Steam Generator for a Pressured Water Reactor(PWA)

Project Report

Submitted in fulfilment of the course of

ME 310: Thermo-Fluid System Design

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Submitted by

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Background on Steam Generators

Steam generators are heat exchangers used to convert water into steam from heat produced in a nuclear reactor core. In the steam generator, the primary fluid is transported through tubes while its heat is transferred from the primary fluid to the secondary fluid in the shell. Inside the shell, the secondary fluid temperature increases to its saturation temperature and steam is produced. This steam exits the steam generator and drives the turbine generator before condensing and returning to the steam generator as feed-water. There are two main types of vertical steam generators in nuclear power plants: recirculating steam generators and once-through steam generators. Steam generators of the vertical recirculating type are widely used in both PWR and CANDU reactors.
In a PWR, the primary coolant is water which is pumped under high pressure to the reactor core where it is heated by the energy released by the fission of atoms. The heated, high pressure water then flows to a steam generator, where it transfers its thermal energy to lower pressure water of a secondary system where steam is generated. The steam then drives turbines, which spin an electric generator.
The CANDU (Canada Deuterium Uranium) is a Canadian pressurized heavy-water reactor design used to generate electric power. The CANDU reactor use natural uranium as their nuclear fuel.
The project at hand demanded optimum design of simple steam generator.

Problem Statement

The problem statement was given as follows:

Design of a Steam Generator for a Pressured Water Reactor (PWA)—600 MWe Specifications:

Power to be generated:	150 MWe
Reactor coolant inlet temperature:	300°C
Reactor coolant outlet temperature:	337°C
Reactor coolant pressure:	15 MPa
Secondary circuit water inlet temperature:	200°C
Secondary circuit steam outlet temperature:	285°C
Secondary fluid pressure:	6.9 MPa

Fig 1: Problem Statement

Based on the given statement analysis is performed in the following procedure:
A design has been proposed to meet the given criteria of the problem statement by detailed thermal-hydraulic as well as mechanical analysis. Scope of design parameters include: ID_t (tube inner diameter), OD_t (tube outer diameter), P_t(tube pitch), N_t (tube count), L (effective heat transfer area length), ID_s (Shell inner diameter) and finally, OD_s (Shell outer diameter). While, selection of design parameters TEMA standards [1], ASME standards [2] and other sources are also taken into consideration.

For thermal performance, LTMD method [3], Dittus Boelter equation [3] (for tube side heat transfer coefficient h_t), Shah correlation [3] (shell side heat transfer coefficient h_s for two phase flow). For tube side pressure drop Δp_t modification of Darcy Weisberg equation for vertical U-tube flow [3] and for shell side, combination of Kern method [3] (for frictional pressure drop) and gravitation pressure drop is taken into account. Afterwards, in mechanical design from thick wall cylinder [4] along with constraints introduced according to von mises stress [4].

Both hand calculation Python programming language and HTRI simulation software[*] are preferred for thermo-hydraulic calculation. The results obtained in both ways are reported. Further analysis (mechanical, total cost) is done in Python environment.

Afterwards, cost of the total project is calculated considering manufacturing cost, as well as operating cost. Finally, parametric study is introduced to evaluate performance of the steam generator by changing design parameters.

These calculations are done considering the following assumptions:

- Design suggested in this scope is for go thorough steam generators.
- The analysis is based on steady operating conditions of the steam generator.

- Negligible heat leakage from shell to ambient is considered.
- The thermal properties of fluid for both sides are evaluated at the average temperature of both ends (inlet, outlet) and given working pressure. They are assumed to be constant.
- Changes in the kinetic and potential energies of the entire system are negligible.
- Leakage flows are non-dominant with respect to other flow parameters.
- Water level at shell side is always at a constant height which is at the tip of U-tube.
- Blowdown water has same thermo-hydraulic conditions as the feed water inlet.
- From separator to steam drum the pressure drop and density variation of steam is negligible.

Governing Equations

Heat Transfer Coefficient:
Since, working condition of inlet and outlets are known, the flow rate for both shell $\dot{m}_s$ and tube $\dot{m}_t$ can be evaluated by the given Heat duty by the following formulas.

$$\dot{m}_t = \frac{\dot{Q}}{h_{t,o} - h_{t,i}} \tag{1}$$

$$\dot{m}_{steam} = \frac{\dot{Q}}{h_{s,o} - h_{s,i}} \tag{2}$$

$$\dot{m}_s = \dot{m}_{steam} \times circulation\ rate \tag{3}$$

Here, $\dot{Q}$ is heat duty, $h_{t,o}$ and $h_{t,i}$ are enthalpy of water at 15 MPa pressure tube side for inlet and outlet temperatures, respectively. $\dot{m}_{steam}$ is mass of steam produced; $h_{s,i}$ is enthalpy of water (single phase) at shell side at 6.9Mpa for inlet temperature.
While $h_{s,o}$ is enthalpy of water vapor mixture (two phase) at 6.9MPa at outlet conditions.

