



# OPEN CIRCUIT HEAT EXCHANGER DYNAMICS DURING FLOW REDUCTION TRANSIENT IN THEIR SECONDARY LOOPS

Dr. Akram. W. Ezzat  
Mechanical Engineering Department  
University of Baghdad  
Email: [akramwahbi@yahoo.ie](mailto:akramwahbi@yahoo.ie)

## ABSTRACT

An experimental and theoretical analysis was conducted for simulation of open circuit cross flow heat exchanger dynamics during flow reduction transient in their secondary loops. Finite difference mathematical model was prepared to cover the heat transfer mechanism between the hot water in the primary circuit and the cold water in the secondary circuit during transient course. This model takes under consideration the effect of water heat up in the secondary circuit due to step reduction of its flow on the physical and thermal properties linked to the parameters that are used for calculation of heat transfer coefficients on both sides of their tubes. Computer program was prepared for calculation purposes which cover all the variables that affect such type of transient mechanisms. The effect of the power density in the primary circuit and the water flow reduction percentage on the average temperature build up of the water in the primary circuit was investigated. The elapsed time required for the primary circuit average temperature to reach a steady state value was also calculated. These calculations were supported with experimental measurements conducted on a standard cross flow heat exchanger apparatus. The experimental results were compared with the theoretical results for certain power density value at different flow reduction percentages which show a reliable agreement. This relative agreement was necessary to consider the mathematical model with certain assurance for calculating transient parameters for higher power densities that are out of apparatus ranges. The results proved that water average temperature build up in the primary circuit has sharp tendency when the percentage of flow reduction in the secondary circuit reach 25% of its nominal values.

## الخلاصة

تم إجراء تحليل عملي ونظري لمحاكاة ديناميكية المبادل الحراري من نوع الجريان المتقاطع أثناء الحالات غير المستقرة لنقصان جريان المبرد في دوراتها الثانوية. حيث تم إعداد موديل رياضي مبني على الفروقات المحددة لتغطية ميكانيكية عملية التبادل الحراري ما بين الماء الحار في الدورة الأساسية والماء البارد في الدورة الثانوية. أخذ الموديل بنظر الاعتبار تأثير زيادة تسخين الماء في الدورة الثانوية بسبب النقصان الدرجي لجريانه على الخصائص الفيزيائية والحرارية ذات العلاقة بالحدود المستخدمة في احتساب معاملات التبادل الحراري على جانبي أنابيب المبادل الحراري. تم إعداد برنامج حسابي على الحاسوب الذي يغطي جميع العوامل المؤثرة لمثل هكذا نوع من ميكانيكية الحالات غير المستقرة. تم بحث تأثير كثافة الطاقة في الدورة الأساسية والنسبة المئوية لنقصان معدل جريان المبرد في الدورة الثانوية على زيادة معدل درجة حرارة الماء في الدورة الأساسية. تم كذلك حساب الزمن المستغرق لوصول معدل درجة حرارة الماء في الدورة الأساسية الى قيم مستقرة. هذه الحسابات تم تدعيمها بقياسات عملية تم اجرائها على جهاز مختبري لمبادل حراري من نوع الجريان المتقاطع. تم إجراء مقارنة بين الحسابات العملية والنظرية لكثافة طاقة معينة لنسب انخفاض في معدلات جريان مختلفة وكانت نتائج المقارنة مقبولة. هذا التطابق النسبي كان ضروريا لاعتماد الموديل والبرنامج الرياضي في احتساب النتائج لقيم كثافات طاقة أعلى من حدود تحمل الجهاز. النتائج برهنت أن زيادة معدل درجة حرارة الماء في الدورة الأساسية تأخذ ميلا حادا عند وصول قيم النسب المئوية لانخفاض معدل الجريان الى 25% من قيمها الطبيعية.

**Keywords: open circuit heat exchanger, cross flow, flow reduction, transient**

## INTRODUCTION

In process industries, heat is transferred by radiation, by the mixing of hot and cold fluids or most frequently, by conduction through the walls of heat exchanger.

For our purposes, the characteristic feature of these devices, heat exchangers, is that the capacities on one or both sides are distributed along the length of the exchanger.

Because the exchanger parameters are distributed and interacting the exact dynamic equations for an ordinary exchanger are quite complex, and lengthy calculations are required just to get the open-loop frequency response. In general, heat exchangers are fairly easy to control, and except where very close control is needed, simplified methods of dynamic analysis give answers that are accurate enough for the practicing engineer.

The approach taken here is to present exact transferred functions for a few of simpler cases in order to show the parameters that determine the lags and to explain the resonance effects sometimes found with distributed systems.

### Types of Heat Exchangers

One type of heat exchanger is that of a double pipe arrangement with either counter or parallel flow and with either the hot or cold fluid occupying the annular space and the other fluid occupying the inside of the inner pipe type of them. Heat exchangers are widely used in the chemical process industries is that of the shell and tube arrangement.

One fluid flows in the inside of the tubes, while the other fluid is forced through the shell and over the outside of the tubes to insure that the shell side fluid will flow

across the tubes and thus induce higher heat transfer, baffles are placed in the shell.

Cross flow heat exchangers are commonly used in air or gas heating and cooling applications where a gas may be forced across a tube bundle, while another fluid is used inside the tubes for heating or cooling purposes. In the relevant study the cross heat exchanger used water in both their primary and secondary circuits.

### Loss of Heat Sink

Loss of heat sink occurs due to a decrease in heat removal by the secondary cooling system due to pump failure, blockage in piping or heat exchanger, valve failure or a break in the main pipe of the secondary cooling system.

