Fouling of a Double Pipe Heat Exchanger

By

Antonio Llinares Fontdevila

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Under the Supervision of
Dr. Brian G. Corcoran

School of Mechanical and Manufacturing Engineering
Dublin City University
Ireland

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Declaration

I hereby certify that the material, which I now submit for assessment on the programme of study leading to the award of Degree of Master of Engineering, is entirely my own work and has not been taken from the work of others save and to the extent that such work has been cited and acknowledged within the text of my work.

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Abstract

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Author: Antonio Llinares

Every single heat exchanger in operation in modern industries is exposed to fouling to a greater or lesser extent depending on the surface temperature, surface condition, material of construction, fluid velocity, flow geometry and fluid composition. The fouling phenomenon is time dependent and will result in a decrease in thermal effectiveness of a heat exchanger. Once the thermal effectiveness decreases to a minimum acceptable level, cleaning of the equipment becomes necessary to restore its performance.

This thesis investigates the effects of fouling in a double pipe heat exchanger. The first part consisted of the design and construction of a double pipe heat exchanger that corresponds to our budget and meets our demands. These demands are the possibility of the change in fluid velocity, the ability to use different fluid processes, to have an easy way to get the inlet and outlet temperatures and to design a double pipe system easy to strip out in order to analyse fouling on the heat transfer surface.

The main lines of research are carried out to establish a comparison between the normal development of fouling in a heat exchanger shown in theory books, and the development of fouling in a heat exchanger designed by us. The results of the research also indicate how fouling affects heat transfer, especially heat transfer coefficients and hot and cold fluid outlet temperatures.

Dairy products were used as process fluid. Tests were carried out for periods of up to eight hours. The evolution of temperature, heat transfer, overall heat transfer coefficient and fouling resistance over time were investigated. The evolution of the temperatures with time and the overall heat transfer coefficient values are affected by fouling. The evolution of heat transfer with time occurred according to the different fouling mechanisms. Fouling deposits increased with time until reaching the point where they produced a blockage of the system.

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Chapter 1 Introduction

The aim of this project is to predict how fouling progresses with time in a double pipe heat exchanger. This research is an introduction to heat exchangers and the main problem that remains unresolved in the majority of modern industries and in the entire history of heat exchangers: Fouling. [1]

Table 1.1 shows the economic loss due to fouling. The average cost of fouling in highly industrialized nations has been estimated at 0.25% of the GNP. [2]

Country	Fouling Cost (Euro million)	2000 GNP (Euro billion)	Fouling Cost/GNP (%)
UK	3200	1300	0.25
US	18000	7300	0.25
New Zealand	80	55	0.15
Australia	600	400	0.15
Germany	6500	2500	0.25
Japan	13000	5100	0.25

Table 1.1 Loss due to fouling in highly industrialized nations.

Fouling is used specifically to refer to undesirable deposits on the heat exchange surface. It has been recognised as a nearly universal problem in design and operation. Heat Exchangers affected by fouling are designed with excess heat transfer capacity, in order to offset the losses in efficiency caused by fouling. Although fouling is time dependent, generally a fixed value is prescribed during the design stage. [3]

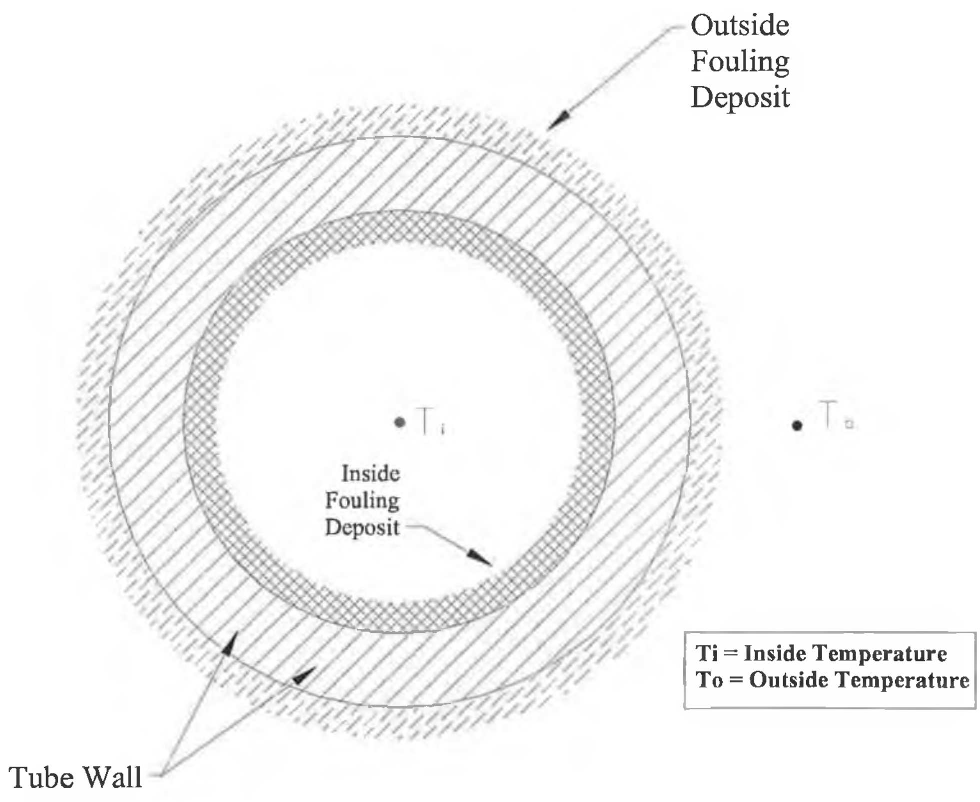


Fig. 1.1 Fouled tube

In the food industry, fouling deposits can also act as sites to support bacteria, causing product safety concerns. In the chemical, petroleum and pharmaceutical industries, the production, transportation and processing of fluids could be affected by the deposition of heavy organic and other solids dissolved or suspended in the fluid flow systems. When fouling becomes substantial, cleaning needs to be carried out, which takes a similar amount of time as that of normal production. [4]

Therefore, fouling may significantly influence the overall design of a heat exchanger and may determine the amount of material employed for construction as well as performance between cleaning schedules. Consequently, fouling causes an enormous economic loss as it directly impacts the initial cost, operating cost and heat exchanger performance. [5]

Heat exchangers are one of the most common pieces of equipment found in all plants. Their purpose is very simple: to transfer heat from a hot source to a cool one or vice versa. Generally, a heat exchanger is a device that transfers heat energy from one fluid (or gas) to another fluid (or gas). [6]

The scope of the heat exchanger is broad. It includes power production, process, chemical and food industries, electronics, environmental engineering, waste heat recovery, manufacturing industry and air-conditioning, refrigeration and space application. Figure 1.2 shows some applications of Heat Exchanger technologies. [7]

A common example is automotive radiators or an air conditioning system. In an automotive vehicle, heat released by the hot engine is transferred to the water, which is pumped into the radiator, while air is blown through the radiator fins. In the radiator, the heat from the water is transferred to the air. Thus, cooler water goes back to the engine and the cycle starts again, keeping the engine at the right temperature. An air-conditioning system and refrigerator system have at least two heat exchangers, one for cooling and one to expel heat (evaporators and condensers).

The design and consequently shape and characteristics of heat exchangers are unique. The construction of a heat exchanger is based on a complete study of the customer's demands. These demands include dimensions, materials, functionalities, heat capacity, the company budget and so on. However, the way in which a heat exchanger works follows some rules or criteria. Some authors have given their own classifications of heat exchangers and they have been accepted to a certain extent in the heat transfer world. One of the most accepted heat exchanger classifications will be explained in Chapter 2 (section 2.2).

According to this classification, the heat exchanger used in this project (double pipe heat exchanger) is a recuperator heat exchanger with indirect contact transfer processes. The geometry of the construction is a circular tube and these units can be used in a parallel or counter current flow.

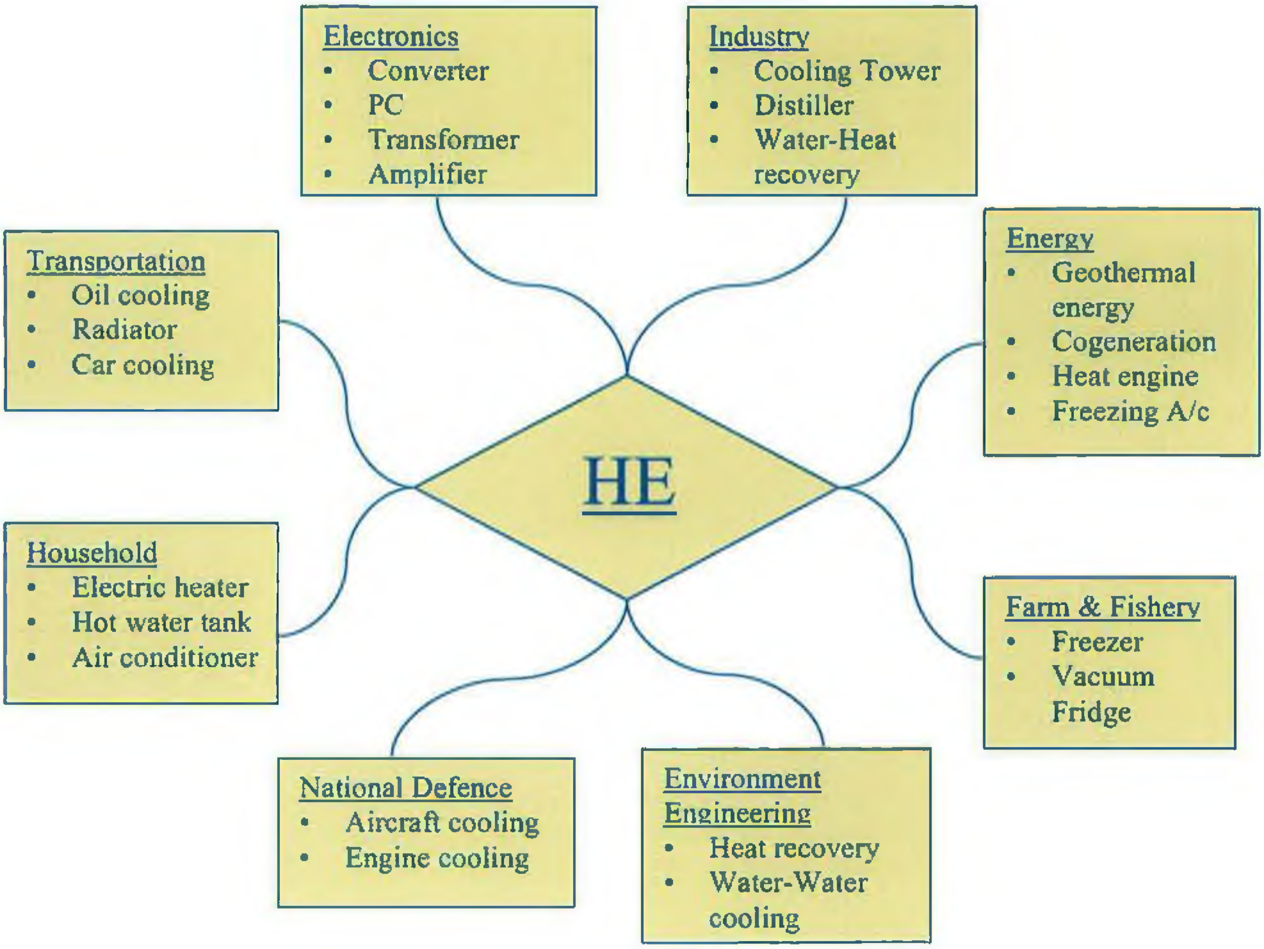


Fig. 1.2 Application of Heat Exchanger (HE) technologies

The fouling that will be analysed is situated in an inner pipe and does not build up in a constant shape along the pipe. This will show how difficult it is to measure fouling. This research will be based on how temperatures change with time along the pipes. The equations to be used in this research are shown in Chapter II (2.5): Theory and literature review.

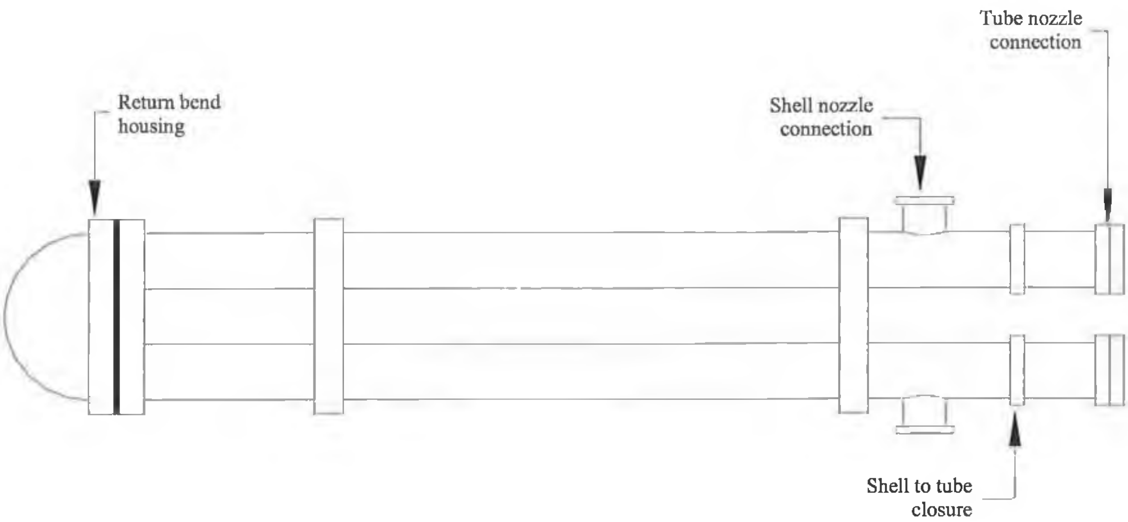


Fig. 1.3 Single hairpin double pipe heat exchanger

The two fluids used in the heat exchanger will be:

1. Process fluid: Dairy products. The main advantages are they are cheap, easy to obtain and produce fouling easily at low temperatures. One last advantage is that the concentration of proteins and fat can be easily modified in some dairy products by adding more water to the mix.
2. Heating fluid: Water. An existing heat exchanger was used to provide the hot water reservoir and heaters, flow meters were provided to control the volume of hot water in the double pipe system and finally a pump was also used.

The aim of this thesis is to be able to predict the evolution of fouling in a double pipe heat exchanger. However, the design and construction of such a heat exchanger will be of even greater importance, since the way the rig is built will show us the path to follow in our research and the equations to use to fulfil the thesis' aims.

Therefore, the objectives are directed to the acquisition of the inlet and outlet temperatures for both, the hot water and the different dairy products used in the tests. With these temperatures and applying the Log Mean Temperature Difference method for heat exchanger analysis, the heat transfer, the overall heat transfer coefficient and the log mean temperature difference values will be obtained. Consequently, calculations of fouling resistance rates in the rig will be carried out.

Once all the main values are calculated, a complete study and analysis of the rig performance will be carried out. The analysis of all the main values will be based on how fouling affects these values over time. Therefore, a detailed analysis will be carried out on:

1. Evolution of temperature with time.
2. The effect of fouling on heat transfer.
3. The effect of fouling on the overall heat transfer coefficient.
4. The effect of flow rate on the induction, transportation period.
5. Reynolds number evolution with temperature.
6. The effect of deposition on fouling resistance.
7. The effect of flow rate on fouling resistance.
8. Evolution of fouling on the heat transfer surface.
9. The effect of concentration on fouling resistance.

Chapter 2 Theory and Literature Review

2.1 Heat Transfer: Basic Equations

2.1.1 Introduction

In this chapter, the basic rating and sizing equations used in the design of heat exchangers are shown. Using the rating equations, the heat transfer rate and the fluid outlet temperatures for specific fluids flow rates, as well as the inlet temperatures and allowable pressure drop for an existing heat exchanger can be calculated. The dimensions of the heat exchanger required for specific values of flow rates, inlet and outlet temperatures and pressure drops using the sizing equations can be determined.

2.1.2 Heat transfer

Heat can be defined as the energy transferred by a thermal process. There are three transfer mechanisms: conduction, convection and radiation. [8]

1. Conduction: Is the transfer of heat from/through solids and fluids when there is no movement of the fluid in the heat flow direction.

$$\dot{Q} = \frac{KA(T_h - T_c)}{L} \tag{2.1}$$

2. Convection: Is any transfer of heat through a fluid caused by the motion of the fluid. The movement in the fluid can be produced in two ways: caused by an external mechanism such as a pump (forced convection) or caused only by a change in the fluid properties such as the density (natural convection) caused by the heating process. Both natural and forced convection can occur simultaneously in the same application.

$$\dot{Q} = Ah(T_w - T_\infty) \tag{2.2}$$

3. Radiation: Is the transfer of heat by electromagnetic waves/particles. Objects emit or and absorb electromagnetic waves/particles. There is no need for any medium through which this form of heat is transported.

   $$H = e\sigma AT^4 \tag{2.3}$$

2.1.3 First law of thermodynamics

From the first law of thermodynamics for an open system, under steady state and steady flow conditions, the change of enthalpy of one of the fluids streams is [9]

$$\delta Q = \dot{m}di \tag{2.4}$$

$$\dot{Q} = \dot{m}(i_2 - i_1) \tag{2.5}$$

where Q is the rate of heat transfer, $\dot{m}$ is the mass flow rate and i_1 represents the inlet specific enthalpy of the fluid and i_2 outlet specific enthalpy of the fluid stream. If the fluids do not undergo a phase change, the enthalpy can be written as

$$i = c_p(T_2 - T_1) = c_p\Delta T \tag{2.6}$$

Therefore equation (2.5) can be written under conditions of the hot fluid

$$Q = (\dot{m}c_p)_h(T_{h1} - T_{h2}) \tag{2.7}$$

or cold fluid

$$Q = (\dot{m}c_p)_c(T_{c2} - T_{c1}) \tag{2.8}$$

As the temperature difference between the hot and cold fluids varies with the position along the heat exchanger, so does the heat transfer. A mean value of temperature difference between fluids stream must be established so that the total heat transfer Q can be determined

$$Q = UA\Delta T_m \tag{2.9}$$

Where A is the total hot or cold side heat transfer area, and U is the average overall heat transfer coefficient based on that area. ΔT_m is a function of T_{h1}, T_{h2}, T_{c1} and T_{c2}. A specific form of ΔT_m will be defined further on in this chapter, in section 2.1.5 as log mean temperature difference (LMTD) method.

Therefore, the problem is calculating the overall heat transfer coefficient U and the mean temperature difference ΔT_m.

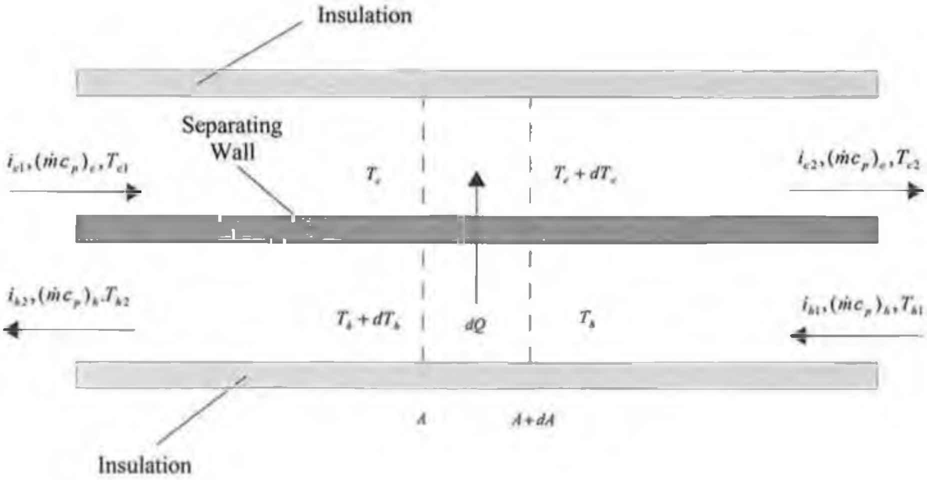


Fig. 2.1.1 Overall energy balance

2.1.4 Overall heat transfer coefficient

The overall heat transfer coefficient U for a single smooth (no fins) and clean plane wall, assuming uniform heat flux, can be calculated as [10]

$$UA = \frac{1}{R_t} = \frac{1}{\frac{1}{h_i A} + \frac{t}{KA} + \frac{1}{h_o A}} \tag{2.10}$$

Where R_t is the total thermal resistance, t is the thickness of the wall, h_i is the heat transfer coefficient for the inside flow and h_o is the heat transfer coefficient for the outside flow.

The overall heat transfer coefficient U for a smooth (no fins) and clean tubular heat exchanger can be calculated as

$$U_o A_o = U_i A_i = \frac{1}{R_t} = \frac{1}{\frac{1}{h_i A_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi kL} + \frac{1}{h_o A_o}} \tag{2.11}$$

Fouling can be produced on one or more surfaces of a heat exchanger. Fouling introduces an additional thermal resistance R_t to the heat transfer. R_t value depends on the type of fluid, fluid velocity, type of surface and length of service of the heat exchanger. The total thermal resistance of a tubular heat exchanger with the inner pipe fouled on both the inside and the outside surfaces, can be expressed as [11]

$$R_{ft} = \frac{1}{UA} = \frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi kL} + \frac{R_{fi}}{A_i} + \frac{R_{fo}}{A_o} + \frac{1}{h_o A_o} \tag{2.12}$$

The overall heat transfer coefficient varies along the heat exchanger and is dependent on the heat transfer surface geometry, fluid properties and the Reynolds number (laminar or turbulent flow).

2.1.5 The log mean temperature difference method (LMTD)

This particular method of calculating ΔT_m is only applicable to parallel flow and counter flow heat exchangers and is not applicable to multipass flow and cross flow heat exchangers.

ΔT_m may be determined by applying an energy balance to a differential area element dA in the hot and cold fluids. Therefore, from the equation (2.7) and (2.8), the energy balance is [12]

$$\delta Q = -(\dot{m}c_p)_h dT_h = \pm(\dot{m}c_p)_c dT_c \tag{2.13}$$

$$\delta Q = -C_h dT_h = \pm C_c dT_c \tag{2.14}$$

where C_h is the hot fluid heat capacity at constant pressure and C_c is the cold fluid heat capacity. The + sign will be used for parallel flow and – sign for counter flow. For counter flow, the equation is

$$d(T_h - T_c) = dT_h - dT_c = \delta Q\left(\frac{1}{C_c} - \frac{1}{C_h}\right) \tag{2.15}$$

The heat transfer rate δQ can be expressed from the differential form of equation (2.9)

$$\delta Q = U(T_h - T_c)dA \tag{2.16}$$

By joining equations (2.15) and (2.16) the result is

$$\frac{d(T_h - T_c)}{(T_h - T_c)} = U\left(\frac{1}{C_c} - \frac{1}{C_h}\right)dA \tag{2.17}$$

which, when integrated with constant values of U, C_c and C_h over the entire length of the heat exchanger, can be expressed as

$$\ln\left(\frac{T_{h2} - T_{c1}}{T_{h1} - T_{c2}}\right) = UA\left(\frac{1}{C_c} - \frac{1}{C_h}\right) \tag{2.18}$$

$$T_{h2} - T_{c1} = (T_{h1} - T_{c2})\exp\left[UA\left(\frac{1}{C_c} - \frac{1}{C_h}\right)\right] \tag{2.19}$$

For a parallel flow heat exchanger, the equation becomes

$$T_{h2} - T_{c2} = (T_{h1} - T_{c1})\exp\left[-UA\left(\frac{1}{C_c} - \frac{1}{C_h}\right)\right] \tag{2.20}$$

Substituting C_c and C_h values in equation (2.20) and solving for Q, the result is

$$Q = UA\frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln\left(\frac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}\right)} \tag{2.21}$$

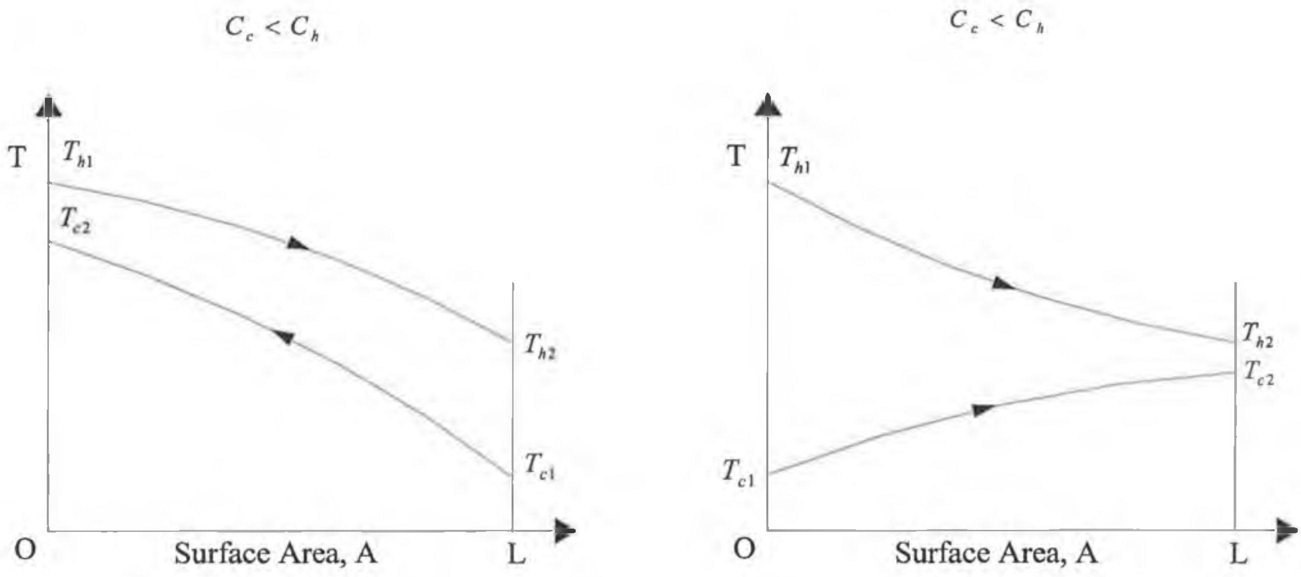
$$Q = UA\frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{2.22}$$

where ΔT_1 is the temperature difference between the two fluids at one end of the heat exchanger and ΔT_2 is the temperature difference between the two fluids at the other end of the heat exchanger.

If we compare equation (2.9) with equation (2.22) we get the average temperature difference between the hot and cold fluids over the entire length of the heat exchanger, called LMTD

$$LMTD = \Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{2.23}$$

$$Q = AU\Delta T_{lm} \tag{2.24}$$



(a) **(b)**

Fig. 2.1.2 Fluid temperature variation in: (a) counter flow; (b) parallel flow

2.1.6 The number of transfer units method (NTU)

The log mean temperature difference method (LMTD) can produce some errors and erroneous results if the inlet or outlet temperatures of the fluid stream are not known precisely. In order to avoid errors in the procedures, the number of transfer units (NTU) based on the heat exchange effectiveness may be applied. The rating analysis with the ε-NTU method is as follows: [13]

1. Calculate the capacity rate ratio and NTU
2. Determine the effectiveness ε from the appropriate charts or ε-NTU equations for the given heat exchanger and specified flow arrangement.
3. Calculate the total heat transfer rate once ε is known.
4. Calculate the outlet temperatures.

As will be discovered further on in the tests, the inlet and outlet temperatures of the cold and hot fluids streams are given. Therefore the NTU method will not be used in the calculation of the main rate values.

2.1.7 Forced convection

As we have already mentioned, forced convection is the transfer of heat between a moving fluid (produced from an external force as a fan or a pump) and a solid surface or another fluid. In general, it is nearly impossible to find a mathematical solution to all forced convection problems. This is due to the huge amount of factors that affect convective heat transfer. The geometry of the heat exchanger (tubular, plate, extender surface…) is just one of the factors that can modify the forced convection values from one heat exchanger to another. The change of flow arrangement in a heat exchanger is another factor that can modify the forced convection values in the same heat exchanger.

A l arge num ber o f e xperimental a nd a nalytical c orrelations c an b e f ound f or he at transfer coefficients and flow friction factors for laminar and turbulent flow streams. However, an analysis can be formulated by the following equation [9,12]

$$Nu=f(Pr,Re) \tag{2.25}$$

where,

Nu= Nusselt number
Pr= Prandtl number
Re= Reynolds number

The Prandtl number is a dimensionless parameter of a convecting system. The Prandtl number shows the relative rates of development of the velocity and temperature boundary layers in the entrance region. It is defined as

$$Pr = \frac{\upsilon}{\alpha} \tag{2.26}$$

where υ is the kinematic viscosity of fluid and α is the thermal diffusivity of fluid.

The R eynolds num ber i s a dim ensionless r atio us ed t o de termine if t he f low in a certain system is laminar (the fluid particles move in defined paths called streamlines) or turbulent (the laminar flow becomes unstable due to small disturbances and the fluid flows in a series of eddies, which result in a complete mixing of the flow). The Reynolds number is defined as

$$\mathrm{Re} = \frac{uL\rho}{\mu} \qquad \left({}^{inertia}\!/_{diffusivity} \right) \tag{2.27}$$

where u is the velocity of the fluid stream, ρ is the density of the fluid, μ is the viscosity of the fluid and L is the characteristic length.
At a Reynolds number $\mathrm{Re} \leq 2100$ the flow is fully developed laminar in pipe. [14]
At a Reynolds number $\mathrm{Re} \geq 10^4$ the flow is fully developed turbulent in pipe. [15]
Between the lower and upper limits lies the transition zone from laminar to turbulent flow.

The Nusselt number is a dimensionless coefficient of the heat transfer in a convection process. It is defined as [10]

$$Nu = \frac{hL}{k} = \frac{1}{L}\int_0^L Nu_x dx \tag{2.28}$$

where

$$Nu_x = \frac{h_x d}{k} = \frac{-d(\partial T/\partial y)_w}{(T_w - T_b)_x} \tag{2.29}$$

where

$$T_b = \frac{1}{A_c u_m \int\limits_{A_c} uT dA_c} \qquad (2.30)$$

2.1.8 The effect of variable physical properties

The physical properties in some fluids change with temperature. This change in the physical p roperties in f luids w ill influence c onsiderably t he v ariation o f im portant rates in heat transfer. Mainly, the variation of physical properties in the fluid will affect the velocity and temperature through the boundary layer or over the flow cross section of the pipe. [16]

For liquids, the variation of viscosity with temperature is responsible for most of the properties effects. Therefore, the Nusselt number correlations can be defined as

$$\frac{Nu}{Nu_{cp}} = \left(\frac{\mu_b}{\mu_w}\right)^n \qquad (2.31)$$

$$\frac{f}{f_{cp}} = \left(\frac{\mu_b}{\mu_w}\right)^m \qquad (2.32)$$

where μ_b is the viscosity at the bulk temperature, μ_w is the viscosity at the wall temperature, subscript cp refers to the constant-property solution and f is the fanning friction factor. Fanning friction factor is proportional to shear stress at pipe/conduit wall as number of velocity heads and is used in momentum transfer in general and turbulent flow calculations in particular.

For gases, the viscosity, thermal conductivity and density vary with temperature. Therefore, the Nusselt number correlations can be defined as

$$\frac{Nu}{Nu_{cp}} = \left(\frac{T_w}{T_b}\right)^n \qquad (2.33)$$

$$\frac{f}{f_{cp}} = \left(\frac{T_w}{T_b}\right)^m \tag{2.34}$$

Where T is the absolute mean temperature.

