A STUDY ON THERMAL-HYDRAULICS CHARACTERISTICS FOR DESIGNING A SHELL AND TUBE CONDENSER FOR A 1200 MW$_e$ NUCLEAR POWER PLANT

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ABSTRACT

The study explores the thermal-hydraulics parameters of a condenser of a nuclear power plant with 1200MW$_e$ net electric output and 37% thermal efficiency using empirical correlations of pressure drop and heat transfer coefficient both for the tube and shell sides. Considering a two-phase fluid system, a shell and tube condenser with coolant water on the tube side and condensing steam on the shell side has been selected. For designing a condenser with a thermal load of 2060MW$_{th}$, the input temperature data of cold fluid inlet and outlet temperatures are taken as 29.4ºC and 40ºC while the condensation temperature is taken as 53.97ºC. Transverse, two-pass condenser with 4 shell tanks has been considered in this study and the length of each shell tank is taken as 14m. Based on these input data, this work finds heat transfer area, logarithmic mean temperature difference (LMTD), and convection heat transfer coefficient inside the tubes as 549536m^2, 18.74°C, and 2869.85W/m^2.ºC respectively for 20mm tube outer diameter. Hydrodynamic parameters relating to the friction factors and pressure drops on tube side are found as 0.031 and 14.86kPa respectively. Similar design data have been generated for varying coolant inlet temperatures and tube inner diameters. Results reveal that velocity of flow inside the tubes as well as the number of tubes in a bundle decrease with the increase in tube diameter. Finally, the thermal-hydraulic data may be used to design a large scale commercial condenser to be applicable for a large scale nuclear plant since limited design data are available in the literature.

INTRODUCTION

Surface condenser is an essential component of a Rankin Cycle for electricity generation in a nuclear power plant whose rapid development depends on dynamic characteristic of a condenser (Lebele-Alawa and Egwanwo 2012; Nazarov and Zaekin 2007). The thermal design of a condenser is related to enumerate proper surface range to handle certain thermal load for the designated specifications while the hydraulic analysis decides the pressure drop of the flowing fluid in the system below the maximum allowable values (Bell 1983; Saunders 1988; Thomas 1993).

The logarithmic mean temperature difference (LMTD) is a well-established method for finding the temperature difference and the heat transfer area for designing a counter flow as well as a parallel flow condenser (Incropera *et al.* 2007). An analytical expression for evaluating effective temperature difference was developed by Underwood *et al.* (1934) and later modified by Bowman *et al.* (1940). Heat transfer characteristics evaluation was done for turbulent pulsating fluid flow. And the effects of flow geometry variables for a circular tube are changed with the transient heat transfer and turbulent conditions. The enhancement of the Nusselt number with the increase in the Reynolds number is presented by Zohir (2012), Tandiroglu (2006) and Promvonge (2010). In

