MAURITIUS RESEARCH COUNCIL

# DESIGN A HEATING SYSTEM FOR SNAPPY FISHPOND AT FUEL

## Final Report

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# PROJECT TITLE: Design a heating system for snappy fishpond at FUEL.

## BACKGROUND

FUEL has invested millions of rupees in a fish farm by using APT PAZ technologies for the breeding of red snappy fish. The objective of the farm is to produce fifty tons of fish annually but this is not being attained. The reason for this under-production had been investigated and it was found that this was due to the water temperature.

## INTRODUCTION

The aim of this project is to raise the temperature of water in the fishpond by using energy available at the factory. Study has demonstrated that for fish production to be fruitful, there are some conditions that need to be satisfied. Beside a good alimentation, the temperature of water plays an important role on the rapid growth of these fish. The result of the increase in the water temperature will automatically increase the fish production in the farm.

## SURVEY OF TEMPERATURE AGAINST PRODUCTION THROUGH THE YEAR

### Investigation of the fishpond water temperature

The temperature of water in the fishpond had been taken twice a day mainly in the morning and in the afternoon. With all the data that had been gathered by the fish farmer it was found that the fishes took less time in the production ponds, when the water temperature is between 24°C and 28°C. In the morning the temperature of water is at 18°C and the fishes do not seem to have appetite when they are given food. And in the afternoon when the temperature increases it is seen that they begin to feed as usual. Table 1 shows temperature of water for the year1999.

| **Month** | **Jan** | **Feb** | **Mar** | **Apr** | **May** | **Jun** | **Jul** | **Aug** | **Sep** | **Oct** | **Nov** | **Dec** |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Average water temp $^0$C max** | **24.0** | **24.0** | **23.7** | **23.1** | **21.8** | **20.6** | **19.3** | **20.3** | **21.4** | **21.5** | **23.0** | **24.0** |

## SYSTEM DESIGN

The fishpond obtains water from Bassin board at a volume flow rate of $5m^3$/min at a temperature, which varies between $18^0$C and $22^0$C. To heat this large amount of water to the desired temperature a very big heat exchanger and pump would be needed at the factory. It would be therefore an advantage if a certain amount of water could be heated at a higher temperature. The hot water would then be pumped back to the fishpond for mixing with cold water. The result of this will give the desired water temperature.

There are a lot of places in the factory where energy could be recuperated for the heating system, without affecting the performance of the factory. Upon investigation the following potential energy sources have been spotted,

**1. Flue gas**

A heat exchanger was designed from the model of the existing economizer. But the available space in the chimney of the JTA boiler was not enough to house it. A smaller exchanger was designed, but temperature achieved was not enough. Therefore a second heating system was required hence had to find another energy source.

**2. Scrubber tank (water temperature $\approx 60^0$C)**

The scrubber tank was not large enough to accommodate a heat exchanger to heat all the 5 $m^3$/min. But using this to further heat the water from 1 above was considered. The problem was the deposit of ash on the heat exchanger, which would affect the heat transfer between the hot and cold water. Cleaning of tank is required very often and this could damage the heat exchanger.

**3. Contaminated cooling tower**

Preheat the water before entering the heat exchanger was considered but the distance between the two sources is too far and also size of the pre-heat exchanger was too large.

**4. Condensate from evaporator**

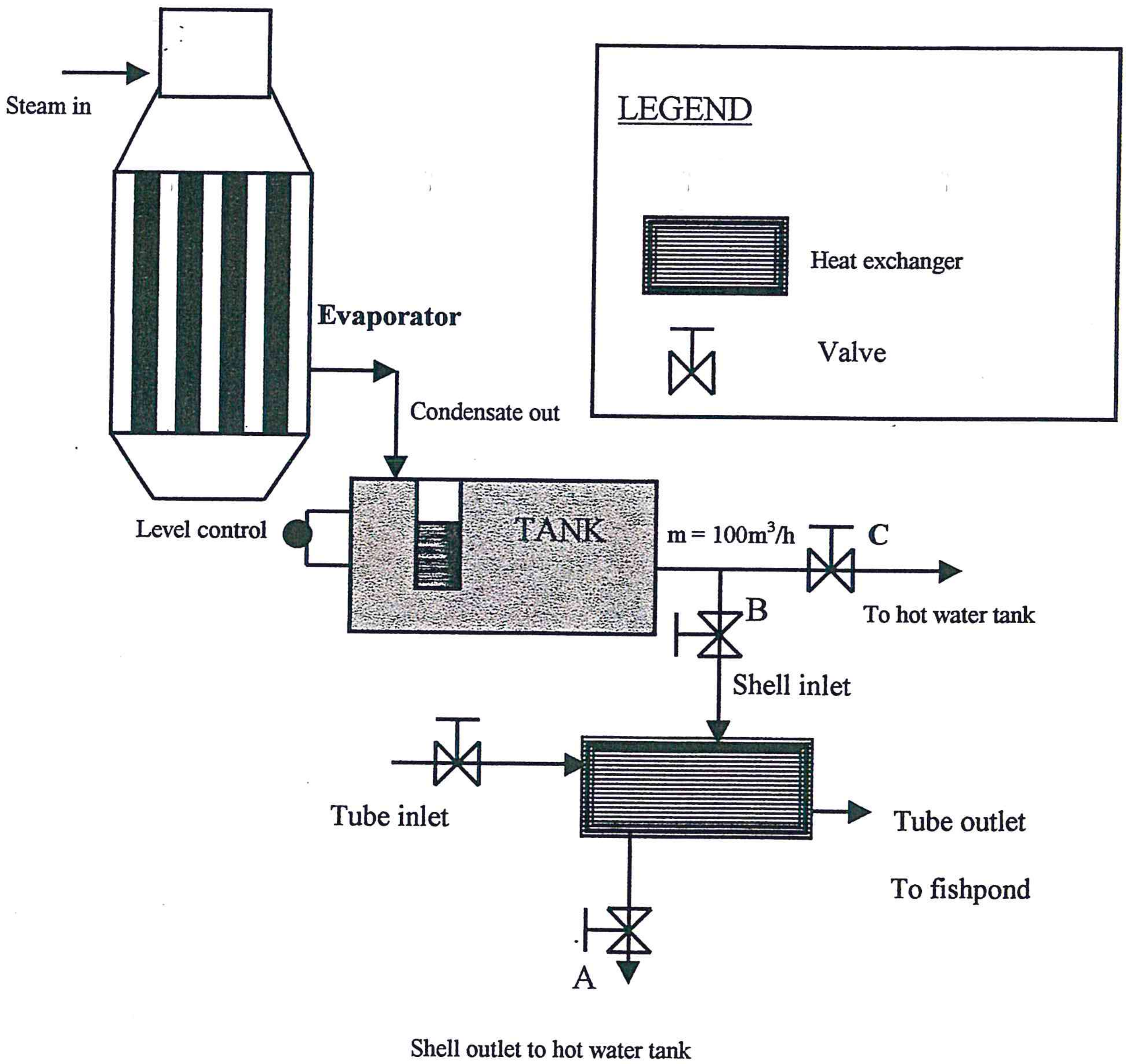
The last design is based on the use of condensate, which is at a temperature of $90^0$C. The design of the heat exchanger is shown on page 3.

