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Optimizing the Flow Rate of Water from a Chiller System for Efficient Operation

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Abstract

During the welding process, there is a process of cooling the welding arm by using water that comes from the chillers. A water chiller system is used to remove heat from the process water through conductive heat transfer with an aid of a shell-tube heat exchanger. The main problem reported in the chiller during this process is the poor heat transfer of heat in the system during operation is due to poor mass flow rate of heat within the system. The system become inefficient due to poor mass flow of heat in the system that often leads to system overheating. There is therefore a need to conduct research and design a system that is more efficient during the mass flow of energy in the chiller system during operation. In the current study the performance of a chiller system is optimized by looking at the major parameters that impacts the flow of energy in the system during operation. The proposed system is modelled and simulated by looking at the parameters of the heat exchanger during optimal operation. The relevant theoretical models are derived using heat flow principles and the tools of Solid-works, Auto-cad and SCADA system are also employed in the study. Heat exchanger was simulated for flow-rate, heat-transfer and pressure difference and all three results were satisfactory to justify the positive outcome of the research. Hydraulic losses must be taken into consideration during simulation of the heat-exchanger for accurate results. Different optimal parameters of the heat exchanger gave different operation variation that impacts the efficiency of the system. The outlet temperature of the machine water was 15.8℃ simulation was as the calculation reflected 18.6℃ which was still acceptable and will not affected the cooling duty of 4MW of the evaporator. This is evident on the SCADA system that the inlet temperature refrigeration plant evaporator is 19℃. Through obtaining the results and simulation, evaporator performance was projected to be restored back to its operating designed limits and by that so, mass flow rate was improved, and heat transfer was optimized.

Keywords: heat exchanger, length, temperature, flowrate, design limit and optimize

I. INTRODUCTION

During car manufacturing process at Nampak through a welding machine, the chiller supply water to the welding machine which is a machine that rolls and welds a straight sheet of plate to a cylinder using a wire copper [1-3]. When the manufacturing process is in continuation the machine needs to cool down and cleans the system because it gets contaminated during the process from wire copper Debry [2]. During the welding process there is a process of cooling the welding arm with the water that comes from the chillers and the process cleans the system, so problem arises when the chillers cannot supply with the sufficient mass flowrate and that impacts the heat transfer efficiency [1-5]. Figure

1 is the setup of the system analysis as fluid circulate in the system during operation.

Figure 1: system analysis of the fluid circulation during operation

Heat exchangers are equipment used to exchange heat between two processes. Heat is transferred when these two processes are in direct contact with a thin wall. Heat Exchangers can be grouped into 2, direct and indirect heat exchangers. In direct heat exchangers there is direct contact between the two fluids, in indirect heat exchangers a heat transfer medium separates the fluids [1-6]. Pre cooling towers and cooling towers are an example of direct heat exchangers, while shell and tube or plate heat exchangers are indirect. In large refrigeration plants there are 2 main types of indirect heat exchangers used, the shell and tube heat exchanger and the plate type heat exchanger.

1.1 Shell and Tube Heat Exchanger

A shell and tube type design is most commonly used. This heat exchanger consists of a bundle of tubes inside a shell. One fluid is passed through the tubes, and the other fluid flows inside the shell and around the tubes. Heat is transferred through the walls of the tubes. The tubes are often finned to, increasing the surface area allowing great heat transfer to take place and thus increasing the duty. These heat exchangers can be made in a number of passes, with one pass being when the fluid flows into the tubes at one end, through the tubes and out at the other end. Multiple pass shell and tube heat exchangers can be made by redirecting the fluid back through the tubes again, this in effect doubles the exposure time of the fluid, making heat transfer more effective [1-9].

II. DESIGN CONSIDERATIONS

There are a number of considerations that need to be taken into account when designing a shell and tube heat exchanger such as:

- Tube diameters, using smaller tube diameters is cheaper, however this makes cleaning more difficult.
- Length. The important characteristic of any heat exchanger is the surface area, this can be increased by either adding more tubes or by increasing the length of the tubes. In general increasing the length of the tubes is more cost effective.
- Materials. The tubes are susceptible to corrosion and the buildup of scale, thus the material that they are made out of must be carefully considered. In most refrigeration systems cupronickel (90% Copper, 10% Nickel) or stainless steel are used.
- Maintenance. Dirty tubes (buildup of scale) can drastically reduce the performance. The tubes must be cleaned on a regular basis. There are several advantages of using a Shell and Tube heat exchanger in a refrigeration cycle, these include:
- Condensation or boiling heat transfer can be accommodated in either the tubes or the shell.
- The pressure drop can be varied and is generally less than that of a plate type heat exchanger. The pressure drop of the fluid inside the shell is generally less than that of the fluid in the tubes.
- The heat exchanger can be designed for relatively high pressures.
- For one shell, the surface area can be varied by removing tubes