Now from Logarithmic Mean Temperature Difference (LMTD)[3]

$$\dot{Q} = F\ U\ A_o\ LMTD \tag{4}$$

U is overall heat transfer coefficient, A heat transfer area, F correction factor for 2 tube pass and 1 shell pass[3], LMTD is Logarithmic Mean Temperature Difference which is further calculated from the following equation.

$$LMTD = \frac{\left(T_{t,i} - T_{s,o}\right) - \left(T_{t,o} - T_{s,i}\right)}{\ln\frac{\left(T_{t,i} - T_{s,o}\right)}{\left(T_{t,o} - T_{s,i}\right)}} \tag{5}$$

Now for tube-side heat transfer coefficient, h_t is calculated from Dittus Boelter method [3].

$$A_o = N_t(\pi OD_t L) \tag{6}$$

$$A_{o,t} = 0.25\frac{\pi {ID_t}^2 N_t}{N_p} \tag{7}$$

$$Re_t = \frac{\dot{m}_t ID_t}{\mu_t A_{o,t}} \tag{8}$$

$$h_i = h_t = \left(\frac{k_t}{ID_t}\right)0.024{Re_t}^{0.8}{Pr_t}^{0.4}\ for\ 2500 < Re_t < 1.24 * 10^5 \tag{9}$$

Where A_o, N_t, OD_t and L are the heat transfer surface area (same as 4), tube count, tube outer diameter and effect heat transfer area length, respectively; $A_{o,t}, ID_t$ and N_p are tube cross sectional area, tube inner diameter and number of pass, respectively; $Re_t, \dot{m}_t, \mu_t$ are tube side Reynolds number, tube side mass flow rate and dynamics viscosity of tube side fluid. Finally, heat transfer coefficient h_t can be determined from (eqn. 9) k_t and Pr_t are as follow: heat transfer coefficient at tube side, thermal conductivity and Prandtl number of tube side fluid.

For determining shell side heat transfer co efficient Shah correlation [3] is taken into account. This correlation can be applied on for convective, nucleate and stratified boiling region.

$$ID_s = 0.637\sqrt{\frac{CL}{CTP}\times\left[\frac{A_o(P_r)^2 OD_t}{L}\right]} \qquad (10)$$

Here, ID_s is shell inside diameter, heat transfer surface area A_o, tube pitch ratio $P_r = \frac{P_t}{OD_t}$, CL and CTP are constants where CL= 0.85 for 30°, 60° tube (triangular) arrangements and CL= 1 for 45°, 90° tube (square) arrangements. CTP= 0.93, 0.9 and 0.85 for 1,2 and 3 tubepass, respectively.
Now,

$$Dh_s = \frac{4A}{P} = \frac{4P_t^2}{\pi OD_t} - OD_t \;\; : square\ pitch \qquad 11(a)$$

$$Dh_s = \frac{4A}{P} = \frac{2\sqrt{3}P_t^2}{\pi OD_t} - OD_t \;\; : triangular\ pitch \qquad 11(b)$$

Here, Dh_s is shell hydraulic or equivalent diameter and P_t is tube pitch. Afterwards,

$$A_s = \frac{ID_s CB}{P_t} \qquad (12)$$

Here, A_s is bundle crossflow area, C = $P_t - OD_t$ is clearance, B is baffle spacing. Therefore, Shah correlation can be implemented using four dimensionless numbers.

$$G = \frac{\dot{m}_s}{A_s} \qquad (13)$$

$$F_0 = \frac{h_{tp}}{h_{LO}} \qquad (14)$$

$$h_{LO} = 0.023\left[\frac{G(1-x)ID_s}{\mu_{s,l}}\right]^{0.8} {Pr_{s,l}}^{0.4}\frac{k_{s,l}}{ID_s} \qquad (15)$$

$$C_o = \left(\frac{1}{x}-1\right)^{0.8}\left(\frac{\rho_g}{\rho_l}\right)^{0.5} \qquad (16)$$

$$Bo = \frac{q''}{G\ i_{fg}} \qquad (17)$$

$$Fr_L = \frac{G^2}{\rho_l^2 g d_i} \qquad (18)$$

$N_s = 0.38\, Fr_l^{-0.3}\, Co$; for $Fr_L < 0.04$ and horizontal tubes 19 (a)

$N_s = Co$; for $Fr_L > 0.04$ horizontal tubes and vertical tubes 19 (b)