2.2 Heat Exchanger

2.2.1 Introduction

In almost any nuclear, chemical, process, or mechanical system, heat must be transferred from one place to another or from one fluid to another. Heat exchangers are used to transfer heat from one fluid to another fluid.

Heat exchangers are devices that can be used to transfer heat from a fluid stream (liquid or gas) to another fluid at different temperatures. Heat exchangers are used in a wide variety of applications. These include power production, process, chemical and food industries, electronics, environmental engineering, waste heat recovery, manufacturing industries and air conditioning, refrigeration and space applications. Examples of heat exchangers that can be found in all homes are heating radiators, the coils on your refrigerator and room air conditioner and the hot water tank. [17]

Although heat exchangers come in every imaginable shape and size, some of them with a unique design, many different classifications of Heat Exchangers exist. Shah R. K. offers one of the most complete heat exchangers classifications. He states that heat exchangers may be classified according to the following main criteria: [18]

1. Recuperators and regenerators
2. Transfer processes
3. Geometry of construction
4. Heat transfer mechanisms
5. Flow arrangements

2.2.2 Recuperators and regenerators

In a recuperator heat exchanger, both fluids flow simultaneously through different areas separated by a "wall". For example, in a double pipe heat exchanger, one of the fluids flows through the inner pipe and the other through the annulus pipe. Both fluids are separated by the thickness of the inner pipe. [19]

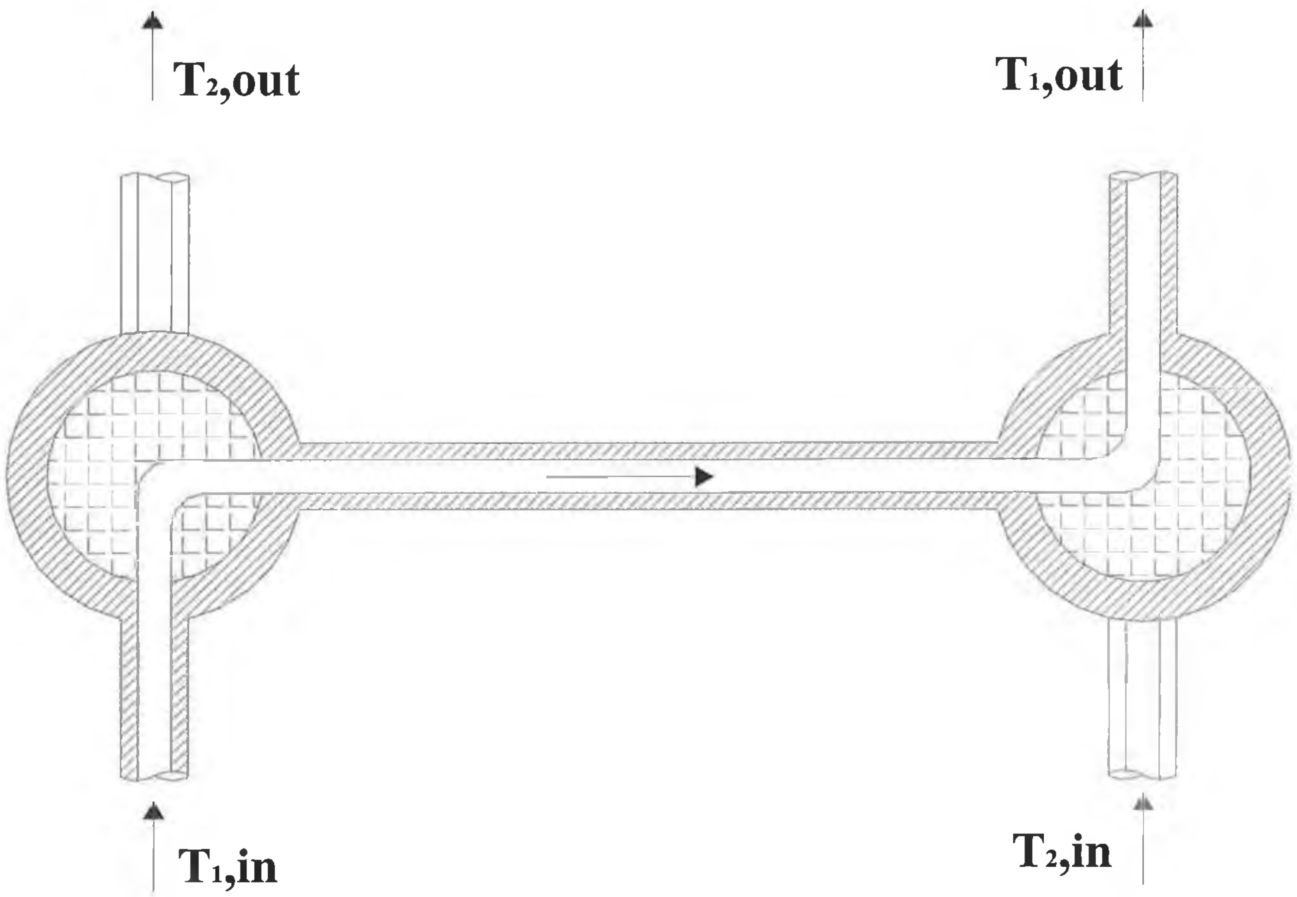


Fig. 2.2.1 Schematic representation of a regenerative heat exchanger

In the regenerator heat exchanger, there is only one area for the two fluids to flow through. One of the fluids passes firstly through the bulk and transmits its heat energy to the "wall". Once the bulk walls are fully charged with heat energy, the valves that control the first fluid close and the second fluid valves open, allowing the fluid to flow through the same space and recover the heat energy stored by the bulk walls.

Regenerative heat exchangers can be classified into two groups: rotary regenerators and fixed matrix regenerators. The fixed matrix regenerators follow the cyclic principle explained above and represented in figure 2.2.1. On the other hand, rotary regenerators do not follow this cyclic principle: firstly one fluid flows and then the second fluid flows through the same path. In the rotary regenerators the two fluids are flowing continuously through different areas, and a rotary matrix with a cylinder shape produces the change of heat energy. Half of the cylinder matrix is in touch with the hot fluid, storing heat energy, while the other half of the cylinder is in touch with the cold fluid delivering the energy. Because the cylinder does not stop in its rotation, the process is continuous and not cyclic. [20]

The fluids used in a regenerative heat exchanger are principally gases. If liquids are used, there is a risk of mixing between fluids, with the consequent change in the original fluid properties.

2.2.3 Transfer processes

There are two kinds of transfer processes: direct contact type and indirect contact type (transmural heat transfer).

In a direct contact heat exchanger, there is no wall between hot streams and cold streams, so heat is transferred through direct contact between hot and cold fluids. The fluids have to be immiscible, a gas-liquid or a solid particle-fluid combination. Cooling towers are a good example of such a heat exchanger. [21]

In an indirect contact heat exchanger, the heat is transferred between fluids through a heat transfer surface such as a wall or a pipe.

2.2.4 Geometry of construction

Indirect transfer heat exchangers are described in terms of their construction features. There are so many different kinds of heat exchangers, and so many different geometries of construction can be used in a heat exchanger design, that we will explain the major construction types: tubular, shell and tube, plate and extended surface.

2.2.4.1 Tubular heat exchanger

The most basic and most common type of heat exchanger construction is the tubular heat exchanger. Tubular heat exchangers are built of circular pipes. One fluid flows inside the tubes and the other flows on the outside of the pipes. The tube diameters, the number of pipes, the tube length, the pitch of the pipes and the pipe arrangement can be changed.

Tubular heat exchangers can be classified as double-pipe heat exchanger, shell and tube heat exchanger and spiral tube heat exchanger. [22]

2.2.4.2 Shell and tube heat exchanger

A shell and tube heat exchanger is used the most, due to its high efficiency. However, they are bulkier and cannot be used in a reduce space. The main bulk where the heat transfer is produced is limited by the shell part. Inside the shell, it is a bunch of pipes separated by baffles. Through the pipes flows one of the fluids while the other fluid flows between the pipes and the inside shell wall. The baffles function is to keep the pipes apart, so the shell side fluid can flow between them.

A double pipe heat exchanger is considered as the simplest shell and tube heat exchanger. The outer pipe functions as the shell while the inner pipe functions as the pipe. [22]

2.2.4.3 Plate heat exchanger

A plate type heat exchanger consists of plates instead of tubes to separate the hot and cold fluids. The hot and cold fluids alternate between each of the plates. Baffles direct the streams flow between the plates.

A plate type heat exchanger, as compared to a similar sized tube and shell heat exchanger, is capable of transferring much more heat. This is due to the large area that plates provide over tubes.

Plate heat exchangers are used for transferring heat for any combination of gas, liquid and two-phase streams. Plate heat exchangers can be classified as gasketed plate, spiral plate or lamella. [23]

2.2.4.4 Extended surface heat exchanger

Extended surface heat exchangers are generally fins or appendages added to the primary heat transfer surface (tubular or plate) with the aim of increasing the heat transfer area.

The two most common types of extended surface heat exchangers are plate-fin heat exchangers and tube-fin heat exchangers.

2.2.5 Heat transfer mechanisms

Heat exchangers equipment can also be classified according to the heat transfer mechanisms such as: [24]

- Single-phase, convection on both sides. The fluid leaves the heat transfer area in the same phase as it has entered.
- Two-phase, convection on one side, two-phase convection on other side.
- Two-phase, two phase convection on both sides.

Figure 2.1.2 shows the temperature evolution on a single-phase heat exchanger while figure 2.2.2 shows the temperature evolution on a two-phase heat exchanger. Two-phase heat exchangers can be classified as boilers or condensers.

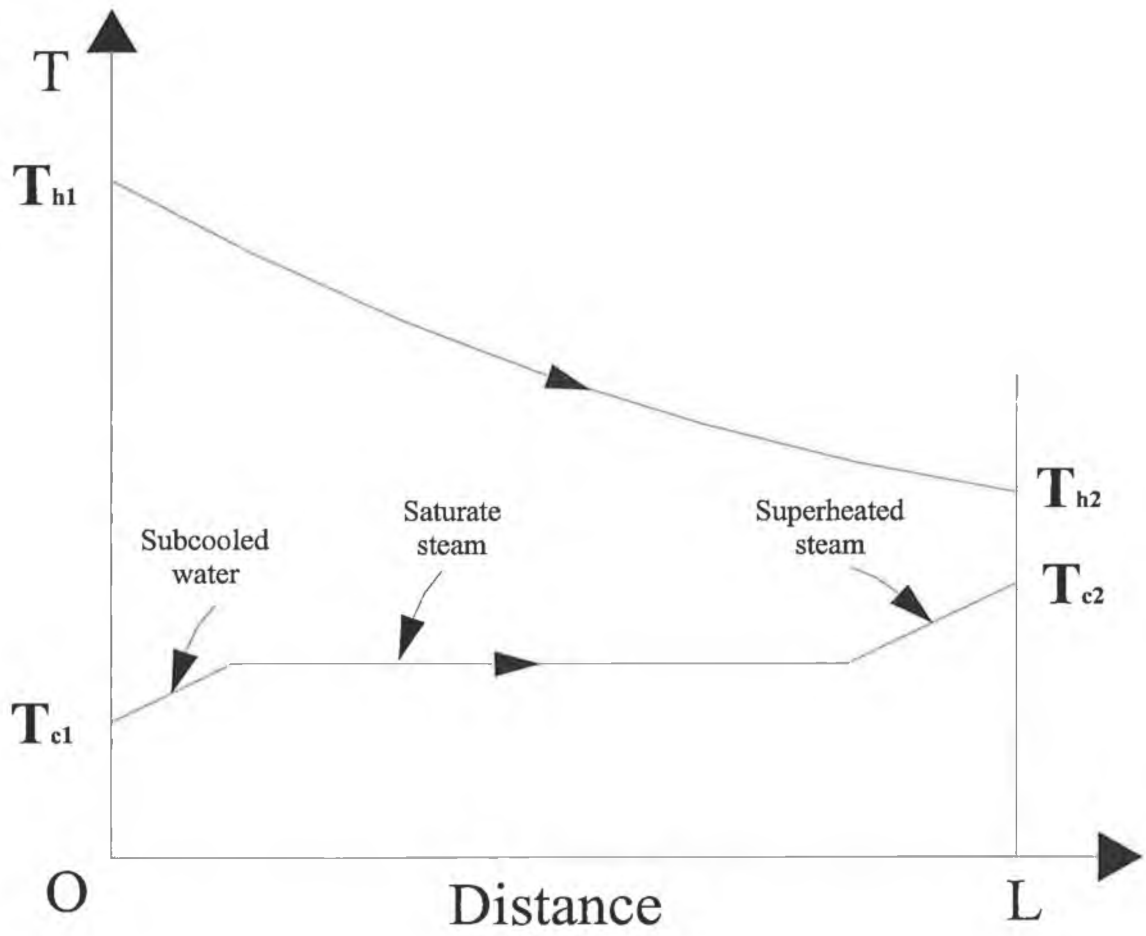


Fig. 2.2.2 Fluid temperature variation for a boiler

2.2.6 Flow arrangement

There are three basic configurations based on the direction of the fluid flow within the heat exchanger. These are: [25]

1. Parallel flow. The two fluids streams in the heat exchanger flow in the same direction.
2. Counter flow. The direction of the flow of one of the fluids streams is opposite to the direction of the other fluid.
3. Cross flow. In a cross flow heat exchanger, one fluid flows though the heat transfer surface at a 90 degrees angle to the flow path of the other fluid.

Having the basic arrangements in a series can also do multipass cross flow configurations.

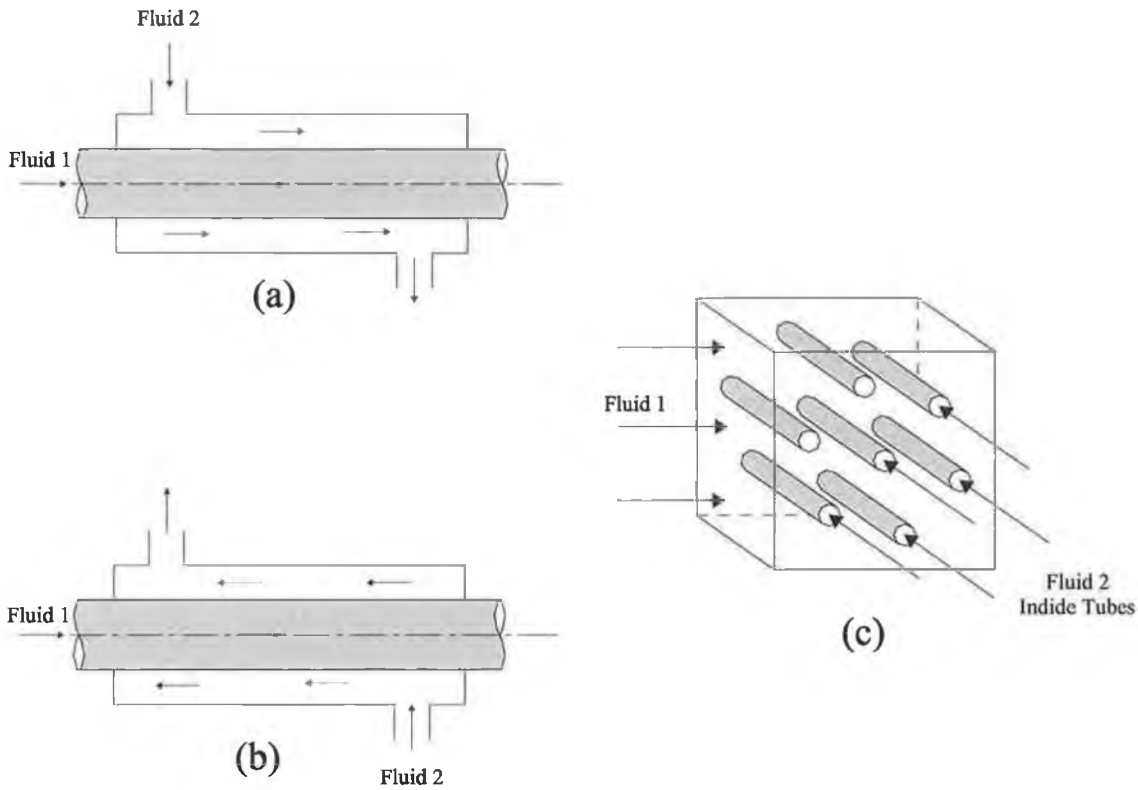


Fig. 2.2.3 Different flow arrangement situations: (a) parallel-flow; (b) counter-flow; (c) cross-flow

The most common arrangements for a heat exchanger are parallel flow and counter flow. See figure 2.1.2. The main difference between the flow arrangements is the temperature distribution along the pipes of the heat exchanger, and therefore the heat transfer. If we compare heat transfer and the overall effectiveness in a counter flow and parallel flow heat exchanger under given temperatures, the result shows that the counter flow configuration is better. This is because the more uniform temperature difference between the two fluids produces a more uniform rate of heat transfer through the heat exchanger. The outlet temperature of the cold fluid in a counter flow arrangement can approach the highest temperature of the hot fluid (the inlet temperature).

2.3 Fouling of Heat Exchangers

2.3.1 Introduction

Fouling is generally defined as the accumulation of undesirable substances on the surfaces of processing equipment. In 1972, Taborek cited fouling as the major unresolved problem in heat transfer. Nowadays, more than 30 years later, fouling remains the major problem affecting the heat transfer industry. [26]

Fouling has been recognised as an almost universal problem in design and operation and affects the operation of equipment in two ways. This accumulation of undesirable substances will change the geometry of the heat exchanger and it will continue to grow with time. This change of geometry reduces the efficiency of a heat exchanger by affecting the pressure drop. Fouling will act as a heat transfer resistance. This new heat transfer resistance reduces the efficiency of a heat exchanger by reducing the heat transfer.

- The fouling layers have a low thermal conductivity. This increases the resistance to heat transfer and reduces the effectiveness of heat exchangers.
- As deposition occurs, the cross sectional area is reduced, which causes an increase in pressure drop across the apparatus.

2.3.2 Cost of fouling

Fouling introduces an additional cost to the industrial sector in the form of: [27]

- Increased maintenance costs
- Reduced service life
- Added energy costs
- Loss of plant efficiency
- Loss of production

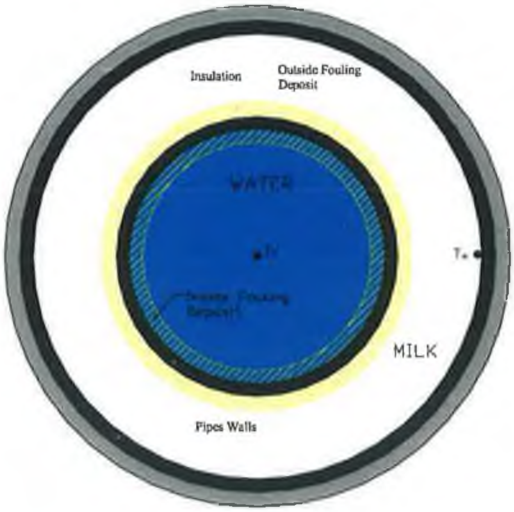


Fig. 2.3.1 Double pipe fouled

There are not many solutions in order to compensate the reduction of efficiency in the heat exchanger due to fouling. However, these solutions increase the additional cost of heat exchanger design, production and maintenance by:

- Special design considerations
- Increased heat transfer area
- Oversized pumps and fans
- Duplicate Heat Exchanger to ensure continuous operations due to maintenance
- Use of high cost materials
- Online cleaning equipment
- Use of chemicals and hazardous cleaning solutions

2.3.3 Types of fouling

Fouling can be classified in a number of different ways. Due to the diversity of process conditions, most fouling situations are relatively unique. However, Epstein has developed a classification widely accepted in the scientific world. [28]

According to Epstein, fouling can be classified as:

- Particulate fouling (sedimentation): is the result of the accumulation of solid particles suspended in the process fluid. Sand, chip, and insoluble corrosion products and many others can be part of these suspended solids.
- Crystallization fouling (precipitation): is the result of the accumulation of dissolved inorganic salts in the fluid streams, producing super saturation when the fluid is heating. These are inverse solubility salts that precipitate in the cooling process.
- Corrosion fouling: is caused by surface reaction with fluids to form corrosion products, which attach to the heat transfer surface to form nucleation sites.
- Biological fouling: the deposition and growth of material of biological origin.
- Chemical reaction fouling: due to chemical reaction within the process stream. Chemical reaction fouling is very common in petroleum refining and polymer production.
- Solidification fouling: due to the formation of ice or wax.

In most fouling situations there is not just one kind of fouling, several different types may be involved. Some of the fouling processes can complement each other.

2.3.4 Fundamental processes of fouling

Fouling is an extremely complex phenomenon, due to the large number of variables that affect it. Again, Epstein gives a sequence of events referred to as the fouling mechanisms. These fouling mechanisms are: [29]

1. Initiation or induction. It is the most critical period. The surface material, temperature, and roughness and the concentration and velocity of the fluid stream have a strong influence in the initial delay, induction or incubation period.
2. Transport. It is the most studied period. This period involves the transport of fouling substances from the fluid stream to the heat transfer surface. The

transport of substances can be carried out by a number of different phenomena such as diffusion, sedimentation and thermophoseris.

3. Attachment: when the formation of the deposit begins.
4. Transformation: when physical or chemical changes can increase deposit accumulation.
5. Removal: depending on the deposit strength, the velocity gradient and the viscosity of the fluid and the roughness surface, some fouling materials can be removed from the surface.

2.3.5 Factors influencing fouling

Fouling can grow in different ways for the same fluids in the same heat exchanger, by changing some of the parameters or properties that can be applied in the same heat exchanger. Some of the factors that influence the conditions of fouling are: [30]

1. Operating parameters. If fluids stream velocity, surface temperature and bulk fluid temperature are changed.
2. Heat Exchanger parameters. If heat exchanger configuration, the surface material and the surface structure are changed.
3. Fluid parameters. If the viscosity, density and concentration of the fluid stream is changed.

2.3.6 Techniques to control fouling

There are various techniques to control and prevent fouling in a heat exchanger. All fouling control techniques can be divided into two kinds: [31]

1. On line or continuous cleaning. Fouling is removed without the interruption of product flow with the use and control of appropriate additives.
2. Off line or periodic cleaning. Fouling is removed by the disassembly of the heat exchanger and by manual cleaning.

ON LINE TECHNIQUES	OFF LINE TECHNIQUES
Inhibitors, Antiscalants	
Dispursants	Liquid jet
Acids, Sponge balls	Steam
Brushes,Sonic horns	Air jet
Soot blowers	Drills
Chains and scrapers	Scrapers
Thermal shock	Chemical cleaning
Air bumping	

Table 2.3.1 On-Off line techniques to control fouling

2.3.7 Thermal analysis

Thermal analysis is based on the conservation of energy. Heat released by hot fluid, must equal heat absorbed by cold fluid plus losses. [32]

$$Q = UA_O \Delta T_m \tag{2.35}$$

It is important to know the difference between a clean surface U_c and a fouled one U_f. U_f can be related to the clean surface as U_c

$$\frac{1}{U_f} = \frac{1}{U_c} + R_{ft} \tag{2.36}$$

where R_{ft} is the total fouling resistance, given as

$$R_{ft} = \frac{A_o R_{fi}}{A_i} + R_{fo} \tag{2.37}$$

Heat transfer under fouling conditions, can be expressed as

$$Q_f = U_f A_f \Delta T_{mf} \tag{2.38}$$

2.3.8 Effect of fouling on heat transfer

Fouling, due to the build up of undesired material on the surface of the pipe, adds an insulating layer to the heat transfer surface. [32]

$$U_f = \frac{1}{\dfrac{A_o}{A_i h_i} + \dfrac{A_o}{A_i} R_{fi} + \dfrac{A_o \ln\left(\dfrac{d_o}{d_i}\right)}{2\pi k L} + R_{fo} + \dfrac{1}{h_o}} \tag{2.39}$$

2.3.9 Effect of fouling on pressure drop

Fouling adds an extra layer to the heat exchanger, changing the original geometry of the pipes. In a tubular heat exchanger, a fouling layer roughness the surface, decreases the inside diameter and increases the outside diameter of the tubes. [32]

The frictional pressure drop in the tube for a single phase flow can be calculate by:

$$\Delta P = 4f\left(\frac{L}{d_i}\right)\frac{\rho u_m^2}{2} \tag{2.40}$$

where f is the fanning friction factor, L indicates the tube's length, d_i is the inner diameter and u_m is the fluid velocity.

2.4 Pressure Drop

2.4.1 Introduction

Every substance (solid, fluid, gas) that is in movement has a friction force or friction effect that acts in the opposite direction than the movement. In the case of fluids, in a forced convection heat exchanger, pumps and fans produce enough work to create the movement of the fluid around the heat exchanger pipes system.

As we have already mentioned, everything that has movement has a friction force. This friction force produces losses all along the heat exchanger system, in the form of a decrease of pressure and consequently, a decrease of fluid velocity within the heat exchanger pipes system. This decrease in pressure is called pressure drop.

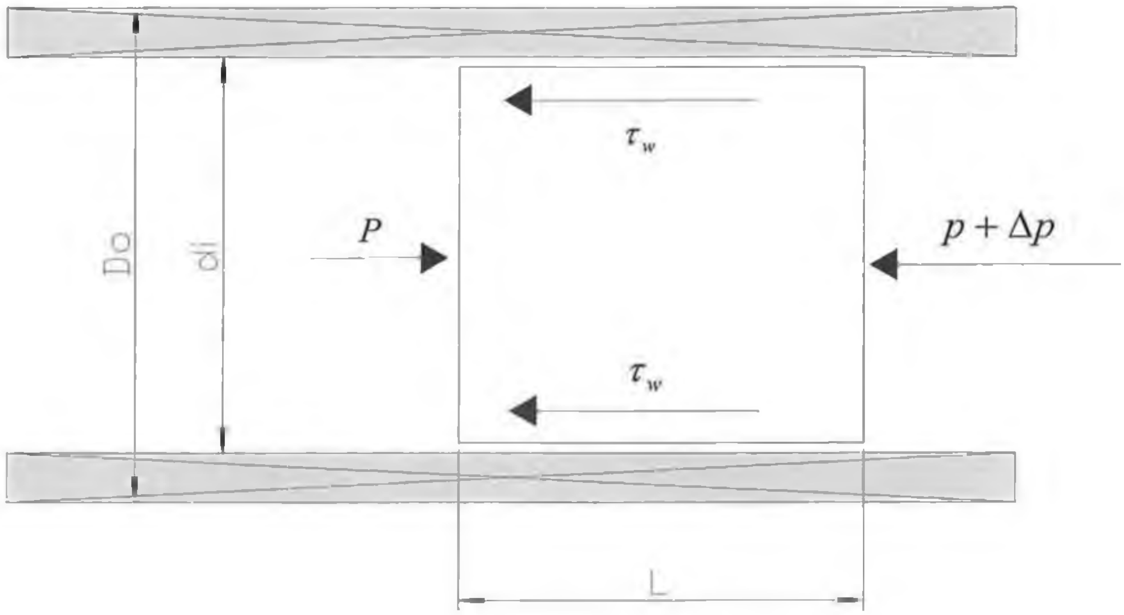


Fig. 2.4.1 Force balance of a fluid inside a pipe

To counter the pressure drop, the pumping power or fan work has to be increased, to maintain a c onstant flow a long t he h eat e xchanger. T herefore, p ressure dr op adds capital costs and is a major part of the operating cost of the heat exchanger. [33]

2.4.2 Circular cross sectional tubes pressure drop

In a fully developed flow in a tube, for either laminar or turbulent flow, the following functional relationship can be written for the frictional pressure drop. [34]

$$\frac{\Delta p}{L} = \Phi(u_m, di, \rho, \mu, e) \tag{2.41}$$

where e is a statistical measure of the surface roughness of the tube and has the dimension of length.
It is assumed that Δp is proportional to the length L of the tube.

$$\frac{\Delta p}{4\left(L/d_i\right)\left(\rho u_m^2/2\right)} = \Phi\left(\frac{u_m d_i \rho}{\mu}, \frac{e}{d_i}\right) \tag{2.42}$$

where the dimensionless numerical constants 4 and 2 are added for convenience. The above dimensionless group involving Δp has been defined as the fanning friction factor, f:

$$f = \frac{\Delta p}{4\left(L/d_i\right)\left(\rho u_m^2/2\right)} \tag{2.43}$$

therefore

$$f = \Phi\left(\mathrm{Re}, e/d_i\right) \tag{2.44}$$

The fanning friction factor for a fluid in a laminar flow, circulating within circular tubes, independent of the surface roughness, can be simply defined as

$$f = \frac{16}{\mathrm{Re}} \tag{2.45}$$

2.4.3 Other situations of pressure drop

Pressure drop is a very important factor in the heat exchanger industry and can have a huge impact on a heat exchanger design. However, in this research, the pressure drop is not important, it takes second place. This is because the calculations of R_f (fouling) are based on the temperatures along the heat transfer surface of the heat exchanger. R_f is calculated from three basic equations: The value of heat transfer, Q, the overall heat transfer coefficient, U, and the log mean temperature difference, ΔT_m. In these three equations, the pressure drop does not affect the outcome.

I have explained pressure drop in a circular cross sectional duct in a previous point in this chapter, because the main geometries of my heat exchanger are circular pipes. I have shown the basic equations and correlations. However, we are not going to use any of these equations in the calculation of R_f. Although studying pressure drop in relation to fouling could be of major interest in our project, the materials used in the heat exchanger fittings make it very complicated. Instead of using straight hard pipes, such as copper or aluminium pipes, we are using pieces of hose to connect all the rig parts. Hoses can take any particular shape and change shape with temperature. This makes the calculation of the pressure drop in our rig very complicated.

Therefore, I will explain some cases where pressure drop can appear but I will not include any equations or correlations to calculate any other kind of pressure drop. It is obvious that there will be a pressure drop in non-circular sectional tubes (not geometrically similar to a circular duct).

The fluid friction effect is very common in circular tube bundles (one of the most common heat transfer surfaces) especially in shell and tube heat exchangers.

Pressure drop also takes place in helical and spiral coils. These are curved pipes, which are used as curved tube heat exchangers in various applications, such as dairy and food processing, refrigeration and air conditioning industries. Experimental and theoretical studies show that coiled tube friction factors are higher than those in a straight tube. [35]

One more example of friction within a heat exchanger is the pressure drop in bends and fittings. Bends are used in heat exchanger piping circuits and in turbulent heat exchangers. Fittings are components used in a heat exchanger system that connect two or more pieces of pipe together. The main uses of fittings are to control the flow in angle valves, gate valves or check valves fittings.

The last example of pressure drop in heat exchangers appears when fluids can experience a sudden contraction followed by a sudden enlargement, when flowing in and out of a heat exchanger core. Therefore, pressure drop takes place when abrupt contraction, expansion and momentum changes appear across a heat exchanger pipes system.

2.4.4 Pumping power relationship

The fluid pumping power is proportional to the pressure drop in the fluid across a heat exchanger. In the design of heat exchangers involving high-density fluids, the pumping power requirement is quite small in relation to the heat transfer, and, thus, the pressure drop has hardly any influence on the design. However, for gases and low-density fluids and very high viscosity fluids, pressure drops are always of equal importance to the heat transfer rate and have a strong influence on the design of heat exchangers. [36]

2.5 Double Pipe Heat Exchanger

2.5.1 Introduction

A typical double pipe heat exchanger consists of one pipe placed concentrically inside another of a larger diameter pipe. One fluid flows through the inner pipe and the other flows through the annular pipes.

