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A MATHEMATICAL MODEL AND COMPUTER AIDED DESIGN FOR THE TEMPERATURE DISTRIBUTION IN AN OPEN HYDRAULIC SYSTEM

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ABSTRACT

Hydraulic systems that operate for long periods of time eventually develop high oil temperatures that have damaging effects on machine performance. If the temperature rise is excessive, the oil viscosity drops, lubricating properties are lost and in the worst cases the whole system can be seriously damaged. A mathematical model for predicting temperature distribution in hydraulic systems has been developed with taking into account the thermodynamic processes and the effects of heat transfer by conduction, radiation and convection. In order to test this model experimentally, a complete hydraulic mixer system has been designed, instrumented and commissioned. A software package for hydraulic systems has been developed to make accurate estimation for unsteady state temperature analysis in hydraulic systems at any time during its operation. The simulation results of this package have shown that this model is more accurate than that reported elsewhere.

KEY WORDS Temperature distribution Hydraulic systems CAD

NOMENCLATURE

A_1	inside surface area of hose [m^2]
A_2	outside surface area of hose [m^2]
A_{1hr}	reservoir base area [m^2]
A_{2tr}	one side of fluid vertical area in the reservoir [m^2]
A_{3tr}	another side of fluid vertical area in the reservoir [m^2]
A_m	logarithmic mean surface area of hose [m^2]
C_{pf}	specific heat of fluid [J/(kg·K)]
C_{ph}	specific heat of hose wall [J/(kg·K)]
D	hose diameter [m]
f	dimensionless friction factor
g	acceleration of gravity [m/sec^2]
h_a	coefficient of heat transfer by natural convection [W/(m^2·K)]
H_f, H_{ff}	head loss [m]
k	thermal conductivity of hose wall material [W/(m·K)]
k_{res}	thermal conductivity of reservoir wall material [W/(m·K)]
K	constant
L	length of hose [m]
m_1	mass flowrate of fluid into reservoir [kg/sec]
m_2	mass flowrate of fluid outlet of reservoir [kg/sec]
M_f	mass of fluid in the system [kg]

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M_{fh} mass of fluid within the hose section [kg]
M_p mass of wall hose material [kg]
ΔP pressure drop [bar (10^5 N/m^2)]
Q flowrate [m^3/sec]
Q_1 heat flow transferred from fluid to hose
Q_2 heat flow transferred from hose wall to surrounding atmosphere [W/(m·K)]
Q_{1R} heat flow transferred from fluid into its surrounding atmosphere at vertical wall of reservoir [W/(m·K)]
Q_{2R} heat flow transferred from fluid into its surrounding atmosphere at horizontal wall of reservoir [W/(m·K)]
Q_{3R} heat flow transferred from the surface of fluid in the reservoir to atmosphere [W/(m·K)]
R_1, R_2 hose inside and outside radius [m]
S width of the base of the reservoir [m]
T_1 temperature of the wall at inside hose surface [K]
T_2 temperature of the wall at outside hose wall surface [K]
T_a constant surrounding atmosphere temperature [K]
T_f temperature of fluid [K]
T_{res} temperature of fluid in reservoir [K]
T_{wrb} outside wall temperature at bottom of reservoir [K]
T_{wrv} outside vertical wall temperature of reservoir [K]
W power losses converted into heat energy [Watts]
x thickness of the wall of the reservoir [m]

Greek symbols

ε emissivity
ν kinematic viscosity [m^2/sec]
σ surface tension [N/m]

INTRODUCTION

Since no hydraulic system can convert mechanical energy to hydraulic energy and transmit it at 100% efficiency, heat will be generated in the system due to the power losses. The higher a fluid's temperature, the lower its viscosity and lubricity which will affect the system performance. However, thermal transient in any hydraulic system is an aspect of system performance and it is generally associated with thermodynamic and heat transfer processes which are related to power losses converted into heat during the operation of the system. Most previous studies for temperature analysis in hydraulic systems were concerned with steady state conditions in which heat flows continuously at a uniform rate[1–3], and unsteady and thermodynamic and heat transfer processes were ignored due to their complexities. The temperature analysis in hydraulic system have also been reported[4–6] but in these studies the authors have not considered the effects of heat transfer or thermodynamics processes under unsteady state conditions and, their analyses were developed based on the assumption that both inside and outside pipe wall temperature of the system are the same as the fluid temperature; in addition the heat generated by the pressure losses due to pipe friction were negligible. Yang[7,8] and Bown and Yang[9] have studied the temperature rise in hydraulic systems considering it as a closed thermal system and compared their results obtained experimentally using a partly closed hydrostatic drive propeller system. However, in their study, the heat transfer by conduction in the system and heat transfer by radiation from the surfaces of reservoir were ignored, which could give rise to errors in estimating the temperature distribution in hydraulic systems. 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 with a uniform fluid and pipe wall temperature. In addition, in this analysis the differences between the fluid and pipe wall temperatures have been considered. Thus, the transient difference in the heat flows transferred from the fluid to the pipe wall and from the pipe wall to the system ambient can be estimated to predict transient variations in the 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.

MATHEMATICAL MODEL

In order to study and verify the temperature behaviour in any hydraulic system, it is important to develop a suitable mathematical model. As a first step, a hydraulic system or one part of it can be treated as a thermodynamic system where the power losses will be converted into heat energy which causes the variations in the system temperatures; and the heat transfer will remove some of this heat, while the remainder will increase or decrease the internal energy of the fluid and the containing metal. In the present analysis, the hydraulic system is considered as two open thermal systems, the main loop is considered as an open thermal system where a uniform temperature is assumed to represent the loop fluid temperature, while the reservoir is another open thermal system as shown in *Figure 1a*. Hence, the fluid energy exchanges between the loop and reservoir can be estimated.