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reality, lot of constraints are appeared in a power plant where there is a tendency to reduce or increase the output power than the designed power and heat rate (Geetea and Khandwawala 2013; Karri 2012). Various techniques and methods of the revolutionary algorithms such as Genetic Algorithms (GA) are uncovered gateways for the solution of heat transfer problems (Gosselin *et al.* 2009). Selbas *et al.* (2006) had utilized a special type of code for the most appropriate design of a shell-and-tube condenser by altering the design parameters for instance outer tube diameter, number of tube passes, tube layout, outer shell diameter, baffle cut and baffle spacing. By applying the algorithm for example GA provides significant advancement in the optimal condenser design which contributes minimum cost and significantly attributes faster methods for getting multiple solutions of the defined situation (Wildi-Tremblay and Gosselin 2007; Babu. and Munawar 2007). Genetic algorithms are used not only for the most appropriate design of a shell and tube condenser but also for optimization of the leading geometric parameters (Ponce-Ortega *et al.* 2009). El-Fawal *et al.* (2011) proposed a computer code for getting the most economic design of a heat exchanger. The minimization cost of the equipment is obtained for the various range of pressure drop. Genetic algorithm is capitalized to solve the optimization problem associating to the decrement of the pumping power (Guo *et al.* 2009). Sanaye and Hajabdollahi (2010) considered several design parameters for achieving the maximum effectiveness as well as the minimum cost of the determining parameters of tube arrangement, tube length, tube diameter, number of the tubes, tube pitch ratio, baffle cut ratio and baffle spacing ratio. The work was developed by applying Genetic algorithms (GA) for the reduction of cost or configuration of the mass along with volume, improvement of heat transfers and number of transfer units (Özçelik. 2007; Allen. Gosselin 2008; Xie *et al.* 2008; Valdevit *et al.* 2006; Ozkol and Komurgoz 2005). Fesanghary *et al.* (2009) analyzed the application of global sensitivity analysis (GSA) and harmony search algorithm (HSA) for the most appropriate design of the heat exchangers. By considering single phase fluid on tube and shell sides, a computer-based model for shell and tube heat exchanger was prepared (Kara and Güraras 2004). To increase the heat transfer rate in a shell and tube type condenser, the segmental baffles with different orientation were introduced inside the cover pipe (Gaddis Gnielinski 1997; Singh and Sehgal 2013). The flow arrangement using this analysis was laminar which is more efficient for counter flow arrangement rather than parallel flow arrangement (Zhang 2010). Xie *et al.* (2007) implemented an experimental arrangement for investigating the performance of E and J types shell and tube condensers. A new strategy was utilized to determine single phase shell side heat transfer coefficient on the basis of Tubular Exchanger Manufacturers Association (TEMA) style (Ayub 2005). Hosseini *et al.* (2007) analyzed heat transfer coefficient and pressure drop on the shell side of a condenser for three types of fins with copper tubes. The basic design details for a variety of condensers are given in Kern's process heat transfer text-book (Kern 1950). Lyczkowski (1984) proposed the outlines on how heat exchanger design techniques evolved over the hemisphere. Than *et al.* (2008) estimated the heat transfer area and pressure drop. After that, the inquiry has been done whether the assumed design satisfies all requirements or not. The primary motivation of this design was to find the maximum heat transfer rate by keeping the allowable pressure drop. Mukherjee (1988) explained the basics of thermal design of the heat exchanger which is covering a lot of topics such as shell and tube
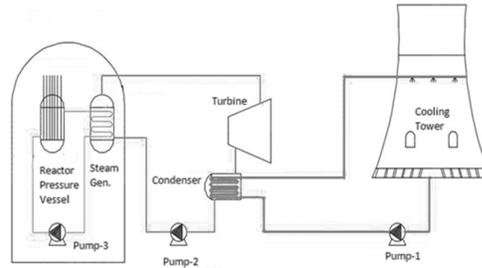
heat exchanger (STHE) components, classification of STHE, according to construction and service, tube side design, shell side design, tube layout, baffling, shell side pressure drop and the mean temperature difference. A compact formula which is based on Bell-Delaware method was used to assess the shell-side pressure drop as well as the film coefficient (Serna. and Jimenez 2005). Costa and Queiroz (2008) surveyed several techniques which were exercised according to the specific problem whose formulations were relating to the heat transfer area, total annualized costs, heat transfer and fluid flow equations, pressure drop, and velocity bound, and decision variable. A non-traditional optimization technique for a shell and tube heat exchanger named as particle swarm optimization (PSO) is adopted to understand economic view point. Minimization of the total annual cost was the goal of this study. Three important design variables such as shell internal diameter, outer tube diameter, and baffle spacing were considered in their design (Patel and Rao 2010; Caputo *et al.* 2008).

There is no particular scientific study that focuses on design and thermal-hydraulic performances a condenser for a 1200MW$_e$ nuclear power plant with heat load of 2060MW$_{th}$. In this work, a linear approach is applied to scrutinize for designing a large-scale condenser with allowable working pressure drop and film heat transfer coefficient based on Kern method (Sinnott *et al.* 1993). Kern method was established based on the experimental task. It provides significant prediction of heat transfer coefficient for a condenser design. However, for the pressure drop, this method does not provide good prediction. The thermo-physical parameters are varied with length, diameter, and temperature where pressure drop on shell side is compromised. For a nuclear power plant, usually titanium-based alloy material of the tube is considered for aggressive brackish water and seawater.

