roughness of
the conductor surface"
LOCATE 12, 12: PRINT "and then reading the friction value from
the left or right margins"
Center "Press Any Key to Continue", 21
PressKey$ = INPUT$(1)
ClrScr 1, " "
MakeWindow 13, 5, 18, 78, 14, 1, 0, 3, 16, 1, "Input Data "
LOCATE 16, 7
INPUT "Please enter the computed value of friction factor for
turbulent flow "; f
ClrScr 2, " "
END SUB
```