## Heat exchanger design

### INTRODUCTION

Several heat exchangers had been designed throughout the course of this project and the one that can be implemented will be described fully in this chapter. The heat exchanger uses condensate that comes, at a temperature of 90°C from the evaporator as the heating source.

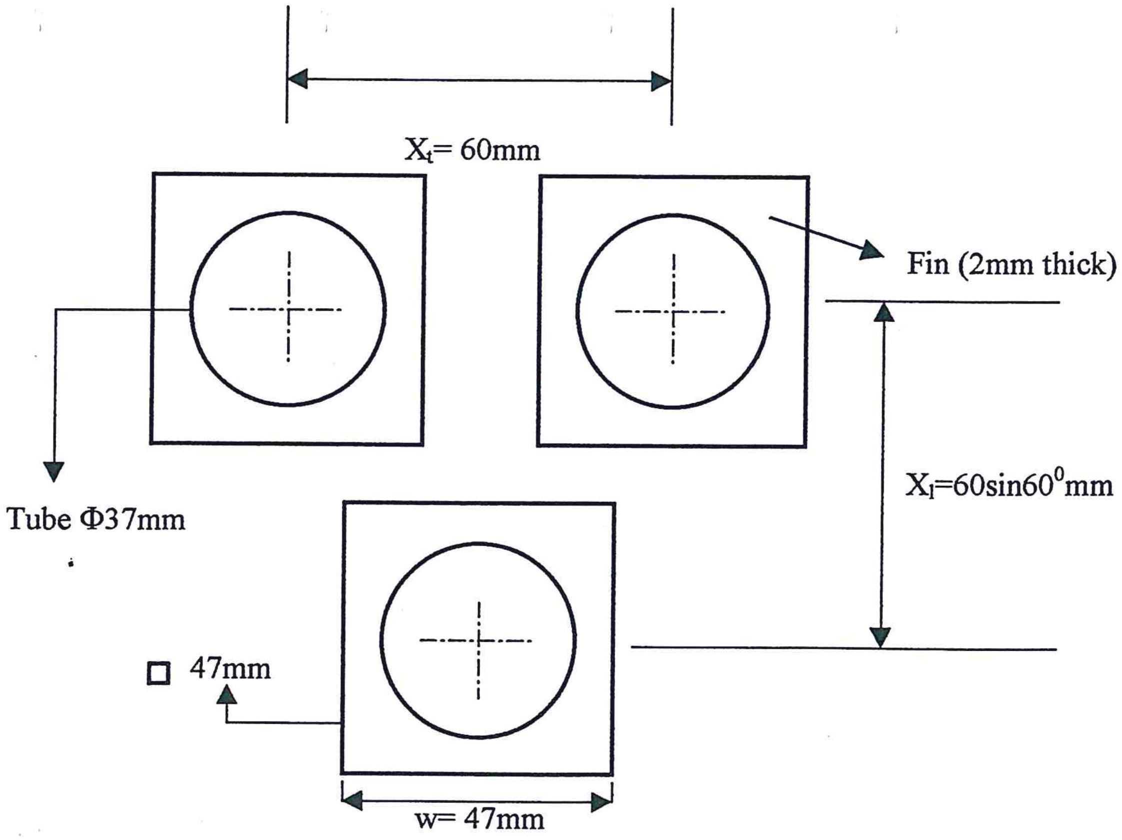
The figure below shows a schematic diagram of the system.



When the heat exchanger is in operation valves A and B are opened and C is closed and vice versa.