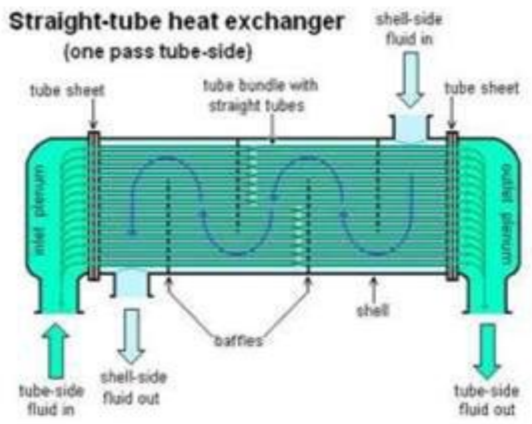


Figure 2: Straight tube heat exchanger being used in most study.

Most studies used the straight tube heat exchanger and the operation of a straight heat exchanger is inefficient due to the fact that the flow of air in the system is efficient. The focus of the research is to improve the rate of flow of the system by eliminating designing a shell-tube heat exchanger. The improve of air flow in the system will led to an optimal and efficient operation of the system during operation.

III. METHODOLOGY

This section of the research addresses a sub problem and a method used to design a suitable heat exchanger to increase the mass flow rate in order to have an improved (reduced) the temperature difference of the process water to from $46℃\ to\ 15℃$ in order to be suitable for use in the welding operation of Nampak. A selection of heat exchanger that will be installed in an inclined manner in the basement floor of the Nampal facilities, to dissipate heat from the water and through conduction and to assist inline coolers to reduce the high temperatures will primarily depended on acquiring theoretical thermodynamic principles and heat transferer equations useful for designing heat exchangers as shown in Fig.3 and their major operating parameters are shown

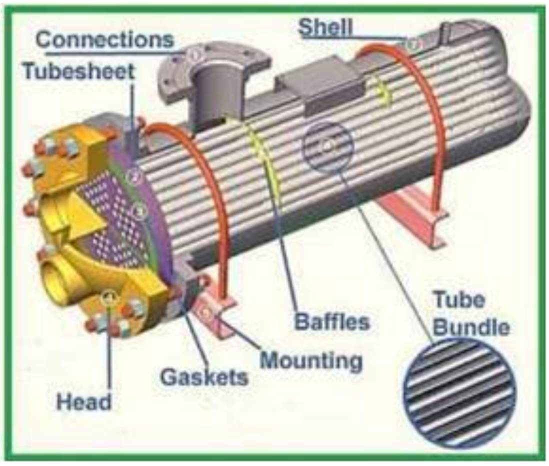


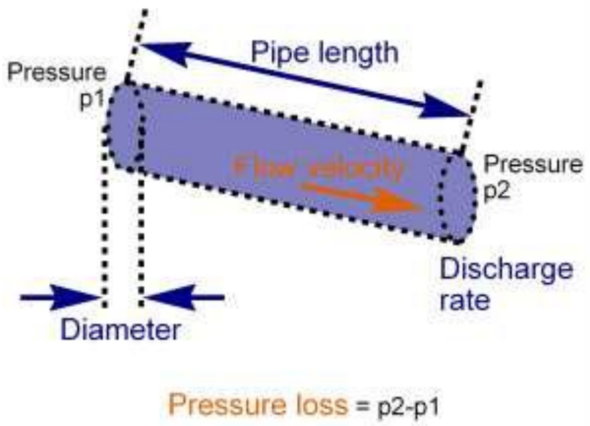
Figure 3: Major parameters that impacts the performance of heat exchanger during operation.

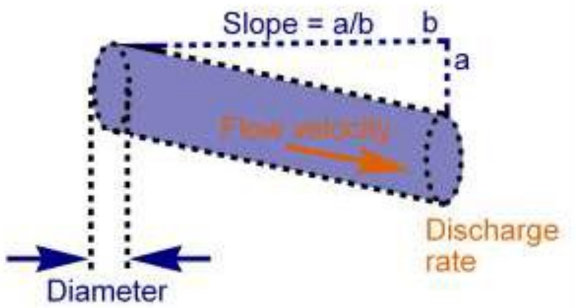
The optimum thermal design of a shell and tube heat exchanger involves the consideration of the many interacting design parameters which can be be summarized in the follows

IV. PROCESS:

1. Process fluid assignments to shell side or tube side.
2. Selection of stream temperature specifications.
3. Setting shell side and tube side pressure drop design limits.
4. Setting shell side and tube side velocity limits.