### Sequence of Events and System Operation

In case of pump failure due to loss of electrical power or pump over heat, the secondary pump failure relay will open without time delay and the heat source device is shut down automatically by proper protection system.

If this relay failed closed or in case of other causes to loss of heat sink, the primary coolant temperature would continue to rise, then the primary coolant temperature protection channel will shutdown the heat source device when its corresponding trip value is reached.

If the safety system related to the heat source device failed to shutdown the heat source, the operator can shutdown it manually when he realize that there is an abnormal condition in the system, i.e insufficient cooling, by monitoring the different signals and indications in the control room.



### System Performance

During the transient of loss of heat sink the water temperature in the primary circuit will rise up until heat source device is shutdown, then these temperature starts to decrease after heat source device shutdown. The maximum increase in these temperatures above the normal operational values depends on the time lag between loss of reduction of flow in the secondary circuit and the heat source device shutdown.

The attempt of this study is to simulate the response of the primary circuit temperature build up according to the reduction or completely loss of secondary circuit flow during the failure of the safety systems to mitigate such type of transients. Such type of transients plays an important role in the evaluation of heat generation system safety and integrity specially those related to nuclear reactors in which self control is a dominant factor due to the effect of negative temperature factor on the reactivity of the reactor core.

**Ezzat & Taki (1987)** investigated in the final safety analysis report, FSAR related to the Iraqi research reactor IRT 5000 on 1987 loss of flow incidents on both primary and secondary circuits and their effect on the reactor safety

**Gvozenac (1992)** applied Laplace Transform Method to solve the transient response of the counter flow heat exchanger with finite wall capacitance problem. The mathematical model is based on three local energy balance equations which are solved assuming that only the fluid 1 inlet condition is perturbed (step change). As any counter flow problem could be reduced to an adequate integral equation, collocation method is used for solving such equation in the presented case.

**Housiadas (2000)** investigated the course of loss-of-flow transients in pool-type research

reactors, with scram disabled. The analysis is performed with a customized version of the code PARET. The focus was on determining the two-phase flow stability boundaries as function of initial reactor conditions, recognizing that flow instability is the basic mechanism responsible for core damage in such type of transients. A useful chart was provided, which describes the stability region in terms of initial reactor power, initial pool temperature, peaking factor, and flow-decay time constant.

**Abdelfatah and Galanis (2000)** Calculated the transient temperature field in a parallel flow heat exchanger numerically assuming fully developed hydrodynamic conditions. This approach uses fewer assumptions than published analytical studies. It shows the influence of physical and operational characteristics on experimentally defined parameters that describe the transient response of heat exchangers.

**Srihari et al. (2004)** focused on the effects of flow mal distribution and conventional heat exchanger parameters on the temperature transients of both U-type and Z-type configurations. It is found that the effect of flow mal distribution is significant and it deteriorates the thermal performance as well as the characteristic features of the dynamic response of the heat exchanger. In contrast to the previous studies, here the axial dispersion describes the in channel back mixing alone, not mal distribution, which is physically more appropriate. Present method is an efficient and consistent way of describing mal distribution and back mixing effects on the transient response of plate heat exchangers using an analytical method without performing intensive computation by complete numerical simulation.

**Manish et al. (2005)** Presented results of temperature response to step and ramp

change in flow rate of hot and cold fluids, and step, ramp, exponential and sinusoidal variation in hot fluid inlet temperature transient temperature response of cross flow heat exchangers having finite wall capacitance with both fluids unmixed is This investigation is conducted numerically for perturbations provided in both temperature and flow. Results are presented for step and ramp change in flow rate of hot and cold fluids, and step, ramp, exponential and sinusoidal variation in hot fluid inlet temperature.

**Thirumarimurugan et al. (2008)** presented the performance evaluation of cross flow plate fin heat exchanger with several different Gas-Liquid systems. Experimental results such as exchanger effectiveness, overall heat transfer coefficients were calculated for the flow systems of Cross flow Heat Exchangers. A steady state model for the outlet temperature of both the cold and hot fluid and overall heat transfer coefficient of a plate-fin cross flow heat exchanger was developed and simulated using MATLAB, which was verified with the experiments conducted.

## THEORITICAL ANALYSIS

### Determination of Heat Convection Coefficient between Water Flows and Tube Surfaces

Heat transfer coefficient for both inner surface and surface of tube bundles could be calculated using the following procedure. This procedure could be conducted for the inner surface and then repeated for the outer surface using the appropriate formula.

The formulas adopted for calculating heat transfer coefficient in our case are recommended by the supplier company for

the type of the heat exchanger selected for experimental work, **PROFIT**.

### 1. Laminar Flow

As far as the laminar flow concerned, there are two zones where the flow is quite different with respect to the other one.

**a.** Entrance length whose length is given by the following:

$$L_i = (Re.Pr.d)/20$$

Where we can use:

$$Nu = 1.86.[Re.Pr.(d/L)]^{0.33} . (\mu/\mu_w)^{0.14}$$

$$\text{For } Re.Pr.(d/L) > 10, L/d > 2, \\ 100 < Re < 2100, 0.48 < Pr < 16700$$

$\mu$ : The dynamic viscosity at average temperature.

$\mu_w$ : The dynamic viscosity at wall temperature.