It is known that the first law of thermodynamics is applicable to a closed thermal system where no mass flow crosses the boundary. In most engineering applications it is difficult to separate a mass of the working substance and treat it as a closed thermal system. Zeuner[10] and Gillespie *et al.*[11] suggested that it is possible to regard the continuous flow process as a series of non-flow processes undergone by an imaginary closed system[12]. However, with an appropriate derivation, the first law of thermodynamics for an open thermal system can be expressed as:

$$\frac{dE}{dt} = Q - W + \sum_{\text{inlet}} m(e + Pv) - \sum_{\text{outlet}} m(e + Pv) \tag{1}$$

where Q is the heat flowrate transferred into the system, W is the rate of work done to the ambient. In this equation the term $m(e + Pv)$ represents the energy transfer associated with the flow of mass across the system boundary. The specific energy, e, in the absence of macroscopic forms of energy storage other than kinetic and gravitational, can be decomposed into $(u + 1/2V^2 + gz)$. The result of this decomposition is that the specific enthalpy $h = u + Pv$ which shows up explicitly in the terms accounting for energy transfer via mass flow, so that

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

In 1966, Kestin proposed an engineering generalization of enthalpy concept under the name of *methalpy*, (h°)[13] [p. 223: $h = e + Pv = h + 1/2V^2 + gz$]. In (2) the open system has a quantity

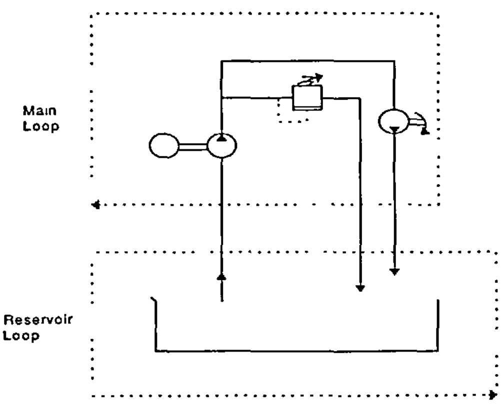

Figure 1a An open hydraulic system as two open thermal systems

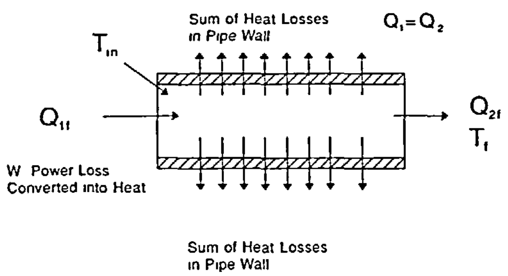

Figure 1b Open thermal system of fluid in a pipe section

of internal energy E which can be defined as:

$$E = M(h + \tfrac{1}{2}V^2 + gz) \tag{3}$$

Any specific length of pipe in a hydraulic system can be regarded as an open thermal system undergoing a thermodynamic process as shown in *Figure 1b*. If a uniform fluid temperature is assumed within the open section of the pipe, and the change in fluid kinetic energy and potential energy for a specific hydraulic pipeline are negligible, the thermodynamic flow process associated with the section of pipe can be studied by applying (1) for open thermal system, so that

$$\frac{\mathrm{d}E}{\mathrm{d}t} = Q_1 - W + m_1 h_1^\circ - m_2 h_2^\circ \tag{4}$$

Substituting (3) into (4) to define the system internal energy by ignoring its kinetic and potential energies, we have:

$$Q_1 - W + m_1 h_1^\circ - m_2 h_2^\circ = \frac{\mathrm{d}(M_{fh})}{\mathrm{d}t} \tag{5}$$

Since in the case being investigated, the effect of pressure on internal energy is small and the heat exchange due to compression or expansion of the fluid is ignored, the fluid enthalpy can be defined as:

$$h = C_v T = C_p T \tag{6}$$

Therefore,

$$M_f C_p \frac{\mathrm{d}T_f}{\mathrm{d}t} = Q_1 - W + m_f C_p (T_{\text{inlet}} - T_f) \tag{7}$$

The study of heat transfer by forced convection is usually concerned with the calculation of rates of heat exchange between fluid and solid because the fluid is flowing inside the wall. A condition of forced convection applies to heat transfer across the wall (*Figure 2*) which shows a view of the pipe section illustrating the heat transfer at any instant of time. It is known that the transfer of heat by forced convection can be written as:

$$Q_1 = h_f A_1 (T_f - T_1) \tag{8}$$

The heat flow transferred from the pipe wall to the ambient by conduction, natural convection and radiation Q_2 can be determined by using Fourier's law and the Stefan–Boltzmann law as

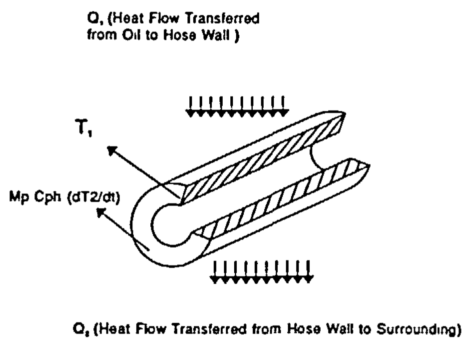

Figure 2 Heat flow losses through the hose in surroundings

follows:

$$Q_2 = \frac{2\pi kL}{\ln \frac{R_2}{R_1}}(T_1 - T_2) = h_a A_2(T_2 - T_a) + \varepsilon\sigma A_2(T_2^4 - T_a^4) \tag{9}$$

In the above two equations the heat transfer coefficients have been used from the result of the experimental work carried out by Rogers and Mayhew[14].

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 state, the difference between Q_1 and Q_2 must equal the increment of the pipe wall internal energy. Hence,

$$Q_1 - Q_2 = \frac{dE}{dt} \tag{10}$$

From (3) and (6) we have:

$$\frac{dE}{dt} = \frac{d(M_p C_p T_2)}{dt} \tag{11}$$

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, (10) becomes:

$$M_p C_p \frac{dT_2}{dt} = Q_1 - Q_2 \tag{12}$$

If the heat flow through the pipe wall is assumed to be transferred under one dimensional condition, then the heat flow radially through the pipe wall can be expressed by Fourier's law as,

$$Q = \frac{kA_m(T_1 - T_2)}{R_2 - R_1} \tag{13}$$

The heat transferred through the pipe wall by conduction must be equal to the heat transferred from the fluid to the pipe wall by forced convection at the boundary of the inner surface of the pipe section. Therefore,

$$Q = \frac{kA_m(T_1 - T_2)}{R_2 - R_1} = h_f A_1(T_f - T_1) \tag{14}$$

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

$$T_1 = \frac{h_f A_1 T_f + \frac{kA_m}{R_2 - R_1} T_2}{h_f A_1 + \frac{kA_m}{R_2 - R_1}} \tag{15}$$

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 to the fluid in the reservoir. If the loop and the 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 (7) so that:

$$\frac{dT_f}{dt} = \frac{\sum W - \sum Q_1 + m_f C_{pf}(T_{res} - T_f)}{M_f C_{pf}} \tag{16}$$