For a given heat transfer service with known mass flow rates and inlet and outlet temperatures the determination of Q is direct and ΔTm can be easily calculated if a flow arrangement is selected (e.g. Logarithmic Mean Temperature difference for pure countercurrent or concurrent flow). The literature has many tabulations of such typical coefficients for commercial heat transfer services.





The Bernoulli equation states that velocity is determined by calculating difference in pressure between two points, multiplying by 2, dividing by the density of water and then taking the square root. You then get the flow rate by multiplying the velocity by the cross-sectional area of the pipe

It can be derived from the Bernoulli's equation that a higher pressure drops acting on a pipe creates a higher flow rate. A wider pipe also produces a higher volumetric flow, and a shorter pipe lets a similar pressure drop provide a greater force.

The final factor controlling a pipe's viscosity is the fluid's viscosity. This factor measures the fluid's thickness in poise, or dyne seconds per square centimeter. A thicker fluid flows more slowly under the same pressure.

The design (thermal) of heat shell-tube exchangers was directed to calculate an adequate surface area to handle the thermal duty for the given specified parameters whereas the hydraulic analysis determined the pressure drop of the water flowing in the system. The pumping power work input necessary to maintain the mass flow was to be considered as well.

Velocity of the fluid (ut), cross-sectional flow area of the tube (Ac), and the number of tubes (Nt).

$$mt = \rho\ tut\ AcNt \tag{1}$$

$$N_t = \frac{4m_t}{\rho_t u_t A_c} \tag{2}$$

A methodology used to select and design a suitable heat exchanger was through having preliminary ideas, experimentation, calculations, simulations and discussing the finding;

V. EXPERIMENTATION

There are two methods used in experimentation to model the heat exchanger, these methods are namely; Kern method and Bell- Delaware method. These methods were uses thermal and hydraulic analysis exclusively for both tube and shell sides. Kern method is a simple way, and this considered in the study.

b) Distinguished between laminar and turbulent flow inside the pipes.

This was achieved by calculating the Reynolds Number.

$$Re_t = \frac{\rho_t u_t d_t}{u_t} \tag{3}$$

where µ is the viscosity of the tube-side fluid, ut is fluid velocity inside the. tubes, and ρt is the density of fluid in the tubes.

4.1.1 Kern Method

To endeavor and associate data for heat exchangers using a simple equation, an equation was developed for fluid flow inside the tubes, this equation was called the Kern Method. It was however its usage is limited to a cut (fixed) 26%. This cannot afford to be provisionally responsible or consider shell-baffle and tube- baffle leaks. Its limitation is that, it does accurately a rapid shell side coefficient calculation including pressure drops

c) Determining the tube side Nusset number.

Nusselt number is a function of Reynolds number (Re) and Prandtl number (Pr). However, there are equations developed according to the type of flow.

$$Nu_t = \frac{(f/2)\mathrm{Re}_t \mathrm{Pr}_t}{1.07+12.7(f/2)^{1/2}(\mathrm{Pr}_t^{2/3}-1)} \quad (4)$$

$$f = (1.58\ln \mathrm{Re}_t - 3.28)^{-2} \quad (5)$$

Thermal Analysis for the Tube Side

a) Determining the number of tubes Inside the tube, the flow rate (mt) is a function of the density of the fluid (ρt), the

d) Tube side Heat Transfer coefficient

$$h_t = Nu_t \frac{k_t}{d_i} \quad (6)$$

3.2.3.2 Thermal Analysis for the Shell Side

a) Shell Diameter

$$N_t = (CTP)\frac{\pi D_s^2}{4A_1} \quad (7)$$

A1 is the area of the tube

Ds is the diameter of shell inside

CTP is the number of tubes

PR is tube pitch ratio

$$PR = \frac{P_T}{d_o} \quad (8)$$

$$D_s = 0.637\sqrt{\frac{CL}{CTP}}\left[\frac{A_o (PR)^2 d_o}{L}\right]^{1/2} \quad (9)$$

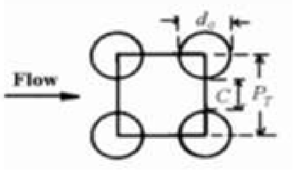
Ao is the outside heat transfer surface area

$$A_o = \pi d_o N_t L \quad (10)$$

b) Shell Equivalent Diameter

$$D_e = \frac{4\times \text{free-flow area}}{\text{wetted perimeter}} \quad (11)$$