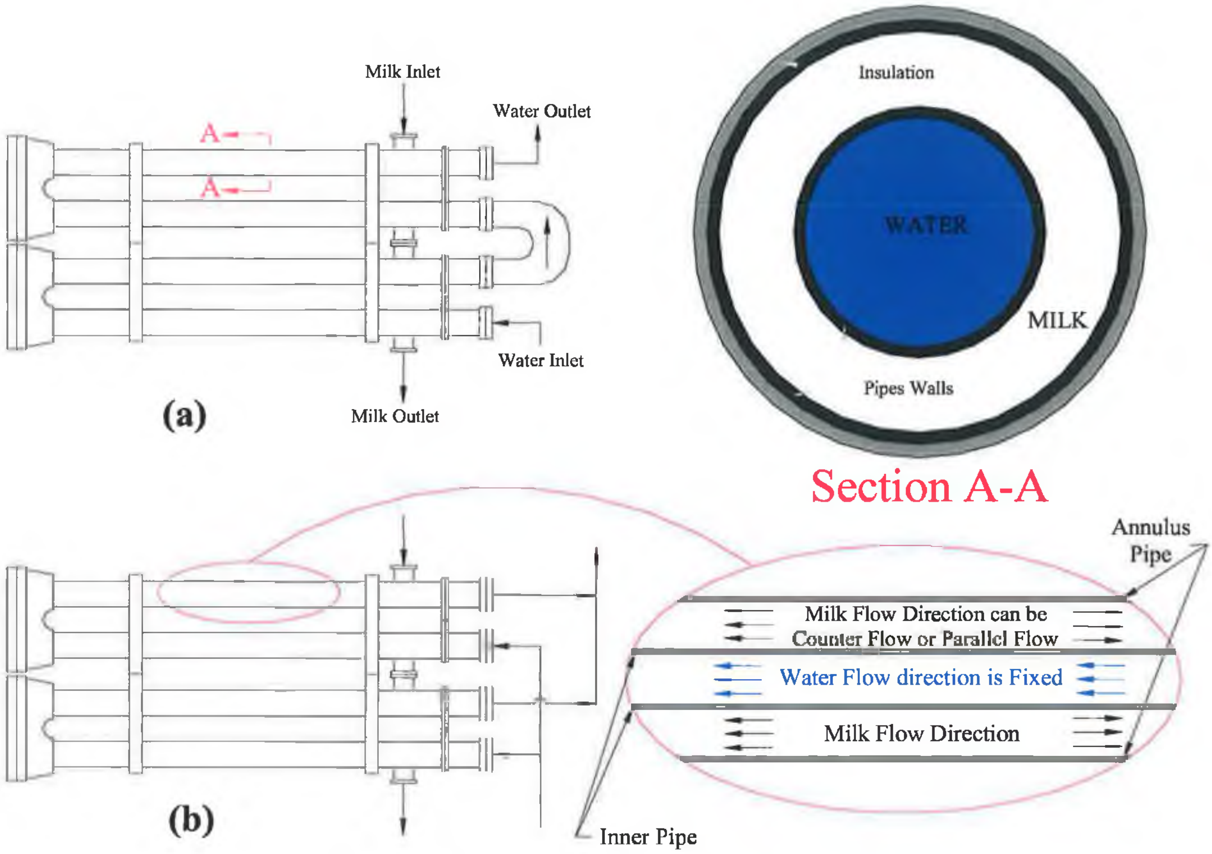


Fig. 2.5.1 Two double pipes with two hairpin sections: (a) two hairpin sections arranged in series and (b) two hairpin sections in series on the annulus side and parallel on the inner tube side

Double pipe heat exchangers are also called hairpin heat exchangers. The inner pipes are connected by U-shaped return bends and the annulus pipes are connected by special fittings as figure 2.5.1 shows. Different hairpin arrangements (in series and parallel) can be used to increase the heat transfer along the heat exchanger by increasing the heat transfer area.

The major use of double pipe heat exchangers is for sensible heating or cooling of fluids where small heat transfer areas are required. The major drawback is that they are voluminous and expensive per unit of heat transfer surface area. This second drawback is due to the U-shaped return bends, where there is no heat transfer area between fluids. The pressure drops and the friction of the fluids increases considerably.

Double pipe heat exchangers can be used as counter flow or parallel flow arrangements. However it is demonstrated in several theory books that the largest heat transfer between fluids occurs in the counter flow arrangements. [37]

2.5.2 Thermal analysis

The thermal design in a double pipe heat exchanger has to be divided into two different parts: [38]

1. The first one will be the thermal and hydraulic analysis of the inner pipe.
2. The second will be the thermal and hydraulic analysis of the annulus pipe.

Some correlations used in the thermal and hydraulic analysis of the inner and annulus pipe will be shown. The main function of these correlations is to calculate the heat transfer coefficients.

The thermal and hydraulic analysis shown in the following points, will give the correlations and equations used to calculate the main values (heat transfer, overall heat transfer coefficient and so on) that this thesis requires.

Fig. 2.5.2 Flow arrangements through the hairpin fittings

2.5.3 Thermal analysis of inner pipes

The aim of the thermal analysis is to determine the heat transfer coefficient in the inner pipe. The first calculation to be made is to determine the Reynolds number to find out whether the flow is laminar or turbulent inside the pipe. In order to calculate the Reynolds number, it is essential to have access to the different property tables of the diverse fluids flowing through the heat exchanger pipes. We should be familiar with the following fluid properties:

- Density, ρ
- Viscosity, μ
- Specific heat at constant pressure, c_p
- Thermal conductivity, k
- Prandt Number, Pr

The next step is to calculate the fluid mass flow rate. To do that, we use equations (2.7) and (2.8) from Chapter 2.1:

$$Q = (\dot{m}c_p)_c \Delta T_c = (\dot{m}c_p)_h \Delta T_h \tag{2.46}$$

$$\dot{m}_h = \frac{(\dot{m}c_p)_c \Delta T_c}{c_p \Delta T_h} \tag{2.47}$$

Once we discover the fluid mass flow rate, we must calculate the fluid velocity. The next fluid velocity equation is only applicable for fluids flowing through straight pipes.

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} \tag{2.48}$$

Therefore, the Reynolds number can be calculated from equation 2.27 for fluids flowing through straight pipes as

$$\mathrm{Re} = \frac{\rho_h u_m d_i}{\pi} = \frac{4\dot{m}_h}{\pi \mu d_i} \tag{2.49}$$

The following equation is taken from correlations shown in references. It is used if the flow is laminar. [39]

$$Nu_H = 1.953\left(\frac{Pe_b d}{L}\right)^{1/3} \qquad \text{range of validity is } Pe_b d/L \geq 10^2 \tag{2.50}$$

$$Nu_H = 4.36 \qquad \text{range of validity is } Pe_b d/L \geq 10 \tag{2.51}$$

The range of validity is $0.5 \leq \mathrm{Pr}_b \leq 500$ and $Pe_b d/L \geq 10^3$ where

$$Pe_b = (\mathrm{Re}\,\mathrm{Pr})_b \tag{2.52}$$

If, on the other hand, the fluid is turbulent, a correlation selected from the references will be used. In this case [40]

$$Nu_b = \frac{\left(\frac{f}{2}\right)\mathrm{Re}_b \,\mathrm{Pr}_b}{1+8.7\left(\frac{f}{2}\right)^{\frac{1}{2}}(\mathrm{Pr}_b-1)} \tag{2.53}$$

Where the fanning friction factor based on correlations from the references is [4]

$$f = (1.58\ln \mathrm{Re} - 3.28)^{-2} \tag{2.54}$$

Therefore

$$h_i = \frac{Nu_b k}{d_i} \tag{2.55}$$

2.5.4 Thermal analysis of annulus pipes

The path to follow in the calculations of the thermal analysis of annulus pipes is the same one used in the thermal analysis of inner pipes. The only changes are the equations taken from previous chapters and the correlations taken from the references.

Once again we need the following property values of the fluids: density, ρ, viscosity, μ, specific heat at constant pressure, c_p, thermal conductivity, k and Prandt number, Pr. The same equations as point 2.5.3 are used to calculate the heat transfer, Q, and the fluid mass flow rate $\dot{m}_{h,c}$.

However, everything else changes. The equations associated with the Reynolds number, the fanning friction factor and the heat transfer coefficient are different.

Firstly, the Reynolds number has to be determined, just as for the thermal analysis of inner pipes was done.

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} \tag{2.56}$$

$$D_h = \frac{4A_c}{P_w} = D_i - d_o \tag{2.57}$$

where D_h represents the hydraulic diameter and P_w is the wetter perimeter.

$$\mathrm{Re} = \frac{\rho u_m D_h}{\mu} \tag{2.58}$$

The following equations based on correlations from references are used if the flow is laminar [41]

$$Nu_T = Nu_\infty + \left[1 + 0.14\left(\frac{d_o}{D_i}\right)^{-1/2}\right] \frac{0.19\left(Pe_b \, {}^{D_h}\!/_{L}\right)^{0.8}}{1 + 0.117\left(Pe_b \, {}^{D_h}\!/_{L}\right)^{0.467}} \tag{2.59}$$

Where

$$Nu_\infty = 3.66 + 1.2\left(\frac{d_o}{D_i}\right)^{-1/2} \tag{2.60}$$

If the flow is turbulent, a correlation selected from references for the Nusselt number and for the fanning friction factor will be used. In this case the equations to use are [40]

$$f = \left(3.64\lg_{10}(\mathrm{Re}_b) - 3.28\right)^{-2} \tag{2.61}$$

$$Nu_b = \frac{\left(\frac{f}{2}\right)(\mathrm{Re}_b)\mathrm{Pr}_b}{1+8.7\left(\frac{f}{2}\right)^{\frac{1}{2}}(\mathrm{Pr}_b-1)} \tag{2.62}$$

The equivalent diameter for heat transfer is

$$D_e = \frac{D_i^2 - d_o^2}{d_o} \tag{2.63}$$

Therefore

$$h_o = \frac{Nu_b k}{D_e} \tag{2.64}$$

2.5.5 Calculation of heat transfer rate in a double pipe heat exchanger

Basic equations from Chapter 2.1 will be used to calculate the heat transfer together with the heat transfer coefficient calculated in points 2.5.3 and 2.5.4. These equations are:

$$Q = UA\Delta T_m \tag{2.65}$$

$$\delta Q = -(\dot{m}c_p)_h dT_h = \pm(\dot{m}c_p)_c dT_c \tag{2.66}$$

$$LMTD = \Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{2.67}$$

2.5.6 Calculation of fouling rate in a double pipe heat exchanger

Basic equations from Chapter 2.1 will be used to calculate the fouling together with the heat transfer coefficient calculated in points 2.5.3 and 2.5.4 and the overall heat transfer coefficient, U, calculated in point 2.5.5 with the equation (2.64). The equation is:

$$R_t = \frac{1}{UA} = \frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi kL} + \frac{R_{fi}}{A_i} + \frac{R_{fo}}{A_o} + \frac{1}{h_o A_o} \tag{2.68}$$

2.5.7 Remarks

The equations shown in this chapter are the ones I am going to use in the calculations of the results for this thesis. The double pipe Heat Exchanger situated in the thermo fluids lab will give the hot and cold fluid temperatures along the heat transfer surface.

There are plenty of references, which show the properties values of the fluids used in the double pipe heat exchanger for this thesis. However, I have given several references about specific books and I will attach graphs and tables relevant to this research.

As regards the calculation of fouling rates in a double pipe heat exchanger, I will consider negligible the value of fouling due to water, R_{fi}. Therefore, the fouling rate equation will be:

$$\frac{1}{U} = \frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k} + R_{fo} + \frac{1}{h_o} \tag{2.69}$$

Chapter 3 Rig Design

3.1 Introduction

In order to analyse fouling on the surface of a double pipe heat exchanger, a special rig was designed. The design is based on some preliminary characteristics:

- Easy to access, manage and control.
- Easy to dismantle for cleaning and maintenance purposes.
- Pipe system easily exchangeable to analyse fouling on the heat transfer surface area.
- Simplification of the design as much as possible, due to economical restrictions.

The new rig consists of a hot water system, a new double pipe system, a process fluid system, a cooling system and the adequate measurement instrumentation.

3.2 Hot water system

As mentioned above, one of the main problems in this rig design is cost. Firstly, we have to analyse the devices, instrumentation and facilities that the thermo fluids lab already possesses. One of these devices is an old water-water turbulent flow double pipe Heat Exchanger part of an undergraduate laboratory. There is one part of this heat exchanger that can be taken advantage of: The Hot Water System. [42]

3.2.1 Hot water system specifications

Figure 3.1 illustrates the diagram of the existing heat exchanger and shows its workings and structure. Everything starts in the water-heating tank (number 1 in Figure 3.1). It is a closed stainless steel tank fitted with 2x1.5 KW immersion heaters (number 5) with individual internal high temperature cut offs. The tank is fitted with an external thermostat, a pressure relief valve (number 3) and a water level sight glass (number 2). The filling cap (number 4) is situated beside the pressure relief valve. The hot water flows using a centrifugal brass and stainless steel pump (number 6).

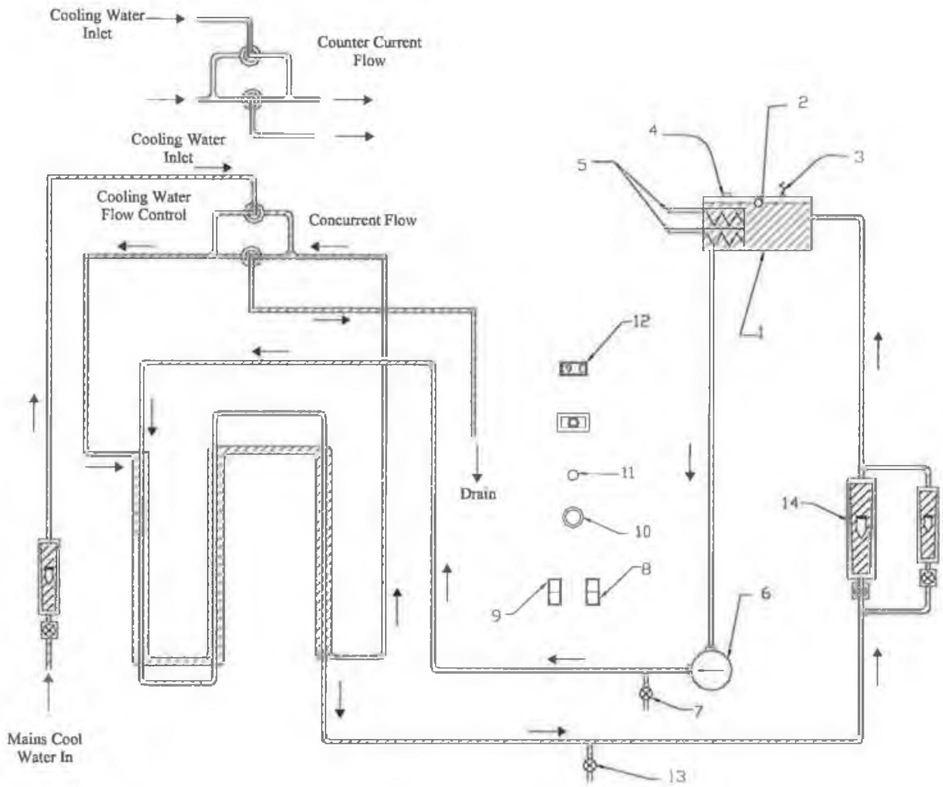


Fig. 3.1 H951 Water-Water turbulent flow heat exchanger diagram

A drain point is kept closed at all times (numbers 7 and 13). It only opens when the system needs cleaning or maintenance work. Finally, valves are provided to control the flow rate. There are two flow meters (number 14) with a variable area type for: Range 4 to 60 $g \cdot s^{-1}$ and 1 to 10 $l \cdot \min^{-1}$ $\left(16g \cdot s^{-1} \cdot to \cdot 167g \cdot s^{-1}\right)$. An electronic control (numbers 10 and 11) regulates the power input to the water heaters and a thermostat, sensing the temperature in the heating tank, limits the water temperature to approximately $90^{\circ}C$.

Some more useful information of the dimensions of this old heat exchanger and the electrical services are required. These are:

Height 920 mm

Width 1060 mm

Depth 430 mm

Either A. 3.1kW 220/240 Volts, Single Phase, 50 Hz. (With earth/ground)

or B. 3.1kW 110/120 Volts, Single Phase, 60 Hz. (With earth/ground)

3.2.2 How the hot water system works

The specifications given in this section can be followed using the Figure 3.1 above. Firstly, the main switch (number 9 in Figure 3.1) must be turned on. This activates the pump and the temperature indicator (number 13). The temperature indicator, which shows the temperatures on the heat exchanger, will not be used in our new rig. Once the pump starts to work, it will produce enough pressure to move the water from the heating tank, through the pump, to the double pipe system, into the control valves and back to the heating tank. The rate of flow volume of hot water moving through the pipes controlled by the control valves (flow meters). Secondly, having checked that there are no leaks in the hot water system pipes and that the level of water in the heating tank is correct, the heating switch (number 8) must be turned on. This will activate the heater that will heat the water in the tank.

Photo 3.1 Existing heat exchanger

3.3 Double pipe heat exchanger system

Another part that could have been used from the old double pipe heat exchanger was its double pipe system. However, there were more disadvantages than advantages that made it unadvisable for the new rig. The main drawback was the inability to dismantle the pipe system so we were not able to take the inner pipe apart from the system and analyse the fouling in its surface. Another drawback, derived from the inability to strip out the pipes, was the cleaning and maintenance of the rig. It was not possible to clean these pipes (inner and annular pipe) by hand, as desired. The only possibility to clean them was to use low acid concentration solution that can produce corrosion in different parts of the rig. [43]

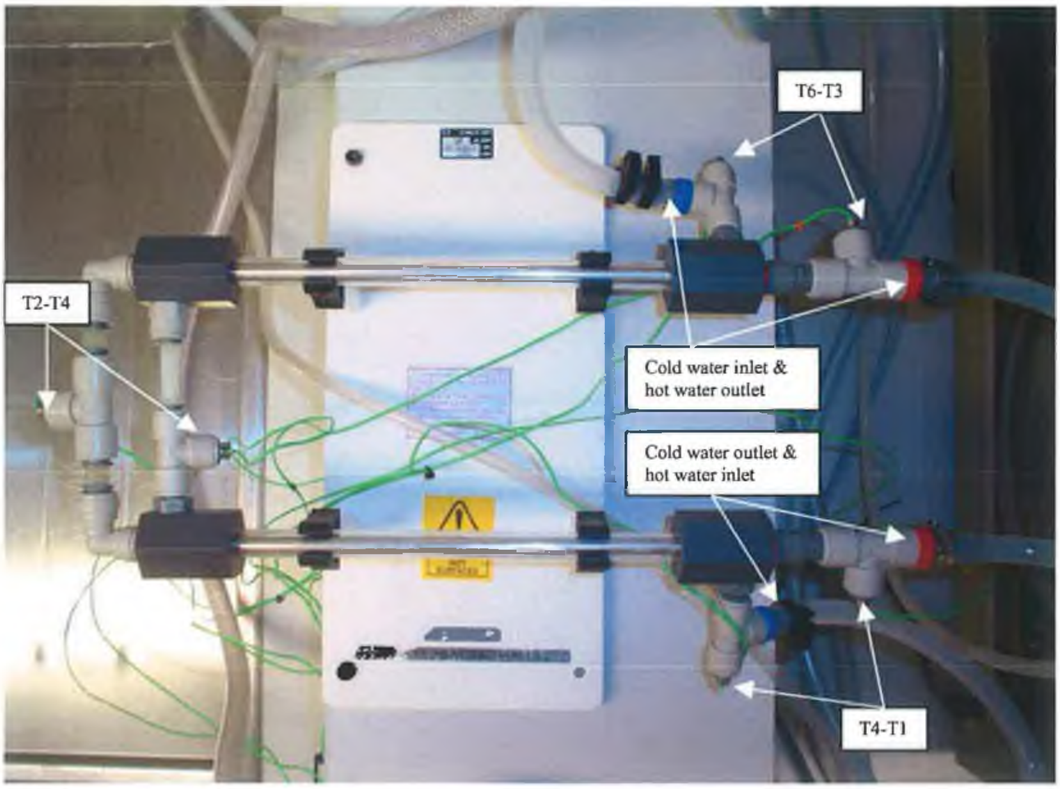


Photo 3.2 HT31 Armfield tubular heat exchanger

The final drawback was the kind of thermocouple that the old double pipe heat exchanger possessed. They were not compatible to connect to a computer, so the data had to be recovered by hand, a task impossible for one person because of the need to take data every minute from six or eight different parts of the double pipe system for more than 6 hours. Therefore, it was decided to use a HT31 Armfield tubular heat exchanger.

3.3.1 HT31 Tubular heat exchanger

This new double pipe (or tubular) system consists of two concentric tubes arranged in series in a U shape, to reduce the overall length and allow the temperature in the hairpin fittings to be measured. The double pipe system is mounted on a PVC base plate that can be fixed in various locations.

In normal operation, the hot water coming from the hot water system flows through the inner stainless steel pipe, and the process fluid passes through the annulus pipe created between the inner metal pipe and the acrylic outer pipe. This arrangement minimises heat loss from the heat exchanger without the need for additional insulation and allows the inside of the annulus pipe to be viewed. PVC housing, bonded to each end of the clear acrylic outer tubes and incorporate O-rings, which close up between each inner and annulus pipes, provide a liquid seal. This allows different expansions between the metal and the plastic parts and the inner pipe can be removed.

The end housing also incorporates the fittings for sensors, to measure the fluid temperatures and a flexible connection to the hot water system and to the process fluid system supplies. The six-temperature thermocouples that the system possesses are labelled T1 to T6 for identification and each lead is terminated with a miniature plug to be connected to the computer.

3.3.2 HT31 Technical details

The heat exchanger technical details are as follows. Each inner pipe is constructed from stainless steel. The dimensions are 9.5 mm outside diameter and 0.6 mm wall thickness. Each outer pipe is constructed from clear acrylic. The dimensions are 12 mm inner diameter and 0.3 mm wall thickness. Each heat transfer section is 330 mm long giving a combined heat transfer area of approximately $20000 \cdot mm^2$.

Temperatures are measured using type K thermocouples. Thermocouples are located at the following six positions.

- Hot fluid inlet T1-Channel 1
- Hot fluid mid-position T2-Channel 2
- Hot fluid outlet T3-Channel 3
- Cold fluid inlet T6-Channel 6
- Cold fluid mid-position T5-Channel 5
- Cold fluid outlet T4-Channel 4

3.4 Process fluid system

The design and characteristics of the double pipe system are shown above, where the heats transfer between two fluids takes place. We have also mentioned the design and characteristics of the hot water fluid, taken from the old double pipe heat exchanger. Only one part is left to complete the construction of a full double pipe heat exchanger; the process fluid system.

The aim of the process fluid system is to circulate the fluid we want to heat (or cool in some situations) through the double pipe system. The old double pipe heat exchanger lacks a process fluid system. It functions using tap water through a connection to the sink, does not have any pump to provide the pressure and no tank to store the water. It uses continuous fresh water. It is therefore impossible to benefit from the old double pipe heat exchanger or any other device in the thermo fluid lab.

It was decided to acquire a Lauda Thermostat Type E 103 circulating water bath. [44]

Photo 3.3 Lauda thermostat type E103

These are the main parts required for the new process fluid system:

- Tank (reservoir): Due to the fact that tap water cannot be used as process fluid because the fouling it produces is nearly negligible, we are forced to use a fluid that will produce fouling in a short period of time. We will be using some dairy products as a process fluid and we will not have unlimited access to them so we need a tank to store the process fluid to use it over and over again.
- Pump: Every system needs a pump that will produce enough pressure to move the fluid around the pipe system.
- Mass flows rate controller: We must be able to control the volume of fluid that enters the double pipe system.

- Temperature selector: It will be greatly beneficial if we can control the temperature of the fluid in the tank.

3.4.1 Lauda thermostat type E103

The E 103 consists of two main parts: The control unit E100 and the type of bath 003. These two parts together produce the Thermostat Type E 103.

The bath 003 is made of stainless steel and is able to tolerate temperatures of up to 150 degrees Celsius. The inner dimensions of the bath are (WxDxH) 135x240x150. It is able to hold a volume of fluid of between 2.5 and 3.5 litres. The bath 003 is supplied with a bath cover made of stainless steel.

The control unit E 100 is supplied with all the other main parts mentioned above, i.e. a pressure pump with various drives, mass flow rate controller and the temperature selector. The pump has an outlet with a rotatable bend, which is connected, to the pump nipple for external circulation. An additional outlet provides circulation inside the bath. By turning the setting knob, it is possible to choose between both outlets or to divide flows. The pump chamber is rotatable in a restricted way to reach optimal circulation. The pump can be used up to viscosities of $150mm^2/s$ during heating. One of the five pump output steps can be selected using the operating menu. The maximum discharge pressure is 0.4 bars and the maximum flow rate of $17L/\min$.

The unit is provided with a 7 segment LCD-Display with additional symbols for indicating bath temperature and settings as well as operating states. The set point is input, and additional adjustments can be made using two or three keys.

There is a tubular heater and a cooling pipe in the unit to make the control of the fluid temperature in the bath easier. The tubular heater can be controlled within the unit. However, the cooling system consists of a tube going through the bath, and so external connections have to be added. At bath temperatures to just above ambient temperature (approximately 10 to 15 Celsius degrees), only it is possible to work connected to a tap water supply. Additional cooling in the way of an external cooling circuit is required to reach lower temperatures.

3.5 Cooling system

In the LAUDA E103 there is a small pipe going through the bath that can be used as a cooler. We will be using fresh tap water as a cooling fluid. The first test we did with all the parts assembled was unsuccessful. The structure of the rig is shown in figure 3.2. The figure shows how the rig works: The hot water flows through the inner pipe in the new double pipe system. The process fluid stored in the bath goes through the annulus pipe, coming back to the bath once it leaves the double pipe system. The small cooling system provided by LAUDA E103 will try to keep the process fluid in the bath at a constant temperature as close to the ambient temperature as possible.

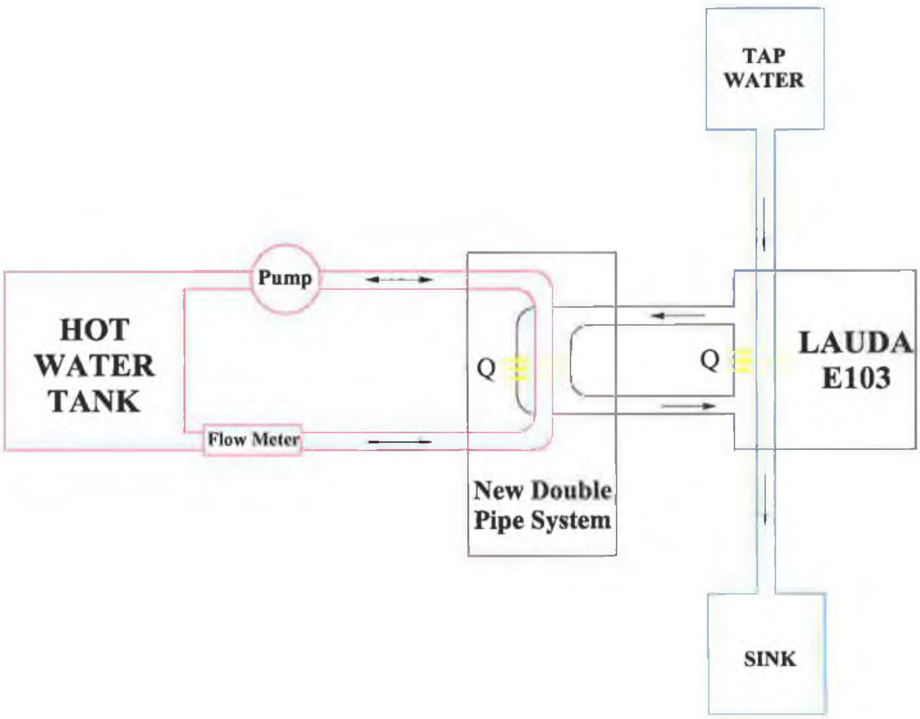


Fig. 3.2 Schematic representation of original rig design

The results were unsuccessful, and the reason is outlined below. As already mention in the above point, the process fluid will be used over and over again. This means that the process fluid leaving the double pipe system will return immediately to the bath. We want to keep the fluid process temperature in the bath constant but this is not possible if the fluid returns to the bath approximately 20 degrees higher than the bath temperature. This problem was not considered beforehand, and it turned out to be the major problem we had to deal with during the design of the new rig. As can be seen in figure 3.3, the bulk temperature in the bath keeps increasing as much as the hot water temperature, reaching inappropriate values for a proper analysis.

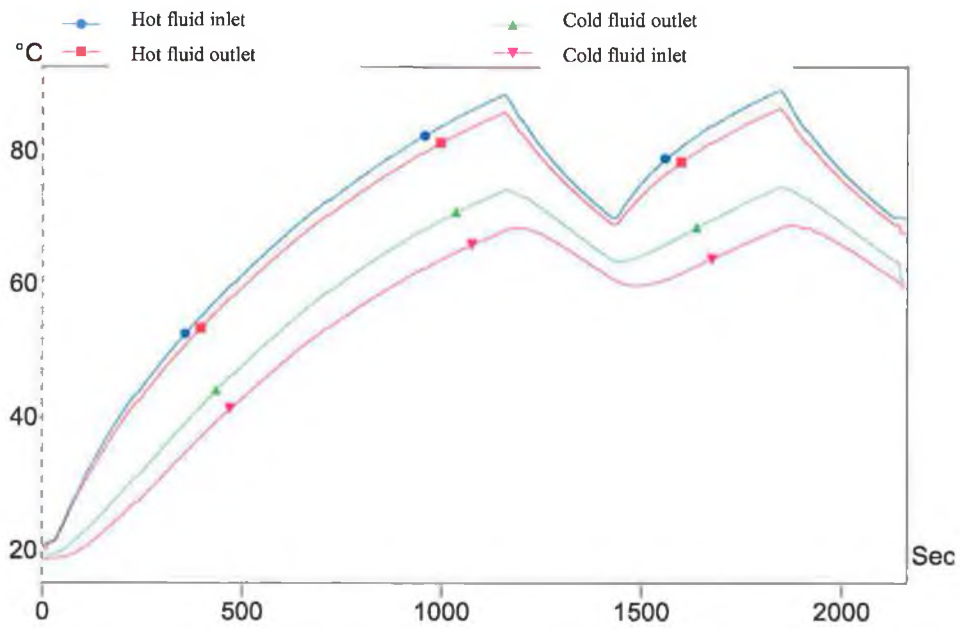


Fig. 3.3 Temperature distributions along the double pipe system corresponding to fig 3.2

The fluid temperature in the bath must be brought to an adequate value in order to analyse it and the temperature in the bath must be kept constant. It was decided to try to take advantage of the existing heat exchanger. Some modifications in its piping were made so the hot water system could be used for both the new rig and the old heat exchanger by adjusting some valves. However, the best solution turned out to be the addition of a new part to the rig: A new cooling system. Here, are some of the modifications we tried and the final decision we made.