In (16), $\sum W$ is the total losses and $\sum Q_1$ is the total heat flow transferred from fluid to the pipe walls by forced convection. T_{res} is the uniform reservoir transient fluid temperature which is determined as follows. The heat flow from fluid in the reservoir into its surrounding atmosphere at the vertical wall of the reservoir can be written as:

$$Q_{1R} = \frac{T_f - T_a}{\dfrac{S}{2k_{res}(A_{2vr} + A_{3vr})} + \dfrac{1}{2h_{avr}(A_{2vr} + A_{3vr})}} \tag{17}$$

The heat flows from the fluid into the surrounding atmosphere at the horizontal wall of the reservoir can be expressed as:

$$Q_{2R} = \frac{T_f - T_a}{\dfrac{S}{k_{res}A_{1hr}} + \dfrac{1}{h_{abr}A_{1hr}}} \tag{18}$$

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

$$Q_{3R} = h_{abr1}A_{1hr}(T_f - T_t) \tag{19}$$

where T_t is the temperature of air inside the reservoir above the fluid surface 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 T_a but constant value of T_t may be assumed. After an appropriate derivation, the outside vertical wall temperature T_{wrv} and the outside reservoir wall temperature at the bottom T_{wrb} can be written as follows:

$$T_{wrv} = \frac{h_{avr}T_a + \dfrac{k_{res}}{S}T_f}{h_{avr} + \dfrac{k_{res}}{S}} \tag{20}$$

$$T_{wrb} = \frac{h_{abr}T_a + \dfrac{k_{res}}{S}T_f}{h_{abr} + \dfrac{k_{res}}{S}} \tag{21}$$

Therefore, fluid temperature in the reservoir with one line flow in and one line flow out can be obtained from:

$$\frac{dT_{fr}}{dt} = \frac{W + m_1C_{pf}T_{\text{inlet}} - m_2C_{pf} - Q_{1R} - Q_{2R} - Q_{3R}}{M_fC_{pf}} \tag{22}$$

In (22), when the temperature in the reservoir is higher than that recommended, it would be easy to involve the effect of a heat exchanger in the above equation just by adding a subtraction term of the amount of heat being dissipated through the heat exchanger during the operation of the system. Also, in (22) the fluid temperature and the reservoir wall temperature are assumed to be uniform. However, in reality the reservoir fluid and the wall temperatures are different from one point to another, but these temperature differences are relatively small.

CALCULATION OF POWER LOSS IN HYDRAULIC SYSTEM

When fluid is pumped through the hydraulic power system a certain amount of the energy in the fluid is lost by friction and then as heat. The losses occur in pipes, fittings and valves. The power losses in pipes are calculated from a given length of the pipe while the calculation of losses in fittings, first must be converted to losses through an equivalent length of a straight pipe using various experimental friction factors. To arrive at the total power loss for a circuit, the losses in pipes and fittings 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:

$$H_f = f\left(\frac{L}{D_1}\right)\left(\frac{v^2}{2g}\right) \tag{23}$$

The head losses in pipe fittings, valves and bends can be computed from:

$$H_{ff} = K\left(\frac{v^2}{2g}\right) \tag{24}$$

After that, the total head losses in pipes and pipe fittings, valves are converted into pressure drop. Finally, the total power loss directly increasing the temperature of the fluid is given by:

$$W = Q\,\Delta P \tag{25}$$

where 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.*[15] under atmospheric pressure it is expressed as:

$$\nu = \mathrm{e}^{10\mathrm{e}^{10[A - B\log(T+273)]}} + 0.7$$

where constant 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 Reference 16.

EXPERIMENTAL PROCEDURE

In order to test and verify the mathematical model experimentally, a complete open hydraulic mixer system has been designed, instrumented and commissioned. A fixed displacement external gear pump and a fixed displacement external hydraulic motor with external drain have been used in this system as shown in *Figure 3*. This system is a combination of a compact sized reservoir, control system, cooler arrangement and associated equipment forming the heart of the hydraulic circuit.

In this system the oil is fed from the pump to the circuit into the drive motor. At this point a small amount of the oil flow is used to drive the cooling fan motor. This hydraulic system 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 the pipework. When the drive 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 the reservoir to be used.

The pipe wall temperatures were measured by four fine wire thermocouples (Type K) brazed onto a shaped metal disc of 7 mm diameter supplied with self adhesive tape to attach to the surface of the hoses, with integral digital output device.

These thermocouples were firmly stuck onto the surface of the hoses as shown in *Figure 3*. Fluid temperature in the reservoir was measured with a dial mounted gauge driven by temperature sensitive fluids that are metal encased and immersed in the hydraulic fluid. The fluid level in

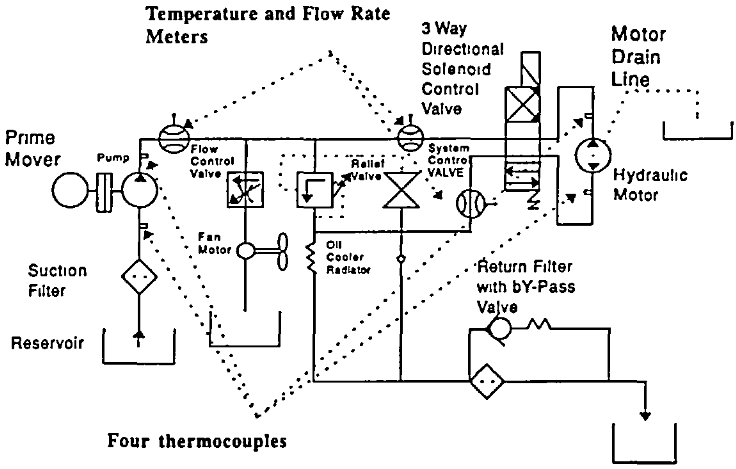

Figure 3 An open hydraulic drive mixer test rig and position of the measurements equipment

the reservoir is monitored by a fluid level gauge which was mounted in the reservoir to indicate the level and amount of fluid necessary to fill the hydraulic system. The temperatures of fluid flowing through the hoses were measured by temperature meters which were incorporated in the lines to record the change of the fluid temperature during operation of the system.

COMPUTER SIMULATION PROGRAM

A number of general purpose simulation programs and specialist application packages exist today[17–21] which handle a wide variety of problems. However, their generality leads to some inefficiency and difficulty in using them. The user must develop the model for each type of component to be used in the circuit. For instance CSMP (Continuous Simulation Modelling Programme)[19] provides for the inclusion of user written subroutines. These 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 program in any order. However, the simulation program used in this research has the advantage of having been developed specifically for simulating hydraulic systems. The main advantages of the developed package are:

— it is friendly and highly interactive;
— it provides a facility which allows addition of new components without any modification in the program;
– it provides a computational tool to be used by a user who has very little knowledge of programming and the computer system.