**Design calculation**

*Tube pitch and layout.* The selection of tube pitch is a compromise between a close pitch for increased shell-side heat transfer and surface compactness, and an open pitch for decreased shell-side plugging and ease in shell-side cleaning. The tube pitch that had been considered was to increase the heat transfer and the tube layout was a triangular one of 60°. The figure below shows the layout.

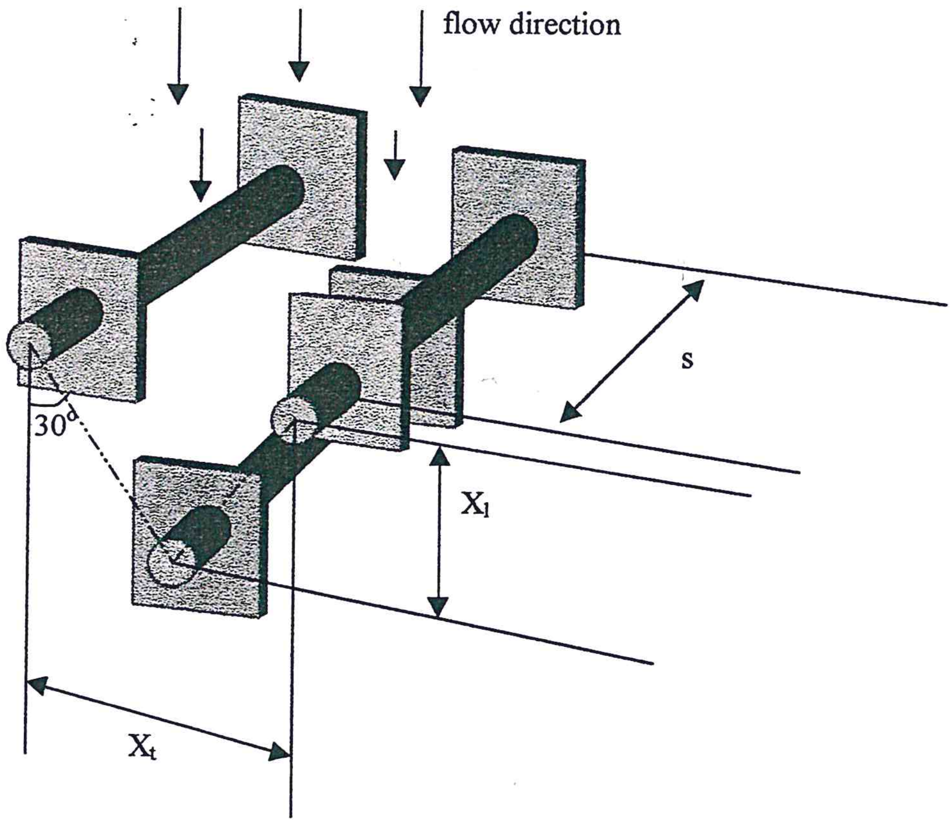



Let s be the distance between adjacent fins and is equal to 12mm.
Fin thickness, $\delta = 2$mm
Tube outside diameter, $d_o$= 37mm
Tube thickness, t = 1.5mm

The hydraulic diameter, $D_h$ of the flow passage is calculated as follows

$$\boxed{\mathbf{D_h = [4 \times \text{area of flow} \times \text{length of flow}] \div [\Sigma \text{ of heating surface area}]}} \quad \mathbf{....(1)}$$

To find the area of flow and the overall heating surface area, let consider a small portion of the configuration as shown below.




The area of flow found in the equation (1) is the minimum free-flow area of the finned passages i.e. the cross-sectional area perpendicular to flow direction and is given by

$$\boxed{\mathbf{Area\ of\ flow = [(X_t - d_o) \times s]\ mm^2}} \quad \mathbf{....(2)}$$

The overall heating surface area is the surface, which is always in contact with the hot fluid and is given by

$$\boxed{\Sigma \text{ of heating surface area} = 2 \times [(w^2 - \pi/4d_o^2) + \pi d_o s]} \quad ....(3)$$

From the information above the following equation can be computed

Flow passage hydraulic diameter, $D_h = \dfrac{4 \times (Xt - d_o) \times s \times X_l}{2 \times (w^2 - \pi/4\, d_o^2) + \pi d_o s}$

$= 15.7mm$

Fin area/ heating surface area area, $\chi = \dfrac{2 \times (w^2 - \pi/4\, d_o^2)}{2 \times (w^2 - \pi/4\, d_o^2) + \pi d_o s}$

$= 0.62$

Free flow area/ frontal area, $\sigma = \dfrac{(Xt - d_o) \times s}{Xt\,(s + \delta)}$

$= 0.33$

$\varphi$ is the ratio of the total cold-side area to the hot- side surface area.

Assume fin thickness to be negligible $\varphi$ is given by

$$\boxed{d_i/\, d_o\, (1-\chi)} \quad ....(4)$$

$\therefore\ \varphi = 0.35$

The mass flow rate, m of the hot fluid entering the shell-side through a frontal area, $A_{fr}$ of $0.167m^2$ is 27.8kg/s.

