# Computer Aided Thermal Design of Condenser

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***Abstract: -*** **Condensers are vital in power producing industries, process and chemical industries, refrigeration and air-conditioning systems. A large number of industries are engaged in designing various condensers. This paper covers computer aided thermal design of condenser of shell and tube type with one shell and one tube pass and a case-study pertaining to power plant condenser. The optimum design of condenser is more effective in thermal design stages compared to mechanical design. The manual thermal design of condenser is an iterative process which consumes a lot of time of heat exchanger designer and is tedious to obtain optimum solution. This monotonous and laborious thermal design of condenser has been worked-out using C programming language considering various parameters such as counter flow arrangement, fouling factor and pressure drops. This way design through digitization has gained high importance where high degree of human interventions is involved. It is essential to use computer aided design to reduce cumbersome and iterative traditional design.**

***Index Terms: -*** **Computer aided thermal design of condenser, designing condenser, Kern's Method**

## I. INTRODUCTION

The most common problems in condenser design are rating and sizing. The rating problem is concerned with determination of heat transfer rate and fluid outlet temperatures for prescribed flow rates, inlet temperatures and pressure drops for an existing condenser. The sizing problem, on other hand, is concern with the determination of dimensions of the condenser to meet the specified hot and cold fluid inlet and outlet temperatures, flow rates and pressure drop requirement. The C program has been developed by using thermal design algorithm for designing.

## II. CONDENSER THERMAL DESIGN METHODOLOGY

By putting following assumptions, thermal design of condenser has been worked out.

1. Shell and tube type condenser with one shell and one tube pass.
2. Condenser operates under steady state and steady flow conditions.
3. The change in kinetic and potential energy is neglected.
4. Heat losses to or from the surroundings are negligible.
5. Individual and overall heat transfer co-efficient remains constant.
6. Phase change takes place at constant temperature.
7. Film wise condensation was assumed over horizontal tube bundle.

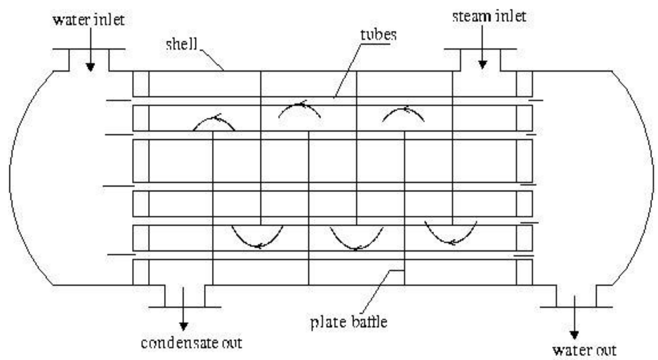


Fig.1-shell and tube type condenser with one shell-one tube pass

From the first low of thermodynamics for an open system, under steady state and steady flow condition, heat transfer rate required of condenser was found out and hence the mass flow rate of unspecified fluid can be found out.

$$Q = \dot{m}_s (x h_{fg}) = \dot{m}_w C_p (T_{Co} - T_{Ci}) \qquad \ldots (1)$$

Task of a heat exchanger designer is to determine parameters of sizing and rating by simultaneously assuming required parameters of either sizing or rating.

The general equation adopted for heat transfer across surface as follows:

$$Q = U_o A_o (\Delta T)_{LMTD} \qquad \ldots (2)$$

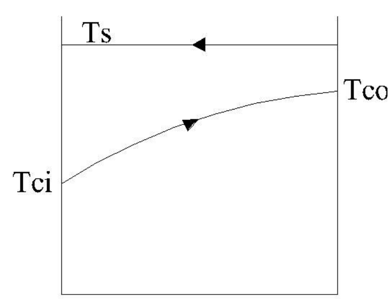

Fig. 2. Temperature variation

By taking LMTD and overall heat transfer coefficient, surface area required for the specified heat duty was worked out.

$$\theta_{in} = T_s - T_{ci}$$
$$\theta_{out} = T_s - T_{co}$$
$$(\Delta T)_{LMTD} = \frac{\theta_{in} - \theta_{out}}{\ln\left(\frac{\theta_{in}}{\theta_{out}}\right)} \quad \ldots (3)$$

Following figure shows the various thermal resistances for the heat transfer. By considering all resistance, over all heat transfer coefficient based on shell side area was found out.