3.5.1 First modification: Two fully open parallel valves

In figure 3.4 we can see where the new valves were situated in order to make use of the old heat exchanger: for the student using it in lab experiments and for use in this project. If valves 1 and 3 were opened and 2 and 4 closed, the heat exchanger was set for use by students. If valves 1 and 3 were closed and valves 2 and 4 opened, the heat exchanger was set for use as a research rig.

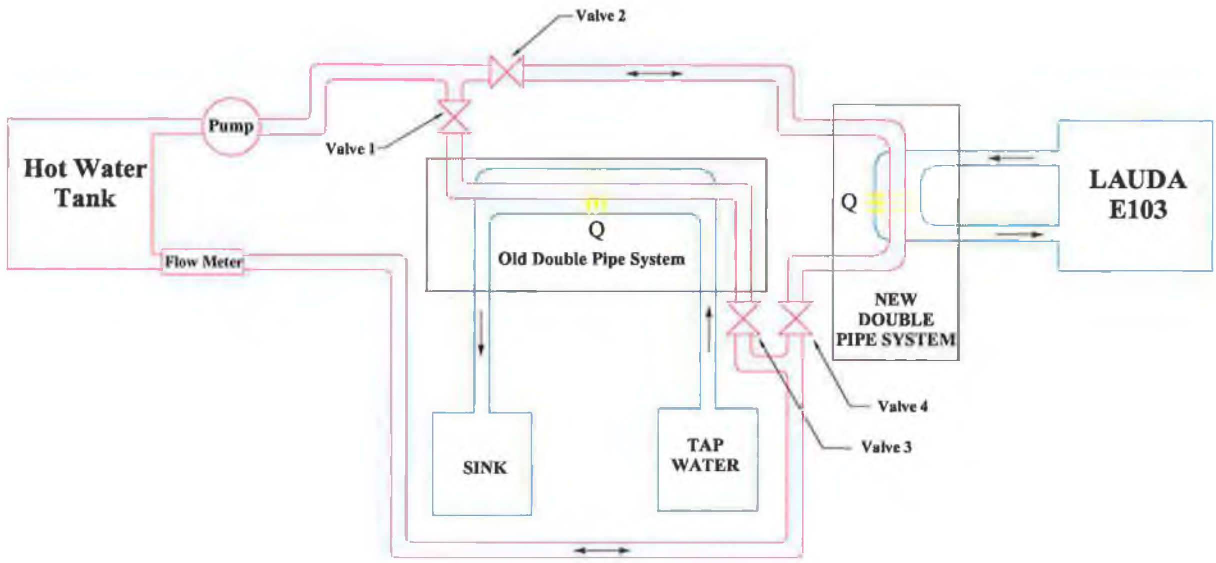


Fig. 3.4 Schematic representation of first modification rig design

The first modification we made was to keep all the valves fully open, so half of the hot water mass flow rate will go to the new double pipe system and the other half will go to the old double pipe system situated in the old heat exchanger. The idea of this new structure was to try to use the old heat exchanger to decrease the temperature in the hot water tank and to try to keep the temperature in the bath at a constant value. The results are shown in figure 3.5.

As can be seen in figure 3.5, we were able to keep the temperature constant in the bath and decrease the temperature in the hot water tank. However, the temperatures with all the valves fully open are too low and therefore, not enough to produce fouling in the annulus pipe. It was possible to reach higher temperatures of up to 50 degrees for hot water by adjusting the flow meters and changing the mass flow rate of both fluids. This however, was not sufficient.

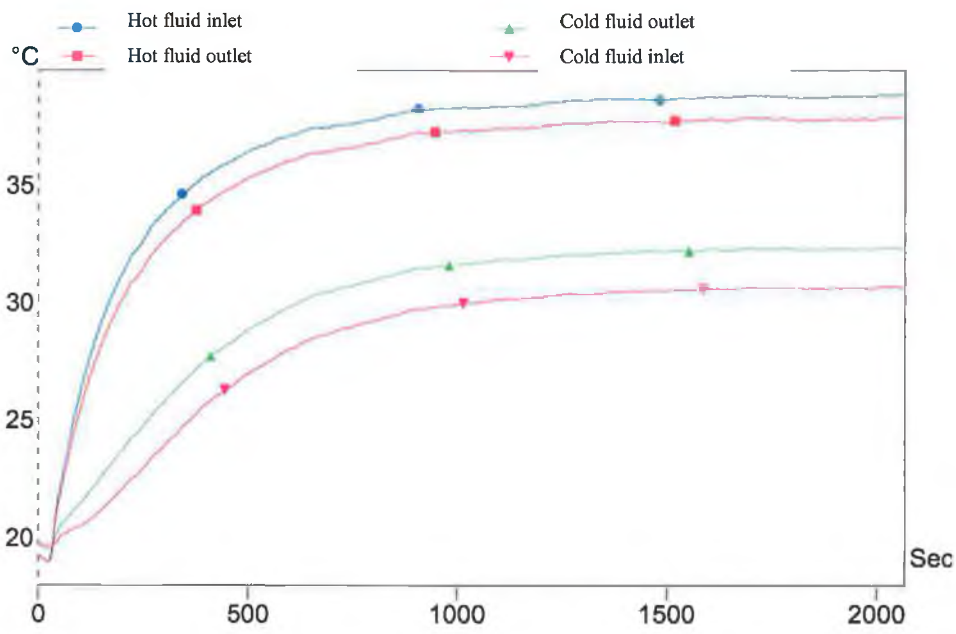


Fig. 3.5 Temperature distributions along the double pipe system corresponding to fig 3.4

3.5.2 Second modification: Two heat exchangers in series

Figure 3.6 corresponds to the second modification made in the old heat exchanger in an attempt to keep the bath temperature constant, and, at the same time, to increase the temperature difference between the hot and cold fluids.

As can be seen in the diagram, the 4 valves we used in the first modification are gone. In this second modification, the rig functions as follows:

1. The hot water is pushed through the new double pipe system by the pump, where the heat transfer between the process fluid and the hot water will take place.
2. Once the hot water leaves the new double pipe system, it enters the old double pipe system, where we will try to cool the hot water before it returns to the tank.

The second modification is quite similar to the first. In the latter, the mass flow rate of the hot water is split in half by means of valves going through two different double pipe systems and joining again at the outlet of each double pipe system. In the second, the whole mass flow rate of hot water goes through the new double pipe system and enters the old double pipe system. By way of an electrical comparison, we can say that the first method is a parallel arrangement and the second modification is a series arrangement of two heat exchangers.

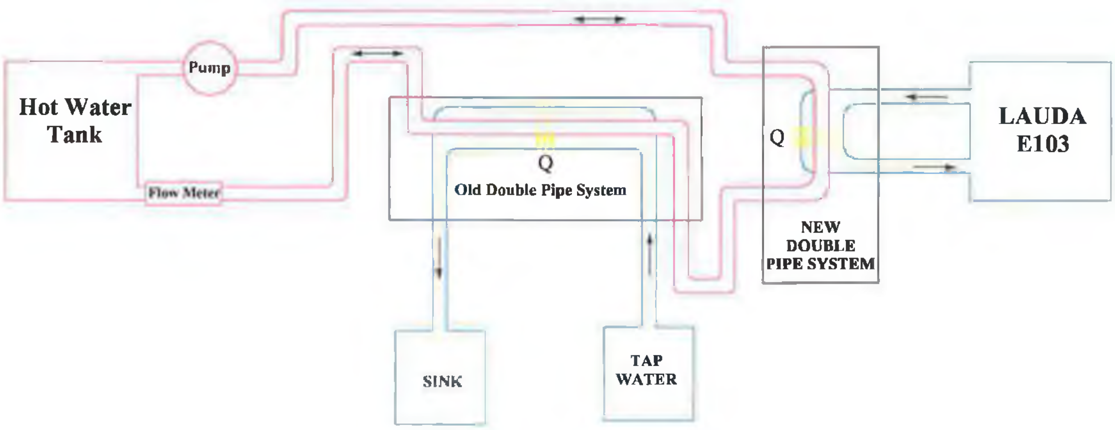


Fig. 3.6 Schematic representation of second modification rig design

The results of this new rig structure are shown in the figure 3.7. As can be seen, the results are nearly the same as those of the first modification. We reach higher temperatures in the second method, but the process fluid temperature is higher also, increasing the temperature in the bath. The temperature difference between the two fluids is not enough to compare our rig with a real heat exchanger in today's industries, and the hot water temperature is not high enough to produce fouling in the heat transfer surface of the new double pipe system.

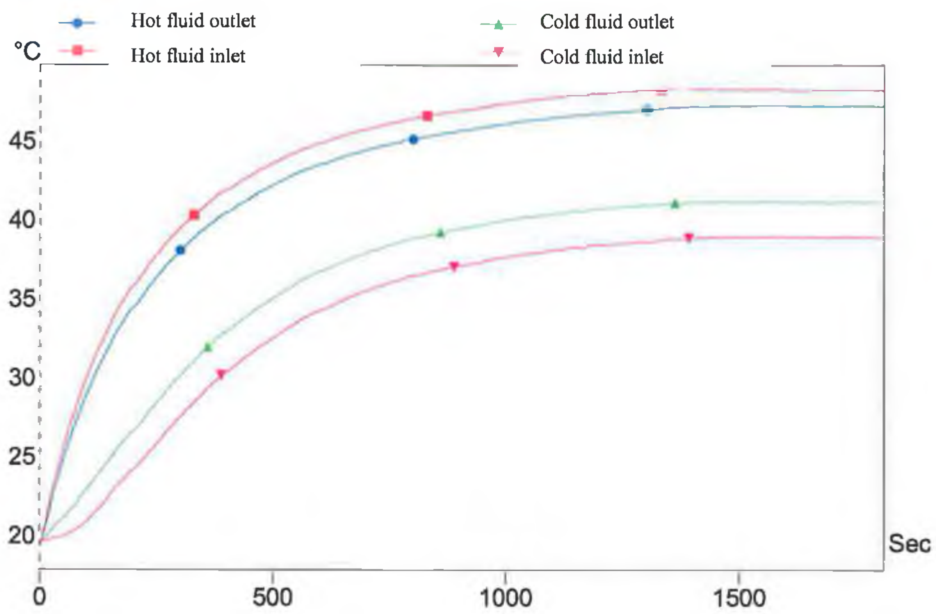


Fig. 3.7 Temperature distributions along the double pipe system corresponding to fig 3.6

3.5.3 Third modification: Adding a new cooling system to the rig

After attempting some modifications in the old heat exchanger pipes, and having tried to fulfil our temperature demands in both the hot fluid and the process fluid with no success, it was time to change our way of thinking. Therefore, we left the old heat exchanger to one side and thought about adding an extra part to the rig.

A device was sourced from the lab, which acted as a condenser and evaporator. We thought that one of these condensers would be perfect to use as a cooling system for the rig. The condenser, shown in photo 3.4, is a glass cylinder of approximately 25 cm in length and 10 cm of diameter with 3 holes in the top lid and one more in the bottom lid. Also, a spiral brass pipe runs through the cylinder pipe. Two of the holes in the top lid are used as the inlet and outlet for the spiral brass pipe.

The condenser was used as a cooler in the following way:

- The cooling fluid was fresh tap water, flowing through the spiral brass pipe. Both the inlet and the outlet of the spiral pipe are situated in two of the top lid holes. The water flows continuously with the aim of keeping it as refreshed as possible at all times.
- The process fluid will be located in the main cylinder. It will enter the cylinder through the hole in the bottom lid and will go out through the remaining top lid hole, returning to the bath.
- We thought that the cooling system would work better if we swapped the location of both the process and the cooling fluids. In this case the cleaning and maintenance of the pipes is easier. Otherwise, if the process fluid goes through the spiral pipe it will be impossible to clean the fouling, building up in the inner surface of the spiral pipe.

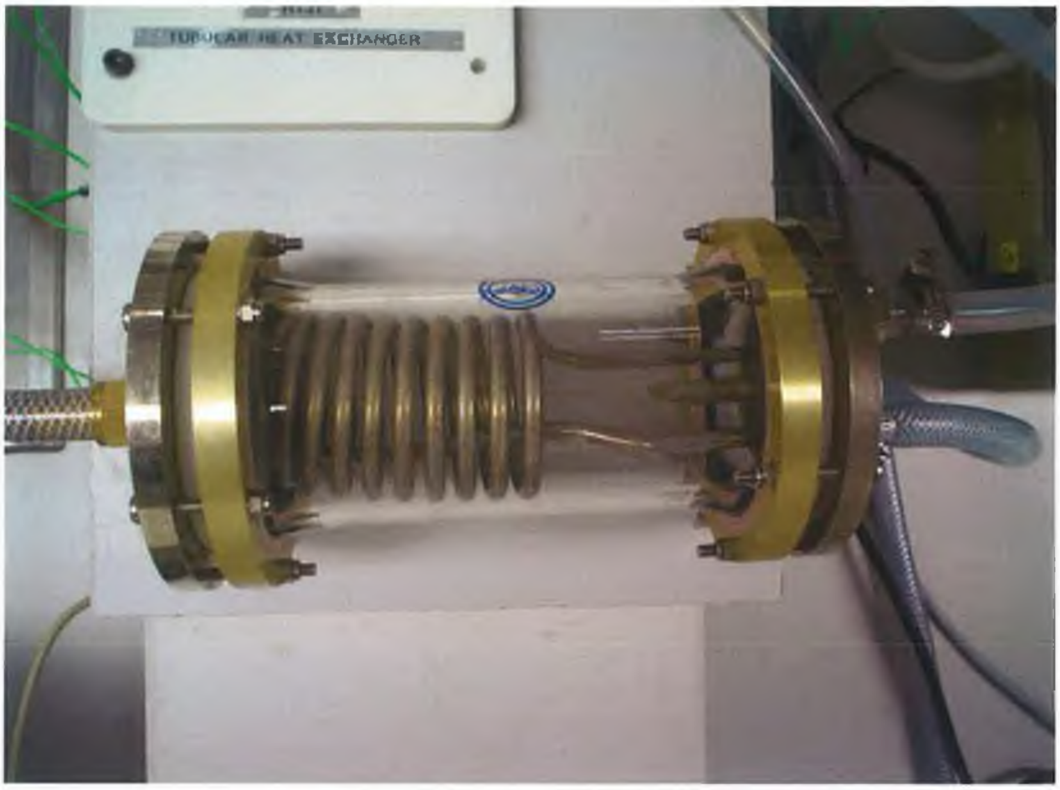


Photo 3.4 New cooling system

Thereby, the new rig structure is shown in figure 3.8, where the old double pipe system is no longer in use and the new cooling system has been added. In this new rig, the hot water will only go from the hot water tank to the new double pipe system and back again to the tank. A pump pushes the hot fluid and the mass flow rate is controlled by a flow meter. The process fluid follows the same path as the last modification, however, because the new cooling system has been added, once the process fluid leaves the new double pipe system, it enters the cooler system and afterwards returns to the bath.

Adding a cooling system to the rig had a significant effect. We almost reached our objective. The hot water temperature value reaches an acceptable high value to produce fouling in the heat transfer area. We met the requirements of keeping the temperature constant for both the hot water and the process fluid. The temperature difference between the two fluids increases by nearly 30-Celsius degrees, making it possible to simulate a real process in industry. These results are shown in figure 3.9.

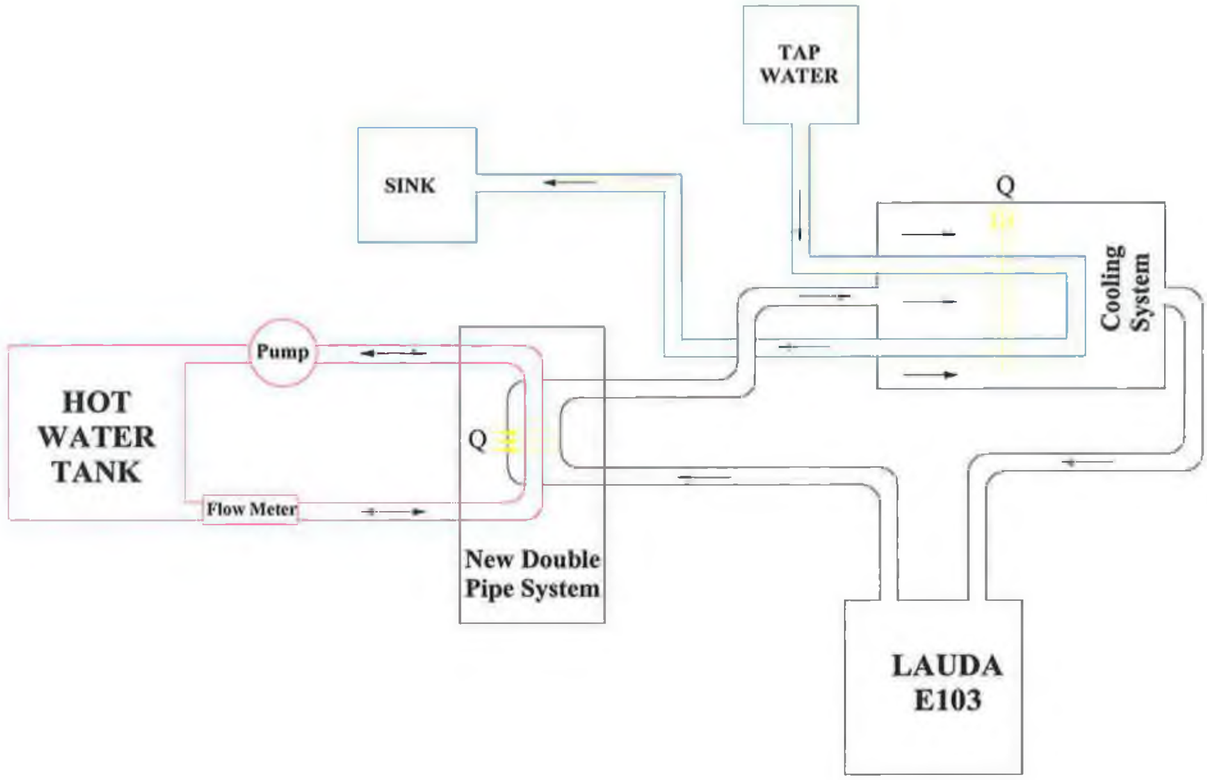


Fig. 3.8 Schematic representation of the third modification rig design

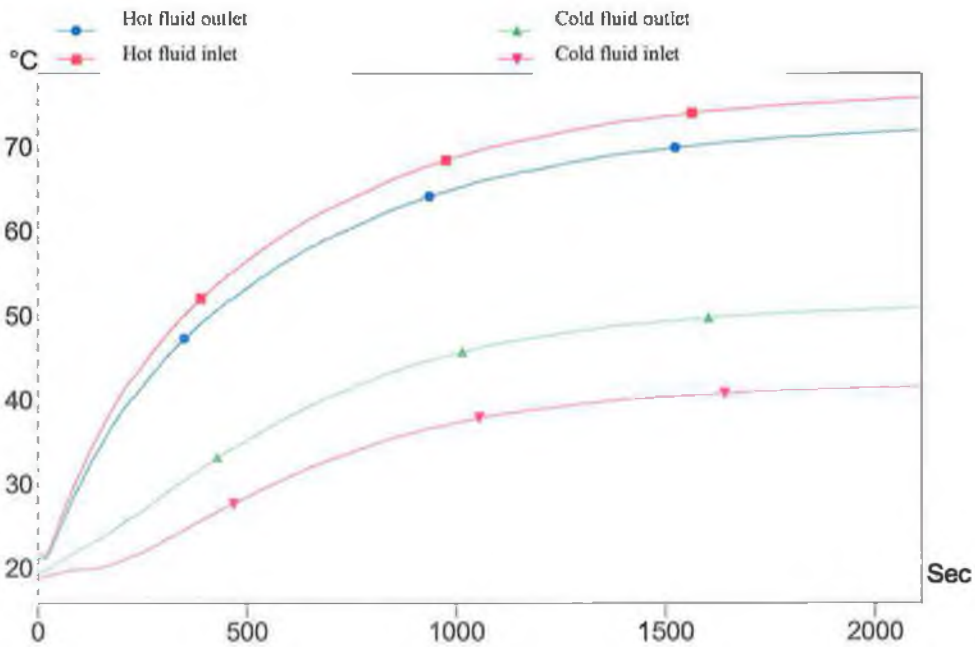


Fig. 3.9 Temperature distributions along the double pipe system corresponding to fig 3.8

3.5.4 Fourth modification: Adding the LAUDA E103 cooling system to the rig

We have two cooling systems and both of them have to be fed with fresh tap water as a cooling fluid. However, there is only one outlet of tap water close to the rig. We have to use the same tap for both cooling systems. The last decision and the final modification made in the rig concerns the cooling systems fittings. They are situated the two cooling system in series. The last modifications are shown in figure 3.10. From the tap, the fresh water flows first through the LAUDA E103 cooling system and then through the new cooling system ending up in the sink.

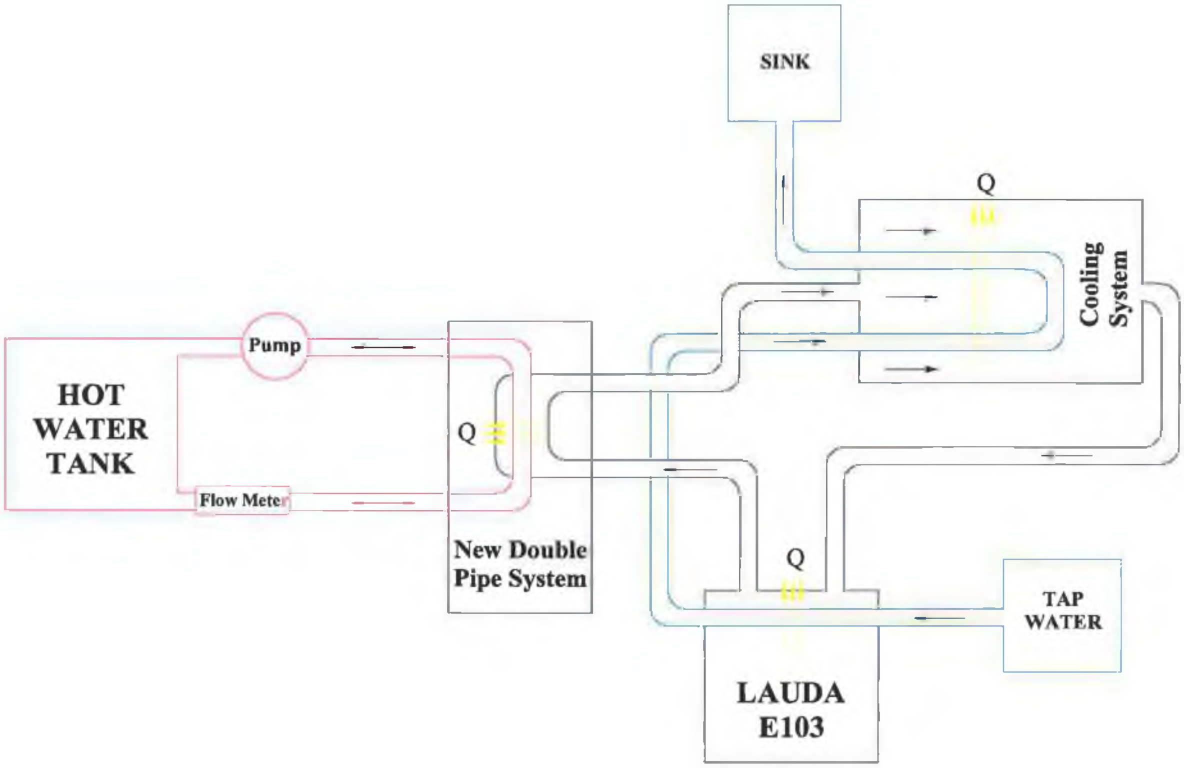


Fig 3.10 Schematic representation of the fourth modification rig design

Once the rig is working with the two cooling systems, we can observe a general improvement. Figure 3.11 shows an increase of hot water temperature. The temperature in the bath decreases. The hot water temperature values almost reach the maximum temperature allowed by the security system before switching off automatically.

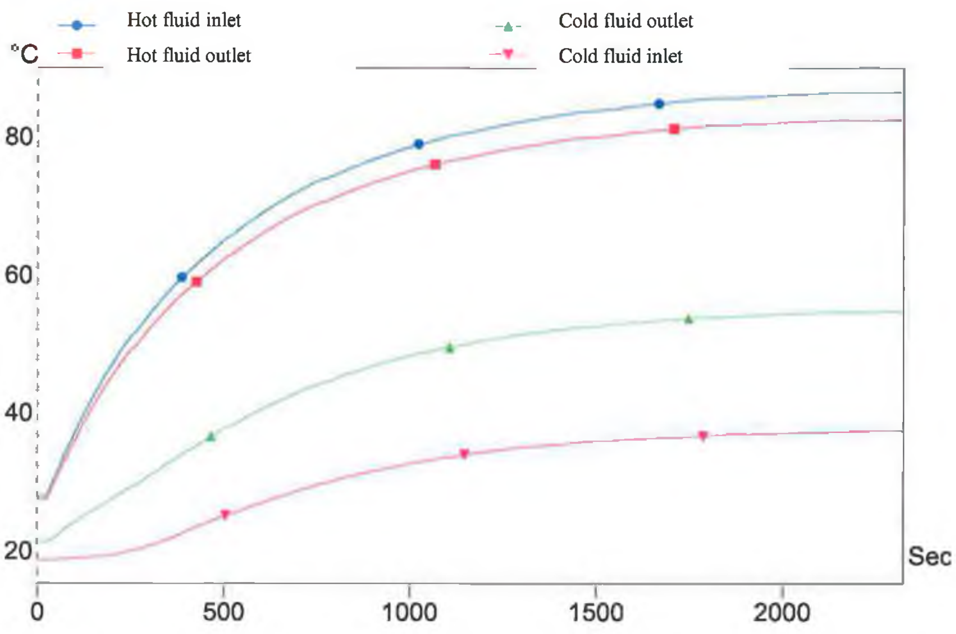


Fig. 3.11 Temperature distributions along the double pipe system corresponding to fig 3.10

3.6 Measurement Instrumentation

The basic tool in the measurement instrumentation was the data-logging computer. We needed exact temperature values for a long period of time to analyse the fouling formation. An average test in this rig would last for more than 5 hours. The new double pipe system includes thermocouples in six strategic points: the inlet, outlet and middle point for both the inner and annulus pipe. With a PICO device and the TC-08 Thermocouple to PC Data Logger, we will be able to transfer the data from the thermocouples to the computer. [45]

The TC-08 comes with PicoLog data software, which allows the computer to display and record temperatures. This data software can collect data rates from one sample per second to one per hour and up to one million samples can be recorded for one simple test. Data can be displayed in graphical or spreadsheet format, both during and after data compilation.

Once the data of a test is collected, it will be transferred to an excel sheet, where all the necessary equations will be previously added to calculate the main rates: Heat transfer, overall heat transfer coefficient, LMTD and fouling.

Photo 3.5 Pico device

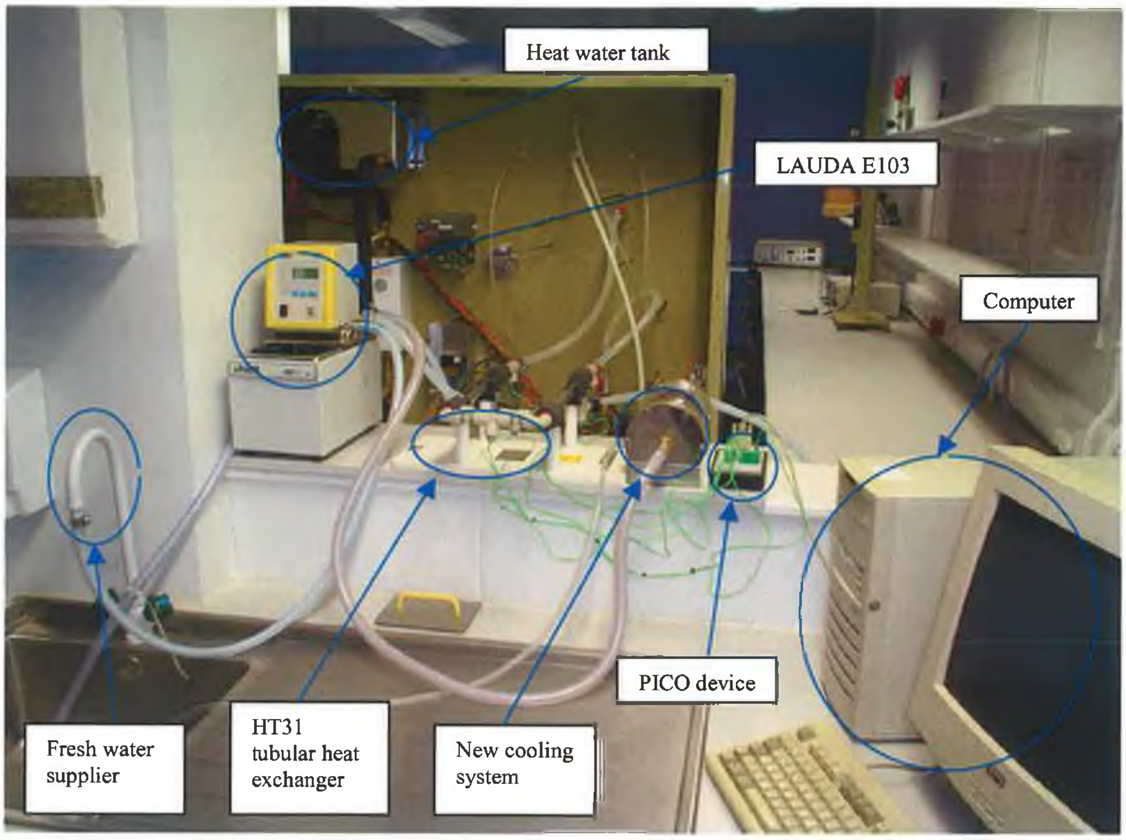


Photo 3.6 Final rig design

Chapter 4 Process fluids selection

4.1 Introduction

Once the rig was modified, it was time to choose the process fluid. In all the previous tests carried out up to now, when the modifications were taking place, water was used for both the hot fluid and the process fluid. However, water does not produce enough fouling for a decrease in efficiency to be noticed.

4.2 Milk. The process fluid

Nowadays, one of the industries most affected by fouling is the dairy industry. Fouling, the undesirable formations of deposits on the heat transfer area, is one of the most unresolved problems of the dairy industry. [46]

In such an industry, both the farms where the milk is taken from the cows and the process industries, where the dairy products are produced, are affected. In the farm, heat exchangers are used to chill the milk to a low temperature to conserve the milk properties while it is transported to the factory. In the dairy factories, heat exchangers are used as both coolers and heaters to produce all kinds of dairy products: from just UHT milk and yoghurts to cheese and ice cream. In both, cooling and heating, fouling is produced and milking equipment has to be cleaned and sanitized on a regular basis, several times per day in some cases, with the consequent additional costs. [47]

The additional costs in the dairy industry caused by fouling are estimated at 260 million euro per year in Europe. [48]

Therefore, the fluid chosen for this thesis is milk. These are some of the reasons why we chose milk as a process fluid:

- The dairy industry is a very important part of the Irish economy. We will later investigate one of the biggest problems in the Irish dairy industry.

- All dairy products produce fouling independent of temperature, and including very low temperatures.
- There are cheap and easy to find.
- The same dairy product can be found in an average supermarket with a lot of variety, therefore we will be able to carry out different analyses for the product.