**b.** The zone where the steady state is generated completely .Here as the convective flow is constant. We have:

$$Nu = 3.66 \quad \text{for constant wall temperature}$$

$$Nu = 4.53 \quad \text{for constant heat flux.}$$

### 2. Turbulent Flow

**a.** Entrance length whose length is given always by the previous formula where you can apply:

$$Nu = 0.036 Re^{0.8} . Pr^{0.33} . (d/L)^{0.125}$$

With:



$10 < (L/d) < 400$  ,  $Re > 10000$  ,  $0.7 < Pr < 16700$  .

**b.** The zone where steady state is generated completely. Here you can obtain the convective coefficient by:

$$Nu = 0.023 Re^{0.8} . Pr^{0.4} [1+(d/L)^{0.7}]$$

$$H_{av} = [(h_m \cdot L_m) + (h_g + L_g) / (L_m + L_G) ] .$$

For  $10000 < Re < 120000$ ,  $0.7 < pr < 120$ ,  $(L/d) > 60$

Where

$h_m$  = mouth zone coefficient convection, (W/m. C).

$h_g$  = Steady state completely generated zone coefficient convection .

$L_m$  = Mouth zone length.

$L_G$  = Steady state completely generated zone length.

The thermal and physical properties of the water flowing in both primary and secondary circuits are obtained by interpolation among their values listed in specified tables.

Interpolation is conducted at mean water temperature between input and output values for each of primary and secondary circuits.

$$T_m = (T_{inlet} + T_{outlet}) / 2$$

**Assumption Studied Case**

We assume that the total mass of water in the primary circuit is (M). Then balance between the heat added to this amount of water by electrical heater and the heat rejected to the secondary circuit could be

represented by the following equation:

$$Q_i - Q_c = M_{pr} * C_{pr} * .(dT_{pa}/dt) \tag{1}$$

$$Q_c = U_o * A_o * (T_{pa} - T_{sa}) \tag{2}$$

$$T_{pa} = (T_{pi} + T_{po}) / 2 \tag{3}$$

$$T_{sa} = (T_{si} + T_{so}) / 2 \tag{4}$$

Where:

$A_o$ : Heat transfer area between primary side and secondary side of the heat exchanger base on outer surface.

$U_o$ : Overall heat transfer coefficient of the heat exchanger based on the outer surface.

$$U_o = 1 / [(1/h_i).(r_o/r_i) + [r_o.ln(r_o/r_i)] / K_c + (1/h_o)] \tag{5}$$

Where:

$h_i$ : Heat transfer coefficient between the tube wall and primary circuit water inside the tube.

$h_o$ : heat transfer coefficient between the tube wall and the secondary circuit water outside the tube.

$r_i$  &  $r_o$ : The inner and outer radius of the tube respectively.

$K_c$ : Thermal conductivity of the copper tube.

According to the range of the experimental work the calculations in the secondary side is based on the laminar flow. Nusselt No. is calculated using following equation:

$$Nu_o = 1.86 * [Re_o * Pr_o * d_o/L]^{0.33} * (\mu_o / \mu_w)^{0.14} \tag{6}$$

$$Nu_o = 1.86 * [(\rho_o * d_o^2 / \mu_o)^{.33} * (Pr_o / L)]^{0.33} * (V_s)^{0.33} * (\mu_o / \mu_{wo})^{0.14} \tag{7}$$

$$h_o = (K_o * Nu_o) / d_o \tag{8}$$

The range of experimental work ensures turbulent flow in the primary circuit of the heat exchanger. Nusselt No. is calculated according to the following equation:

$$Nu_i = 0.023 * (Re_i)^{0.8} * (Pr_i)^{0.4} * (1 + d_i / L)^{0.7} \tag{9}$$

$$Nu_i = 0.023 * (\rho_i * d_i / \mu_i)^{0.8} * (Pr_i)^{0.4} * V_p^{0.8} * (1 + d_i / L)^{0.7} \tag{10}$$

$$h_i = (K_i * Nu_i) / d_i \tag{11}$$

Here we assume the following:

1. The secondary circuit is open loop i.e. the inlet temperature of the water in the circuit is constant.
2. The heat source (power input) is constant during transient period and this source is continuous during the transient course of secondary flow reduction..
3. The average water temperature in the secondary side of the heat exchanger is calculated based on the temperature difference across the secondary side of the heat exchanger.

$$\Delta T_s = T_{so} - T_{si} = Qi / (m_s * C_s) \tag{12}$$

Where:

ms: mass flow rate of the water in the

secondary circuit.

Cps: heat capacity of the water in the secondary circuit.

$$T_{sa} = (T_{si} + T_{so}) / 2 = T_{si} + \Delta T_s / 2$$

$$T_{sa} = T_{si} + (1/2 * Qi) / (m_s * C_s) \tag{13}$$

4. The heat exchanger selected for this calculation is the same that will be used in the test unit. The type of flow selected is parallel.