Mass velocity, $G = m \div \sigma A_{fr}$

$= 27.8 \div 0.33 \times 0.167$

$\approx 504.5\ kg/s.m^2$

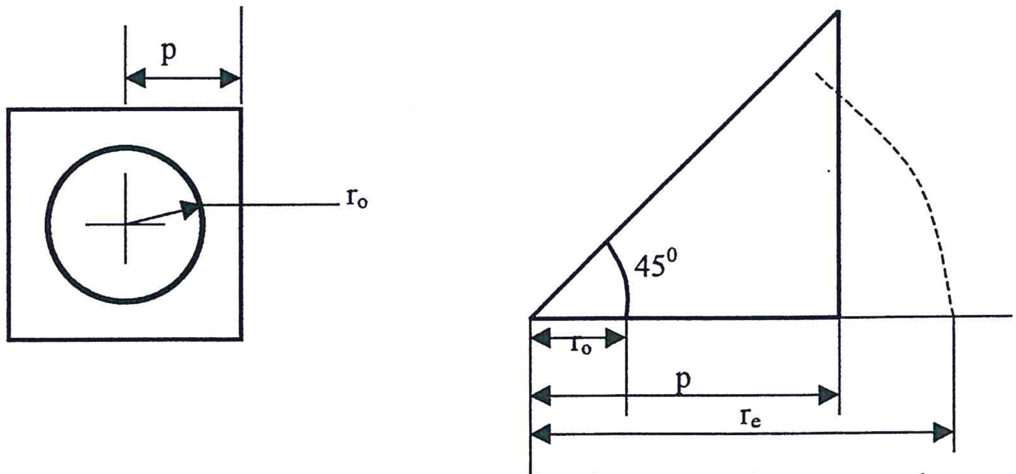
$Re = G D_h/\mu$ $\mu = 306 \times 10^{-6}\ N.s/m^2$

$\approx 25858$

The flow is turbulent.

For crossflow over low-height finned tubes, Rabas and Taborek, Ganguli and Yilmaz, and Chai have assessed the pertinent literature. A simple but accurate correlation for the heat transfer is given by Ganguli and Yilmaz as,

$\mathbf{j = 0.255 Re^{-0.3} (d_e/s)^{-0.3}}$ where $d_e$ is the fin tip diameter of a disk(radial) fin.




The fictitious radius, $r_e$ that corresponds to radial fin having the same surface area, as the square and is a function of the dimension p.

$r_e = (2/\sqrt{\pi})p$

$= 26.5mm$

$d_e = 2 \times r_e = 53mm$

Therefore from equation above, $j = 0.0077$

Heat transfer coefficient on the hot side. $h_h = jGc_p/Pr^{2/3}$

$\approx 11000\ W/m^2.\ K$

From graph the fin efficiency $\approx 50\%$

Overall surface efficiency, $\eta_o = 70\%$

Ten tubes pass is used and the internal diameter of the tube, $d_i = 34mm$.

$Re = 4m/\pi d_i\ \mu$ $\quad\quad\quad\quad$ $\mu = 739 \times 10^{-6}\ N.s/m^2$

$= 89945$

$Nu_d = 0.023Re^{0.8}Pr^{0.4}$ $\quad\quad\quad\quad$ $Pr = 4.97$

$= 401$

Heat transfer on the cold side, $h_i = Nu_d\ k/d_i$ $\quad\quad\quad\quad$ $k = 0.624\ W/m.\ K$

$= 7360\ W/m^2.K$

Wall conduction resistance, $A_hR_w = \dfrac{d_i \ln (d_o/d_i)}{2k\varphi}$ $\quad\quad\quad\quad$ $k = 37.2\ W/m.\ K$

$= 110.4 \times 10^{-6}\ m^2.\ K/W$

Neglecting fouling effects, the overall heat transfer coefficient, $U_h$ is given by

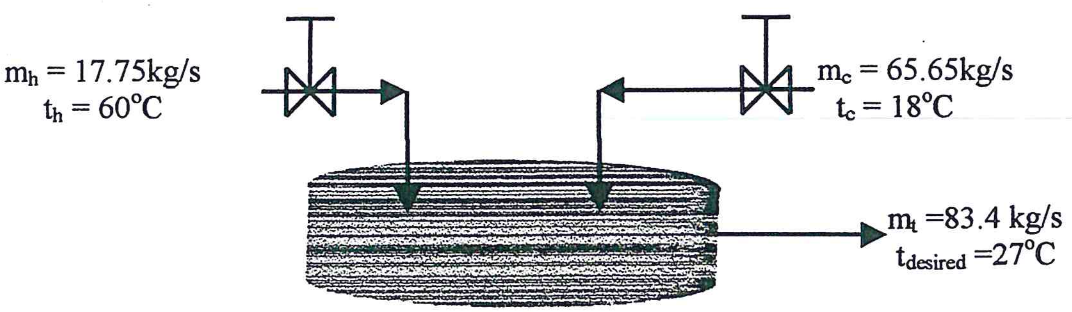
$$1/U_h = 1/h_i\varphi + A_hR_w + 1/\eta_o\ h_h$$

Therefore $U_h = 1591\ W/m^2.K$

AT MIXING ZONE

The mixing zone is simply a pond where the hot water from the heat exchanger mixes with the cold water in order to get the required fishpond water temperature.

The figure below shows a schematic diagram of the mixing zone.