THEORY FOR ASSESSMENTS OF THERMAL-HYDRAULICS PARAMETERS

Fig. 1 shows the role of a condenser in order for heat removal characteristics of a nuclear power plant.

Fig. 1. Nuclear Power Plant Cycle

Pressure Drop

Pressure Loss Inside Tube

The pressure loss can appear not only in the tube side but also in the shell side of a shell and tube condenser. The pressure drop happens because of the friction inside the tubes, sudden contraction, expansion, and flow reversals that the fluid experiences when it flows through the tube structure. The loss with regard to velocity heads can be evaluated by measuring the number of flow contractions, expansions, reversals, and using the factors relating to pipe fittings. For this reason, Frank's recommended that the constant value such as 2.5 should be added for each pass in order to get the most pragmatic value of pressure drop (Sinnott *et al.* 1993). Putting this constant factor in pressure drop Eq. (1) for inside tube, it appears as;

$$\Delta P_t = N_p \left[8 j_f \times \left(\frac{L}{d_i}\right) \times \left(\frac{\mu}{\mu_w}\right)^{-0.14} + 2.5 \right] \times \rho \frac{u_t^2}{2} \quad (1)$$

Velocity of fluid inside tube is calculated by the following Eq. (2);

$$u_t = \frac{G_t}{\rho} \quad (2)$$

The value of fanning friction factor (j_f) for the tube side has been taken from Appendix-A for commercial pipes corresponding to the Reynolds number (Re). High velocity is suitable to block any suspended solids settling but it should not be so high that causes erosion. Typical design velocities inside the tube is allowed to 1 to 2 m/s for process fluid and to be maximum up to 4 m/s if it is necessary to reduce fouling.

Heat Transfer
Heat Transfer Inside the Tube

The physical properties of fluid are conveniently incorporated into the correlation for computing heat transfer coefficient. Heat transfer coefficient of water (h_i) is calculated by Eq. (3);

$$h_i = \frac{\{4200 \times (1.35 + 0.02t) \times U_t^{0.8}\}}{d_i^{0.2}} \quad (3)$$

The general equation for heat transfer across a surface is given by Eq. (4);

$$Q = U \times A \times \Delta T_m \quad (4)$$

Heat Transfer Inside the Shell

Usually, the velocity of steam inside the shell is very low in order to prevent turbulence. At lower velocities, the condensation heat transfer coefficient becomes almost independent of the velocity of the flow. Considering the condenser shell to be horizontal, the shell side average heat transfer coefficient due to condensation over horizontal tube surface ($h_{condensation}$) can be calculated from the equation derived by Nusselt that is shown in Eq. (5),

$$h_{condensation} = 0.729 \left[\frac{g \rho_l (\rho_l - \rho_v) h_{fg} k_l^3}{\mu_l (T_{steam} - T_{wall}) D_o}\right]^{0.25} \quad (5)$$

This equation does not account for the presence of non-condensable gases in the steam. The presence of even 1% air in the steam may result in reduction of the heat transfer coefficient to half. In order to account for the effect of presence of air on the condensation heat transfer coefficient, the partial pressure of steam at the liquid-vapor interface is needed to be calculated, which is given by Eq.(6),

$$\frac{P_{v.i}}{P_v} = \frac{1 - x_{a.i}}{1 - \left(1 - \frac{m_a}{m_v}\right) x_{a.i}} \quad (6)$$

Here $x_{a,i}$ is the mass fraction of air in the liquid-vapor interface. For bulk mass fraction of air in the range 0.01-0.05, the partial pressure of vapor at the interface may be as low as 50% of the actual condenser pressure, which will reduce the interface temperature and thus the heat transfer coefficient. The corrected shell side heat transfer coefficient can be given by Eq. (7), $h_{shell} =$

$$\frac{1}{N_{blank}^{0.25}} h_{condensation} \left[\frac{T_{steam,i} - T_{wall}}{T_{steam} - T_{wall}}\right]^{0.75} \quad (7)$$