Using proper notifications for the multiple variables and after necessary substitutions the following set of equations were used to calculate the steady state and transient water average temperature in the primary circuit based on its steady state values:

$$d\theta_p(t) / dt * [C11 * V_s(t)^{1.33} + C12 * V_s(t)] = [C13 * V_s(t)^{1.33} + C14 * V_s(t)] - [C3 * \theta_p(t) * V_s(t)^{1.33}] + [C9 * V_s(t)^{1.33}] + [C10 * V_s(t)^{0.33}] \tag{14}$$

For steady state case:

$$d\theta_p(t) / dt = 0$$

Then we can use equation (14) to find primary side average temperature,  $\theta_{ps}$  as a function of steady state secondary side velocity,  $V_{ss}$ :

$$\theta_{ps} = \frac{C13 * V_{ss}^{1.33} + C14 * V_{ss} + C9 * V_{ss}^{1.33} - C10 * V_{ss}^{0.33}}{C3 * V_{ss}^{1.33}} \tag{15}$$

Formulate equation (14) to finite difference equation:



$$\theta_p(i+1) = \theta_p(i) + \Delta t * [(C13 * V_{si}^{1.33} + C14 * V_{si} - C3 * \theta_p(i) * V_{si}^{1.33} + C9 * V_{si}^{1.33} + C10 * V_{si}^{0.33}) / (C11 * V_{si}^{1.33} + C12 * V_{si})] \quad (16)$$

Where:

$$\theta_p(i) = \theta_{ps} \text{ when } i = 0$$

Suitable computer program was prepared for iterative calculations of the steady state and transient values of water average temperature in the primary and secondary circuits. The structure of the program allows easy interpolation among the values of the physical properties based on the values of water temperature in both circuits.

The empirical formulas adapted for the recent case study could be easily adapted for investigating cases that covers other type of heat exchangers.

## EXPERIMENTAL APPROACH

### System Description

The test rig consists of shell type cross flow heat exchanger. The hot water flows in shell side, primary side, while the cold water flows in tubes, secondary side. The secondary side consists of 4 tubes.

Thermal conductivity of the copper tubes,  $k=349$  W/m. C. The outer radius of the tubes,  $r_o=8$  mm, the inner radius of the tubes,  $r_i=7$ mm. The total heat exchange area  $A_h= 67380$  mm. Shell diameter=50 mm. Heat exchanger length=355 mm.

Water flow in both primary and secondary sides are ensured by 2 circulating pumps, one in each side which ensures a maximum flow rate of 300l /hr. Each side consists of water flow meter for volumetric flow measurement. Water added and drained

from both sides using proper isolation valves.

Water heating in the primary circuit is ensured by electrical heater that is adjusted by proper resistances from 0-800 Watt. Fig. (1) show the main components of the test rig.

### Test Procedure

1. The heat exchanger package is connected to the water source. Both primary and secondary circuits are filled of water. Air is relieved from the circuits using the specified air relief valves fixed on both circuits. The primary circuit inlet and outlet pipes are isolated from the water source and drain using the specified valves, while the secondary circuit side is kept connected to the water source to ensure open circuit condition during transient course.
2. The circulation pumps of both circuits are put on to circulate the water inside the circuits. Water flow rates in both circuits are fixed at their operational values using the specified valves. The volumetric flow rate of the water in each circuit is monitored using the specified volumetric flow meters fixed in each circuit.
3. The electrical heaters are turned on to heat up the water in the primary circuit to its operational steady state values. The steady state heat input is controlled using the specified knobs.
4. After the water in the primary and the secondary circuits reach their steady state values, the water flow rate in the secondary circuit is reduced to 0.83 of its steady state value and water input and output temperate to the heat exchanger on both circuits are measured versus time using stop watch till they reach their steady state values.

5. Step no. 4 is repeated by reducing the water flow rate in the secondary circuit to 0.5 & 0.33 of its steady state value. Temperature buildup in both primary & secondary circuits is measured using the stop watch.

## RESULTS AND DISCUSSION

All the results and graphs are based on constant water inlet temperature to the secondary circuit equals 25°C and constant water flow rate in the primary circuit equals 190 liter/ hr.

**Fig. (2, 3 and 4)** show the behavior of the water average temperature in the primary circuit versus time at different percentage of flow reduction in the secondary circuit and constant flow rate in the primary circuit. The graphs show reliable coincidence between the experimental and theoretical results. The negative slop of the plotted graphs within the first 60 seconds of the transient course is justified by the effect of flow turbulence in the secondary side of the heat exchanger accompanies the step decrease of the secondary flow which in turn enhance the heat transfer coefficient in that side and decrease the temperature difference for a short period.

**Fig. (5)** illustrate the relationship between the elapsed time required for the water temperature in the primary circuit to reach its steady state values after the end of the transient course versus the percentage of flow reduction in the secondary side of the heat exchanger. The graphs show reliable coincidence between the experimental and theoretical results. It is clear that the elapsed time is proportional to the percentage of flow reduction due to the increase of temperature differences between the steady state values of the water in the primary

circuit before and after the transient courses.

**Fig. (6)** show theoretical relationship between average temperature build up of the water in the primary circuit during transient course normalized to its value when the flow reduction percentage is 0% versus the percentage of flow reduction in the secondary circuit. The results show sharp slop of such increase when the percentage of flow reduction reaches 25% of its nominal values.

**Fig. (7)** show theoretical relationship between the average temperature build up in the primary circuit during transient course normalized to its value at 0% percent flow reduction versus the specific power rate added to the water in the primary circuit at flow reduction percentage in the secondary circuit equals 33%.

## CONCLUSIONS

Investigation of heat exchangers transient conditions during operation is very important in the design of safety engineering feature systems related to any power production or chemical process industries. According to this point of view the following were concluded from the research.

1. The results obtained from this study proved reliable modeling for such transients based on the coincidence between the results obtained from both experimental and theoretical analysis.
2. The analysis shows that the slop of temperature buildup in the primary circuit reaches critical values for flow reduction percentage in the secondary circuits above 25% of their nominal values.
3. The investigation and the mathematical modeling takes under consideration the effect of power adding rate to the primary circuit on the temperature build in this





circuit during flow reduction course in the secondary circuit which ensures coverage of this study for wider range of heat exchangers used in power plants.