This package consists of the following parts:

(1) the first part involves the characteristics of pumps and hydraulic actuators including torque, speed, power and efficiency;

(2) the steady state flow analysis part predicts the steady state flows and pressures in the system, determines the flow rate through the circuit and whether the flow is laminar, transitional or turbulent in order to apply the appropriate resistance factors. In addition, this part can

calculate the equivalent length of the system and then determine either the total pressure drop or fluid horsepower loss resulting from the friction of the total equivalent length. Other factors such as the velocity at the suction line becomes evident, and the pipe can be resized as required to limit the velocities at critical places in the circuit;

(3) power losses: this calculates the power losses in the system including the power loss in the pump, hydraulic motor, fittings and pipes;

(4) the hydraulic transient thermal analysis: this predicts the fluid temperatures in the system loop, the reservoir and pipes and also the reservoir wall temperatures under unsteady state conditions. It also predicts the effects of system heat generation and dissipation of temperature on the performance of an industrial hydraulic system.

SIMULATION DIAGRAM

The simulation program developed for the dynamic and thermal simulation of open hydraulic systems is shown in *Figure 4*. The block HP represents the subroutine for the pump which determines the flowrate and input torque to the pump and the efficiencies of the pump. The hydraulic motor is simulated by model HM which gives the torque at the output shaft of the motor, 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 dynamic performance of the system has been modelled by block DM through subroutines HC (hydraulic circuit model). These subroutines predict the pressures in the system the velocity of the flow in each different diameter of the pipes, whether the flow is laminar or transitional or turbulent in order to apply appropriate

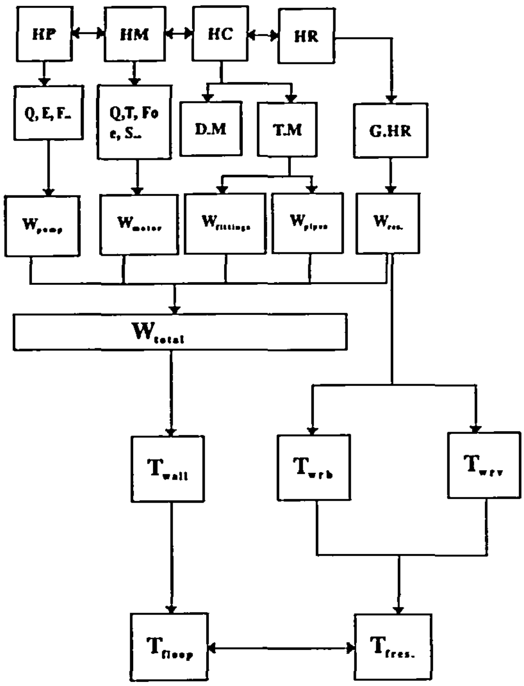

Figure 4 Block diagram for the dynamic and thermal simulation of the open hydraulic systems using the developed programme

Table 1 Results from the developed software

Time	$T_1(i)$	$H_1(i)$	$H_2(i)$	$DT_2(i+1)$	$T_2(i+1)$	$h_a(i)$	$DT_fR(i+1)$
1	295.87	13.58	11.46	0.000208	295.01	3.804	0.003860
2	296.19	18.29	11.54	0.000661	295.07	3.810	0.003701
3	296.67	24.96	11.90	0.001278	295.21	3 836	0.003627
4	297.35	33.38	12.80	0.002013	295.48	3.900	0.003524
5	298.25	43.16	14.57	0.002798	295.93	4.015	0.003390
6	299.38	53.87	17.54	0.003555	296.59	4.185	0.003221
7	300.76	65.02	22.02	0.004209	297.47	4.402	0.003020
8	302.36	76.20	28.22	0.004696	298.57	4.651	0.002789
9	304.16	87.07	36.23	0.004975	299.87	4.915	0.002531
10	306.11	97.37	45.94	0.005033	301.30	5.178	0.002252
.	.	.	.	.	.	.	.
.	.	.	.	.	.	.	.

Time	T_{wv}	Q_1	T_{wp}	Q_2	Q_3	$DT_f(i+1)$	$T_f(i+1)$
1	295.00	1.07	295.30	0.17	0.19	0.01176	297.71
2	297.70	3.12	297.70	0.42	0.31	0.01221	298.74
3	298.74	4.01	298.74	0.54	0.42	0.01254	300.12
4	300.12	5.24	300.12	0.70	0.58	0.01269	301.84
5	301.83	6.86	301.83	0.92	0.79	0.01265	303.86
6	303.86	8.88	303.86	1.19	1.06	0.01243	306.16
7	306.15	11.29	306.15	1.51	1.37	0.01205	308.69
8	308.68	14.07	308.68	1.89	1.74	0.01150	311.39
9	311.38	17.16	311.38	2.30	2.15	0.01081	314.20
10	314.19	20.49	314.19	2.75	2.59	0.00998	317.05
.	.	.	.	.	.	.	.
.	.	.	.	.	.	.	.

Results

Comp. no.	Comp. name	Pressure drop [bar]	Head loss [m]	Heat G
1	Relief valve	3.00	34.48	12.96
2	Filter	0.05	0.57	0.22
3	Check valve	0.5	5.75	2.16
4	Flow valve	0.6	6.90	2.59
.	.	.	.	.
.	.	.	.	.

resistance factor. However, the power losses in all the components are added together 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 models W_{pump}, W_{motor}, and $W_{fittings}$, respectively. The outside wall temperature of the vertical walls and at the bottom surface of the reservoir are represented in blocks T_{wrv}, T_{wrb} respectively. The model T_{wall} represents the outside wall temperature of the pipes. The temperature of fluid in the reservoir and the loop are represented in models $T_{fres.}$ and T_{floop} *Table 1* shows a sample of the results from the simulation program.

THEORETICAL RESULTS

The developed equations were solved by using numerical integration techniques for known initial parameters and have been widely used in the simulation. The detailed discussion about the numerical integration techniques used in this analysis have been documented[22]. Theoretical results have shown that the geometry of the reservoir plays a significant role in determining the temperature distribution in the system loop. The fluid height in the reservoir determines the vertical side areas of the reservoir as indicated in (17), the amount of the heat flow dissipated through the reservoir side walls Q_{1R} is affected by these areas of the reservoir. This heat dissipation through the side walls of the reservoir will affect the fluid temperature inside the reservoir as expressed in (22). Moreover, the fluid temperatures 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. *Figure 5* illustrates the effects of the fluid height in the reservoir on the fluid temperature in the system loop. This Figure suggests that any change of the fluid height in the reservoir will affect the fluid temperature. However, under steady conditions the rise in temperature shows a non-linear relationship with the height of the reservoir.