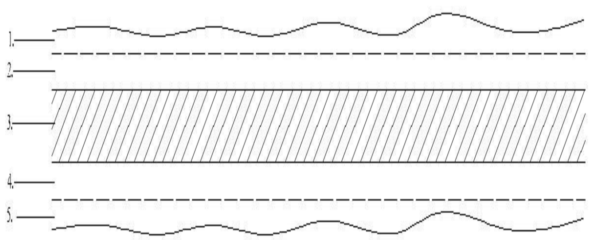

Fig. 3 Total Thermal Resistance

1. condensing heat transfer thermal resistance
2. shell side fouling thermal resistance
3. tube material thermal resistance
4. tube side fouling thermal resistance
5. cooling water side heat transfer thermal resistance

$$\frac{1}{U_o A_o} = \frac{1}{h_o A_o} + \frac{R_{fo}}{A_o} + \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi k_w l} + \frac{R_{fi}}{A_i} + \frac{1}{h_i A_i} \quad \ldots (4)$$

The tube side convection heat transfer coefficient was found out by determining tube side Reynolds number and comparing it with flow pattern whether laminar or turbulence and accordingly available correlations were used to determine (hi).

The following correlation given by Petukhov and Kirillov for turbulent floe through circular tube was used.

$$\mathrm{Re}_t = \frac{\rho u_t d_i}{\mu} \quad \ldots (5)$$

When $10^4 < \mathrm{Ret} < 5X10^6$

$$Nu = \frac{h_i k_l}{di} = \frac{(0.5f)\mathrm{Re}_t.\mathrm{Pr}_t}{1.70 + 12.7(0.5f)^{0.5}\left(\mathrm{Pr}_t^{\frac{2}{3}} - 1\right)} \quad \ldots (6)$$

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

As the fluid phase change took place over shell, appropriate method to determined shell side heat transfer coefficient was used. The correlation developed by Kern was used which is as follows.

$$h_o = 0.728\left[\frac{1000\rho_l(\rho_l - \rho_v) g h_{fg} k_l^3}{\mu_l (T_{sat} - T_{wall}) d_o}\right]^{\frac{1}{4}} \frac{1}{Nr^{\frac{1}{6}}} \quad \ldots (7)$$

Where $N_r = \frac{2}{3}\left(\frac{D_b}{P_t}\right)$ and $D_b = d_0\left(\frac{N_t}{K_1}\right)^{\frac{1}{n1}}$

Constants K1=0.319 and n1=2.142 for single pass triangular pitch arrangement,
Where, pitch $P_t = 1.25 \times d_o$

Kern developed correlation based on experimental work. This correlation yields conservative results than other available correlations for the condensations process.

After determining individual convective heat transfer coefficients, overall heat transfer coefficient and hence surface area required was determined and than task is to find number of tubes and length of tube. Number of tubes can be found out by considering recommended tube side fluid velocity in the literature.

$$N_t = \frac{4\dot{m}_w}{\rho u_t \pi d_i^2} \quad \ldots (8)$$

The required length found by following equation:

$$L = \frac{A_o}{Nt \pi d_o} \quad \ldots (9)$$

Now that the length of condenser has been determined, the shell size of the condenser, tube-side and shell-side pressure drop and tube-side pump power can be calculated.

$$D_s = 0.637\sqrt{\frac{CL}{CTP}}\left[\frac{A_o (PR)^2 d_o}{L}\right]^{\frac{1}{2}} \quad \ldots (10)$$

Where,
CL= 0.87 for triangular pitch arrangement (tube layout constant)
CTP= 0.93 for 1-tube pass (accounts for incomplete coverage of shell diameter)

Tube-side pressure drop and pump power:

$$\Delta P_t = \left(\frac{4fLN_p}{d_i} + 4N_p\right)\frac{\rho u_t^2}{2} \quad \ldots (11)$$

Where,

Friction factor, $f = 0.079(\mathrm{Re}_t)^{-0.25}$

$$P = \frac{\dot{m}_t \Delta P_t}{\rho \eta_{pump}} \quad \ldots (12)$$

Shell side pressure drop;

$$\Delta P_s = \frac{f G_s^2 (N_b + 1) D_s}{2 \rho_l D_e \phi_s} \quad \ldots (13)$$