4.3 Characteristics of milk

Milk is the basis of all dairy products. Milk uses water as the medium in which the other milk constituents are dissolved or suspended. These milk constituents are mainly proteins, fat, lactose sugar and small amounts of various minerals. [49]

Whole milk contains between 3 and 5 % of fat. Milk fat is a liquid above 35 degrees Celsius but below this temperature it tends to solidify and form a film on the surface. Raw milk contains approximately 3.2 % of proteins. Proteins in milk can be divided into caseins and whey proteins. As regards fouling, β-Casein, α-Casein and β-Lactoglobulin are the most important. Minerals are found present in small quantities. Some of the minerals include potassium, sodium, magnesium, calcium, phosphate, citrate, chloride sulphate and bicarbonate.

These quantities are for raw milk. Of course we will not be using raw milk in our tests due to different reasons. For example, we would need to go to a farm to buy the raw milk and we do not have a proper storeroom in the mechanical department to keep the milk. However, pasteurised milk in all its variants, powder milk, evaporated milk and condensed milk, contains the same amount of proteins. Some of them have added minerals, mainly calcium and sugars, and others have decreased proportion of fats.

4.4 Milk fouling

Milk fouling has been studied for a long time. However, it is only partially understood because of its very complex nature. It is a widely studied area and a number of major contributions have been reported. Of course, fouling cannot be eliminated, but strategies to reduce fouling are very valuable. [50,51]

According to Burton, 1967, there are two different kinds of milk fouling, where temperature is an important factor: [52]

1. The deposit formed between 80 and 105 degrees Celsius on the surface of a heat exchanger is mainly proteinaceous. The major protein present in fouling is β-lactoglobulin and the remaining part is composed of calcium and phosphate. The overall composition of this deposit is 50-60% protein, 30-50% minerals and 4-8% fat.
2. At temperatures above 100 degrees Celsius, the deposit consists of 70-80% minerals, 15-20% protein and 4-8% fat. The proteins present in this case are mainly β-Casein (50%) and α-Casein (27%).

Nowadays, it is commonly agreed that fouling can be split into three phases: the induction phase, the transient phase and the severe fouling phase. During the induction phase, no significant changes can be observed. The heat transfer surface suffers small modifications due to minor depositions, not affecting the pressure no the heat transfer values. Depending on the heat exchanger characteristics and on the process fluids properties, a more or less pronounced transient period follows the induction period. In the case of milk, heavy fouling begins for both proteins and minerals when a monolayer of fouling has been formed. [53]

The protein depositions are mainly due to the denaturising of the whey protein β-lactoglobulin. This protein starts to be thermally unstable at temperatures above 65-Celsius degrees. Meanwhile, the mineral precipitation is important for temperatures higher than 100-Celsius degrees. [54]

Another important factor apart from the fluid composition, temperature, and geometry of the heat transfer area, is the presence of air bubbles, which can influence the formation and rate of fouling.

4.5 Cleaning of milk fouling

Many valuable studies have examined the cleaning of milk. An understanding of cleaning is still developing. Milking equipment demands frequent and expensive cleaning due to fouling. Since milk deposits contain proteins and minerals, a two stage wash is frequently used, where the proteins are removed by a caustic solution, typically sodium hydroxide, and the minerals are dissolved by an acid, such as nitric acid. [56,57]

However, in our rig we cannot use any acid or caustic solution. Because after consulting the maintenance department from companies we acquired the double pipe system and the process fluid bath, including pump and cooling system. The companies were not sure that corrosion would or would not appear. We do not want to take any risks. Therefore the cleaning process in our rig will be done by hand, using suitable brushes.

4.6 Process fluid selection

Following this brief introduction to dairy products and their problems in the dairy industry, and due to the wide range of dairy products on the market, we decided to use 3 different kinds of milk-based products for the analysis of fouling in a double pipe heat exchanger. There are: skimmed powder milk, full cream evaporated milk and fresh milk (full fat, low fat and free fat).

4.6.1 Powder milk

Powder milk was the first dairy product we thought of using in our studies. The biotechnology department in Dublin City University carried out some experiments using powder milk in a heat exchanger. The amount of fouling produced in these experiments was huge and it gave us the idea of using powder milk in our rig.

Powder milk is very easy to use. It only needs to be dissolved in water and stirred. To obtain the equivalent of fresh milk, we dilute 57 grams of Marvel dried skimmer milk with 1 pint of fresh water. We can adjust the amount of powder milk added to the dissolution to obtain different concentrations. Table 4.1 shows some of the powder milk properties we will be using in our tests. Powder milk is also very cheap, can be found in any shop and does not need any special cold room to store it.

4.6.2 Evaporated Milk

The reasons why we are using evaporated milk are quite similar to the reasons above. It is very simple to use. We add some water to the evaporated milk, stir and it is ready to use. To obtain the equivalent of full cream milk, we dilute 400 grams of evaporated milk with 0.6 litres of water. It is very easy to find in a supermarket, and does not need any special cold room. We can change the amount of water we add to the solution and compare the concentration results.

In this case we do not have any personal references that evaporated milk produces fouling. However, fouling due evaporated milk is mentioned in some studies and, as a dairy product, the main properties are quite similar, so it should foul the heat transfer area. Table 4.1 shows the evaporated milk properties.

4.6.3 Fresh Milk

Fresh milk is the most basic of dairy products and the closest to raw milk properties. In this case, it is easier to use than powder milk and evaporated milk. It does not have to be mixed with water. The main drawback is that it needs a special cool room. However we decided to use it. The fresh milk was purchased on the day for a single test.

We will use the three variants of fresh milk available in shops: Whole milk, low-fat milk and fat-free milk. See properties in table 4.1 [49]

	Energy $\left({KJ}/{Kcal}\right)$	Proteins (g)	Carbohydrate (g)	Fat (g)	Minerals (mg)	Fibre (g)
Marvel Dried Skimmed Milk	1535/361	36.1	52.9	0.6	1000	0
Milbona Full Cream Evaporated Milk	676/162	8.5	11.7	9	850	0
Whole Milk	272/65	3.3	4.9	3.5	230	0
Low-fat Milk	205/49	3.4	5.2	1.5	240	0
Fat-free Milk	173/41	3.6	5.7	0.1	250	0

Table 4.1 Properties of several dairy products. Information based on 100 grams of each product

Chapter 5 Results and Conclusions.

5.1 Introduction

The first results were shown in Chapter 3. Although these tests were carried out with water-water as process fluid and did not produce any fouling, they highlighted the pathway to follow in rig modifications.

The results shown in this chapter correspond to the use of the dairy products explained in Chapter 4.

5.2 Evolution of temperature with time

The Log Mean Temperature Difference method was chosen in Chapter 2.1 to analyse the fouling in this rig. Therefore, it is necessary to begin by monitoring the temperature along the double pipe heat exchanger.

Once a test is being processed, the PICO software automatically generates a graph of the evolution of the temperature with time in six different points of the double pipe system: three for hot water and the other three for milk.

Despite the graphs being as good as expected, there was a problem. The hot water temperature was brought to almost the maximum temperature allowed for the safety of the old heat exchanger, that is approximately 90 degrees Celsius. If the hot water reaches a temperature higher than the set-point temperature, the hot water tank heaters turn off automatically.

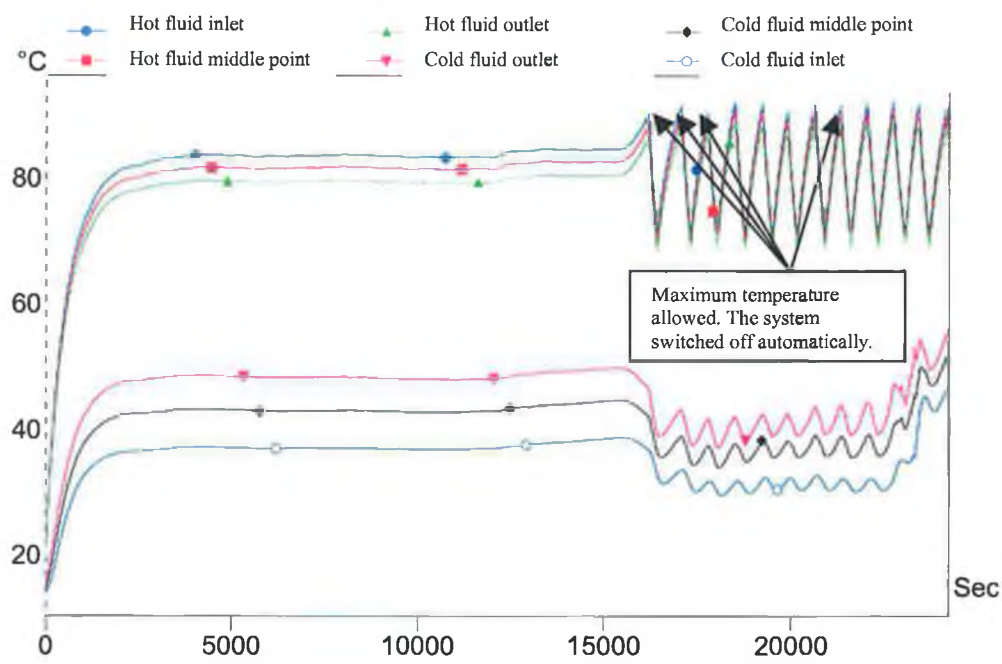


Fig. 5.1 Evolution of temperature for water –milk test, taken with PICO software at 8.5 l/min of hot water and 3.6 l/min of full fat milk

In the test results shown in Chapter 3, which were carried out with water-water processes fluid, this situation does not occur. However, once the tests were running with water-milk, the heat transfer rates were obviously different, due to the different physical properties of milk against water. These different heat transfer rates added to the fouling effect permitted the hot water to reach, in some occasions, the maximum temperature allowed. See figure 5.1.

Figure 5.1 shows the evolution of the temperature of the six heat exchanger thermocouples against time for a typical test of approximately 7 hours. Channel 1 (inlet), channel 2 (middle point) and channel 3 (outlet) represent the evolution of the hot water temperature. Channel 4 (outlet), channel 6 (middle point) and channel 7 (inlet) represent the evolution of milk temperature.

After 4 hours and 30 minutes, fouling grew considerably and started to affect the rates of fluid temperature. The temperature of hot water increased and the temperature of milk decreased. In order to better explain the effect of fouling on temperature, a new graph, figure 5.2, was drawn based on the temperatures of figure 5.1. This new graph is easier to explain and the temperature evolution with time and the different phases of fouling can be seen more clearly. The different phases (mechanisms) are explained in Chapter 2.3.

The graph in figure 5.2 has been divided into 4 parts. The first part, number 1, corresponds to the induction period. It takes between 1 hour and 1 hour and 30 minutes to reach the constant temperatures where the rig works at its maximum efficiency. In this period the fouling is minimum but critical. A fine layer of fouling modifies the smooth outer surface of the stainless steel pipe. In this period, the changes on the main values of heat transfer are minimum and can be negligible.

Number 2 in figure 5.2 is the transportation and light attachment period. In this phase, the milk properties start to change due to the denaturising of some milk protein. These milk proteins are transported from the bulk fluid to the outlet surface of the inner pipe where the attachment is produced. Pressure drop and a decrease in heat transfer cannot be neglected during this period.

Heavy fouling is produced in number 3 in figure 5.2. The evolution of temperature is not constant and an inflection point is produced in the graph for both the hot water and the milk. The hot water temperature also increases. Following the basic laws of thermo fluids, if the temperature of the hot water increases, it should cause an increase in the milk temperature. However, due to the heavy fouling on the surface of the inner pipe, the thermal resistance increases and the milk temperature decreases to low proportions. This period is not longer than 30 minutes. Before this period occurs, the rig should be stopped and cleaned. However, instead of this, we keep the rig working so the next fouling mechanisms can be appreciated.

Points 4 and 5 in figure 5.2 correspond to the removal and blocked phases. Both the hot water and milk temperature maintain a constant value for around 2 hours. In these 2 hours, the exterior fouling is not properly attached to the surface. Due to heavy fouling in the outlet of the inner pipe producing a decrease in cross section area of the annulus, the milk velocity in the annulus increases. Both the increase in milk velocity and the weak fouling attached to the surface, produce some fouling removal, which is accumulated in the cooling system and in the bath. The amount of removed fouling in the bath and cooling system cause, on some occasions, a blockage in the pipes. These blockages can be seen in the graph with an increase in the milk temperature while the hot temperature remains constant.

Even though the effect of fouling causes an increase in the inlet temperature of the hot water by almost 8 degrees Celsius, it is not enough to keep the inlet milk temperature constant at approximately 40 degrees Celsius. It can be appreciated in figure 5.2 how the inlet milk temperature also decreases by approximately 8 degrees.

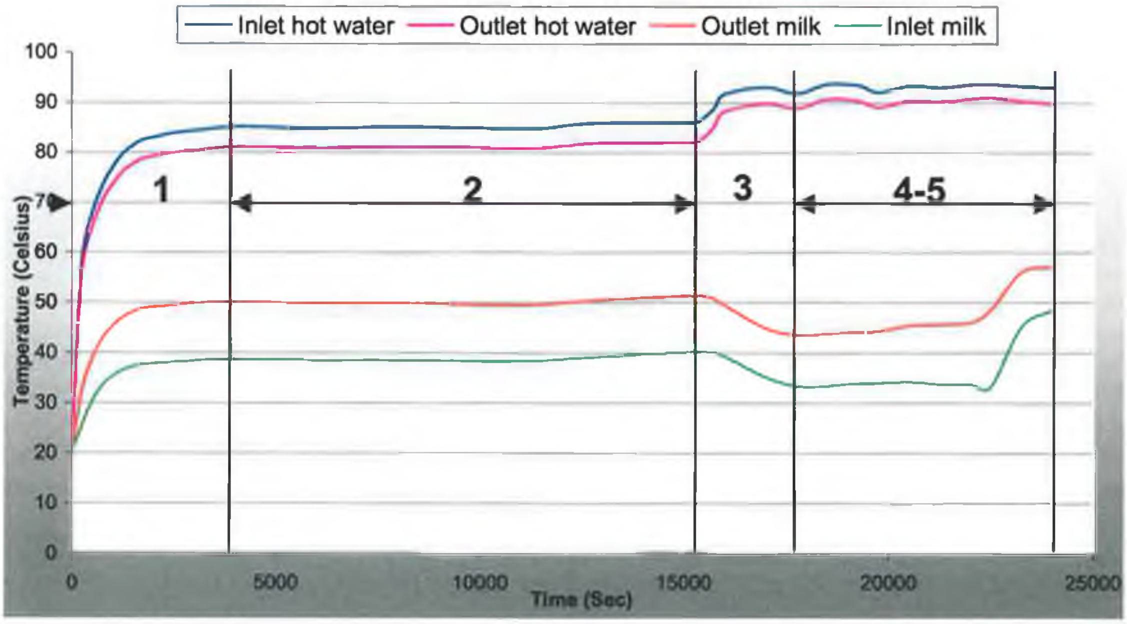


Fig. 5.2 Fouling mechanisms in a Temperature-Time Graph at 8.5 l/min of hot water and 3.6 l/min of full fat milk

5.3 Effect of velocity on heat transfer

Figure 5.3 shows, in red, the increase of temperature between the inlet and the outlet milk thermocouple, ΔT_m, and in blue, the values of the rates of heat transfer, Q, for different flow rates. ΔT_m values are taken after 1 hour of starting the tests, when the induction period is finished and the attachment period has only started. The calculation of Q is based on equation 2.13, where c_p is a constant value, $\dot{m}$ is the fixed value and ΔT_m is taken from the thermocouple notes for different $\dot{m}$ rates.

My first and wrong impression was that, when the flow rate decreases so does ΔT_m, and according to equation 2.13, Q should also decrease. However, figure 5.3 proved the opposite: an increase in heat transfer.

The main reason is that due to the small dimensions of the annulus and inner pipes, the mass flow rate plays a more important role than ΔT_m. Another reason to support the importance of $\dot{m}$ over ΔT_m, is the Reynolds number. In point 5.7 of this chapter, the importance of the Reynolds number influencing the heat transfer is taken into account.

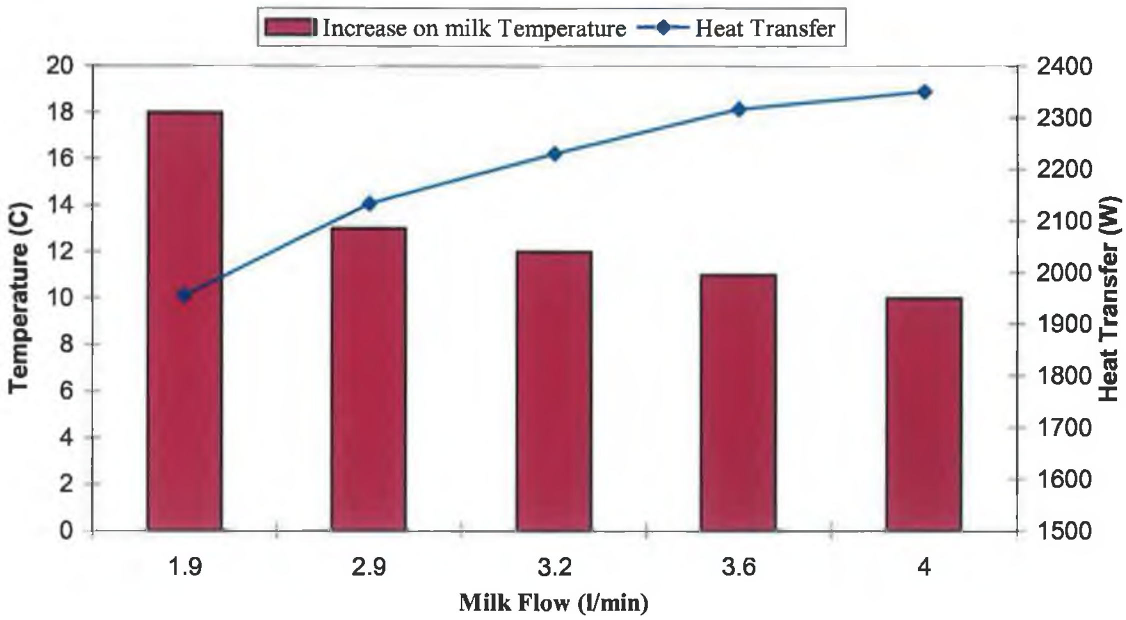


Fig. 5.3 Increase of milk temperature vs. bath pump flow rate & evolution of heat transfer at different milk flow rates

5.4 The effect of fouling on heat transfer

Heat transfer development with time (figure 5.4) must confirm the explanations of temperature vs. time (figure 5.2) and give more details about how fouling affects the rig and its most important value: heat transfer. Three well-defined phases can be appreciated in figure 5.4; the induction, transportation and attachment period. However, the last period, the removal phase, has to be analysed in depth.

In the induction period, (see figure 5.4), the warming up of the hot water tank takes place. Therefore, the rates of heat transfer keeps increasing until it reaches the rates that the rig was designed for. The fouling in this period is nearly negligible and does not affect the heat transfer rates. However, it had a similar evolution for all flow rates. It is at this point when the real values of Q can be appreciated. Maximum heat transfer was produced at maximum milk flow rate, 4 l/min. As the flow rate decreases, Q does also. Therefore, minimum rate of heat transfer corresponds to minimum flow rate, 1.9 l/min.

In the transportation phase, the heat transfer remains at almost constant rates. Fouling starts to affect the heat transfer but cannot yet be taken into account. The small changes that can be seen in this period are mainly due to external factors instead of fouling. One of these external factors is the cooling system. The fresh water used in the cooling system is tap water, provided from a normal sink situated beside the rig. In some stages the pressure of the supply can change, and less or more fresh water enters the cooling system affecting the heat transfer rates.

The third fouling mechanism is attachment, produced by heavy fouling. Figure 5.4 shows an inflexion point in the attachment period. The heat transfer values suffer a significant and constant decrease. If the rig was used in an industrial field, this would be the point when we would stop the heat exchanger and start the maintenance and cleaning of the equipment. Otherwise, we would generate losses in both the economic and sanitation fields. Generally, induction-attachment time decreases with flow rates.

In the last part of the graph, the removal affects heat transfer in different ways. In figure 5.4, 3.6 l/min milk flow rate is the only one in which the heat transfer rates increase after the removal period has started. The main reason for this behaviour is that the removal of fouling is produced quicker than the heavy fouling. For a milk flow rate of 1.9, 2.9 and 3.2 l/min, the decrease in heat transfer remains constant at the same value as the attachment period. For these flow rate values, both removal and heavy fouling are produced simultaneously and in harmony. However, the result will be the same for all of them as shown in the graph below, for 4 l/min milk flow rate, where a critical decrease in heat transfer takes place. This situation is due to the blockage of the pipes. The removed fouling is situated in the new cooling system and in the bath. Therefore, a time will come when the accumulation of removed fouling will block the system.

Other characteristics can also be observed in the graph below. A more gradual decline in performance was produced at low flow rates. On the other hand, high flow rates resulted in an easier removal of fouling and a sudden blockage of the system.

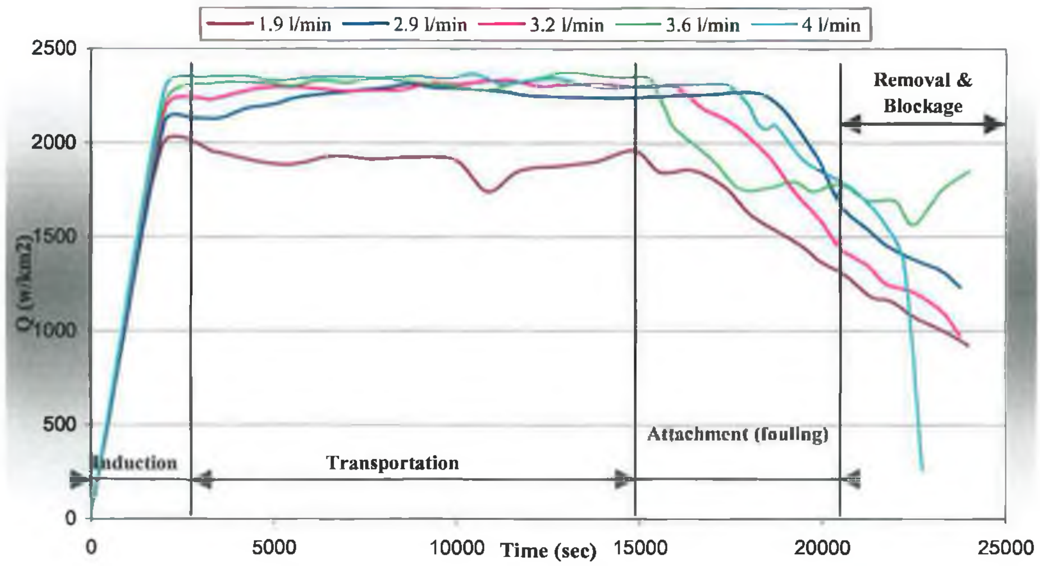


Fig. 5.4 Heat transfer vs. Time Graph for Full Fat Milk at 8.5 l/min of hot Water and Different Milk Flow Values

5.5 Comparison of milk heat transfer and hot water heat transfer

Equation 2.13 is the key to better understanding how fouling affects heat transfer in this rig. This equation is based on the energy balance in a system. Our rig is a two fluid system: hot water and milk. Hot water, expressed with the subscript h, releases heat that is transmitted through the wall to the milk, expressed with the subscript m, which absorbs the heat released by the hot water.

$$Q_h = -(\dot{m}c_p)_h \Delta T_h = \pm(\dot{m}c_p)_m \Delta T_m = Q_m$$

In a perfect system, Q_h and Q_m should be the same. However, some factors such as friction, pressure drop, fouling and insulation, influence the values in a real system. Figure 5.5 shows the evolution of the hot water heat transfer and milk heat transfer in a test carried out with 8.5 l/min of hot water and 4l/min of milk. Once the induction period is finished, both the hot water and milk heat transfers follow the same evolution with time, keeping their values at constant rates.

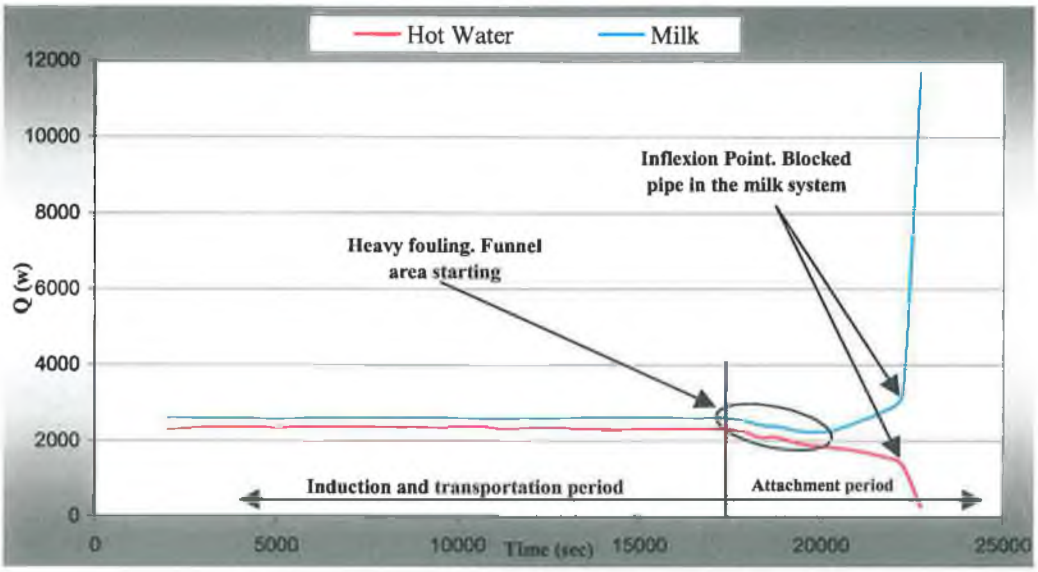


Fig. 5.5 Comparison of hot water heat transfer and milk heat transfer in a test conducted with 8.5 l/min of hot water and 4l/min of milk

It is not until the end of the transportation period, and the commencement of heavy fouling, that the graph undergoes some modifications. These changes take the shape of a "funnel". Figure 5.6 shows an enlarged graph of the funnel area in figure 5.5. In the enlarged graph three periods can be differentiated:

1. Attachment. In this part, both Q_h and Q_m decrease with time in similar proportion, due to heavy fouling.
2. Removal. In this phase, Q_h value continues to decrease while Q_m undergoes a change and begins to increase. The main reason for this behaviour is due to the removal of fouling from the heat transfer surface. It is significant how the removed fouling only affects Q_m and not Q_h. The removed fouling from the heat transfer area is transported by the milk flow to the cooling system and the bath. In these new locations, the removed fouling gets stuck into the cooling system heat transfer surface, decreasing the efficiency of both cooling systems. Consequently, the milk temperature increases, as do the heat transfer values.
3. Blockage. This is the final and critical period, critical because if the rig is not turned off, it can get broken. The milk pipe system gets blocked from the action of the removed fouling in the cooling and bath system. This means that the milk is not moving through the pipes and the consequences are the inlet milk thermocouple, situated close to the bath, takes the bath temperature. The bath cooling system is still working, even though the removed fouling reduces its efficiency. This brings the temperature to almost 20 degrees Celsius. Meanwhile, the milk that is stuck in the double pipe system keeps absorbing the heat liberated by the hot water. Therefore, the temperature in the outlet milk thermocouple reaches the hot water temperature: approximately 90 degrees Celsius. There is a huge difference of temperature between the milk inlet and outlet points, bringing the Q_m rates to absurdly high values. On the other hand, the hot water does not transmit any heat to the milk, bringing the Q_h values to nearly zero.

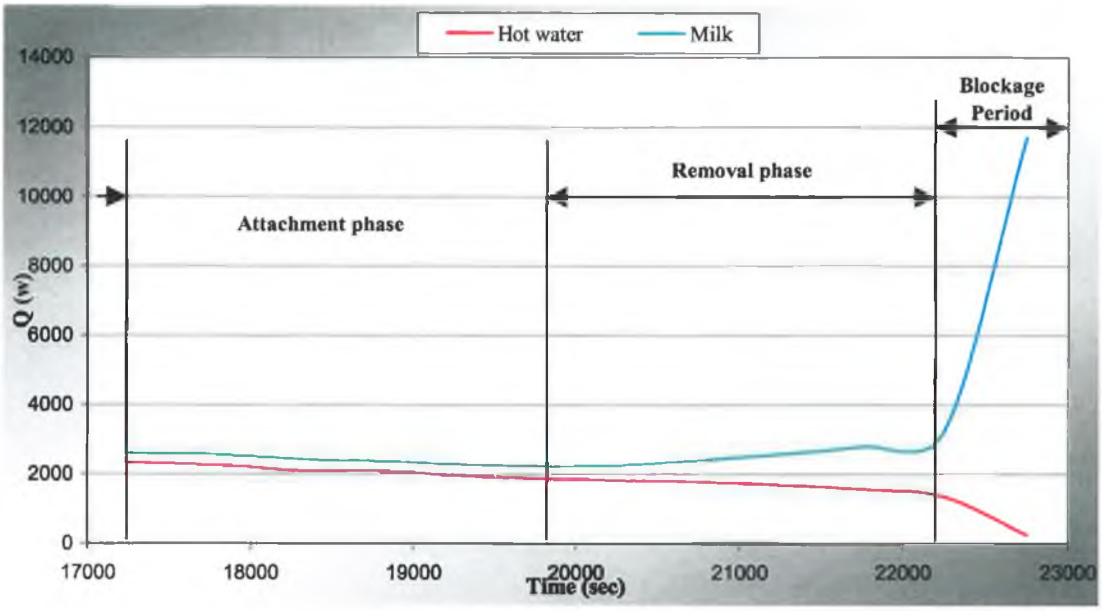


Fig. 5.6 Enlarged graph of funnel area of figure 5.5

5.6 The effect of fouling on the overall heat transfer coefficient, U

According to a study from Belmar-Beiny and Fryer, three phases of fouling can be appreciated in the effect of fouling on the overall heat transfer coefficient. These are: [58]

1. Induction period, in which the fouling is not yet a determinant factor keeping the U-values almost constant. A small increase occurs followed by a rapid decrease.
2. The fouling period. When the heavy fouling starts, U-value decreases to minimum rates. This period can be split into two sub periods: (1) A rapid linear decrease and (2) A slight decrease where U can be considered almost constant.
3. The post fouling period. During this period, U-value is affected by a small increase.