Using energy balance,

$$m_c c_{p,c}(t_2 - t_1) = m_h c_{p,h}(t_3 - t_2)$$

$m_c$ = mass of cold water in kg/s
$m_h$ = mass of hot water in kg/s
$c_{p,c}$ = specific heat capacity of cold water = 4.179 kJ/ kg.k
$c_{p,h}$ = specific heat capacity of hot water = 4.179 kJ/ kg.k
$t_1$ = temperature of cold water = $18^0C$
$t_2$ = desired temperature of water = $27^0C$
$t_3$ = temperature of hot water = $60^0C$

$m_c \div m_h = 3.7$ .........(1)

$m_c + m_h = 83.4$ kg/s .........(2)

Equating (1) and (2), we get $m_h = 17.75$ kg/s.

**Heat exchanger design calculations**

- Mass flow rate of hot water, $M = 27.8$ kg/s
- Mass flow rate of cold water, $m = 17.75$ kg/s
- Shell inlet temperature, $T_i = 90^0C$
- Tube inlet temperature, $t_i = 34^0C$
- Tube outlet temperature, $t_o = 74^0C$
- Specific heat capacity of hot water, $c_{p,h} = 4.209$ kJ/kg.k
- Specific heat capacity of cold water, $c_{p,c} = 4.179$ kJ/kg.k

By using energy balance the shell outlet temperature, $T_o$ can be found.

$M c_{p,h} (T_i - T_o) = m c_{p,c}(t_o - t_i)$

$T_o = 64.6^0C$

The heat transfer, $q = m c_{p,c}(t_o - t_i)$

$= 2.97$ Mw.

To find $m_c$ (the mass flow rate of cold water) that must be added to the $m_h$ (the mass flow rate of hot water) in order to get the desired temperature, the law of energy conservation is used.

Using the heat transfer equation, $q = UAF\Delta T_{lm}$

$\Delta T_{lm} = 22.5°C$

Total surface area, $A = 98m^2$

$A_f = 61\ m^2$ number of fin required = 26901.

$A_t = 37\ m^2$ total length of bare tube = 320m.

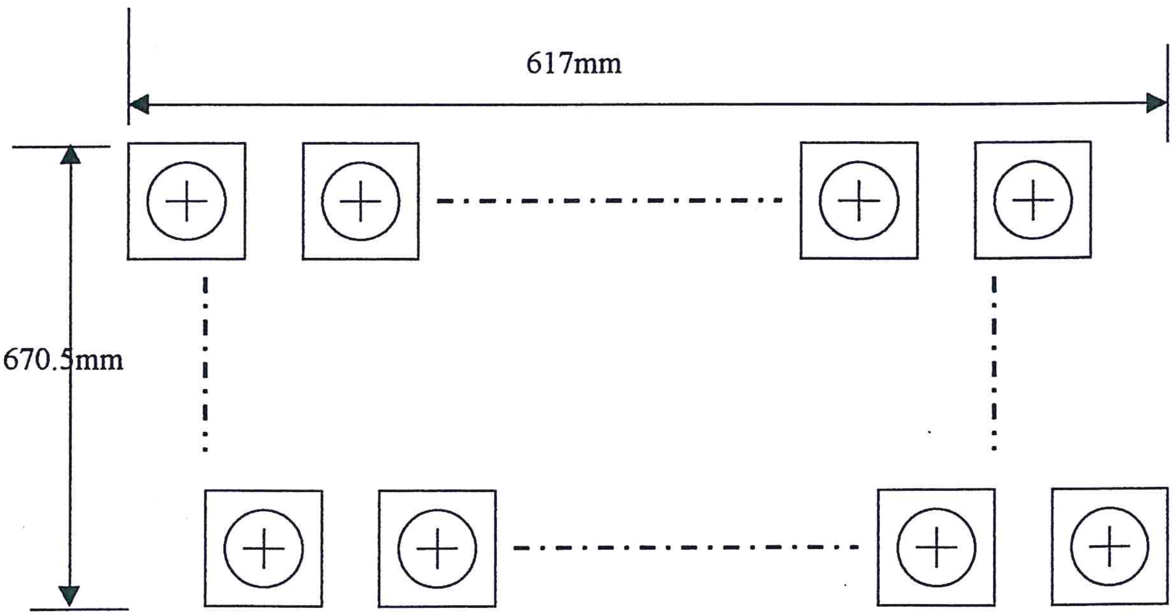
The total length of tube calculated above is the summation of the distance between adjacent fins.