Here, the average number of tubes in condenser tube blank is approximated by Eq. (8),

$$N_{blank} = \sqrt{N_{pass} N_{tube}} \quad (8)$$
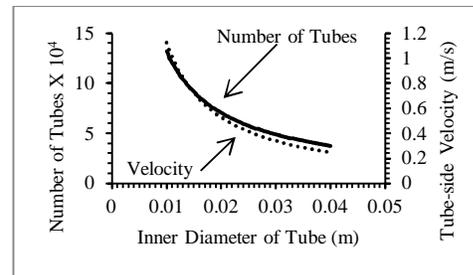
Other Related Physical Parameters

Shell Diameter

Tube Count Constant (CTP) is required for the incomplete coverage of the shell diameter by the tubes. It is also necessary for making clearances between the shell and the outer tube circle. Shell diameter D_s is calculated by the following Eq. (9);

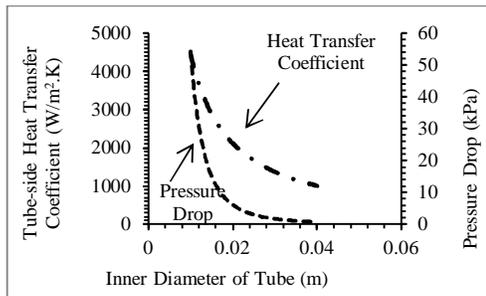$$D_s = 0.637 \left(\frac{CL}{CTP}\right)^{0.5} \times [3.14 \times N_t \times (1.25 \times d_o)^2]^{0.5} \quad (9)$$

RESULTS AND DISCUSSION

For the suggested shell and tube condenser design, parameters associated with heat transfer coefficient and pressure losses are evaluated and their performances are produced in plotting graphs by varying length of the tube keeping the diameter constant and vice versa.
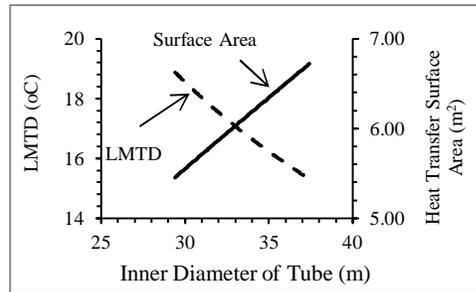

Fig. 2. Velocity of fluid inside a tube vs. inside diameter of a tube and number of tubes of a condenser vs. inside diameter of a tube for a constant tube length of 14 m.

Fig. 2 shows the velocity of fluid inside the tube and the number of tubes of a condenser for a constant tube length of 14 m. The number of shell tanks is taken as 4 and number of tube pass is taken as 2. Also, the coolant inlet and outlet temperature is taken as 29.4°C and 40°C respectively while the condensation temperature of steam is selected as 53.94°C. If the inside diameter of the tube is increased infinitely, then the velocity of fluid inside the tube will be lowered and the number of the tubes will also be reduced significantly.
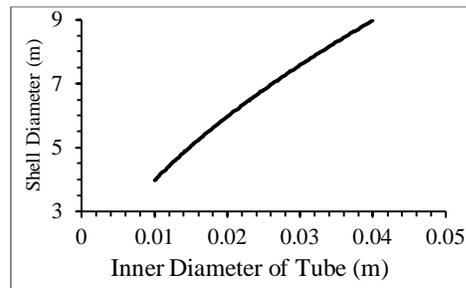

Fig. 3. Heat transfer coefficient inside the tube vs. inside diameter of a tube and pressure drop inside A tube fluid vs. inside diameter of tube for a constant tube length.

Fig. 3 shows the heat transfer coefficient and the pressure drop of fluid inside a tube. Heat transfer coefficient inside the tube relates to the average fluid temperature, the velocity of the fluid, and the cross-section of the tube. In order to calculate the pressure drop inside the tube, the Reynolds number and friction factor are needed to be known. The pressure drops significantly with the increase of the inside diameter of the tube. Same effect is also being observed for the heat transfer coefficient which is unwanted for a good design criterion. For better performance of a condenser, two effects need to be scrutinized carefully.