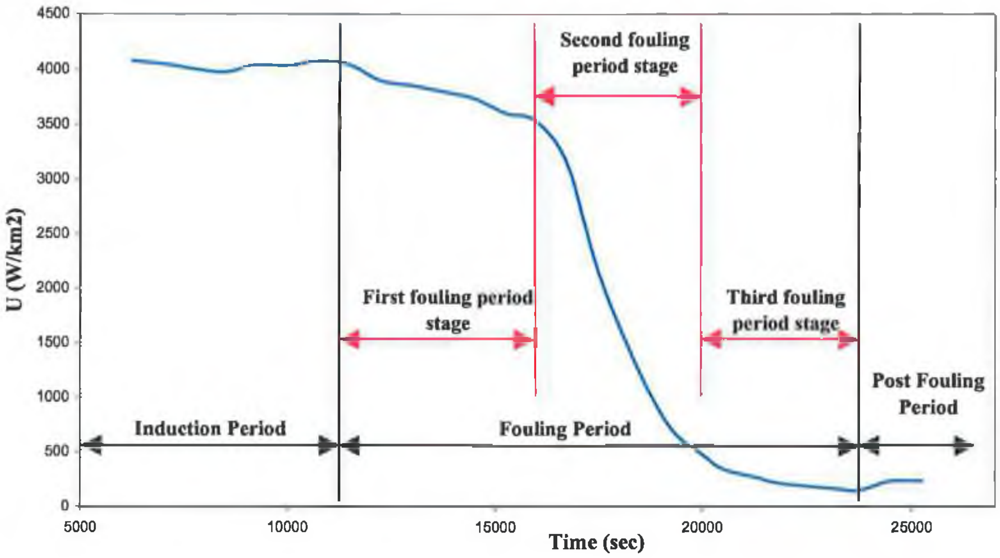


Fig. 5.7 U-value vs. Time for a milk flow rate of 3.2 l/min and hot water flow rate of 8.5 l/min

Figure 5.7 represents the evolution of U with time for a test carried out with full fat milk flow of 3.2 l/min and hot water flow of 8.5 l/min. The three stages related by Belmar-Beiny and Fryer are clearly visible. [58]

Phases 1, 2 and 3 were observed in the experiments. However, period 2 offers some variations. A third fouling sub period stage has to be added. It corresponds to the earlier part of the fouling period and is characteristic of a small but constant linear decrease. Therefore, the fouling period can be split up into three sub periods: (1) a small linear decrease, (2) a rapid linear decrease and (3) a slight decrease.

An obvious relation can be made between the evolution of U and the evolution of heat transfer, Q, with time. Figure 5.8 shows the evolution of both U and Q with time for a test conducted with full fat milk flow rate of 3.2 l/min and hot water flow rate of 8.5 l/min.

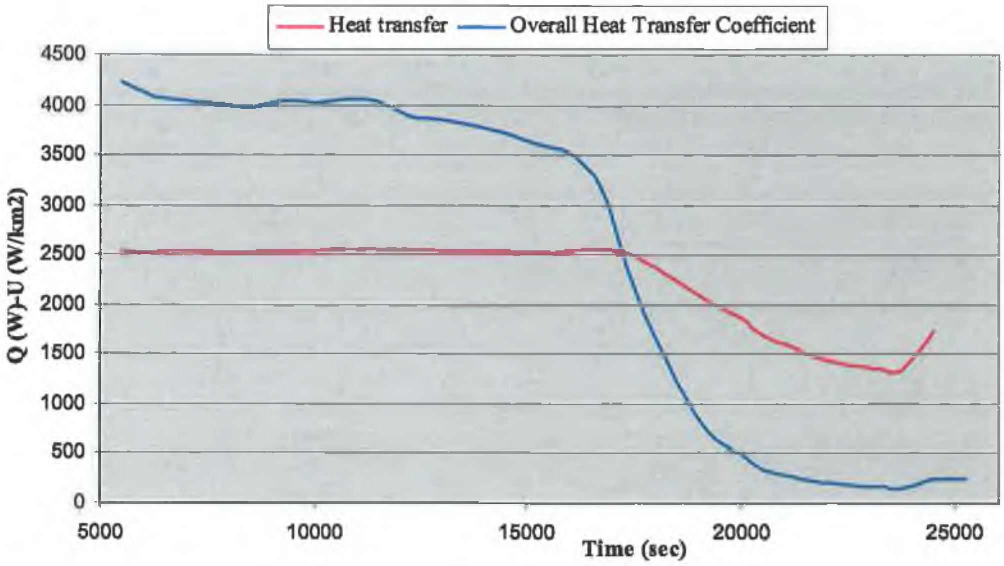


Fig. 5.8 Comparison of U and Q with time for a test carried out with full fat milk flow rate of 3.2 l/min and hot water flow rate of 8.5 l/min

The graph can be divided into four parts:

- In the first part, the induction period of both the overall heat transfer coefficient and the heat transfer, happened at the same time.
- In the second part, the transportation phase was taking place in the heat transfer evolution, while simultaneously, the first stage of the fouling period was happening.
- The third part is when the heavy fouling appeared. At this moment, the attachment period occurred in the evolution of Q and stage two and three of the fouling period affected U evolution.
- The fourth and last part joins the removed period affecting Q evolution and the post-fouling period affecting U evolution.

5.7 Effect of flow rate on the induction-transportation period.

Bird and Fryer carried out a detailed analysis of the effect of flow rate on the induction period. The induction period was shown to increase with increasing velocity and to be strongly dependant on surface roughness. When the flow rate was increased, the turbulent flow held material in suspension and quickly removed fouling attached to the surface, extending the induction period. [59]

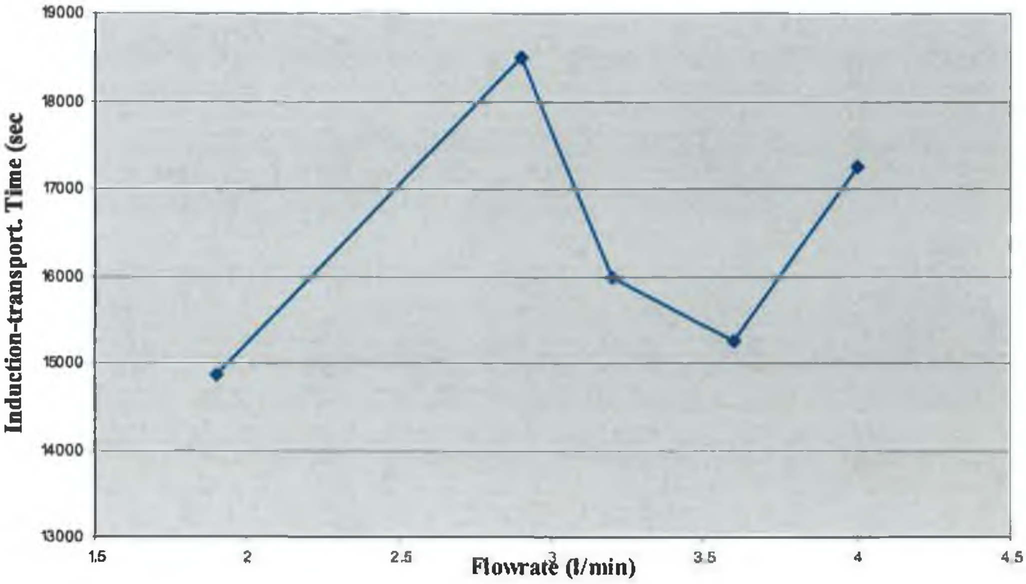


Fig. 5.9 Graph of the effect of flow rate on the induction-transportation period

It should be noted that in the tests carried out in this rig, the induction period is very difficult to determine. Therefore, the transportation period was taken into account because we knew when it was finished. Figure 5.9 demonstrates the effect of flow rate in the induction-transportation period for full fat milk. The results contradicted the majority of the studies in this field. The results did not follow any order to reach a conclusion. It can be seen in figure 5.9 how, for a lowest flow rate of 1.9 l/min, the induction-transportation period came first. However, the last induction-transportation period to be reached corresponds to flow rate 2.9 l/min instead of 4 l/min, producing a contradiction.

5.8 Reynolds number evolution with temperature

The Reynolds number is a very important value in both the analysis of fouling and heat transfer. Once the heat transfer rates Q, the overall heat transfer coefficient, U, and the LMTD are calculated, the only values needed to work out fouling resistance R_f, are h_i and h_o. Both values depend highly on the Reynolds number, as indicated in chapter 2.5.

In Appendix B, the Reynolds number values for hot water can be appreciated when flowing through the inner pipe. The values range go from 27000 at 30 degrees Celsius until 70000 at 95 degrees Celsius. This means that the flow will always be turbulent. The heat transfer between the hot water and the inner pipes wall will be maximum but the risk of fouling formation will be high.

Figure 5.10 shows the Reynolds number vs. the temperature for the milk flowing through the annulus pipe. The first result of such graph is that laminar flow is produced at low milk temperatures. Meanwhile, the milk temperature range in the tests is between 30 and 50 degrees Celsius. At these values, the flow becomes turbulent.

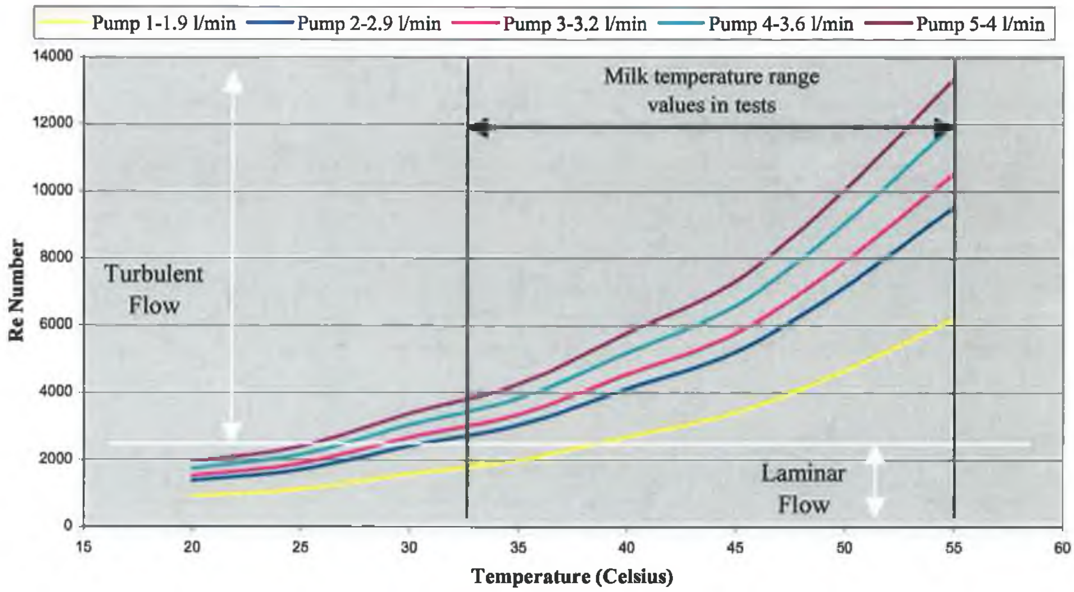


Fig. 5.10 Milk Reynolds number vs. the temperature

When the bath pump is set in number 1 (1.9 l/min) the Reynolds number is at its lowest value. The flow is, in some stages, laminar. For that reason, the heat transfer between the milk and the inner pipe will be the lowest. On the other hand, for a mass flow rate higher than 2.9 l/min, the flow is always turbulent. Therefore, when the milk flow rate increases, so does the Reynolds number, making the flow more turbulent and having a better heat transfer within the milk, which is expressed as higher heat transfer rates.

5.9 The effect of deposition on fouling resistance

Overall heat transfer coefficient values were used by Lund and Bixby to measure fouling resistance evolution with time. A relation was established between the fouling resistance graph and the overall heat transfer coefficient graph. It can be said that the behaviour of both graphs are nearly horizontally symmetric. When the overall heat transfer coefficient decreases with time, fouling resistance was found to increase. The effect of fouling on heat transfer surfaces increases fouling resistance. [60]

Figure 5.11 represents a graph of fouling resistance with time for a test run with 1.9 l/min of full fat milk and 8.5 l/min of hot water. There are no reliable values for the induction period. This is because both the temperatures of the hot water and the milk increased until reaching constant values. After approximately 40 minutes, the temperature values started to be constant and the induction was period considered to be complete. It was at this point when the attachment period started and fouling depositions grew enough to produce some fouling resistance. The attachment period lasted around 3 hours and 30 minutes, where a slight increase in fouling resistance was produced in a linear way. It was not until the heavy fouling started when an inflection point was produced and the fouling resistance rapidly increased, until the test was stopped.

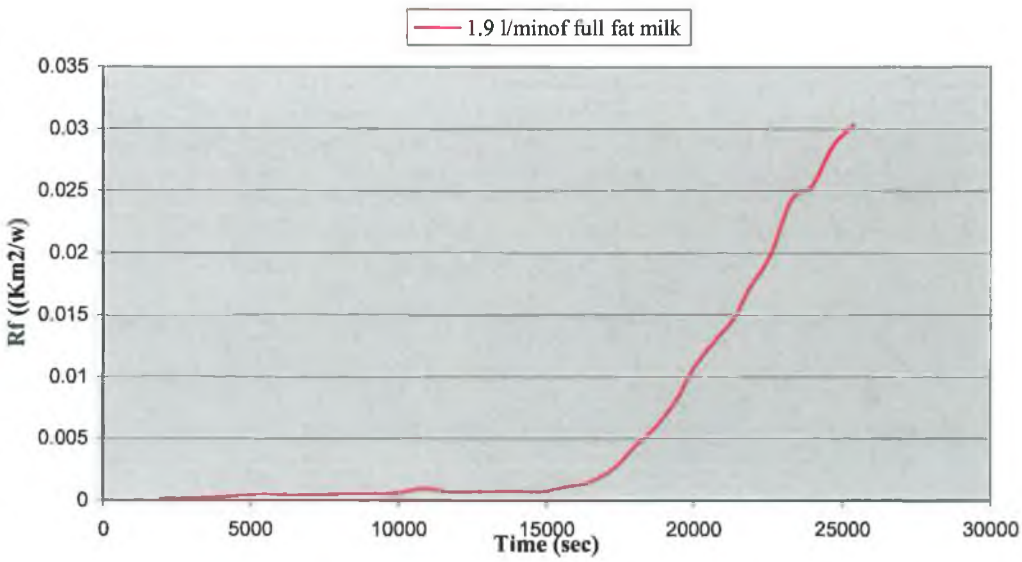


Fig. 5.11 Fouling resistance vs. time for a test run with 1.9 l/min of full fat milk and 8.5 l/min of hot water

5.10 Effect of flow rate on fouling resistance

Figure 5.12 shows the effect of flow rate on fouling resistance for full fat fresh milk. In all the different flow rates, the evolution of fouling resistance in the induction-transportation period followed the same way: a slightly linear increase with time until the heavy fouling period started.

An inflection point was produced for all flow rates when the heavy fouling started. This was represented by a rapid linear increase. However, the main differences were produced in the heavy fouling period. The lower flow rates, from 1.9 l/min to 3.2 l/min, maintained their rapid increase while the rig was on. On the other hand, higher flow rates values, from 3.6 l/min to 4 l/min, were characteristic of a short but rapid increase lasting 40 minutes. After that period of time, the reduction of the cross sectional area of the annulus and the high flow rates produced an important increase in the fluid velocity. The result was equilibrium between the created fouling and the removed fouling. This is represented in the graph below as constant fouling resistance values.

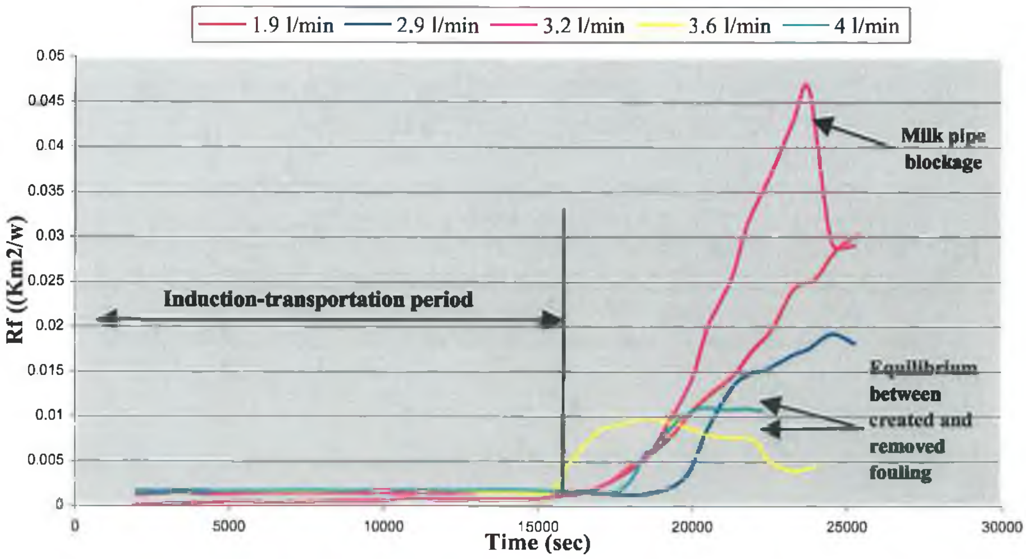


Fig. 5.12 Effect of flow rate on fouling resistance vs. time graph

The last part of the graph shows, for some of the flow rates values, a decrease in fouling resistance. For flow rates of 3.6 l/min and 2.9 l/min this was because more fouling was removed than created. However, for a flow rate of 3.2 l/min, a rapid decrease was produced. The main reason for this behaviour was the blockage of the milk pipes due to fouling which produced quick and erroneous changes on the fluids' temperatures.

5.11 Evolution of fouling on the heat transfer surface

Fouling in a heat exchanger surface follows the mechanisms or phases explained in Chapter 2.3: Induction, transport, attachment, transformation and removal. However, external factors can appear that affect the way fouling grows on the heat exchanger surface. In our rig, this external factor includes air bubbles.

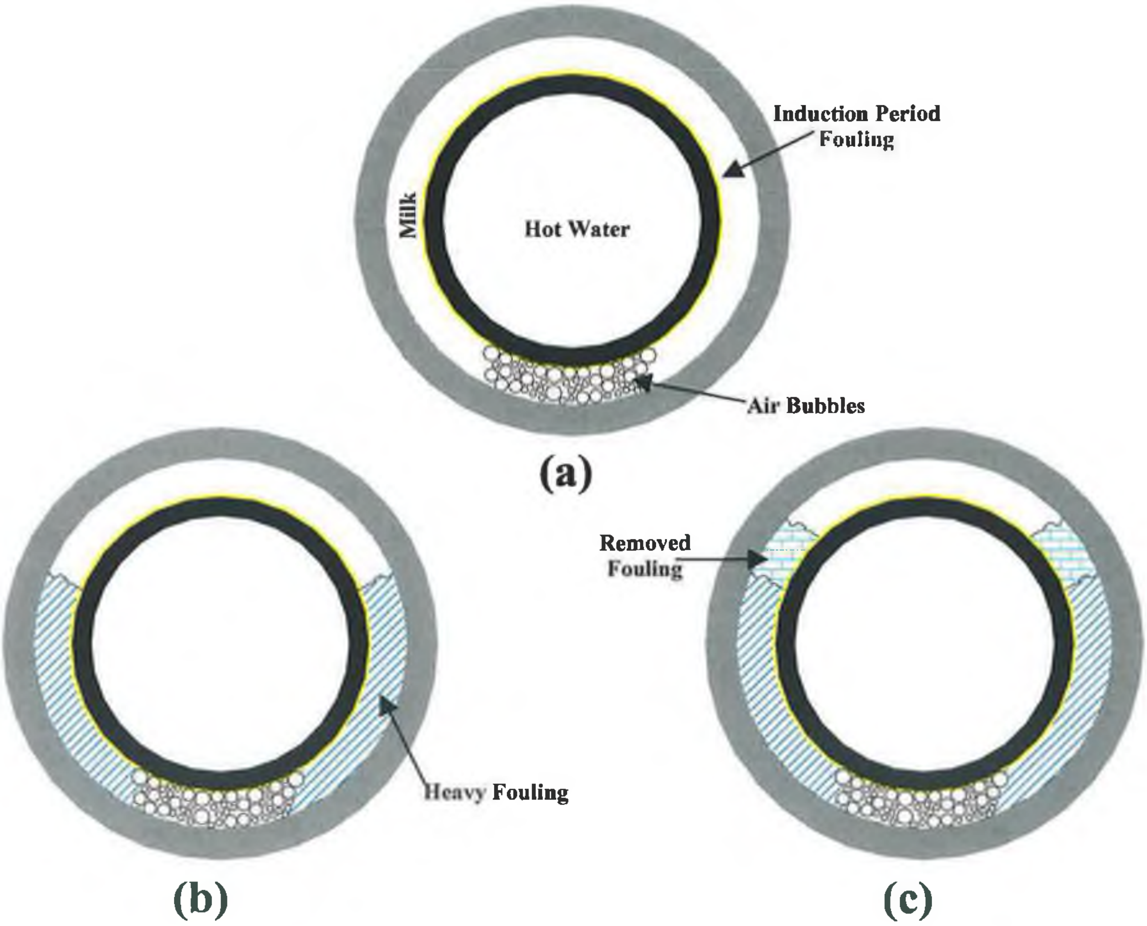


Fig. 5.13 Growth of fouling in the double pipe cross sectional area

Figure 5.13, shows a scale drawing of the cross sectional area of the double pipe heat exchanger. In drawing (a) of figure 5.13, it can be seen how fouling grows in the first two phases: induction and transport. In this period, between 3 and 4 hours since the test started, a thin layer of fouling is formed around the outside of the inner heat pipe (yellow annulus circle). Meanwhile, small bubbles, no more than 1mm of diameter, started to scale up in the lower part of the annulus pipe.

In normal conditions, the bubbles should pass through the annulus pipe without getting stuck, because between D_i and d_o there is a distance of 1.75 mm and the bubbles are small enough to get through. However, the annulus pipe, made from clean acrylic, gets affected at high temperatures and changes its shape from a straight tube to a lightly bent pipe. This small modification on the annulus pipe shape changes the geometry of the double pipe cross sectional area.

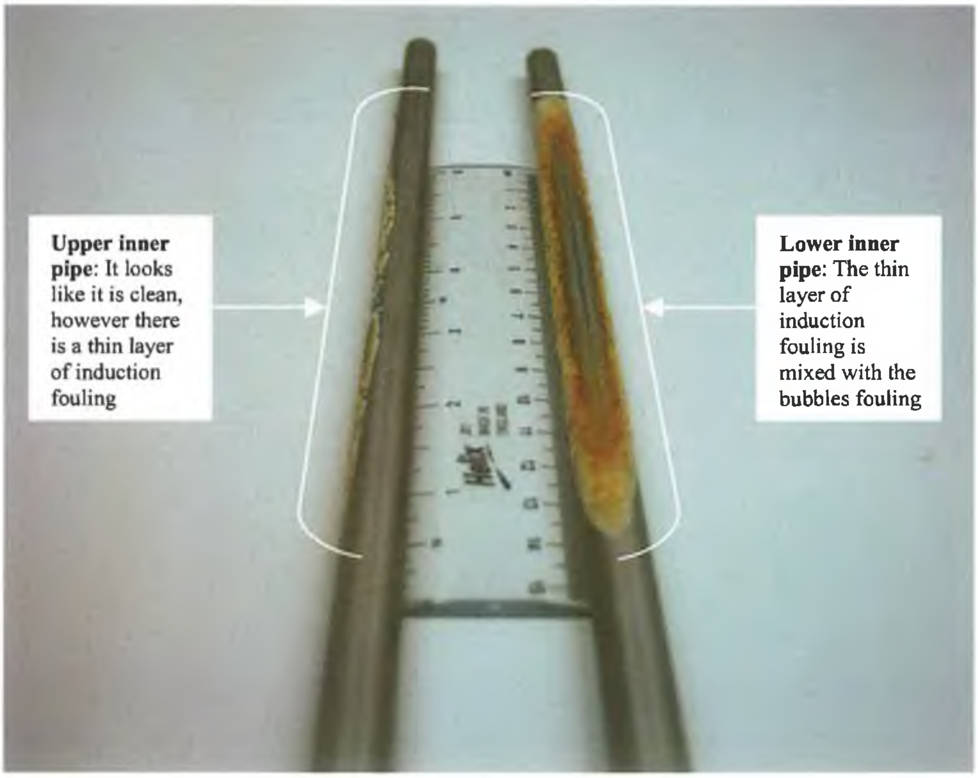


Photo 5.1 Bubble fouling on the heat transfer surface for a test carried out with full fat fresh milk

Due to this small change in the geometry, the distance in the lower part of the double pipe between D_i and d_o is not bigger than 0.5 mm. In some parts, the inner and annulus pipe are even in contact, promoting the stagnation of bubbles. On the other hand, the distance in the upper part of the double pipe between D_i and d_o is bigger than 2.5 mm. Therefore; no bubbles will get stuck in this part.

Drawing (b) of figure 5.13 shows how fouling grows in the attachment period. This phase is not longer than 1 hour, however, heavy fouling is produced. The first signs of fouling are produced around the air bubbles, and it expands around the inner pipe, blocking the middle and lower part of the annulus pipe to the milk flow. Only a small area of the initial cross section area is left in the upper part of the annulus pipe.

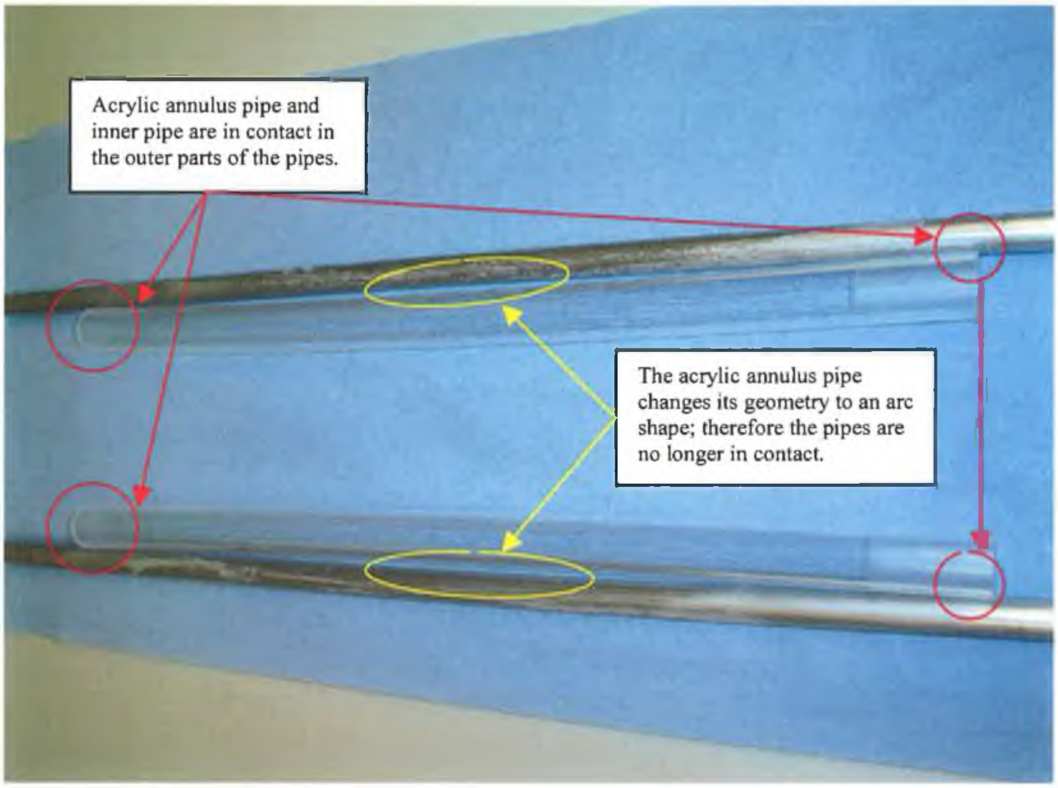


Photo 5.2 Inner pipe and bended acrylic annulus pipe

One more problem was noted: the heavy fouling is weakly attached to the surface, the cross sectional area has been reduced by two thirds and, because of this, the velocity of the milk flow increases. Once all of these conditions exist at the same time, drawing (c) of figure 5.13 appears. The heavy fouling attached to the heat transfer surfaces situated in the upper part of the double pipe gets removed and is carried out to different parts of the rig. The removed fouling scales up in the new cooling system and in the bath. After less than 2 hours, the removed fouling in the cooling system and in the bath reaches a large scale, producing at some stages, the blockage of the whole milk system.