Assuming thirteen number of passes

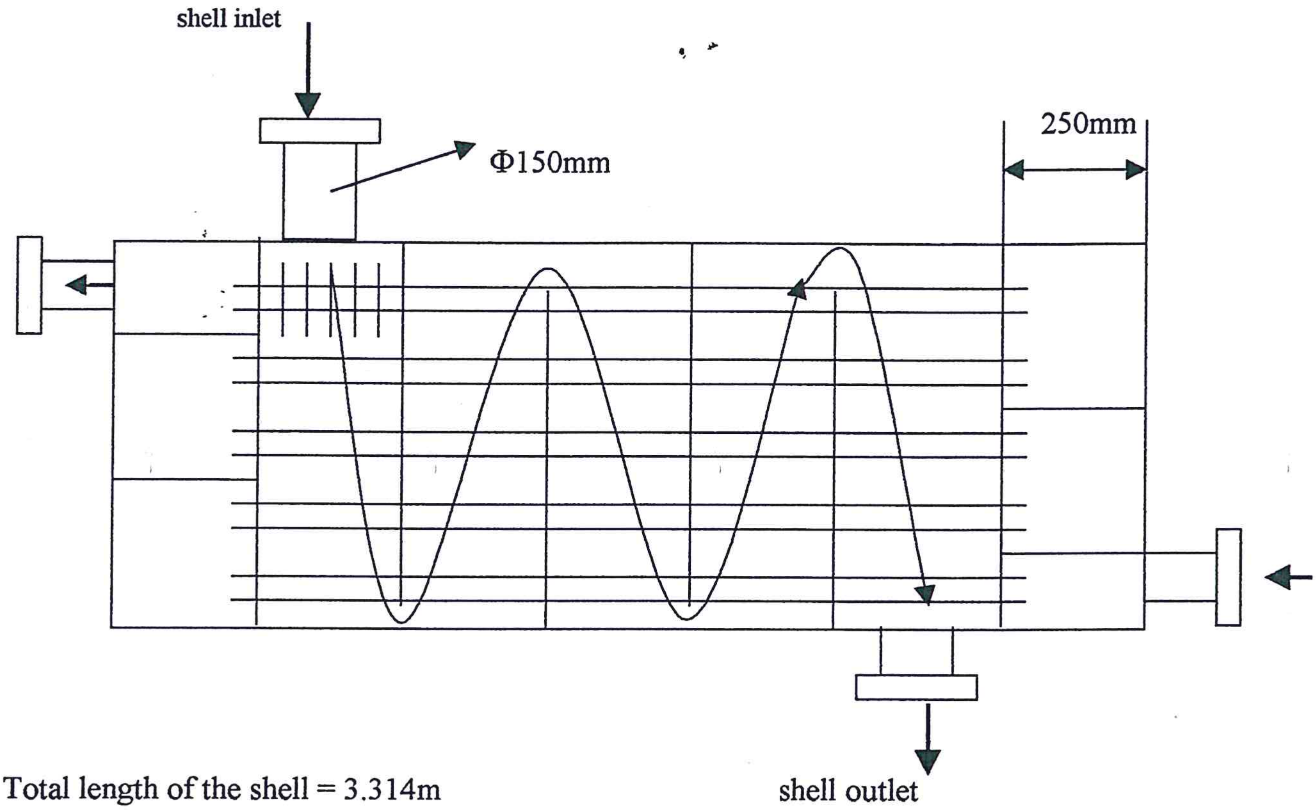
The number of tubes needed are $13 \times 10 = 130$

The number of fins on a tube = 26901/130 = 207
The length of a bare tube is 312/130 = 2.4m
So length of a tube containing fins = $2.4 + 207 \times 0.002 = 2.814$m

A front view of the fins arrangement is shown in the figure below



A schematic of the heat exchanger




Total length of the shell = 3.314m

Width of the shell = 620mm

Height of the shell = 0.871mm

Total length of tube needed = 365.82m

**Baffle**

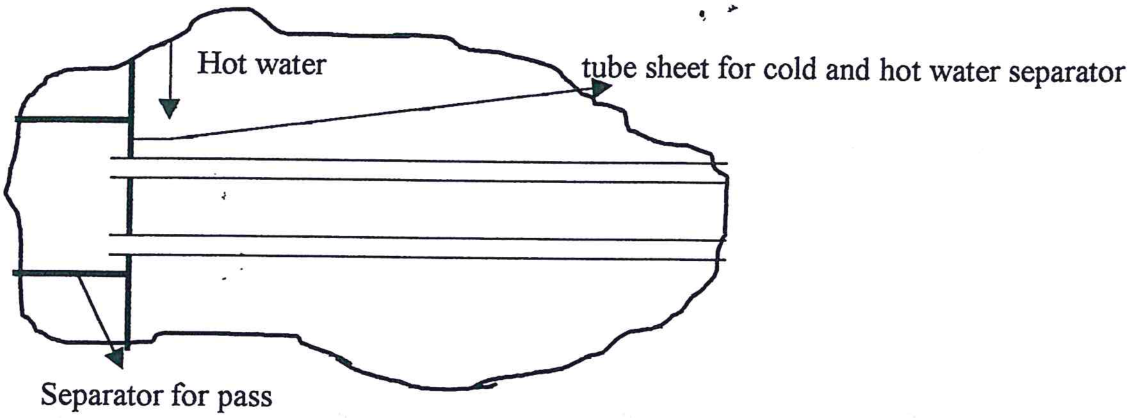
Baffles are used in the shell to direct the fluid stream across the tubes, it increase the fluid velocity and so improve the rate of heat transfer. The baffles will be made with aluminum rods.

Number of baffle needed = 9

Number of tube sheet needed for the passes = 12

Width and length of tube sheet are 250mm and 620mm respectively.

Total area of steel sheet needed = total surface area of shell + total surface area of tube sheet for passes ≈ $11m^2$




**RESUME**

Total surface area of aluminum sheet needed for fin ≈ $61m^2$

Total surface area of steel sheet needed ≈11 $m^2$

Total length of tube Φ37mm needed ≈ 365.82m

Flanged pipe used as hot and cold-water inlets and outlets &150mm.

## Calculating pressure loss in pipe.

When dealing with internal flows, it is important to be cognizant of the extent of the entry region, which depends on whether the flow is laminar or turbulent. The Reynolds number for flow in a circular tube is defined as

$$Re_D = \rho u_m D/\mu \ldots(5)$$

Where $u_m$ is the mean fluid velocity over the tube cross section and D is the diameter.