4. From other hand this study gives an indication to the estimated mission time required for the operation of safety systems based on the elapsed time of each transient course to avoid the heat generation systems from approaching the boiling crisis accompanied by the poor heat transfer mechanisms.

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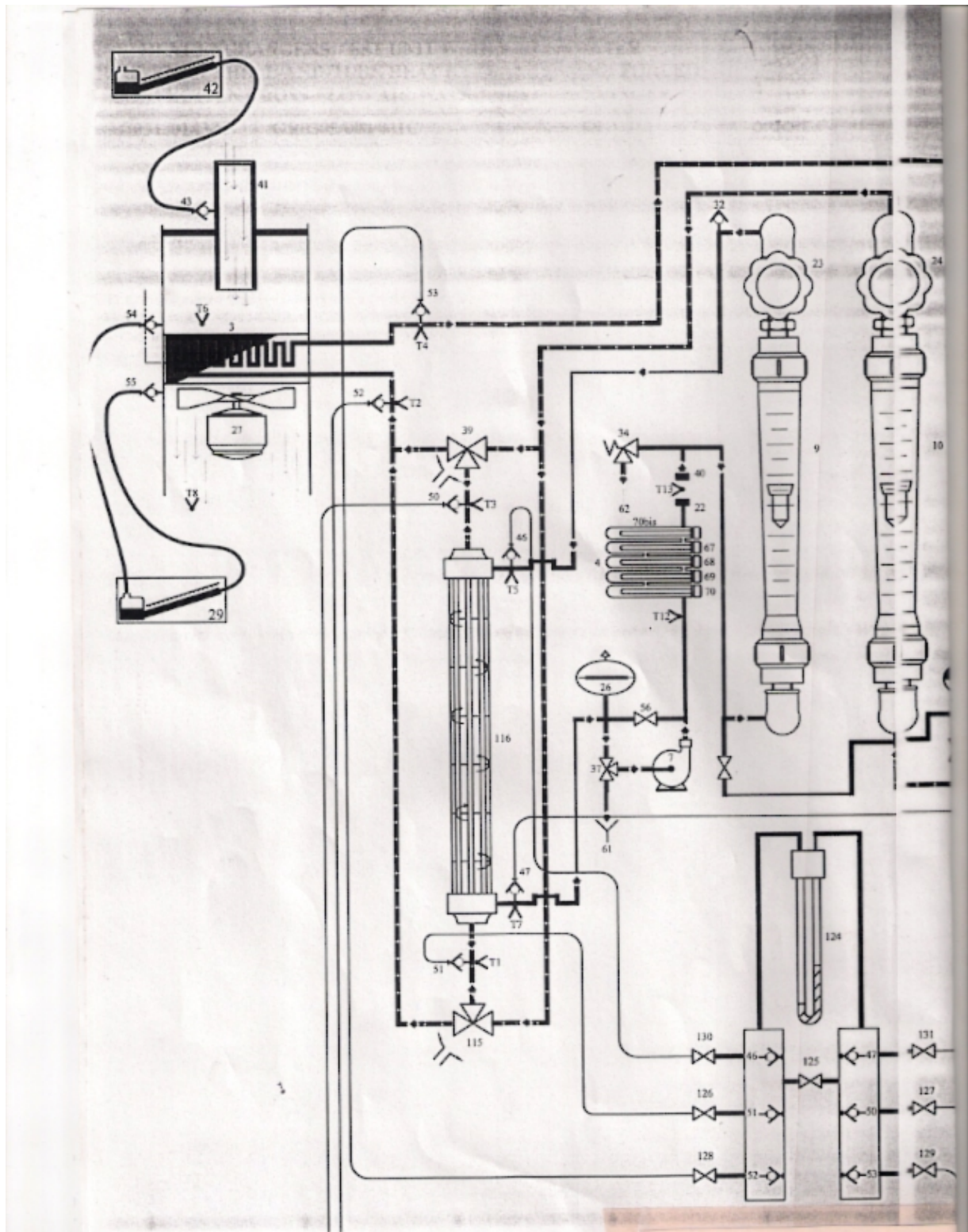
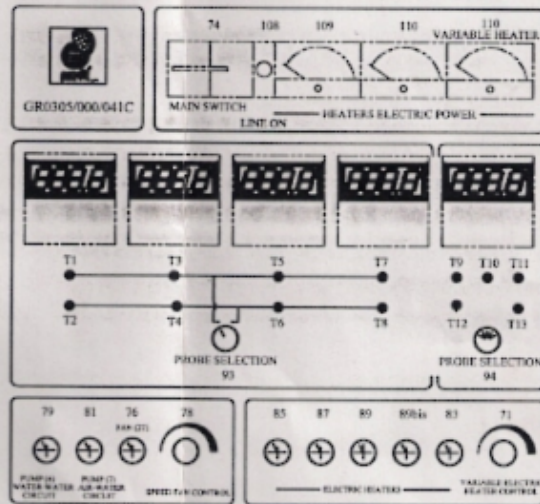
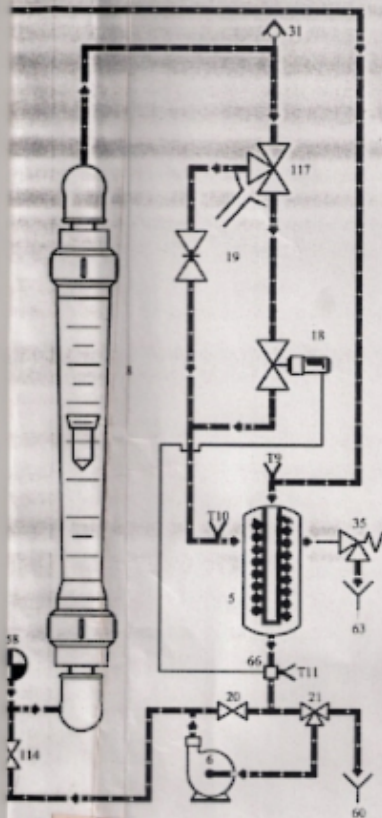