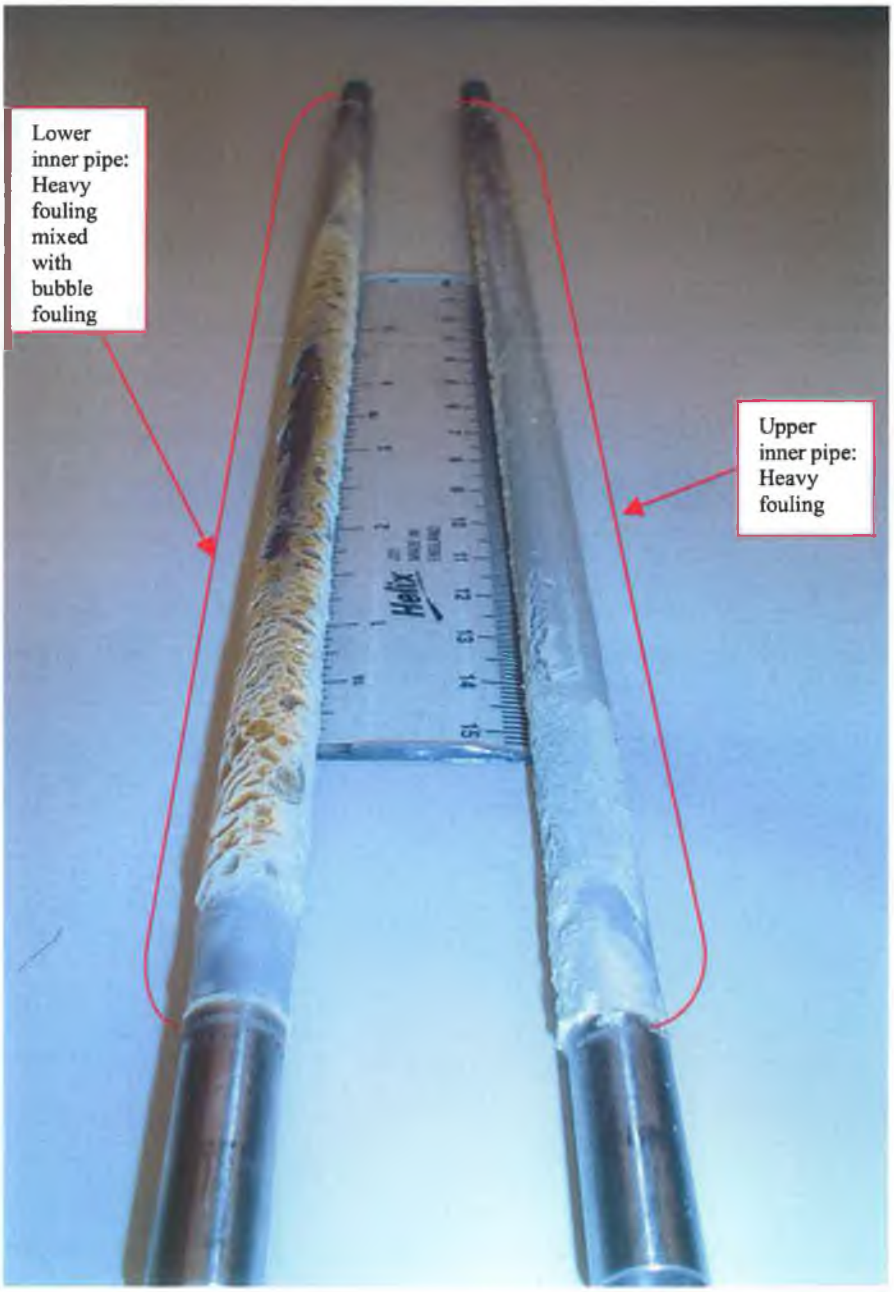
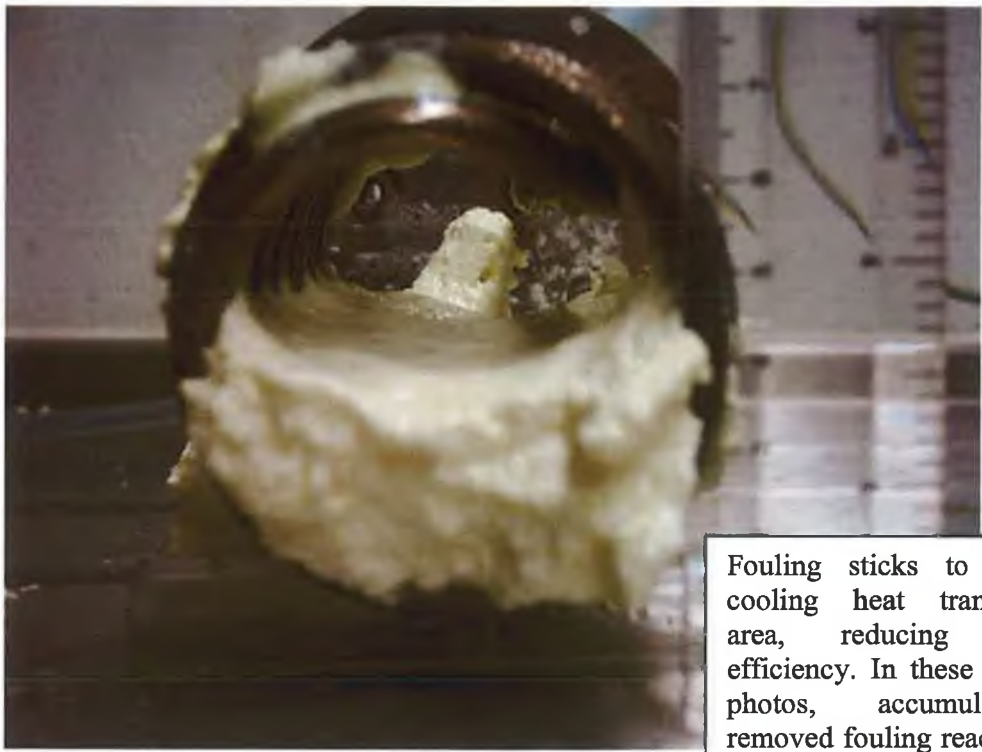
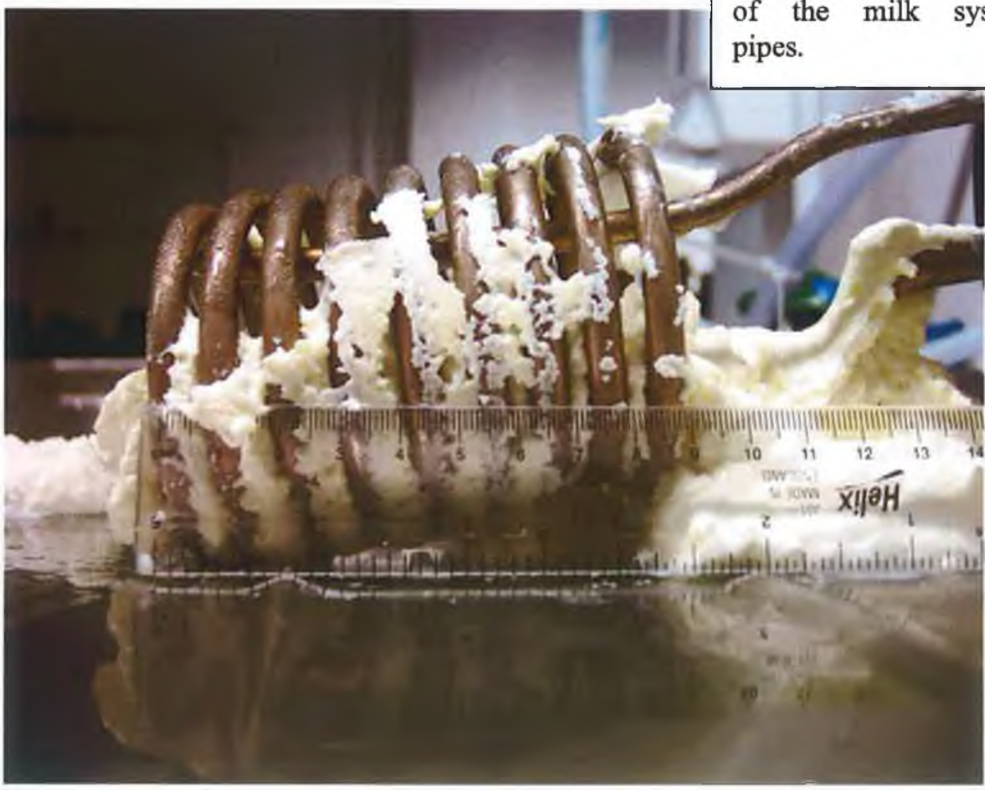


Photo 5.3 Heavy Fouling



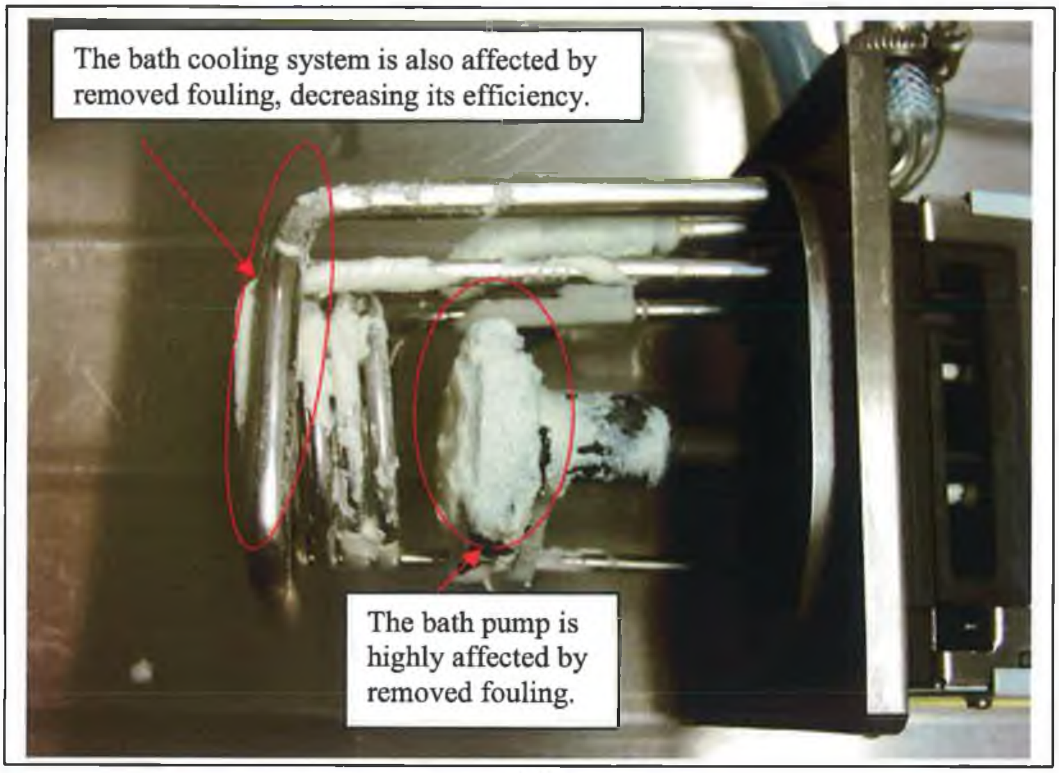
Fouling sticks to the cooling heat transfer area, reducing its efficiency. In these two photos, accumulated removed fouling reaches high proportions, with the consequent blockage of the milk system pipes.

(a)

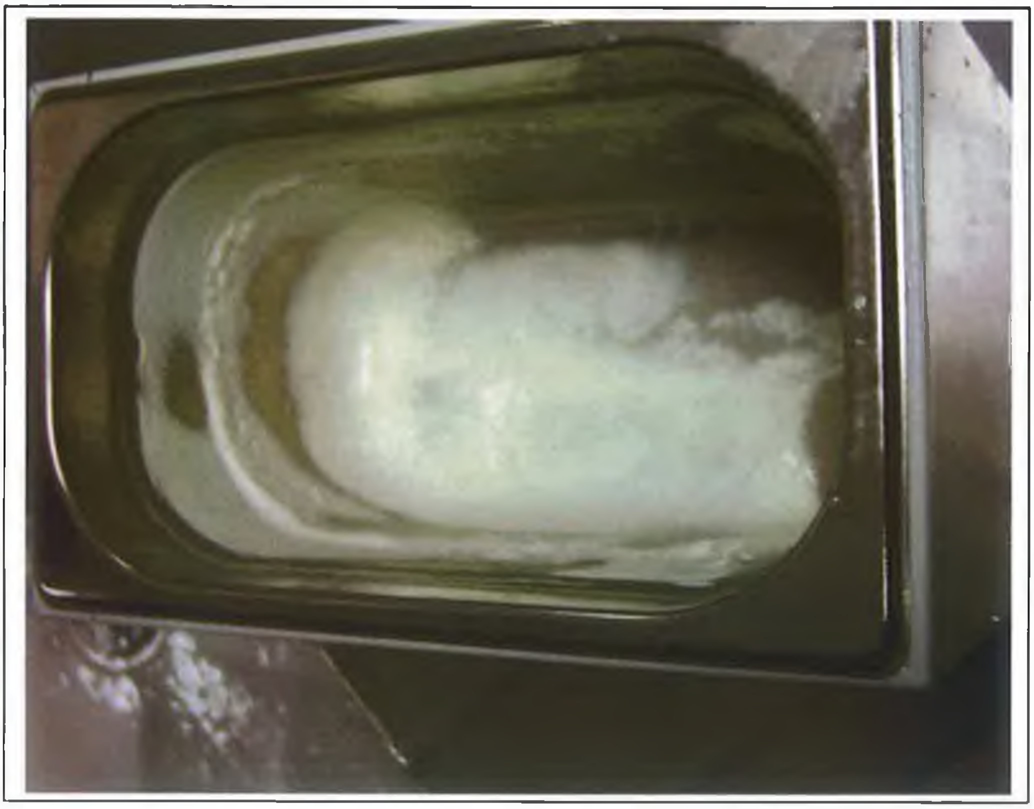


(b)

Photo 5.4 (a) Front view of fouled new cooling system. (b) Side view of fouled new cooling system



(a)



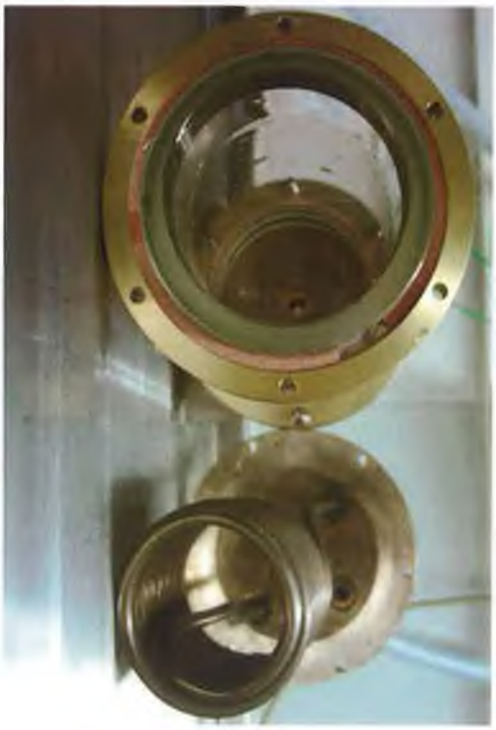
(b)

Photo 5.5 (a) Fouled bath pump and bath cooling system. (b) Fouled Bath

(a)

(b)

(c)



(d)

Photo 5.6 (a) Clean Bath. (b) Clean Pump and bath cooling system. (c) Top view of new clean cooling system. (d) Front view of new clean cooling system

5.12 The effect of concentration on fouling resistance

Different kinds of milk were used to carry out the effect of concentration on fouling resistance. The kinds of milk used were full fat fresh milk, low fat fresh milk and free fat fresh milk. Therefore, the variable value was "fat". The concentrations of fat were: 3.5 % for full fat milk, 1.5 % for low fat milk and 0.3 % for fat free milk.

Figure 5.14 shows the effect of fat concentration on fouling resistance. All tests lasted 8 hours and not all the concentrations were used. Only full fat milk and low fat milk tests were enough to reach a conclusion. While full fat milk produced enough fouling to decrease the efficiency of the rig, low fat milk only produced bubble fouling and did not change the efficiency of the heat exchanger. It was proven that low fat milk needs more time to reach the heavy fouling period and therefore, to affect the heat transfer surface.

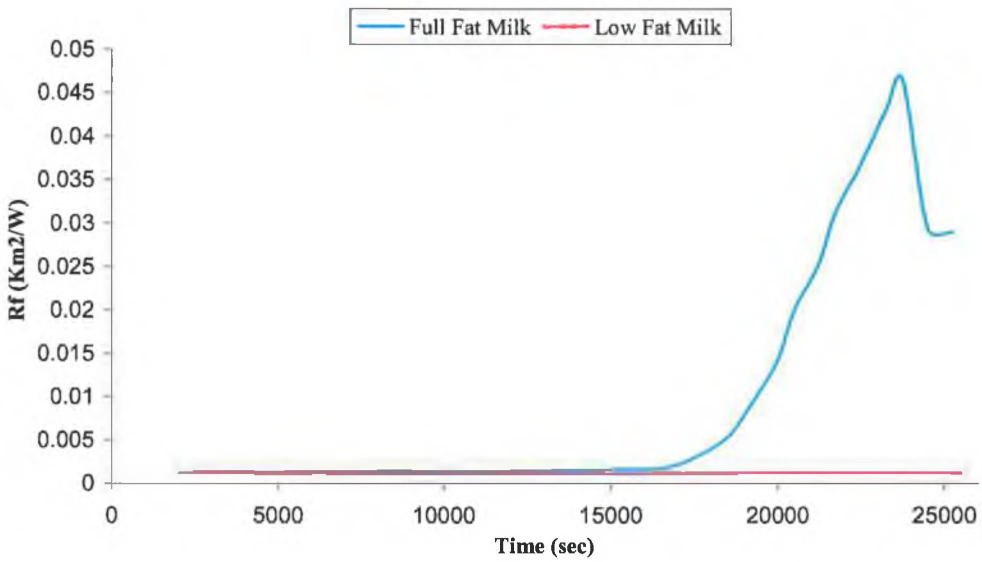


Fig. 5.14 Fouling resistance vs. Time for full fat fresh milk and low fat fresh milk

The effect of high concentration of fat was the increase on fouling resistance. Also, at high concentrations the induction-transportation period occurred earlier. Therefore, due to the geometry and materials of the double pipe heat exchanger, and the range of temperatures reached by the hot water and milk, the most influential milk property was fat. Due to this reason, a free fat milk test was not carried out and considered useless to provide important results.

5.13 Evaporated milk and powder milk

All the results from point 5.2 to point 5.12 were carried out with pasteurised milk. The main results were carried out with full fat milk, and some comparisons were made between full fat, low fat and free fat milk in point 5.12. However, some more tests were conducted with different dairy products: evaporated milk and powder milk.

It was assumed that the results, as dairy products, were going to be similar to those of fresh milk. Therefore, the aim was to try to make an evaluation on the effect of fouling on the induction period and the effect of fouling on heat transfer.

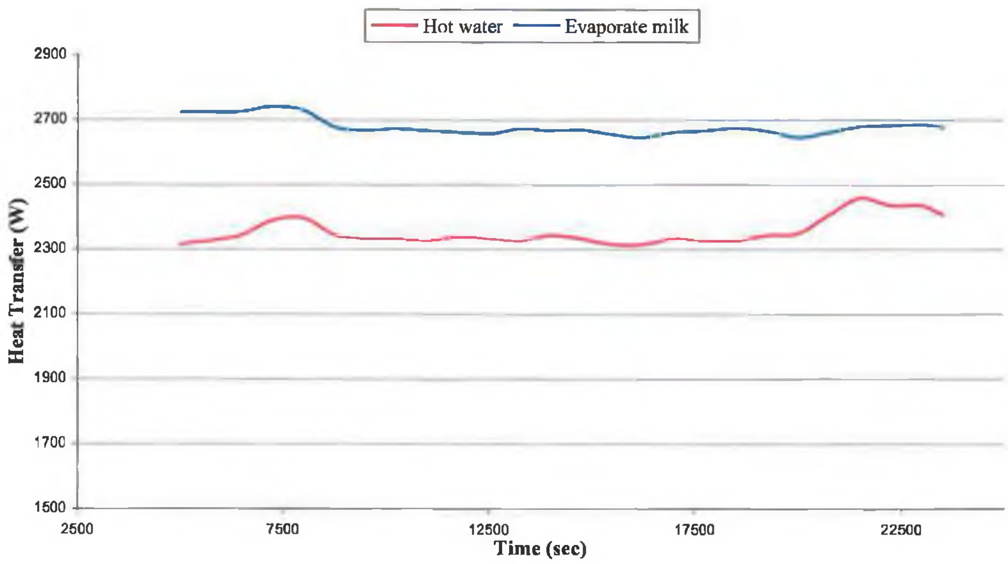


Fig. 5.15 Evaporated milk heat transfer evolution with time

Figure 5.15 shows the evolution of Q_h and Q_m for a test carried out with evaporated milk, diluting 1 part of evaporated milk with 1 part of water. As can be seen in figure 5.15, Q_h and Q_m kept constant values until the rig was stopped. For safety reasons, a test cannot run for more than 7-8 hours, therefore, fouling did not grow enough in the heat transfer surface to produce any significant changes in the heat transfer rates.

However, once the double pipe system was stripped out for maintenance and cleaning, it was observed that fouling was starting to build up in the lower parts of the inner pipe. The fouling shape was very similar to the one for fresh milk in the early stages, induction and transportation period. Photo 5.7 shows the kind of fouling affecting the heat transfer surface, bubbles fouling.

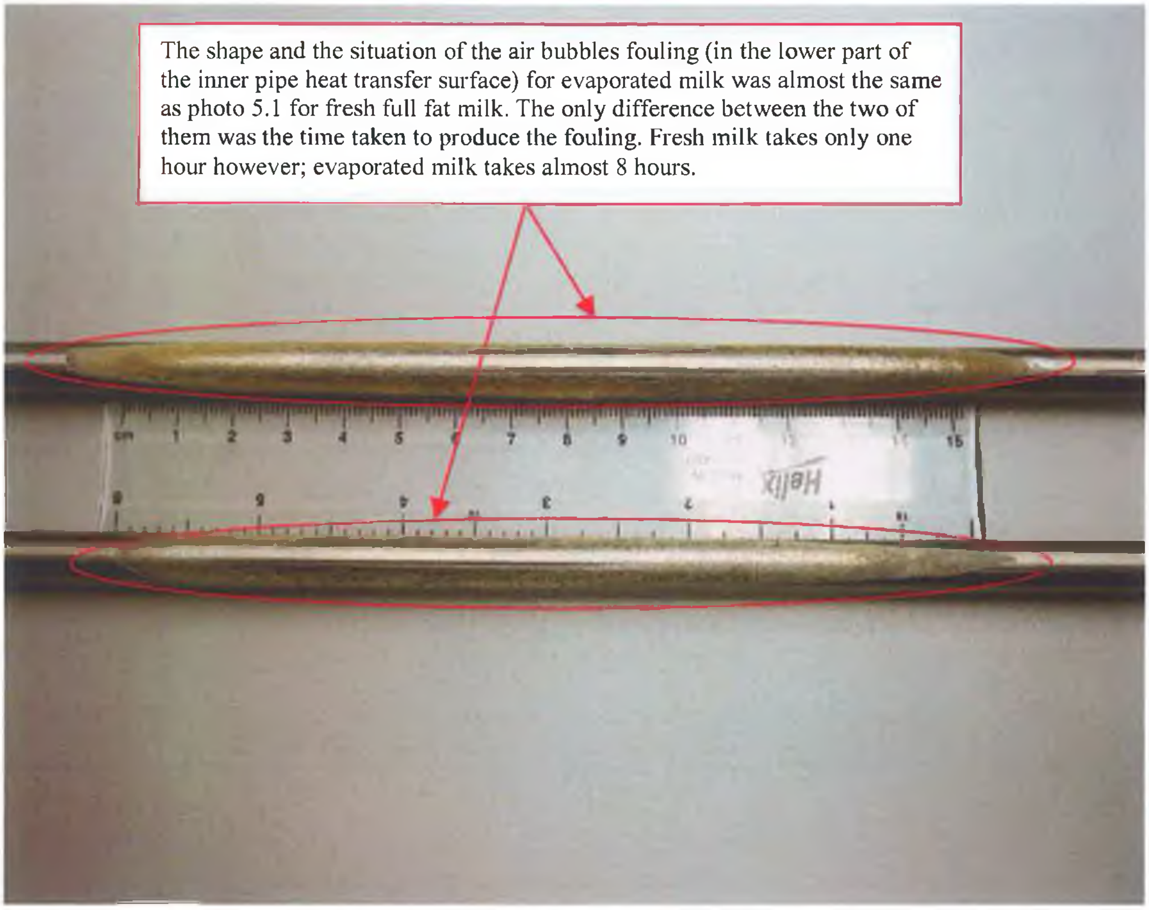


Photo 5.7 Fouling on the heat transfer surface for a test carried out with evaporated milk after 8 hours

Something similar happened with powder milk. Tests were carried out with two different concentrations of powder milk: medium and high concentration. The concentration was based on the amount of powder milk added to water. For a medium concentration, 400 grams of powder milk were added to 4 litres of water and for high concentration, 800 grams of powder milk were added. The concentration of powder milk in normal use is 200 grams to make approximately 2 litres.

The results for a medium concentration were very similar to evaporated milk, where the values of Q remained constant and no decrease was observed after 8 hours. Once the inner pipe was removed for cleaning, the same bubble fouling as shown in photo 5.1 for fresh milk and in photo 5.7 for evaporated milk was present.

Figure 5.16 represents the values of Q_h and Q_m for a test carried out with high concentration of powder milk with flow rates of 8.5 l/min of hot water and 1.9 l/min of milk. The values of Q_h and Q_m remained constant as with medium concentration of powder milk. The test was run for nearly 8 hours. However, once the test was stopped and the inner pipe was analysed, some differences were found.

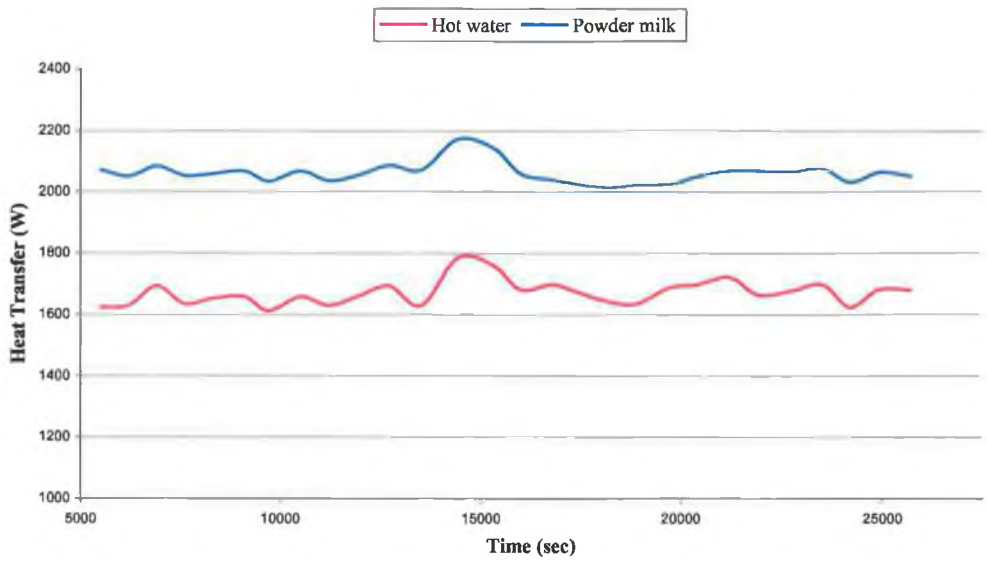


Fig. 5.16 Powder milk heat transfer evolution with time

Photo 5.8 shows the fouling affecting the inner pipe in a test carried out with a high concentration of powder milk with flow rates of 8.5 l/min of hot water and 1.9 l/min of milk. The typical bubbles fouling were found. However, the shape was wider and longer. Also, more fouling was found in different parts of the inner pipe, and this time, bubbles did not start it.

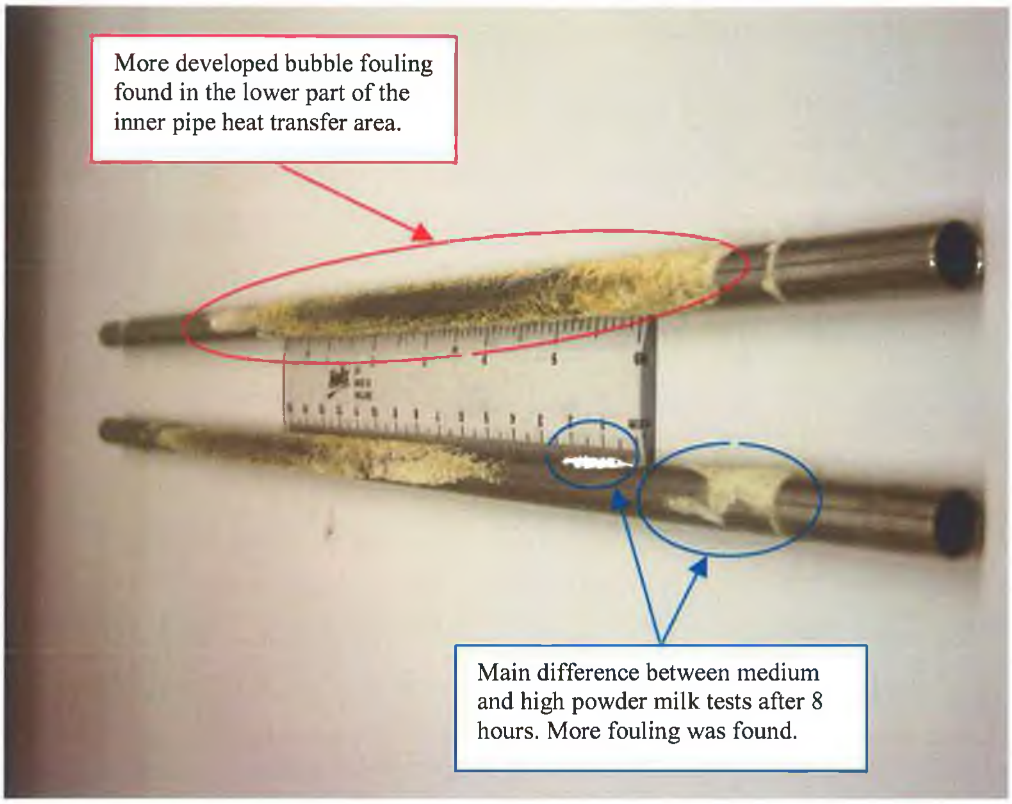


Photo 5.8 Fouling on the heat transfer surface for a test carried out with high concentration of powder milk after 8 hours

Even though more fouling was found, it was not enough to notice a decrease in the Q values. This amount of fouling can be found in a fresh full fat milk test after no more than 3 hours. On the other hand, it takes 8 hours for the powder milk to reach this amount of fouling.

Chapter 6 Conclusions

The following conclusions can be drawn from this work:

- Successful construction of a double pipe heat exchanger.
- Formation of fouling on the heat transfer surface even though the milk did not reach high temperatures (between 20 and 55 Celsius degrees).
- The evolution of the temperatures with time was consistent with the information found in the references.
- The evolution of heat transfer with time occurred symmetrically to the different fouling mechanisms.
- The overall heat transfer coefficient values are affected by fouling as references indicate.
- Milk fluid velocity had relevant importance on the heat transfer values. Reynolds numbers for the 5 different set points of the milk pump indicate that the milk fluid was laminar at some stages.
- Fouling deposits increased with time until reaching the point where they produced a blockage of the system.

However, the effect of concentration on fouling resistance could not be properly analysed. In Chapter 6, Future work, some recommendations were included to obtain better results regarding the effect of concentration on fouling resistance.

Chapter 7 Future Work

A double pipe heat exchanger was built and successful tests were carried out during the time dedicated to this Masters. However, some changes can be made with the aim of increasing the efficiency of the heat exchanger, to improve the accuracy of the results and to reduce heat loss from the system.

The main modifications, in my opinion, are the design and construction of a new hot water tank instead of using the old heat exchanger hot water tank. The reasons for these modifications are:

- The possibility of using a different fluid instead of water with the aim of increasing the inlet temperature by a few degrees. Therefore, results can be obtained in a shorter period of time. Dairy products with a low concentration of proteins will produce significant amounts of fouling in the time dedicated to the test.
- To analyse the evolution of fouling with different hot fluid flows.
- To compare the heat exchanger efficiency using different process hot fluids.

Another very interesting modification is to improve the fittings and insulation of the heat exchanger. Currently, the fittings are pieces of hose of different lengths and diameters producing a huge pressure drop. There is no insulation at all; therefore big heat losses are produced. With these modifications a new path in the double pipe heat exchange is opened: the study of pressure drop in relation to fouling.

In order to study pressure drop, the hose pieces have to be replaced by hard pipes, such as copper or aluminium. Many references can be found for the calculation of pressure drop due to fittings, bends, sudden contractions and sudden expansions. Parallel to pressure drop due to fouling, the pumping power relationship can also be analysed. CIP (Cleaning in Place) of a double pipe heat exchanger before blockage of the tubes takes place is another interesting point for future investigation.