The area of the base of the reservoir has its influence on the fluid temperature as shown in simulation results in *Figure 6* which shows the effect of the change in area of the base of the reservoir, due to the change in the length and width, on the fluid temperature in the system loop which indicates that the lower the value of the area, the higher is the value of the temperature. In addition, this Figure indicates that the change in temperature for the same change in the area is different if the change in area is achieved by changing the length or width.

Moreover, the theoretical results have shown that the effect of the thickness and type of material of the reservoir have very little influence on the fluid temperature distribution in the system loop as shown in *Figures 7* and *8* respectively.

The effects of the rate of flow between the loop and the reservoir on the loop temperature are illustrated in the simulation results in *Figure 9*. As can be seen, when no fluid returns to the reservoir, the highest temperature of the fluid occurs in the system loop during the load application. As this flow rate rises, the flow energy transferred from the system loop to the

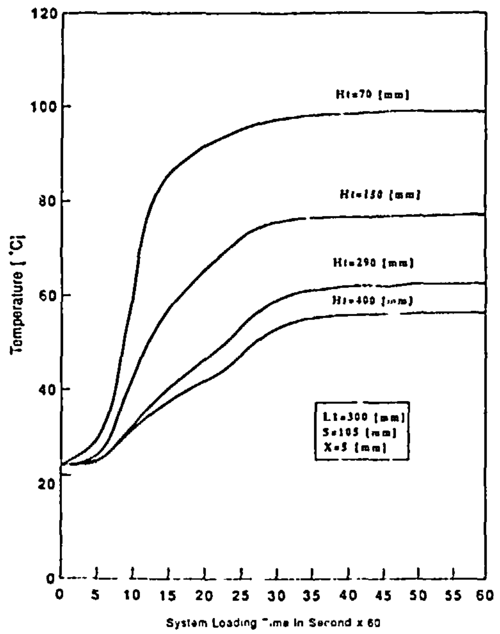

Figure 5 Effects of fluid height in the reservoir on fluid temperature distribution in the open hydraulic drive mixer system

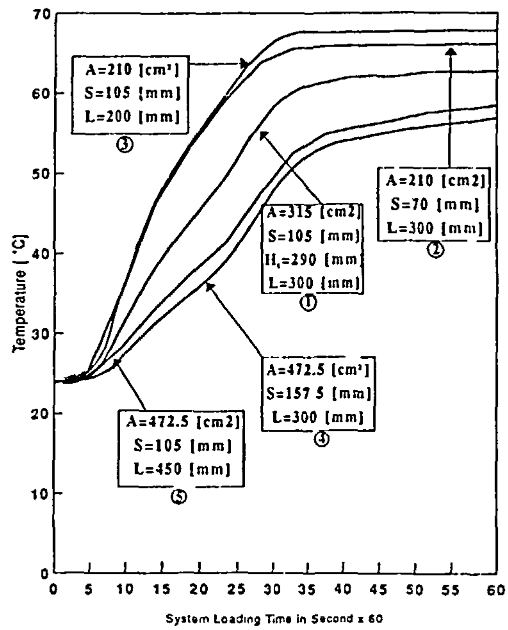

Figure 6 Effects of the change in area of the base of the reservoir on the fluid temperature in the open hydraulic drive mixer system

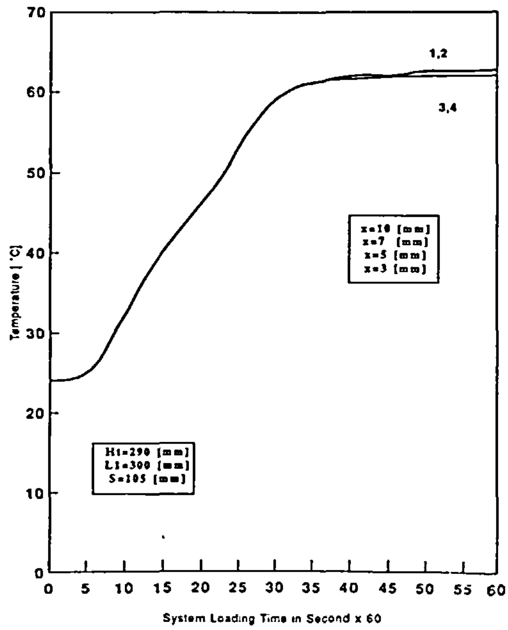

Figure 7 Effects of thickness of the wall of the reservoir on the fluid temperature distribution in the open hydraulic drive mixer system

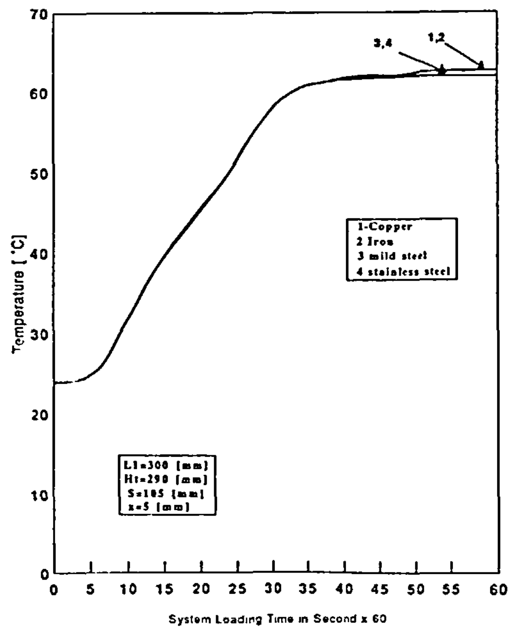

Figure 8 Effects on material of the reservoir on fluid temperature distribution in the open hydraulic drive mixer system