Fig. 4. LMTD vs. cold fluid inlet tempera-ture of a condenser and heat transfer surface area of a condenser vs. cold fluid inlet tempera-ture of a condenser for a constant tube length.
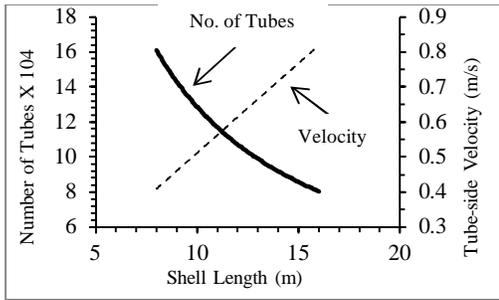
Fig. 4 shows the heat transfer surface area and the LMTD of a condenser. Heat transfer surface area depends on the heat duty, specific heat, temperature difference, and mass flow rate. On the other hand, the LMTD is used to determine the temperature driving force for heat transfer in flow systems. The value of heat transfer surface area is increased marginally with the increase of cold fluid inlet temperature. There is a reciprocal relationship between the LMTD and the inlet temperature of cold water.

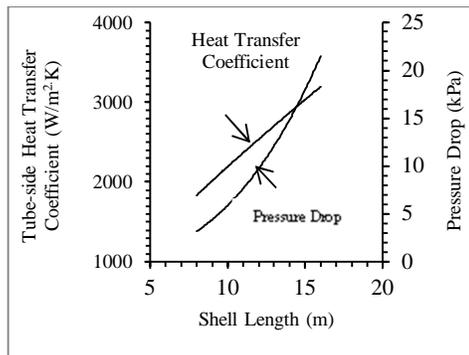**Fig. 5.** Diameter of a shell of a condenser vs. outside diameter of tube for a constant tube length.

Fig. 5 shows the diameter of a shell and diameter of a tube bundle. This graph represents the shell diameter over the outside diameter of tube for constant tube length in the range of diameter of 0.01m to 0.04 m. There is a continual increment outside diameter with the marginal consistent increase of shell diameter. To preserve optimum

design condition, diameter of the shell and tube bundle of a condenser is selected as 6.68 m and 5.40 m respectively.

Fig. 6. Velocity of fluid inside a tube vs. length of a tube and number of tubes vs. length of a tube for a constant inside diameter of a tube.
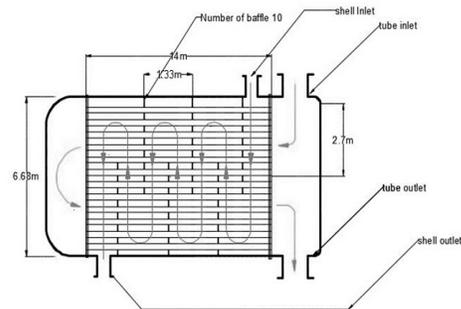
Fig. 6 shows the velocity of fluid inside of a tube and the number of tubes versus length of the tube for a constant inside tube diameter. Economic point of view should be considered carefully for the tube through where hot fluid flows. Using the value of the length of the tube, one may calculate those two parameters. If the length of the tube is increased, then the velocity inside the tube will be increased. Commercial grades smooth tubes without surface roughness are considered for thermal-hydraulic analyses. From Fig. 6, it has been figured out that there is an inverse relationship between a number of tubes and length of the tube.
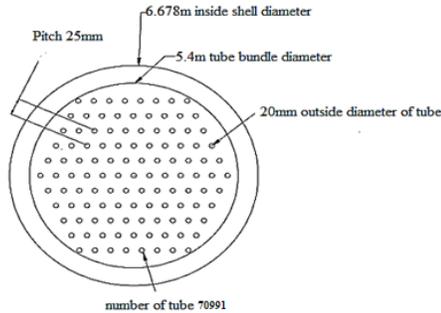

Fig. 7. Heat transfer coefficient inside a tube vs. length of the tube and pressure drop inside a tube vs. length of the tube for constant inside diameter.

Fig. 7 shows the heat transfer coefficient and the pressure drop inside the tube fluid for a constant tube diameter contour. Heat transfer coefficient inside the tube relates to the average fluid temperature, the velocity of the fluid and the cross-section of the tube. In order to calculate the pressure drop inside the tube, the Reynolds number and friction factor of the tube need to be calculated. There is a positive effect on the heat transfer coefficient inside the tube with the increment of the tube length. There is also a positive effect on the pressure drop inside the tube with the increment of the tube length.