Triangular Pitch



$$D_e = \frac{4\left(P_T^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o} \quad (12)$$

$$D_e = \frac{4\left(\frac{P_T^2\sqrt{3}}{4} - \frac{\pi d_o^2}{8}\right)}{\pi d_o/2} \quad (13)$$

The number of tubes in the center line

$$N_t = \frac{D_s}{P_T} \quad (14)$$

$$A_s = \frac{D_s}{P_T} C B \quad (15)$$

Tube Clearance: C

$$C = P_T - 2r_o = P_T - d_o \quad (16)$$

Shell side mas flow rate

$$G_s = \frac{\dot{m}_s}{A_s} \quad (17)$$

c) Shell side Raynolds Number

$$\mathrm{Re}_s = \left(\frac{\dot{m}_s}{A_s}\right)\frac{D_e}{\mu_s} \quad (18)$$

d) Shell side Heat Transfer coefficient

$$h_o = \frac{0.36k_s}{D_e}\mathrm{Re}_s^{0.55}\mathrm{Pr}_s^{1/3}$$

$$\text{for} \quad 2\times10^3 < \mathrm{Re}_s = \frac{G_s D_e}{\mu} < 1\times10^6 \quad (19)$$

k the thermal conductivity of the shell- side fluid

Overall Heat Transfer of the Coefficient of the Heat Exchanger

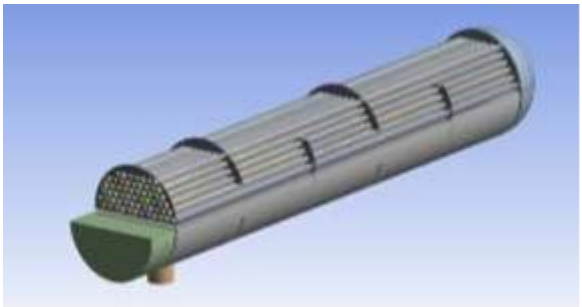
$$\frac{1}{U_c} = \frac{1}{h_o} + \frac{1}{h_i}\frac{d_o}{d_i} + \frac{r_o \ln(r_o/r_i)}{k} \quad (20)$$

Equation (1-20) are solved and simulated using the relevant software and results are obtained and discussion below.

V. RESULTS

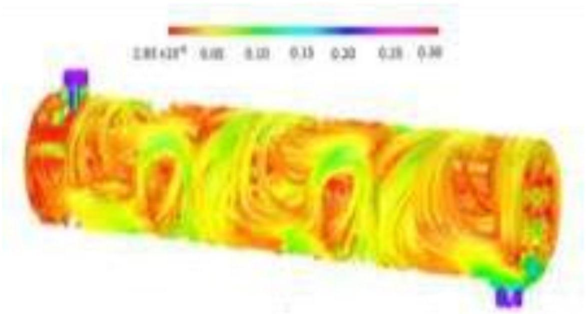
The first drawing is a drawing assembly of a shell tube heat exchanger, the process chilled water is inside the tubes while the cooling water flows outside the tubes. The blue color indicates cold water travelling outside the tubes to chill hot process machine water travelling inside the tubes indicated by the red color. The green like color indicates the heat transfer occurring within the

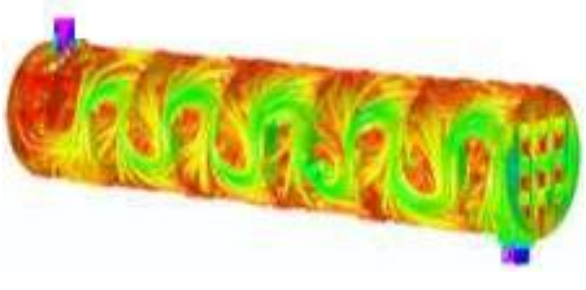
3m distance from the inlet of the heat exchanger. The orange like color indicates the heat gained by the cold water during heat transfer, this shows that the heat transfer is complete.