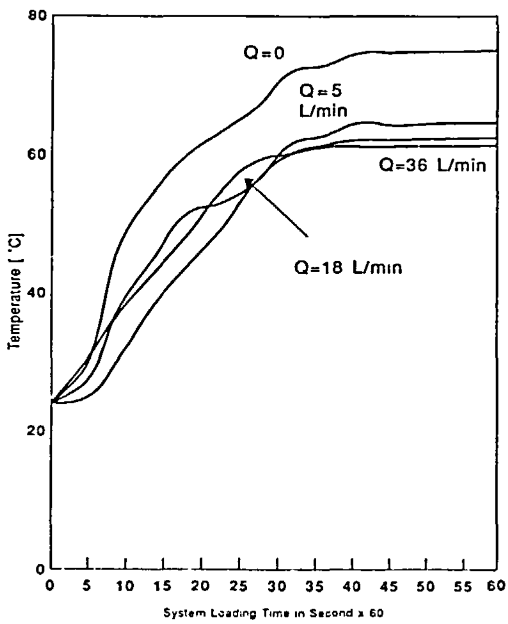

Figure 9 Simulated results of the effects of exchange flow rate between loop and reservoir on loop fluid temperature in the open hydraulic drive mixer system

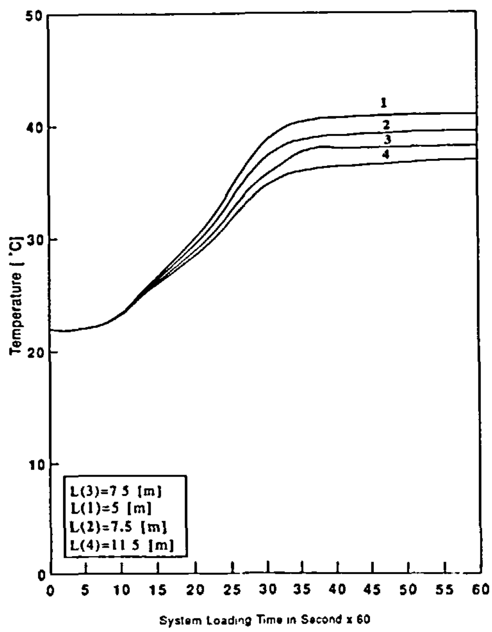

Figure 10 Effects of pipelength on the outside hose wall temperature distribution in the open hydraulic drive mixer system

reservoir increases and so the loop temperature will decrease. Meanwhile, the proportion of heat dissipated through the loop hoses is expressed by Q_1 in (16), the simulation results of the heat dissipated through the pipes are shown in *Figure 10* where 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 *Figure 10.*

Simulation results have shown also that the fluid and pipe wall temperatures of the system are affected by a number of parameters such as length, material and diameter of the pipe. *Figure 11* shows the effect of the pipe length on the outside hose wall temperature distribution in the open hydraulic drive mixer system. This Figure suggests that when the total pipelength is changed the outside wall temperature of the hose is also changed. Similarly, when the inside diameter of the hose is changed the fluid temperature would be affected as shown in *Figure 12*. However, the change in the pipe inside diameter is seen to have more influence on the outside pipe wall temperature as shown in *Figure 13.* On the other hand, the size of the pipe outside diameter has more effects on the pipe wall temperature as shown in *Figure 14*. Furthermore, it is noted that the temperature distribution depends on the type of pipe wall material since the values of specific heat, thermal conductivity and density are related to the pipe wall material. *Figures 15, 16* and *17* show the effect of specific heat, density and thermal conductivity on the outside wall temperature of the pipe respectively. *Figure 15* suggests that the increase in the specific heat does not significantly affect the temperature distribution. The value of the specific heat C_p, of mild steel $C_p = 0.489$ [kJ/kg·°C]; copper $C_p = 0.3831$ [kJ/kg·°C]; rubber $C_p = 2.010$ [kJ/kg·°C]. *Figure 16* shows that for higher values of thermal conductivity k, the temperature should increase. The value of the thermal conductivities k, of mild steel $k = 60$ [w/m·K]; copper $k = 386$ [w/m·K]; rubber $k = 0.013$ [w/m·K]. *Figure 17* demonstrates the results of density ρ, which shows a similar trend to that in *Figure 15*. The value of density of mild steel = 7833

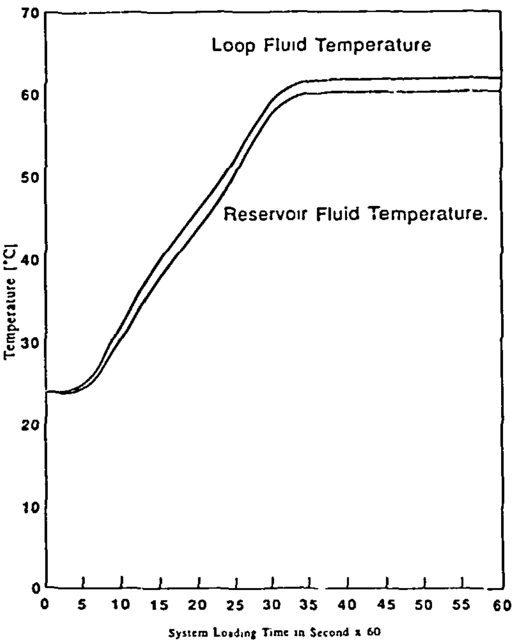

Figure 11 Comparison of simulated results of loop and reservoir fluid temperature in the open hydraulic drive mixer system

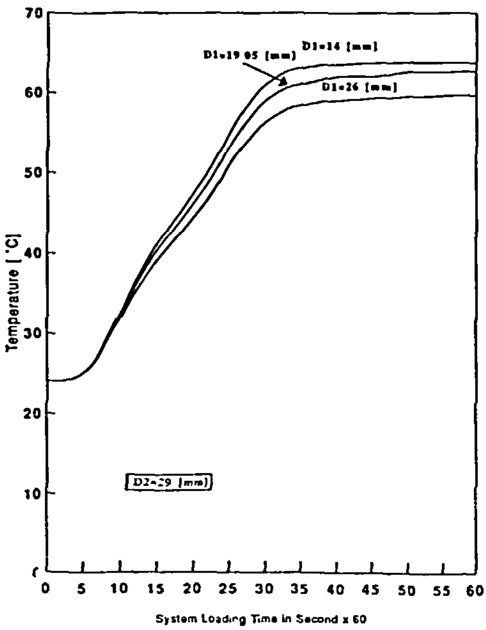

Figure 12 Effects of internal diameter of hose on the fluid temperature distribution in the open hydraulic drive mixer system