Fig. (1-A): Layout of the Test Apparatus



used for conduction the experimental work

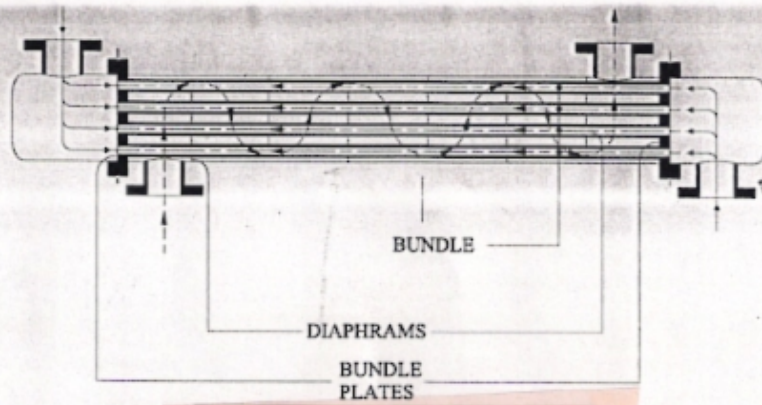


Fig. (1-B): Cross Flow Heat Exchanger

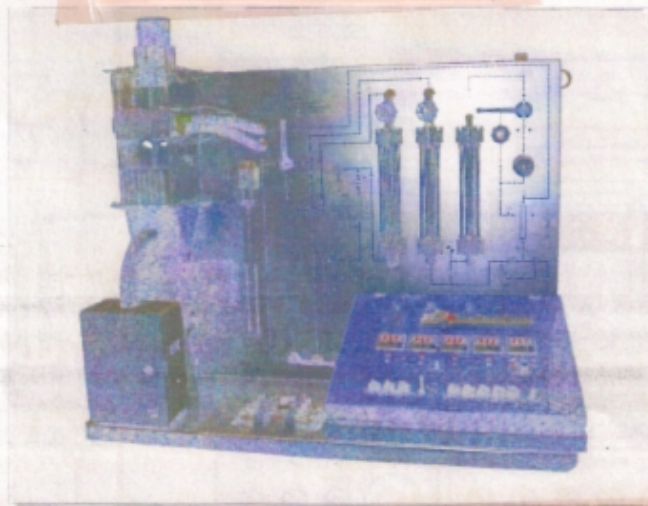


Fig. (1-C): Test Apparatus used for conduction the experimental work

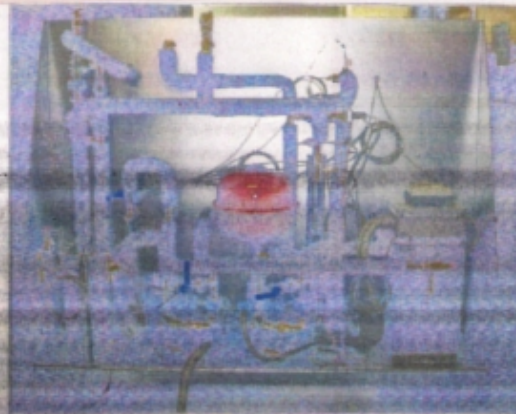


Figure 1.1 - Back view

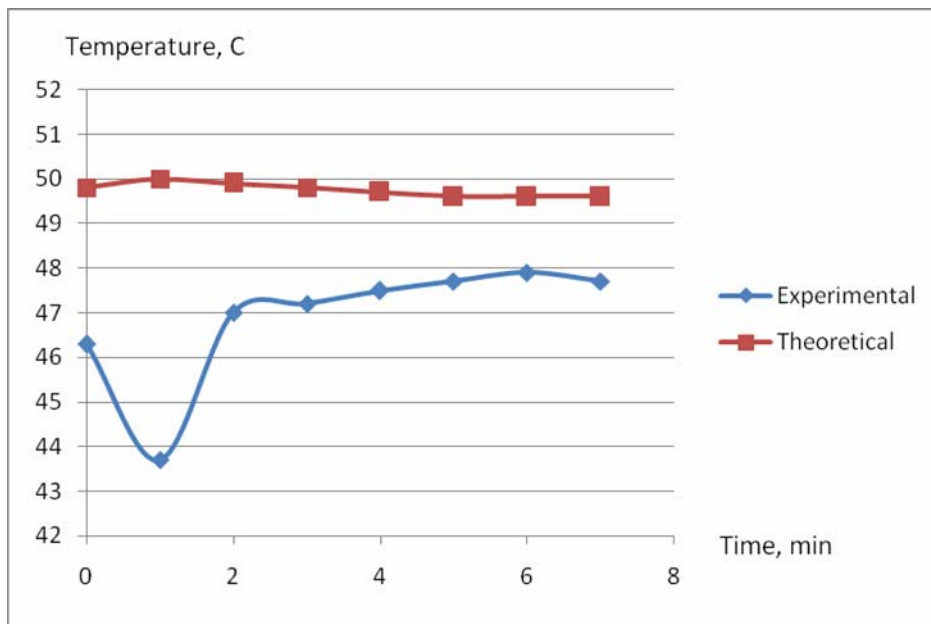


Fig. (2): Experimental & Theoretical average water temperature,  $T_{pa}$  buildup in the primary circuit of the heat exchanger versus time during secondary circuit flow reduction from (300 l/hr to 250 l/hr).