**The mean velocity**

Because the velocity varies over the cross section and there is no well-defined free stream, it is necessary to work with a mean velocity $u_m$ when dealing with internal flows. This velocity is defined such that, when multiplied by the fluid density $\rho$ and the cross-sectional area of the tube $A_c$, it provides the rate of mass flow through the tube. Hence

$$\bar{m} = \rho u_m A_c \ldots(6)$$

For steady, incompressible flow in a tube of uniform cross-sectional area, m and $u_m$ are constants independent of length. From equations 1and 2 it is evident that for flow in a circular tube ($A_c = \pi D^2/4$), the Reynolds number reduces to

$$Re_D = 4\bar{m}/\pi D\mu \ldots\ldots\ldots\ldots\ldots\ldots(7)$$

For a mass flow rate of 17.75 kg/s and a pipe of inside diameter of 135mm, the Reynolds number is 49890 and the mean velocity $\approx$ 1.24m/s : $\mu = 453 \times 10^{-6}$ N.s/m$^2$

Representative equivalent lengths in pipe diameter (l/d) of fittings

Fittings 90° standard elbow-------------------------- 30

45° standard elbow---------------------------16

Seven 90° standard elbow correspond to (l/d) =210

Two 45° standard elbow correspond to (l/d) =32

Total equivalent length of pipe(CPVC) = (210+32)*0.16 = 39m

$$J = \lambda u^2/2gD$$

J (pressure loss m/m)= 0.0075 from technical sheet 7.4

The head loss in pipe (external Φ160mm) including fittings = (900+39)*0.0075 ≈ 7.05m.

Difference in level between Bassin Board and the heat exchanger ≅ 13m

## HEAT EXCHANGER

There are two major sources of pressure loss on the tube side of the heat exchanger. The friction loss in the tubes and the losses due to the sudden contraction and expansion and the flow reversals that the fluid experiences in flow through the tube arrangement.

The tube friction loss can be calculated using the familiar equations for pressure drop loss in pipes. The basic equation for isothermal flow in pipes (constant temperature) is

$$\Delta p = 8 j_f (l/d_i)(\rho u_t^2/2) \quad \text{....(8)}$$

where, $j_f$ is the dimensionless friction factor and l is the effective pipe length.

The flow in the heat exchanger will clearly not be isothermal, and this allowed for by including n empirical correction factor to account for the change in physical properties with temperature. Normally only the change in viscosity is considered:

$$\Delta p = 8 j_f (l/d_i)(\rho u_t^2/2)(\mu/\mu_w)^{-m} \quad \text{..................................(9)}$$

m= 0.25 for laminar flow Re< 2100

m= 0.14 for turbulent flow Re> 2100

values of $j_f$ for the heat exchanger tubes can be obtained from fig 12.24 pg 541.

The pressure losses due to contraction at tube inlets; expansion at exits, and flow reversal in the headers, can be a significant part of the total tube side pressure drop. There is no entirely satisfactory method for estimating these losses. The loss in terms of velocity heads can be estimated by counting number of flow contractions, expansions and reversals, and using the factors for pipefitting to estimate the number of velocity heads lost. For fourteen tube-passes, there will be fourteen contractions, fourteen expansions and seven reversals. The head loss for each of these effects is: contraction 0.5, expansion 1.0, 180° bend 1.5; so for thirteen passes maximum loss will be $12\times0.5 + 12\times1.0 + 6\times 1.5$ = 27 velocity heads

= 2.25 per pass

So one can assume a value of 2.5 per pass.

Combining this factor with equation (9)

$$\Delta p = N_p(8j_f(l/d_i)\,(\mu/\mu_w)^{-m} + 2.5)\,(\rho u_t^2/2) \qquad \text{.........................(10)}$$

where $\Delta p$ is the tube side pressure drop in Pascal(Pa).

$N_p$ is the number of tubes-side passes.

$u_t$ is the tube side velocity m/s.

l is the length of one tube.

Another source of pressure drop will be the flow expansion and contraction at the exchanger inlet and outlet nozzles. This can be estimated by adding one velocity head for the inlet and 0.5 for the outlet, based on the nozzle velocities.

Practical analysis of flow in noncircular ducts is based on the idea of finding an "equivalent" circular pipe flow. The equivalent diameter, $D_{eq}$ is the diameter of a circle that would have the same cross sectional area ($A_{NCD}$) as the actual non-circular duct.

$$\boxed{D_{eq} = \sqrt{4}\, A_{NCD}/\pi.} \qquad \text{...(11)}$$

Using the idea above, one can find an equivalent pipe diameter for each of the passes.

Number of tubes per pass = 10

Internal diameter of tube = 34mm

Cross sectional area of tube = $908mm^2$.

$\therefore$ Total cross sectional area = $9080mm^2$.

$D_{eq} \approx 107.6mm$.

Using equation (7)

$Re_D \approx 3 \times 10^5$ $\mu = 739 \times 10^{-6}$ $N.s/m^2$

Using equation (6)

$u_m = 2.04m/s$

$j_f \approx 0.0022$

$\mu_w$ is evaluated at the average value of the mean temperature, $T_m = (T_{mi} + T_{mo})$.

$T_{mi} = 34°C$

$T_{mo} = 90°C$

$\mu_w = 453310^{-6}$ $N.s/m^2$

Using equation (10)

$\Delta p = 1.813$ bar, this is equivalent to a head loss of 18m.