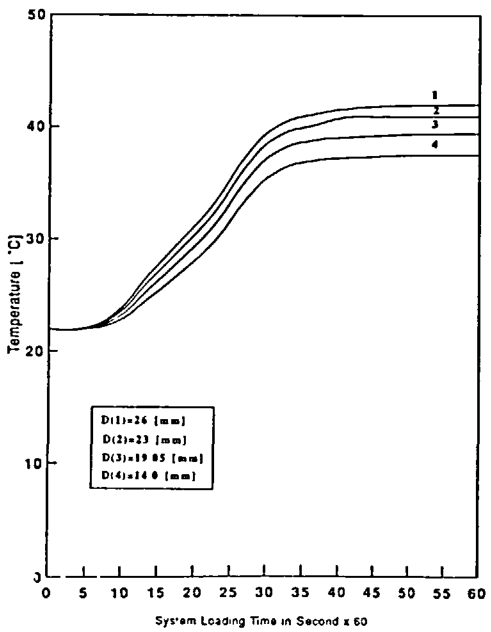

Figure 13 Effects of internal diameter of hose on the outside hose wall temperature distribution in the open hydraulic drive mixer system

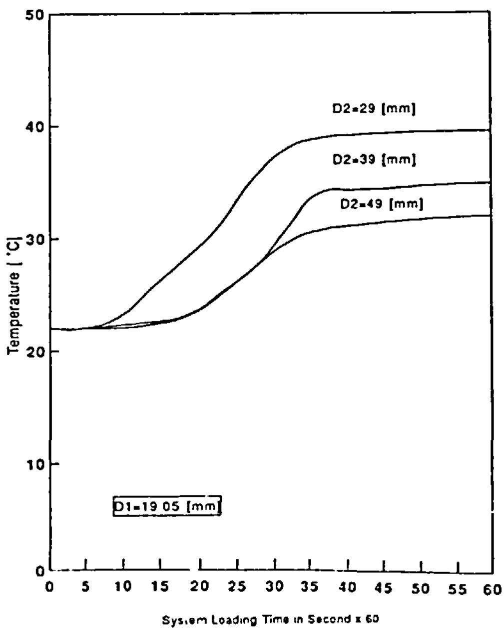

Figure 14 Effects of external hose diameter on the outside hose wall temperature distribution in the open hydraulic drive mixer system

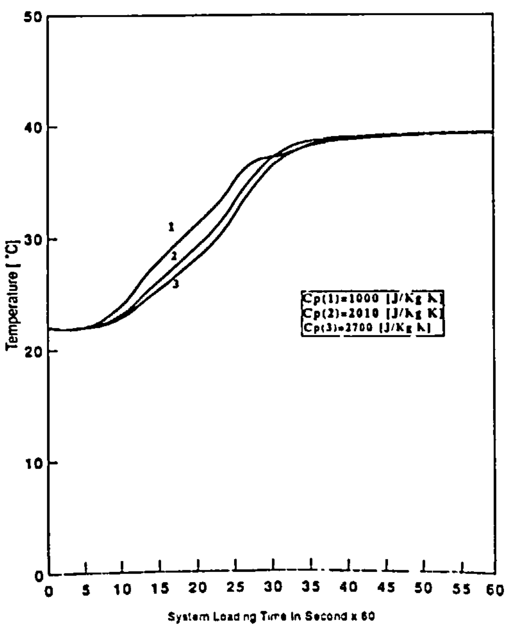

Figure 15 Effects of specific heat of pipe material on the outside hose wall temperature distribution in the open hydraulic drive mixer system

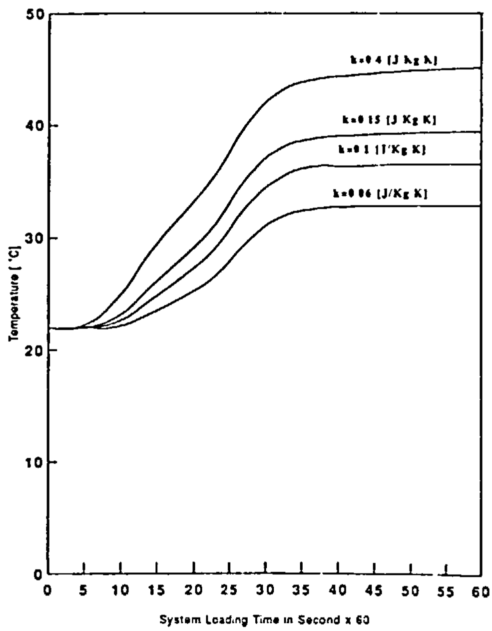

Figure 16 Effects of conductivity of pipe material on the outside hose wall temperature distribution in the open hydraulic drive mixer system

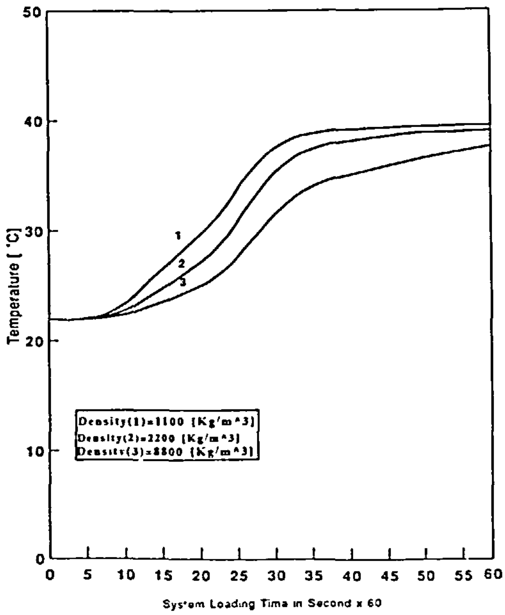

Figure 17 ·Effect of density of pipe material in the outside hose wall temperature distribution in the open hydraulic drive mixer system

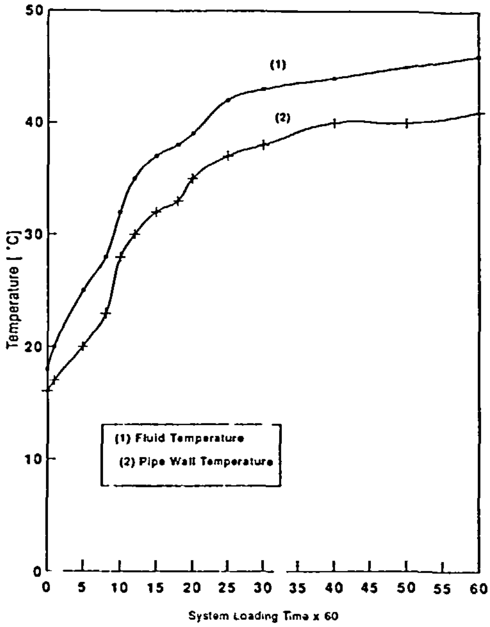

Figure 18 Comparison between fluid and pipe wall temperature in the open hydraulic drive mixer system

[kg/m^3]; rubber = 1100 [kg/m^3]. The values of thermal conductivities, specific heat and density were given according to Reference 23.