Kern method is applied to design a counter flow shell and tube condenser with two passes on tube side and triangular tube pitch arrangement. Length of the tube is chosen as 14 m for optimum pressure drop of 14.86 kPa, whereas inside diameter and tube thickness are found as 20mm and 1mm respectively. The simple design concept of a shell and tube surface condenser is shown in Fig. 8 that can be suitable for a 1200 MW_e nuclear power plant. Fig. 9 shows the tube arrangement inside a tube bundle. The cold fluid is chosen to flow inside the tube side of a condenser whereas the hot fluid (steam-mixture) is chosen to flow inside the shell side.

Fig. 8. Suggested design of a shell and tube condenser.

Fig. 9. Suggested condenser's physical parameters for triangular tube bundle arrangements and a shell.

CONCLUSION

From this theoretical analysis, it is clearly understood that the parameters of heat transfer area, inside and outside fluid film coefficient, and pressure drops on tube and shell sides heavily influence the performances of a condenser. The optimization of these parameters is obtained by iterations. It is found from these results that the value of heat transfer surface area and LMTD has a negative relation with the cold fluid inlet temperature while the size of the condenser is greatly depended on heat transfer surface area. The magnitude of heat transfer surface area and LMTD of 549536 m² and 18.74°C respectively are found from the calculated data. It is seen from the analysis, not only there is a reciprocal relationship between the inside diameter of the tube and the heat transfer coefficient but also the same effect has been found between the inside diameter of the tube and the pressure drop. The tube-side heat transfer coefficient is found as 2869.85W/m².°C. This has happened due to the decrease of velocity inside the tube. For the proper design of a condenser, the velocity cannot be increased beyond a certain limit. For this reason, the pressure drop inside the tube is being kept as

14.86 kPa. If the length of the tube is increased, then the number of tubes will be decreased. So, this parameter has an effect on the sizing of a condenser. The number of the tube bundle is found as 70991 from the calculated data. And tube bundles also vary with the change of inside diameter of the tube. Length and diameter of a tube are estimated carefully by considering the impacts on appropriate overall heat transfer coefficient and pressure drop both for the tube and shell sides. The limitation of these calculated design data is no scope for validation with the experimental data as limited experimental data are available in the literature.

NOMENCLATURE

Symbol

A	Provisional area of heat exchanger (m²)
CL	Tube layout constant
CTP	Tube count constant
d	Diameter (m)
D_s	Shell diameter (m)
$G_t = m_c/A_f$	Mass velocity (kg/m²s)
g	Gravitational acceleration(m/s²)
h	Heat transfer coefficient (W/m² °C)
h_{fg}	Enthalpy(J)
j_f	Tube side friction factor
J_f	Fanning friction factor
J_h	Heat transfer factor
k	Thermal conductivity of fluid (W/m °C)
LMTD	Logarithmic mean temperature difference (°C)
L	Length of the tube (m)
m_c	Mass flow rate (kg/s)
N	Number of tubes/ pass/baffles/tank
p	Pressure (Pa)
Q	Heat transfer rate (W)

Re= $\rho\, u_t\, d_t/\mu$	Reynolds number
T	Temperature (ºC)
U	Overall heat transfer coefficient (W/m^{2}ºC)
$u_t= G_t/\rho$	Linear fluid velocity (m/s)
x	Mass fraction

Greek Symbols

μ	Dynamic viscosity of water (Pa.s)
ρ	Density (kg/m^3)
ΔP_t	Tube side pressure drop (kPa)
ΔT_m	LMTD for condenser (°C)

Subscript

a	Air
c	Cold
e	Equivalent
f	Flow area or Friction
fg	Air-vapor
i	Inside/interface
l	Liquid
o	Outside
p	Pass or Pressure
s	Shell
t	Tube
v	Vapor
w	Wall

APPENDIX-A

Value of different constants used in the calculation of a condenser thermal hydraulic parameters (Sinnott *et al.*1993).

Parameter	Symbol	Constant value
Tube side fanning friction factor	j_f	0.0075
Fanning friction factor	$J_{f,}$	0.05
Heat transfer factor	J_h	0.006

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