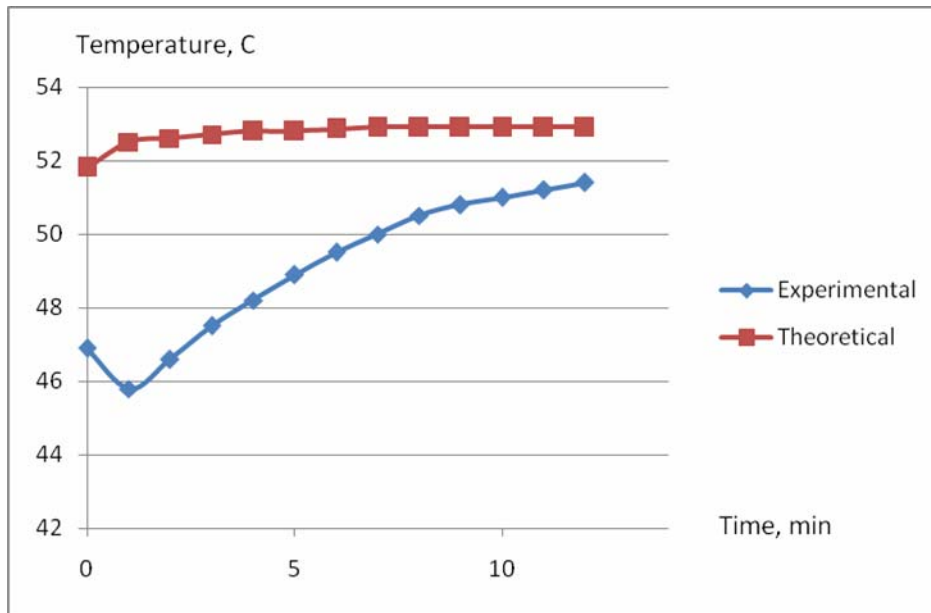


Fig. (3): Experimental & Theoretical average water temperature buildup,  $T_{pa}$  in the primary circuit of the heat exchanger versus time during secondary circuit flow reduction from (300 l/hr to 150 l/hr).

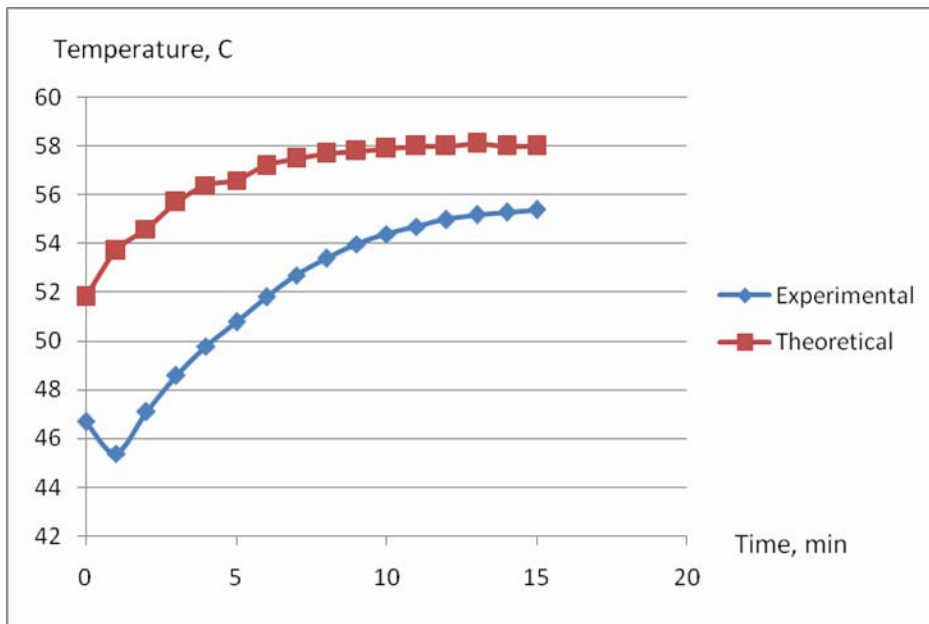


Fig. (4): Experimental & Theoretical average water temperature,  $T_{pa}$  buildup in the primary circuit of the heat exchanger versus time during secondary circuit flow reduction from (300 l/hr to 100 l/hr).