Where,

Dynamic viscosity variation factor($\phi$ s) =1

Friction factor, $f = e^{(0.579 - 0.19 \ln \mathrm{Re}_s)}$

Shell side Reynolds No., $\mathrm{Re}_s = \frac{G_s D_e}{\mu}$

Mass velocity, $G_s = \frac{\dot{m}_s}{A_s}$

Shell side flow area, $A_s = \frac{(P_t - d_o) D_s L_b}{P_t}$

Hydraulic equivalent Diameter,

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

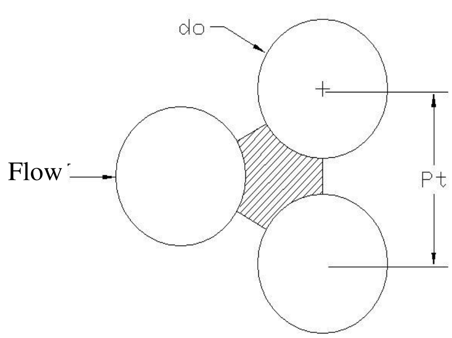


Fig. 4 Triangular pitch-tube layout

## III. Computer Aided Design

The manual design using above methodology requires a lot of time and very careful attention for finding each parameter based on formulae and many graphs, interpolations from tables. It is possible that optimum value of heat transfer co-efficient could be found out by many numbers of iterations. So as to avoid this iteration process, by using algorithm based on above methodology, C program for condenser design has been prepared. The flow chart for the program developed as follows.

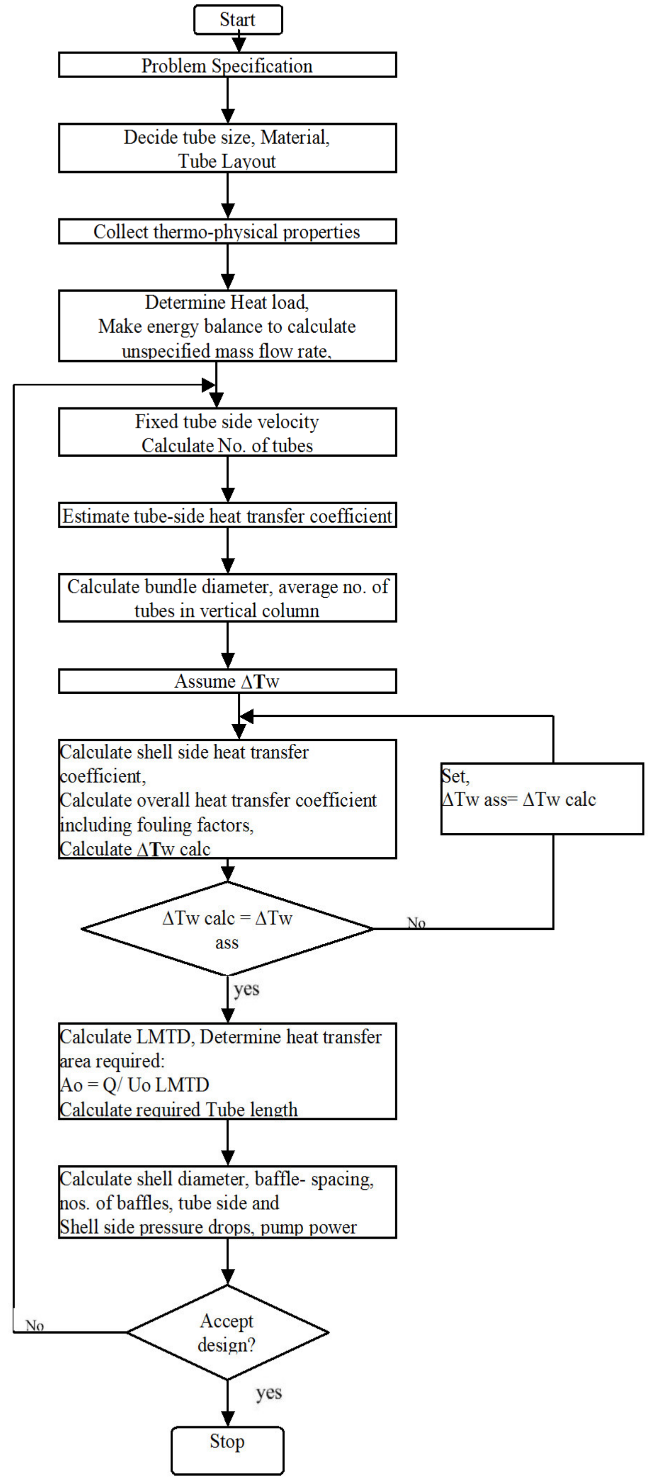


Fig. 5 flow chart for software

*Case study from Thermal power station*

For this paper, the operational and constructional specifications of condensers were gathered from GEB, Gandhinagar and Torrent power AEC Ltd, Ahmedabad for the reference purpose.