Nomenclature

Symbol	Physical Quantity	SI Units
A	total heat transfer area,	m^2
Ac	net free-flow cross-sectional area,	m^2
C	flow stream heat capacity rate, $\dot{m}c_p$,	W/K
c_p	specific heat at constant pressure,	$J/Kg \cdot K$
d_i	tube inside diameter,	m
d_o	tube outside diameter,	m
De	equivalent diameter,	m
Dh	hydraulic diameter for pressure drop,	$4A_c/P_w$, m
e	tube surface roughness,	m
e	emissivity, Eq. 2.3	(0-1)
f	fanning friction factor,	$\tau_w \Big/ \frac{1}{2}\rho u_m^2$
h	heat transfer coefficient,	$W/m^2 \cdot K$
H	energy radiated per second,	W
i	specific enthalpy,	J/Kg
k	thermal conductivity,	$W/m \cdot K$
L	characteristic length,	m
m	exponent	
$\dot{m}$	fluid mass flow rate,	kg/s
n	exponent	
Nu	Nusselt number,	hd/k
P	pressure,	Pa
P	wetter perimeter, Eq. 2.57	m
Pe	Peclet number,	$RePr$
Pr	Prandtl number,	$c_p\mu d/k = \upsilon/\alpha$
Q	heat transfer rate,	W
R	thermal resistance,	$m^2 \cdot K/W$

r	tube radius,	m
R_f	fouling factor,	$m^2 \cdot K/W$
Re	Reynolds number,	$\rho u_m d/\mu, \rho u_m D_h/\mu$
T	temperature,	$^\circ C, K$
ΔT	local temperature difference between two fluids,	$^\circ C, K$
ΔT_{lm}	log-mean temperature difference,	$^\circ C, K$
ΔT_m	true mean temperature difference,	$^\circ C, K$
U	overall heat transfer coefficient,	$W/m^2 \cdot K$
u	fluid velocity component in axial direction,	m/s
u_m	fluid mean velocity in axial direction,	m/s
x	Cartesian coordinate, axial distance,	m

Greek Symbols

α	thermal diffusivity of fluid,	m^2/s
Δ	difference	
δ	differential form,	
ε	heat exchanger effectiveness	
μ	dynamic viscosity,	$Pa \cdot s$
ν	kinematic viscosity,	m^2/s
ρ	fluid density,	Kg/m^3
σ	Stefan-Boltzmann constant,	$5.67 \cdot 10^{-8} J/sm^2k^4$
τ_m	shear stress at the wall,	Pa
Φ	parameter, function of	

Subscripts

b	bulk
c	cold fluid
c	clean, Eq. 2.36
cp	constant-property solution
f	fouling condition
h	hot fluid
h	hot water, Chapter 5
i	inlet condition, inner or inside
m	mean
m	milk, Chapter 5
o	outlet condition, outer or outside
t	total
w	wall condition or wetter
x	local value at distance x
∞	fully developed condition
1	inlet
2	outlet

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Appendix A. Fluid Physical Properties

T (Celsius)	ρ (kg/m3)	μ (Pa*s)	c_p(kj/kgK)	k (W/mk)	Pr
30	995.5	0.0008	4.178	0.619	5.3996769
35	993.5	0.0007	4.178	0.625	4.67936
40	992	0.00062	4.179	0.631	4.10614897
45	990	0.00056	4.18	0.637	3.67472527
50	988	0.00052	4.181	0.642	3.38647975
55	985	0.00049	4.183	0.647	3.16795981
60	983	0.00046	4.184	0.653	2.94738132
65	980	0.000425	4.186	0.658	2.7037234
70	977.5	0.0004	4.189	0.662	2.53111782
75	974	0.00038	4.194	0.666	2.39297297
80	970.5	0.00036	4.197	0.67	2.25510448
85	967.5	0.00033	4.2	0.674	2.05637982
90	965	0.00031	4.205	0.676	1.9283284
95	962	0.000285	4.216	0.678	1.77221239

Water physical properties

T (Celsius)	ρ (kg/m3)	μ (Pa*s)	c_p(kj/kgK)	k (W/mk)	Pr
20	1029.5	0.0021	3.8875	0.555	14.7094595
25	1028.5	0.0017	3.875	0.568	11.5977113
30	1026	0.0012	3.8625	0.575	8.06086957
35	1024.5	0.00095	3.85	0.583	6.27358491
40	1022.5	0.0007	3.8375	0.59	4.5529661
45	1019	0.00055	3.825	0.598	3.51797659
50	1017	0.0004	3.8125	0.602	2.53322259
55	1015	0.0003	3.8	0.608	1.875

Full fat milk physical properties

Appendix B. Calculation of Heat Transfer Coefficient

T(Celsius)	m(kg/s)	um (m/s)	Re	f	Nu	hi (W/km2)
30	0.141027	2.618307	27042.78	0.00606	142.429	9280.376
35	0.140745	2.618307	30843.94	0.00587	154.932	10192.932
40	0.140533	2.618307	34771.22	0.00570	166.655	11069.458
45	0.14025	2.618307	38419.10	0.00557	176.472	11832.932
50	0.139966	2.618307	41290.83	0.005476	183.512	12401.613
55	0.139541	2.618307	43685.79	0.005405	188.846	12861.453
60	0.139258	2.618307	46440.37	0.005329	194.561	13373.551
65	0.138833	2.618307	50111.47	0.005236	201.742	13973.333
70	0.138479	2.618307	53107.62	0.005167	207.103	14431.822
75	0.137983	2.618307	55702.59	0.005112	211.273	14811.408
80	0.13748	2.618307	58585.90	0.005054	215.564	15202.946
85	0.13706	2.618307	63714.32	0.004960	222.931	15816.404
90	0.136708	2.618307	67649.67	0.004895	228.130	16233.261
95	0.136283	2.618307	73355.09	0.004808	235.110	16779.474

Calculation of inner pipe heat transfer coefficient for hot water flow of 8.5 l/min

T(Celsius)	m(kg/s)	um (m/s)	Re	f	Nu	ho (W/km2)
20	0.031742	0.730383	895.1548	0.017945	9.6068	942.3686
25	0.031712	0.730383	1104.705	0.016447	11.2553	1129.930
30	0.031635	0.730383	1561.195	0.014362	14.5630	1480.007
35	0.031588	0.730383	1969.153	0.01317	17.2299	1775.401
40	0.031527	0.730383	2667.205	0.011838	21.2779	2218.848
45	0.031419	0.730383	3383.004	0.010926	24.8244	2623.768
50	0.031357	0.730383	4642.501	0.009867	29.9558	3187.303
55	0.031295	0.730383	6177.829	0.009038	34.6265	3720.988

Calculation of annulus heat transfer coefficient for milk flow of 1.9 l/min

T(Celsius)	m(kg/s)	um (m/s)	Re	f	Nu	ho (W/km2)
20	0.04890	1.12518	1379.022	0.015062	13.45883	1320.218
25	0.04885	1.12518	1701.843	0.01390	15.79475	1585.646
30	0.04875	1.12518	2405.084	0.012271	20.46715	2080.034
35	0.04866	1.12518	3033.560	0.011331	24.21130	2494.778
40	0.04856	1.12518	4108.937	0.01025	29.85481	3113.238
45	0.04840	1.12518	5211.656	0.009518	34.74688	3672.503
50	0.04830	1.12518	7151.962	0.008653	41.75289	4442.507
55	0.04821	1.12518	9517.19	0.007969	48.02727	5161.033

Calculation of annulus heat transfer coefficient for milk flow of 2.9 l/min

T(Celsius)	m(kg/s)	um (m/s)	Re	f	Nu	ho (W/km2)
20	0.05404	1.24362	1524.182	0.014494	14.5672	1428.949
25	0.05399	1.24362	1880.984	0.013399	17.1009	1716.772
30	0.05385	1.24362	2658.251	0.011852	22.1651	2252.593
35	0.05378	1.24362	3352.882	0.010958	26.2175	2701.500
40	0.05368	1.24362	4541.457	0.009935	32.3159	3369.878
45	0.05349	1.24362	5760.251	0.009231	37.5896	3972.959
50	0.05339	1.24362	7904.800	0.008403	45.1248	4801.284
55	0.05328	1.24362	10519.00	0.007747	51.8488	5571.700

Calculation of annulus heat transfer coefficient for milk flow of 3.2 l/min

T(Celsius)	m(kg/s)	um (m/s)	Re	f	Nu	ho (W/km2)
20	0.06177	1.42128	1741.922	0.01378	16.1988	1588.991
25	0.06171	1.42128	2149.697	0.01276	19.023	1909.766
30	0.06156	1.42128	3038.00	0.01132	24.6635	2506.505
35	0.06147	1.42128	3831.865	0.01049	29.1684	3005.570
40	0.06135	1.42128	5190.236	0.00953	35.9335	3747.119
45	0.06114	1.42128	6583.144	0.00886	41.7651	4414.286
50	0.06102	1.42128	9034.057	0.00808	50.0725	5327.723
55	0.0609	1.42128	12021.72	0.00746	57.4504	6173.656

Calculation of annulus heat transfer coefficient for milk flow of 3.6 l/min

T(Celsius)	m(kg/s)	um (m/s)	Re	f	Nu	ho (W/km2)
20	0.068633	1.579208	1935.469	0.01326	17.6218	1728.581
25	0.068566	1.579208	2388.55	0.01230	20.7000	2078.090
30	0.0684	1.579208	3375.557	0.01093	26.8420	2727.901
35	0.0683	1.579208	4257.628	0.01014	31.7405	3270.602
40	0.068166	1.579208	5766.929	0.00922	39.0845	4075.703
45	0.067933	1.579208	7314.605	0.00859	45.3995	4798.410
50	0.0678	1.579208	10037.84	0.00785	54.3744	5785.442
55	0.067666	1.579208	13357.46	0.00725	62.3158	6696.493

Calculation of annulus heat transfer coefficient for milk flow of 4 l/min

Appendix C. Calculation of Fouling Resistance

Time (sec)	T1	T3	T4	T7	Q_h (w)	Q_c (w)	U (w/km2)	R_f (km2/w)
1925	93.58	90.12	57.97	39.5	1983.61	2175.76	12812.8	0.0001219
3325	93.07	89.66	57.18	38.85	1954.95	2159.27	10665.5	0.0002319
4375	93.32	89.99	57.35	39.33	1909.08	2122.75	8783.80	0.0003725
5425	93.28	89.99	56.89	38.91	1886.15	2118.04	7650.93	0.0004905
6650	93.67	90.3	57.19	39.2	1932.02	2119.22	8022.88	0.0004481
7700	93.77	90.43	56.99	38.92	1914.82	2128.64	7523.19	0.0005060
8750	93.79	90.43	56.78	38.92	1926.28	2103.90	6904.04	0.0005895
9975	93.4	90.07	56.53	38.81	1909.08	2087.41	6691.60	0.0006217
10850	91.11	88.07	54.52	37.24	1742.83	2035.58	5132.07	0.0009395
11725	92.59	89.36	55.93	38.36	1851.75	2069.74	6225.83	0.0006999
12950	93.1	89.82	56.15	38.64	1880.42	2062.67	6042.30	0.0007341
13825	93.15	89.83	55.97	38.47	1903.35	2061.5	5959.14	0.0007502
14875	92.86	89.44	55.4	38.01	1960.68	2048.54	5940.45	0.0007539
15575	90.5	87.28	53.29	36.64	1846.02	1961.37	4619.76	0.0010908
16450	93.11	89.87	54.06	37.35	1857.49	1968.43	3699.25	0.0014679
17325	93.51	90.42	51.57	35.55	1771.49	1887.15	2262.21	0.0026699
18025	92.09	89.26	48.09	33.17	1622.43	1757.57	1450.44	0.004401
18725	92.68	90	46.48	31.99	1536.44	1706.92	1094.73	0.005969
19425	93.43	90.89	44.78	30.9	1456.18	1635.06	819.547	0.0081166
19950	93.58	91.19	43.08	29.78	1370.18	1566.74	642.412	0.0104720
20650	93.88	91.62	41.79	28.78	1295.65	1532.57	534.007	0.0126840
21350	92.5	90.44	40.04	27.05	1180.99	1530.22	463.372	0.0146822
21875	93.83	91.81	39.24	26.52	1158.06	1498.41	398.279	0.0171512
22575	93.74	91.88	38.56	25.94	1066.33	1486.63	345.910	0.019812
23275	93.94	92.19	37.38	25.4	1003.27	1411.24	283.979	0.0242256
23975	94.07	92.46	37.42	24.81	923.01	1485.45	271.083	0.0253979
24675	94.08	92.64	37.23	24.35	825.55	1517.26	240.940	0.0286284
25375	94.15	92.84	36.94	23.28	751.02	1609.14	227.978	0.0302803

Main heat transfer values of full fat milk for a flow rate of 1.9 l/min

Time (sec)	T1	T3	T4	T7	Q_h (w)	Q_c (w)	U (w/km2)	R_f (km2/w)
2000	89.77	86.08	53.7	40.95	2115.47	2325.6	3521.05	0.001652
2750	90.6	86.88	54.17	41.17	2132.67	2371.2	3578.96	0.001620
3500	90.85	87.13	54.08	41.06	2132.67	2374.84	3455.34	0.001690
4250	91.18	87.37	54.05	40.88	2184.27	2402.20	3568.63	0.001625
5000	91.78	87.93	54.48	41.11	2207.20	2438.68	3693.74	0.001559
5750	92.44	88.52	54.74	41.33	2247.336	2445.98	3688.71	0.001561
8000	92.49	88.49	54.59	41.26	2293.2	2431.39	3714.26	0.001548
8750	92.6	88.56	54.77	41.38	2316.13	2442.33	3858.57	0.001478
9250	92.66	88.65	54.78	41.44	2298.93	2433.21	3747.77	0.001532
11250	92.07	88.1	54.53	41.29	2276.00	2414.97	3751.75	0.001530
12000	92.44	88.51	54.62	41.44	2253.06	2404.03	3529.20	0.001647
14500	92.42	88.51	54.7	41.66	2241.60	2378.49	3454.60	0.001690
16750	92.56	88.62	55.31	42.13	2258.80	2404.03	3778.82	0.001516
18500	92.69	88.77	56.23	42.95	2247.33	2422.27	4170.39	0.001342
19750	92.9	89.48	53.37	41.28	1960.68	2205.21	1983.19	0.003193
20500	93.08	90.16	46.9	36.76	1674.03	1849.53	777.503	0.008667
21250	93.36	90.67	43.58	34.28	1542.17	1696.32	519.185	0.013146
21750	92.81	90.27	42.53	33.18	1456.18	1705.44	469.618	0.014569
22500	92.75	90.34	42.13	32.51	1381.65	1754.68	442.351	0.015488
23250	93.44	91.14	41.71	31.8	1318.59	1807.58	406.282	0.016893
23750	93.54	91.39	41.49	30.94	1232.59	1924.32	391.370	0.017550
24500	93.85	91.76	40.89	30.42	1198.19	1909.72	358.522	0.019188
25250	93.34	91.32	41.32	30.31	1158.06	2008.22	378.772	0.018145

Main heat transfer values of full fat milk for a flow rate of 2.9 l/min

Time (sec)	T1	T3	T4	T7	Q_h (w)	Q_c (w)	U (w/km2)	R_f (km2/w)
3250	87.35	83.46	52.3	39.96	2230.13	2485.27	4140.83	0.0013715
4500	88.35	84.36	52.78	40.06	2287.46	2561.80	4387.45	0.0012765
5500	88.33	84.32	52.6	40.02	2298.93	2533.61	4243.18	0.0013307
6250	88.25	84.26	52.38	39.87	2287.46	2519.51	4077.32	0.0013978
7000	88.33	84.36	52.38	39.79	2276.00	2535.62	4049.87	0.0014095
7750	88.4	84.42	52.38	39.84	2281.73	2525.55	4003.95	0.0014293
8500	88.44	84.46	52.38	39.85	2281.73	2523.54	3978.36	0.0014405
9250	88.91	84.87	52.54	39.89	2316.13	2547.7	4041.70	0.0014130
10000	89.14	85.11	52.78	40.12	2310.39	2549.72	4031.70	0.0014173
10750	89.31	85.25	52.85	40.15	2327.59	2557.78	4076.01	0.0013984
11500	89.31	85.24	52.76	40.08	2333.33	2553.75	4041.14	0.0014132
12250	89.37	85.35	52.75	40.08	2304.66	2551.73	3897.42	0.0014771
13000	89.49	85.46	52.77	40.13	2310.39	2545.69	3854.39	0.0014971
13750	89.5	85.46	52.68	40.09	2316.13	2535.62	3799.91	0.0015232
14500	89.59	85.57	52.7	40.1	2304.66	2537.64	3737.03	0.0015542
15250	89.65	85.64	52.6	40.08	2298.93	2521.52	3606.04	0.0016222
16000	90.33	86.32	52.89	40.24	2298.93	2547.71	3527.88	0.0016652
16750	90.62	86.81	52.92	40.18	2184.27	2565.83	3139.89	0.0019104
17500	92.68	89	52.12	39.76	2109.74	2489.30	2143.65	0.0029465
18500	92.16	88.78	47.99	36.85	1937.75	2243.59	1232.04	0.0053626
19250	93.44	90.4	44.81	34.71	1742.83	2034.14	718.509	0.0094234
20000	93.61	90.86	41.76	32.54	1576.57	1856.90	480.685	0.0142436
20500	93.78	91.27	39.36	30.94	1438.98	1695.78	344.541	0.0199979
21250	93.66	91.31	37.6	29.83	1347.25	1564.87	272.383	0.0253801
21750	93.42	91.24	36.24	28.97	1249.79	1464.17	221.257	0.0313183
22500	94.06	91.96	35.27	28.38	1203.93	1387.64	188.437	0.0368287
23250	93.88	91.95	34.61	27.89	1106.46	1353.40	161.712	0.0429677
23750	91.53	89.83	34.77	28.11	974.61	1341.32	149.437	0.0465235
24500	94.32	92.72	44.37	35.79	917.28	1728.01	235.787	0.0293688
25250	94.25	92.49	44.67	37.12	1009.00	1520.57	238.316	0.0290534

Main heat transfer values of full fat milk for a flow rate of 3.2 l/min

Time (sec)	T1	T3	T4	T7	Q_h (w)	Q_c (w)	U (w/km2)	R_f (km2/w)
3250	84.54	80.5	49.92	38.55	2316.13	2592.36	4000.68	0.0014513
4500	85.09	81.04	50.04	38.65	2321.86	2596.92	3836.57	0.0015262
5500	84.82	80.8	49.79	38.53	2304.66	2567.28	3703.37	0.0015918
6250	84.87	80.8	49.74	38.45	2333.33	2574.12	3778.02	0.0015544
7000	84.99	80.94	49.81	38.52	2321.86	2574.12	3717.16	0.0015848
7750	85.17	81.08	49.83	38.5	2344.79	2583.24	3752.37	0.0015671
8500	85.14	81.07	49.79	38.5	2333.33	2574.12	3684.76	0.0016013
9250	85	80.96	49.64	38.42	2316.13	2558.16	3582.16	0.0016558
10000	84.89	80.88	49.48	38.31	2298.93	2546.76	3478.80	0.0017138
10750	84.73	80.75	49.42	38.3	2281.73	2535.36	3434.77	0.0017396
11500	84.88	80.84	49.58	38.45	2316.13	2537.64	3553.66	0.0016714
12250	85.56	81.47	50.02	38.81	2344.79	2555.88	3597.86	0.001647
13000	86	81.85	50.51	39.2	2379.19	2578.68	3793.72	0.001546
13750	85.98	81.86	50.77	39.53	2361.99	2562.72	3811.33	0.0015388
14500	85.99	81.89	51.1	39.88	2350.53	2558.16	3900.24	0.0014964
15250	86.06	81.96	51.38	40.24	2350.53	2539.92	3943.44	0.0014767
15500	87.02	83.07	51.3	40.24	2264.53	2521.68	3205.09	0.0018857
15750	89	85.22	50.76	40.01	2167.07	2451	2213.16	0.0028645
16000	91.76	88.14	49.49	39.33	2075.34	2316.48	1396.22	0.0047151
17000	93.11	89.77	44.88	35.21	1914.82	2204.76	808.05	0.008364
17750	91.86	88.79	43.53	33.35	1760.03	2321.04	747.33	0.0090683
18500	93.69	90.62	43.75	33.4	1760.03	2359.8	699.21	0.0097129
19250	93.46	90.33	44.12	33.92	1794.42	2325.6	730.73	0.0092819
19750	92.08	89.04	44.21	33.91	1742.83	2348.4	764.88	0.0088533
20500	93.41	90.3	45.35	34.28	1782.96	2523.96	850.57	0.0079313
21250	93.06	90.1	45.58	33.72	1696.96	2704.08	887.57	0.0075883
22000	93.59	90.64	45.97	33.72	1691.23	2793	912.66	0.0073714
22500	93.63	90.9	48.5	33.17	1565.10	3495.24	1325.63	0.0049821
23250	93.31	90.26	56.05	45.19	1748.56	2476.08	1676.48	0.0038777
24000	92.99	89.76	57.07	48.55	1851.75	1942.56	1492.93	0.0043904

Main heat transfer values of full fat milk for a flow rate of 3.6 l/min

Time (sec)	T1	T3	T4	T7	Q_h (w)	Q_c (w)	U (w/km2)	R_f (km2/w)
3000	80.89	76.79	47.67	37.46	2350.53	2599.46	3994.77	0.001468
4250	81.1	76.99	47.73	37.53	2356.26	2596.9	3937.55	0.001494
5000	81.77	77.7	47.88	37.8	2333.33	2566.36	3554.66	0.001685
6000	82.43	78.33	48.24	38.03	2350.53	2599.46	3567.57	0.001678
6750	82.63	78.52	48.26	38.06	2356.26	2596.92	3509.06	0.001711
7500	82.81	78.71	48.38	38.15	2350.53	2604.55	3484.94	0.001725
8250	82.61	78.51	48.32	38.13	2350.53	2594.37	3516.24	0.001707
9000	82.77	78.66	48.37	38.15	2356.26	2602.01	3508.94	0.001711
9750	82.95	78.86	48.35	38.14	2344.79	2599.46	3391.84	0.001780
10500	82.85	78.72	48.3	38.14	2367.72	2586.73	3453.20	0.001743
11250	82.73	78.69	48.34	38.21	2316.13	2579.09	3337.99	0.001813
12000	82.7	78.63	48.34	38.13	2333.33	2599.46	3446.36	0.001747
12750	82.75	78.66	48.31	38.12	2344.79	2594.37	3440.48	0.001751
13500	83.26	79.23	48.71	38.45	2310.39	2612.19	3333.04	0.001816
14250	83.26	79.26	48.9	38.64	2293.2	2612.19	3350.12	0.001806
15000	83.26	79.24	49.1	38.81	2304.66	2619.83	3478.41	0.001728
15750	83.26	79.23	49.39	39.12	2310.39	2614.74	3601.39	0.001660
16500	83.36	79.32	49.74	39.54	2316.13	2596.92	3682.39	0.001617
17250	83.71	79.67	50.34	40.12	2316.13	2602.01	3805.03	0.001556
17500	83.89	79.87	50.38	40.24	2304.66	2581.64	3652.35	0.001633
17750	84.83	80.89	50.33	40.24	2258.80	2568.91	3111.67	0.001966
18000	86.71	82.87	49.88	40.04	2201.47	2505.26	2272.82	0.002796
18250	89.16	85.49	49.05	39.46	2104.01	2441.61	1553.45	0.004222
18500	92.33	88.71	48.19	38.8	2075.34	2390.69	1128.00	0.005922
18750	93.3	89.65	47.76	38.4	2092.54	2383.05	1046.47	0.006405
19500	93.43	90.1	44.88	36.04	1909.08	2250.66	721.246	0.009421
20250	93.51	90.33	43.53	34.7	1823.09	2248.11	624.056	0.010933
21000	93.77	90.75	43.6	33.84	1731.36	2484.89	633.47	0.010766
21750	93.67	90.95	43.86	32.89	1559.37	2792.96	630.09	0.010826
22250	93.04	90.69	44.07	31.4	1347.25	3225.78	639.79	0.010657

Main heat transfer values of full fat milk for a flow rate of 4 l/min

Time (sec)	T1	T3	T4	T7	Q_h (w)	Q_c (w)	U (w/km2)	R_f (km2/w)
3000	85.62	81.58	52.95	42.11	2316.13	2759.86	4651.77	0.00122
3750	85.84	81.77	53.12	42.2	2333.33	2780.23	4773.34	0.00118
4500	85.92	81.82	53.2	42.26	2350.53	2785.32	4875.10	0.00115
5250	86.18	82.04	53.35	42.38	2373.46	2792.96	4942.36	0.00113
6000	86.26	82.15	53.43	42.42	2356.26	2803.14	4895.91	0.00114
6750	86.36	82.25	53.43	42.44	2356.26	2798.05	4811.19	0.00117
7500	86.35	82.21	53.46	42.46	2373.46	2800.6	4930.52	0.00113
8250	86.39	82.23	53.5	42.53	2384.92	2792.96	4958.30	0.00112
9000	86.45	82.31	53.54	42.57	2373.46	2792.96	4888.67	0.00114
9750	86.54	82.33	53.57	42.56	2413.59	2803.14	5085.42	0.00109
10500	86.57	82.39	53.58	42.58	2396.39	2800.6	4975.86	0.00112
11250	86.7	82.53	53.65	42.64	2390.66	2803.14	4916.88	0.00114
12000	86.86	82.68	53.72	42.69	2396.39	2808.23	4903.72	0.00114
12750	86.93	82.74	53.75	42.73	2402.12	2805.69	4895.60	0.00114
13500	86.92	82.73	53.74	42.71	2402.12	2808.23	4904.96	0.00114
14250	86.97	82.79	53.76	42.75	2396.39	2803.14	4839.27	0.00116
15000	87.01	82.85	53.75	42.74	2384.92	2803.14	4753.33	0.00118
15750	86.99	82.81	53.76	42.73	2396.39	2808.23	4844.70	0.00116
16500	87.06	82.93	53.75	42.76	2367.72	2798.05	4625.91	0.00123
17250	87.15	82.96	53.76	42.74	2402.12	2805.69	4760.35	0.00118
18000	87.16	83.01	53.83	42.79	2379.19	2810.78	4709.65	0.00120
18750	87.28	83.12	53.83	42.83	2384.92	2800.6	4627.37	0.00122
19500	87.42	83.25	53.9	42.91	2390.66	2798.05	4602.47	0.00123
20250	87.41	83.27	53.92	42.9	2373.46	2805.69	4569.35	0.00124
21000	87.56	83.38	53.99	43.02	2396.39	2792.96	4581.24	0.00124
21750	87.78	83.61	54.15	43.14	2390.66	2803.14	4554.33	0.00125
22500	87.91	83.71	54.23	43.23	2407.86	2800.6	4592.15	0.00124
23250	87.96	83.76	54.26	43.28	2407.86	2795.50	4563.60	0.00125
24000	88.14	83.91	54.37	43.39	2425.05	2795.50	4597.89	0.00123
24750	88.62	84.34	54.59	43.54	2453.72	2813.33	4629.38	0.00122
25500	89.56	85.17	55.17	44	2516.78	2843.88	4795.87	0.00117

Main heat transfer values of low fat milk for a flow rate of 4 l/min

Time (sec)	T1	T3	T4	T7	Qh (w)	Qc (w)	U (w/km2)	Rf (km2/w)
5500	92.51	89.68	55.35	37.77	1622.43	2070.92	4306.12	0.00132
6200	91.52	88.68	54.62	37.21	1628.17	2050.89	4356.07	0.00130
6900	92.45	89.5	55.25	37.56	1691.23	2083.88	4769.89	0.00116
7600	92.34	89.49	54.87	37.44	1633.90	2053.25	4047.46	0.00143
8300	92.08	89.2	54.56	37.08	1651.10	2059.14	4143.13	0.00139
9100	92.68	89.79	55.22	37.67	1656.83	2067.39	4268.10	0.00134
9700	91.09	88.28	54.02	36.75	1610.97	2034.40	4041.32	0.00143
10500	92.51	89.62	54.88	37.33	1656.83	2067.39	4162.98	0.00138
11200	91.64	88.8	54.35	37.06	1628.17	2036.76	4013.14	0.00144
11900	92.72	89.83	55.11	37.68	1656.83	2053.25	4077.59	0.00141
12700	94.23	91.28	56.04	38.33	1691.23	2086.23	4129.16	0.00139
13500	92.6	89.76	55.21	37.62	1628.17	2072.10	4198.23	0.00136
14400	96.69	93.58	58.08	39.65	1782.96	2171.05	5002.77	0.00110
15300	95.61	92.54	57.15	38.95	1760.03	2143.96	4747.70	0.00117
16000	92.96	90.03	55.31	37.83	1679.76	2059.14	4207.82	0.00136
16800	92.94	89.98	55.08	37.77	1696.96	2039.11	4032.31	0.00143
17500	92.38	89.47	54.67	37.5	1668.30	2022.62	3877.20	0.00150
18200	91.65	88.79	54.17	37.07	1639.63	2014.38	3818.71	0.00153
18900	92.17	89.32	54.48	37.31	1633.90	2022.62	3734.3	0.00157
19700	92.28	89.34	54.5	37.31	1685.50	2024.98	3932.59	0.00148
20400	93.09	90.13	55.13	37.73	1696.96	2049.72	4045.34	0.00143
21200	93.39	90.39	55.34	37.78	1719.9	2068.56	4231.28	0.00135
21900	92.92	90.02	55.18	37.62	1662.57	2068.56	4133.65	0.00139
22700	93.24	90.32	55.31	37.78	1674.03	2065.03	4054.53	0.00142
23500	93.6	90.64	55.49	37.87	1696.96	2075.63	4131.92	0.00139
24200	91.88	89.05	54.36	37.11	1622.43	2032.05	3828.41	0.00153
24900	92.9	89.97	54.92	37.4	1679.76	2063.85	4045.54	0.00143
25700	92.88	89.95	54.77	37.35	1679.76	2052.07	3898.49	0.00149

Main heat transfer values of powder milk for a flow rate of 1.9 l/min

Time (sec)	T1	T3	T4	T7	ΔTm	Q_h (w)	Q_c (w)	U (w/km2)
5000	90.89	86.85	50.84	40.14	-54.2003	2316.13	2724.22	2136.64
5750	90.85	86.79	50.8	40.1	-53.9724	2327.59	2724.22	2156.28
6500	91.15	87.06	51.21	40.5	-53.0839	2344.79	2726.76	2208.57
7250	91.72	87.55	51.51	40.74	-52.9404	2390.66	2742.04	2257.88
8000	91.61	87.43	51.55	40.83	-52.5246	2396.39	2729.31	2281.21
8750	90.53	86.44	50.9	40.38	-53.056	2344.79	2678.39	2209.73
9500	90.37	86.3	50.83	40.35	-53.1698	2333.33	2668.20	2194.22
10250	90.32	86.25	50.82	40.32	-52.8554	2333.33	2673.3	2207.27
11000	90.17	86.11	50.73	40.25	-52.8467	2327.59	2668.20	2202.21
11750	90.28	86.2	50.74	40.28	-53.1961	2339.06	2663.11	2198.53
12500	90.29	86.22	50.77	40.33	-53.3628	2333.33	2658.02	2186.28
13250	90.11	86.05	50.74	40.24	-52.4027	2327.59	2673.3	2220.87
14000	90.15	86.06	50.76	40.28	-52.2899	2344.79	2668.20	2242.11
14750	90.15	86.08	50.85	40.36	-52.0555	2333.33	2670.75	2241.19
15500	90.08	86.04	50.85	40.42	-52.5065	2316.13	2655.47	2205.56
16250	89.93	85.89	50.74	40.34	-52.5414	2316.13	2647.84	2204.10
17000	90.01	85.94	50.84	40.38	-51.6996	2333.33	2663.11	2256.62
17750	90.21	86.15	51.04	40.56	-51.6739	2327.59	2668.20	2252.19
18500	90.22	86.16	51.11	40.6	-51.209	2327.59	2675.84	2272.64
19250	90.14	86.05	51.05	40.58	-51.0621	2344.79	2665.66	2296.02
20000	90.07	85.97	51.04	40.64	-51.1715	2350.53	2647.84	2296.71
20750	90.73	86.53	51.29	40.83	-51.4112	2407.86	2663.11	2341.76
21500	91.47	87.18	51.7	41.17	-51.3514	2459.45	2680.93	2394.73
22250	91.54	87.29	51.76	41.22	-51.7703	2436.52	2683.48	2353.20
23000	92.17	87.92	51.93	41.38	-53.6929	2436.52	2686.03	2268.94
23500	92.69	88.49	51.94	41.42	-56.7304	2407.86	2678.39	2122.19
24750	93.06	89.1	49.96	40.18	-77.205	2270.26	2489.98	1470.28
25750	92.79	89.29	51.06	43.74	-104.349	2006.55	1863.67	961.462
26250	91.97	88.55	54.05	45.5	-69.361	1960.68	2176.83	1413.39

Main heat transfer values of evaporated milk for a flow rate of 4 l/min