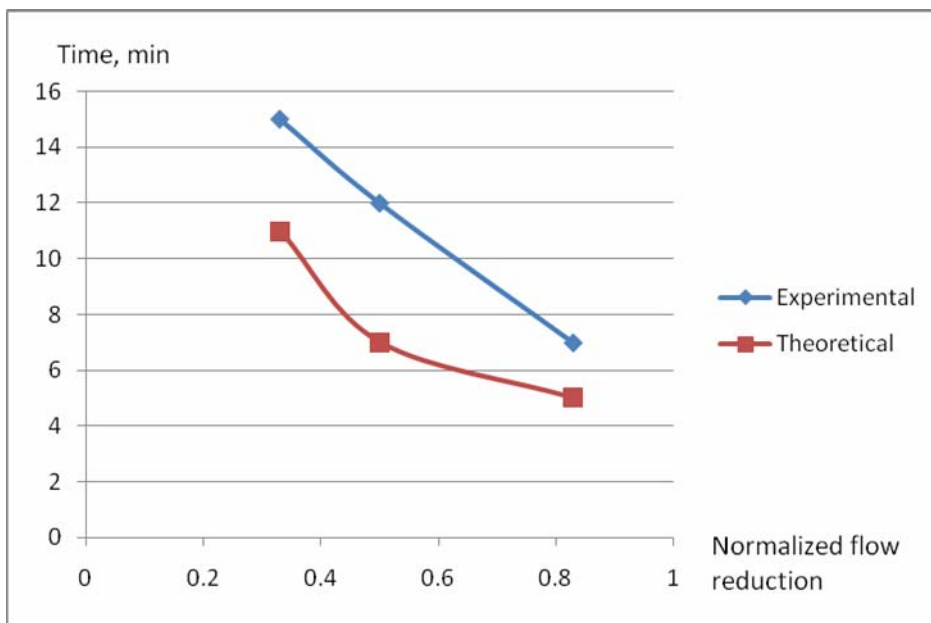
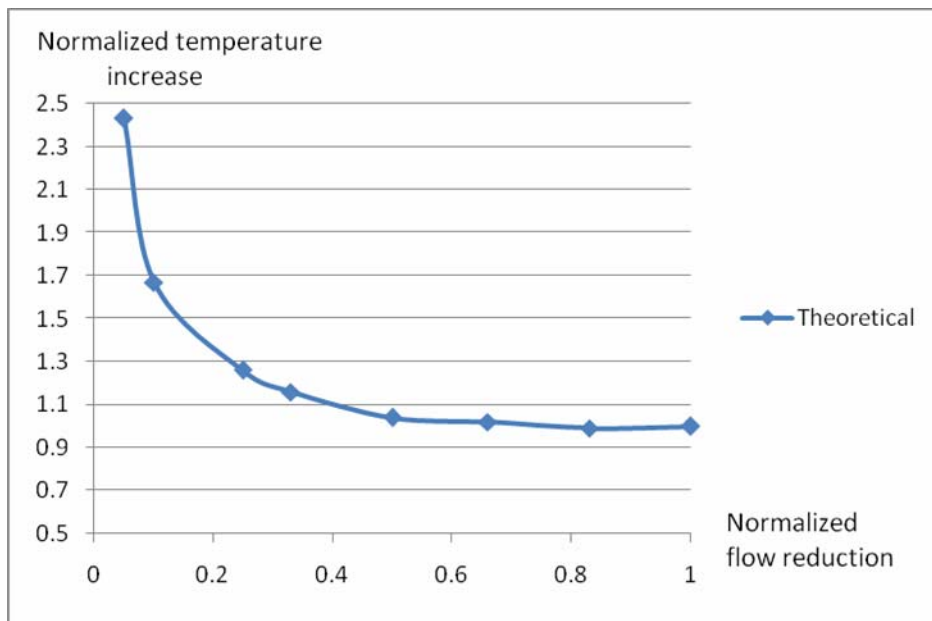
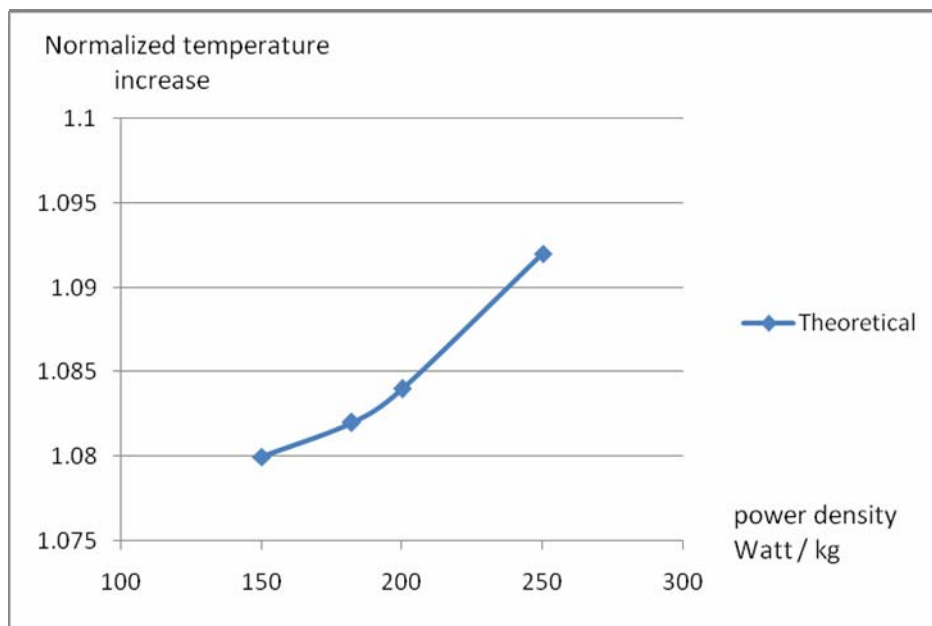


Fig. (5): Experimental & Theoretical results for the elapsed time required to settle the average water temperature in the primary circuit versus normalized flow reduction in the secondary circuit based on its steady state value (300 l/hr).



**Fig. (6): Theoretical results for the normalized water average temperature increase at the end of transient course with reference to its increase during steady state operation fixed parameters versus normalized flow reduction in the secondary circuit based on its steady state value (300 l/hr).**



**Fig. (7): Theoretical results for the normalized water average temperature increase at the end of transient course with reference to its increase during steady state operation fixed parameters versus power density input to the primary circuit.**