The total head loss is equal to 38m.

Therefore a pump of 4 bars is needed in this operation when **CPVC PIPE** is used.

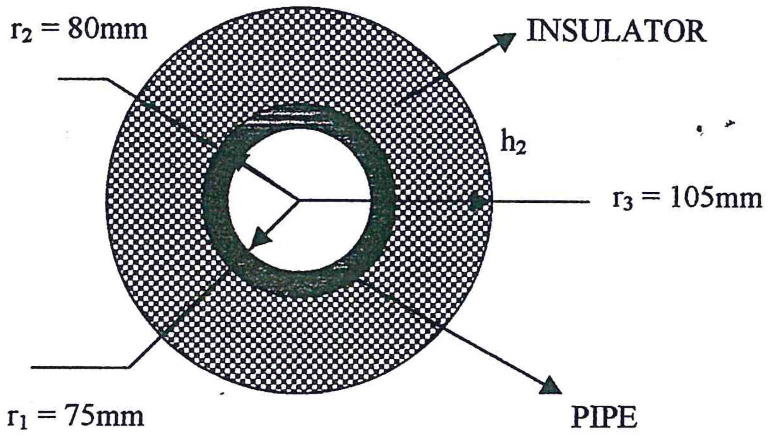
**USING STEEL PIPE**

The head loss in steel pipe including fittings = (900 + (210+32)*0.15)*0.02 = 19m

Total head loss = 50m

Therefore a pump of 6bar is needed in order to be on the safe side.

**Calculating heat loss in steel pipe.**




The heat transfer rate may be expressed as

$$q = \frac{T_{\infty 1} - T_{\infty 2}}{1/2\pi r_1 L h_1 + \dfrac{\ln (r_2/r_1)}{2\pi k_1 L} + 1/2\pi r_3 L h_2 + \dfrac{\ln (r_3/r_2)}{2\pi k_2 L}} \qquad \text{...(12)}$$

$T_{\infty 1}$ is the temperature of the hot fluid and is 70°C.
$T_{\infty 2}$ is the temperature of the ambient air and is approximately 25°C.
L is the length of pipe and is 900m.
$k_1$ is the thermal conductivity of steel pipe and is 60 W/m. K.
$k_2$ is the thermal conductivity of insulating material and is about 0.8 W/m. K.
$h_1$ is the convection coefficient of the hot fluid.
$h_2$ is the convection coefficient of the ambient air (negligible).

The inside convection coefficient may be obtained from the knowledge of the Reynolds number. Using equation (7)

$$Re = (4 \times 17.75) \div \pi \times 0.15 \times 453 \times 10^{-6}$$
$$= 3.325 \times 10^5$$

Hence the flow is turbulent and the Nusselt number is given by

$$\boxed{\mathbf{Nu = 0.023Re^{0.8}Pr^{0.3}}} \quad \textbf{...(13)}$$

$Nu = 0.023(3.325 \times 10^5)^{0.8}(2.45)^{0.3}$

$= 787$ W/m. K

Now using the Nusselt number the convective coefficient can be found in the equation

$$\boxed{\mathbf{h = Nu\ k/D}} \quad \textbf{...(14)}$$

Therefore $h_l = (787)\ 0.668/0.15$

$= 3504.25$ W/m. K

Using the equation above $q = 3.723 \times 10^5$ W/m$^2$

The temperature drop of the fluid can be computed by using the conservation of energy. The heat loss by the hot fluid is equal to the heat gains by the air.

Therefore $q = mc_p\Delta T$

$T = 70 - (3.723 \times 10^5/4200(17.75))$
$= 65°C$

Hence the temperature drop is 5°C.

**Cost analysis**

A cost analysis of the project works had been done in order to know the amount of money needed for implementation. The cost of materials and the cost of labour had been calculated.

The table that follows contains information about the cost of materials, which are needed for the realization of the project.

| MATERIALS | DIMENSIONS | PRICE in Rs |
|---|---|---|
| Steel pipe | 900m Φ 150mm | 267,000 |
| Valves | 4 × Φ 150mm | 72,000 |
| Fittings | 7 × elbow 90°<br>2 × elbow 45° | 11270<br>1610 |
| Pump | $64m^3/h$ at 5bar | 35,000 |
| Aluminum sheet | $61m^2$ | 6355 |
| Steel sheet | 3mm thick | 2250 |
| Insulating material | $452.4m^2$ | 146,810 |
| Aluminum sheet | $297m^2$ | 45,572 |

LABOUR COST (an approximation) considering only one worker.

Time needed for cutting the aluminum sheet in square of $2209mm^2$ is 2mins.

Time needed for drilling a hole of Φ37mm in the square fin of $2209mm^2$ is 2mins.

Time needed for welding a fin on the tube is 4mins.

Time needed for constructing the shell is 4 days.

The total time in days needed for constructing the heat exchanger is approximately 360

Number of hours that the person works per day is 9

The amount of money that he gets per month is Rs6000

Therefore the amount of money that has to be paid to the workers for constructing the heat exchanger is Rs98,000

So an approximation for the cost of the heat exchanger including 10% overheads is Rs 760,000.

## Conclusion

The design of the above heat exchanger together with the selection of the pipes and pump has been accepted by the maintenance manager, for an eventual construction. The cost analysis has shown that project is expensive but indeed it is valuable since an extra revenue would be obtained if the production of fish increases