COMPARATIVE PERFORMANCE OF THE HYDRAULIC SYSTEM

Figure 18 shows the comparison between the fluid and hose wall temperature obtained experimentally. This Figure shows considerable differences 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 as shown in *Figures 19* and *20*. *Figure 19* 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. The theory predicts slightly higher temperature than the experiment. *Figure 20* shows the comparison between the experimental and theoretical results of the hose wall temperature. This Figure also shows similar trends as that in *Figure 19*.

Comparison has shown a significant deference between the results predicted according to Reference 7 and the present mathematical model. According to Bown and Young[7], in which the effect of heat transferred by conduction along the pipe and reservoir in the hydraulic system and by radiation from the surfaces of the reservoir were ignored, a considerable deference in the estimate of the temperature distribution in the hydraulic systems is suggested. Typical examples are given in *Figures 21* and *22*. *Figure 21* shows the comparison of the fluid temperature predicted according to the present analysis and the analysis of Bown and Young[7] 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 underestimates the temperature rise. *Figure 22* shows the same trend as that in *Figure 21* in respect of the wall

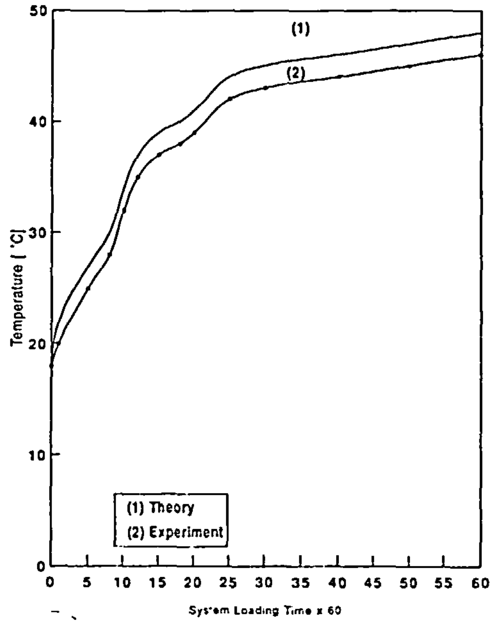

Figure 19 Comparison between experimental and theoretical results of fluid temperature in the pressure line in the open hydraulic drive mixer system

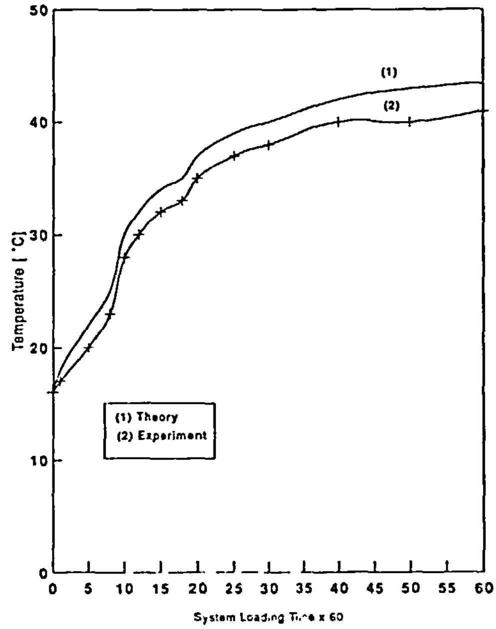

Figure 20 Comparison between experimental and theoretical results of hose wall temperature in the pressure line in the open hydraulic drive mixer system

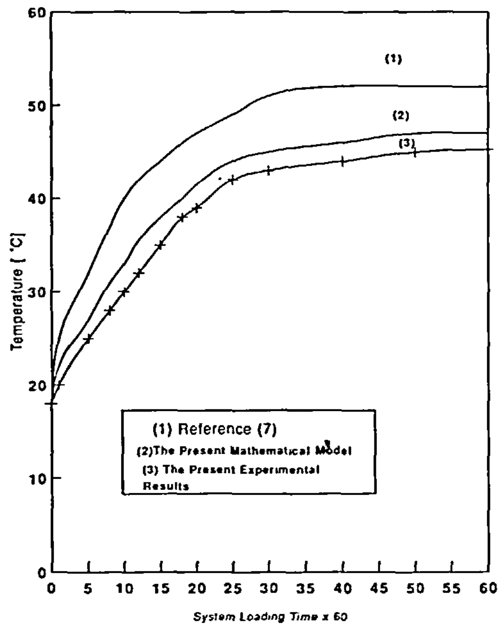

Figure 21 Comparison of the fluid temperature between the present analysis and the analysis in Reference 7 in the open hydraulic drive mixer system

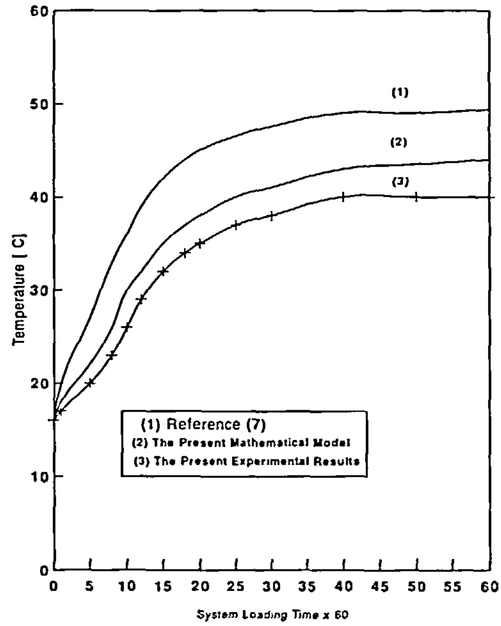

Figure 22 Comparison of the outside wall temperature of the hose between the present analysis and the analysis in Reference 7

temperature. Moreover, *Figures 21* and *22* show a slight difference between the present predicted and experimental results. A possible reason is the gradual increase in the atmospheric temperature surrounding to the test rig over the duration of the tests of the load application or due to pressure drop through the fittings and the errors in the temperature measurements.

CONCLUSION

A mathematical model for the temperature distribution under unsteady state in hydraulic systems has been developed. The thermodynamics process and heat transferred by conduction, convection and radiation have been taken into account. This consideration has given a closer agreement between the experimental and theoretical results. The present analysis gives improved predictions of the temperature distribution in hydraulic system than that reported in previous studies.

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