*Results and Discussion*

Thermal Design of condenser installed at GEB, was carried out, which had following process parameters.
Dry saturated steam at 49°C enters the condenser at a flow rate of 550T/hr. It is required to design a condenser for phase change. A single tube-side pass is used and the cooling-water average velocity is selected as 1.25 m/s as a good compromise between fouling and erosion. Cooling water is available at 36°C and 9.75°C temp. rise of cooling water is allowed.

The designed result was compared with actual available parameters.

Table 1 Result

| Parameters | Designed | Actual |
|---|---|---|
| Outer dia of tube(m) | 0.0254 m | 0.0254 m |
| Inner dia of tube(m) | 0.02291 m | 0.02291 m |
| Thermal conductivity of tube | 52 w/mK | 52 w/mK |
| Tube side fouling resistance | 0.00018 (m^2.k/w) | |
| Shell side fouling resistance | 0.00009 (m^2.k/w) | |
| Condenser heat load(Q) | 364410.84 kw | 364410.84 kw |
| Cooling Water mass flow rate(mw) | 8943.64 kg/s | 7500 kg/s |
| No. of tubes(Nt) | 17497 | 17350 |
| Pitch Ratio / arrangement | 1.25 / Triangular | 1.25 / Triangular |
| Tube side Reynold's no.(Ret) | 44731.78 | |
| Tube side heat transfer co-efficient:(hi) | 6183.24 W/m²K | |
| Bundle dia.(Db) | 4.14 m | |
| Average no of tubes in vertical column | 87 | |
| Shell side heat transfer co-efficient(ho) | 8431.7 W/m²K | |
| Overall heat transfer co-efficient(Uo) | 1620.26 W/m²K | |
| Total surface area required(Ao) | 31978.53 m² | 27966.26 m² |
| Total length required(L) | 22.91 m | 20.2 m |
| Shell diameter(Ds) | 4.59 m | |
| No. of Baffles(Nb) | 13 Nos. | |
| Baffle spacing (Lb) | 1.76 m | |
| Tube side pressure drop | 19940.27 Pa | 47480.4 Pa |
| Efficiency of pump | 0.85 | |
| Pump power required | 211.53 kw | |
| Shell side pressure drop | 6263.09 Pa | |

The result obtained from the software was reliable and was within permissible error limit.

## IV. Conclusion And Future scope

The Thermal design of condenser is carried out as per the Kern method and it will give conservative results. As the manual design procedure is found tedious, time consuming and incurred human errors. The addition of software results into drastic reduction in time for designing condenser compare to traditional designing method.
In future the thermal design of condenser can be extended for the simulation which results into optimization of design of condenser. The integrated software can be then be developed for the computer aided design and simulation of condenser.

## V. Nomenclature

$A_o$ = shell side surface area($m^2$)
$D_s, D_b$ = shell ID, tube bundle dia.(m)
$D_e$ = hydraulic dia(m)
$d_o$ = O.D.of tube(m)
$d_i$ = I.D.of tube(m)
f = friction factor
$h_i, h_o$ = heat transfer co – efficient for tube and shell side respectively($w/m^2K$)
$h_{fg}$ = latent heat of steam(kJ/kg)
$k_w$ = thermal conductivity of tube(w/mK)
L = length of tube(m)
$\dot{m}_s$ = steam flow rate (kg/s)
$\dot{m}_w$ = cooling water flow rate(kg/s)
$N_t$ = Number of tubes
$P_s$ = Presurre corresponding to Ts (bar)
$Pr_t$ = tube side prantl number
$\Delta P_t, \Delta P_s$ = Prssure drop on tube and shell side respectively(Pa.)
$Re_t, Re_s$ = Reynolds no. on tube side and shell side respectively
$R_{fo}$ = shell - side fouling resistance($m^2K/w$)
$R_{fi}$ = tube - side fouling resitance($m^2K/w$)
$T_s$ = Saturaturion temp. of Steam($^0C$)
$T_{ci}$ = cooling water inlet temp.($^0C$)
$T_{co}$ = cooling water outlet temp.($^0C$)

$u_t$ = cooling water velocity(m/s)

$U_o$ = over all heat transfer co – efficient($w/m^2K$)

$\rho$ = density of water($kg/m^3$)

$k_l$ = thermal conductivity of liquid phase(w/mK)

$\mu$ = dynamic viscosity($Ns/m^2$)

$C_p$ = specific heat(kJ/kg K)

$\rho_v$ = density of vapour($kg/m^3$)

$\rho_l$ = density of liquid($kg/m^3$)

$\mu_l$ = dynamic viscosity of liquid($Ns/m^2$)

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