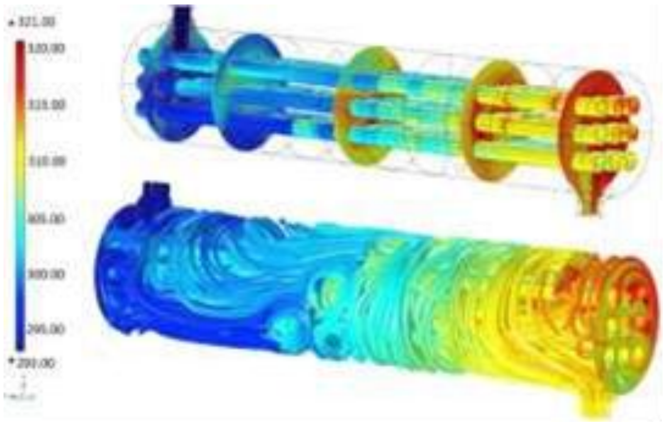


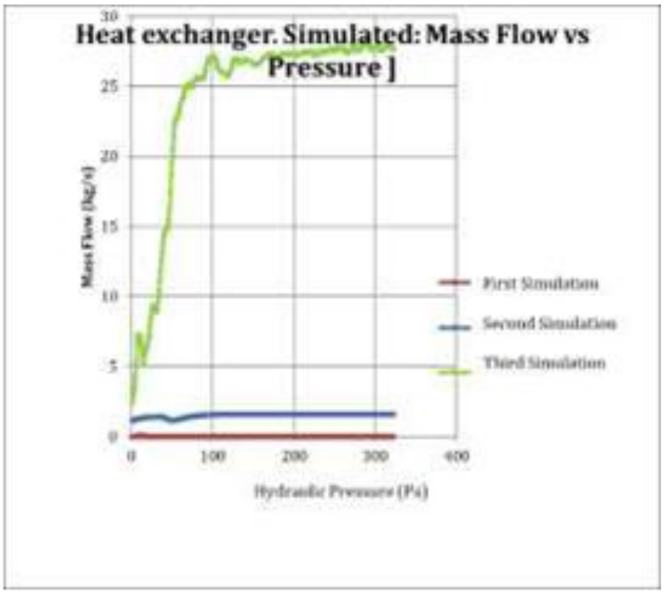
There are 3 simulations completed on it, the first simulation shows poor heat exchange as a result of poor mass flow rate, the second simulation shows better heat exchange but with poor mass flow rate. The third simulation shows a good heat exchange due to increased mass flowrate.

Flow was turbulent inside the shell and in the tubes, mass flow rate was very high on both sides and the contact area with the tubes was 98%, heat transfer rate was observed to be satisfactory for the flow rate, though deliberately, the length of the heat exchanger was 5.2 meters were as in actual reality, the length of the heat exchanger is 6m.









First Simulation: Poor heat transfer due to poor mass flow rate

As seen on the graph of Mass flow rate vs Pressure graph, which was drawn on simulation through solid works, the first simulation was observed to have not yielded the expected mass flow results, though the pressure drop between two points, (inlet and outlet) pressures is different. The learning was that, not only does mass flow rate depend on the pressure hydraulic difference in the system, but also the density of water, size of the tubes. The second simulation (highlighted in green shows a constant mass flow rate of 2.76 kg/s. This value was still not acceptable as the expected mass flow value was any value greater than the 15 kg/s in order to obtain a desirable heat exchange output. The third simulation (highlighted in green) shows the increasing mass flow rate with an increasing hydraulic pressure of the system. This was achieved through correctly sizing the heat exchanger tubes to minimize restriction of the water flow, for the given hydraulic pressure difference.

Pressure and length of the heat exchanger were observed to be directly proportional with keeping the mass flow rate, thickness of the tubes, the material type, inlet and required outlet temperatures the same. The outlet temperature of the machine water was 15.8℃ simulation were as the calculation reflected 18.6℃ which was still acceptable and will not affected the cooling duty of 4MW of the evaporator. The evaporator performance was projected to be restored back to its operating designed limits and by that so, the refrigeration plant was optimized.

This is evident on the SCADA system that the inlet temperature refrigeration plant evaporator is 19℃. Through obtaining the results and simulation, evaporator performance was projected to be restored back to its operating designed limits and by that so, mass flow rate was improved and heat transfer was optimized.

6. CONCLUSION

The current study was aimed optimizing the performance of a heat exchanger. To achieve this objective, the major parameters that impacts the flow of fluids in a heat exchanger were modelled. It was learnt that the contact surface of the water is increased by increasing the Tube length and Tube diameter. Due to the restricted size of the excavation of the water way, the designed shell- tube heat exchanger diameter was fixed to 750mm which through company restricted calculations, the diameter of the tubes was 50mm made out of copper and were machined to 1mm thickness. These tubes were further roughened to increase the contact area of water, the water temperature inlet was 46℃ and it dropped to 15℃ successfully, which was required as an inlet temperature into a 4MW refrigeration plant's evaporator to increase the efficiency to 74%.

7. RECOMMENDATION

Throughout the simulation process, it was discovered that the simulation results are prone to be inaccurate due to one parameter not being taken into consideration, the hydraulic losses. For the results to be accurate, the hydraulic losses must be taken into consideration for all three sets of simulation as mass flow rate results and heat transfer results may be inaccurate.

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