G is flow rate per cross bundle, $\dot{m}_s$ is mass flow rate at shell side. $F_o, h_{tp}\ and\ h_{LO}$ are dimensionless parameter, two phase heat transfer co efficient and heat transfer coefficient for liquid phase, respectively. h_{LO} is determined from Dittus Boelter method (as shown in eqn. 15) where x is steam-water mass fraction, $Pr_{s,l}$ Prandtl number of shell side liquid, thermal conductivity of shell side fluid $k_{s,l}$. Then, C_o is convection number, $\rho_g\ and\ \rho_l$ are density of vapor and density of fluid. Afterwards, from (eqn. 17) Bo is known as bubble number, q'' is heat transfer per unit area and i_{fg} is latent heat of vaporization. Therefore, Fraude number Fr_L is evaluated by G, ρ_l, g acceleration due to gravity and ID_s. Then, another dimensionless parameter N_s.

Now, for $N_s = Co < 1$,

$$F_{cB} = \frac{1.8}{N_s^{0.8}} \qquad (20)$$

$$F_{nb} = 231\, B_o^{0.5};\ \text{for Bo} > 1.9 \times 10^{-5} \qquad 21(a)$$

$$F_{nb} = 1 + 46\, B_o^{0.5};\ \text{for Bo} < 0.3 \times 10^{-4} \qquad 21(b)$$

Now, F_o is the larger of F_{cb} or F_{nb}, for $F_{nb} > F_{cb}$; $F_o = F_{nb}$ else, $F_o = F_{cb}$.

Now, overall heat transfer coefficient U is evaluated.

$$\text{U} = \frac{1}{\frac{1}{h_s} + R_{o,f} + \frac{OD_t\, ln\, ln\, \left(\frac{OD_t}{ID_t}\right)}{2k_w} + R_{i,f}\frac{OD_t}{ID_t} + \frac{1}{h_t}\frac{OD_t}{ID_t}} \qquad 22$$

Here, $R_{o,f}, R_{i,f}$ are fouling factor at tube outer and inner face; k_w is thermal conductivity of tube material.

Pressure Drop:
Now, pressure drop is evaluated for selection of pump and operating cost. Therefore, tube side friction factor, f_t is:

$$f_t = 0.316\, Re_t^{-0.25}\ ; for\ Re_t \leq 2 \times 10^4 \qquad 23(a)$$

$$f_t = 0.184\, Re_t^{-0.2}\ ; for\ Re_t \leq 3 \times 10^5 \qquad 23(b)$$

Therefore, pressure drop at tube side Δp_t is:

$$\Delta p_t = 4\, N_p \left(\frac{f_t\, L}{ID_t} + 1\right) \times \frac{1}{2}\, \rho_t\, u_{t,m}^2 \qquad 24$$

Here, ρ_t is tube side fluid density and u_m is mean velocity of tube side fluid which is calculated by the following:

$$u_{m} = \frac{\dot{m}_t}{N_t\, \rho_t\, A_o} \qquad 25$$

Now, shell side friction factor f_s is:

$$f_s = \exp\left[0.576 - 0.19 \ln(Re_s)\right] \qquad 26$$

$$\Delta P_s = f_s(N_b + 1)\ \frac{D_s}{Dh_s}\left(\frac{1}{2}\rho_{s,l}V_{s,l}^2\right) + \rho_{s,l}\, g\, \frac{L}{2} \qquad 27$$

So, shell side pressure drop Δp_s is calculated from (eqn. 27). Here, N_b is number of baffles, $\rho_{s,l}$ is density of shell side fluid, $V_{s,l}$ is shell side fluid velocity.

Total Cost:
Cost evaluation is separated in 2 parts, Construction Cost C_{con} and operating cost C_{op} [5].

$$V_{t,m} = \pi\, N_t\, (OD_t - ID_t)\, L \qquad 28$$

$$V_{s,m} = \pi\, (OD_s - ID_s)\, L/2 \qquad 29$$

In (eqn. 28 and 29) volume of tube material $V_{t,m}$ and shell material volume $V_{s,m}$ are determined. Now, C_{con} is:

$$C_{con} = \$_{t,m}\, \rho_{t,m}\, V_{t,m} + \$_{s,m}\, \rho_{s,m}\, V_{s,m} \qquad 30$$

Here, $\rho_{t,m}$ and $\$_{t,m}$ are density of tube material and cost of tube material per kg; $\rho_{s,m}$ and $\$_{s,m}$ density of shell material and cost per unit kg. In (eqn. 30) manufacturing cost is neglected for insufficient data.

For C_{op},

$$C_{op} = \sum_{k=1}^{n} \frac{C_o}{(1+i)^k} \quad 31$$

$$C_o = P k_{el} \tau \quad 32$$

$$P = \frac{1}{\eta}\left(\frac{m_t}{\rho_t} \Delta P_t + \frac{m_s}{\rho_s} \Delta P_s\right) \quad 33$$

where n is the life time of equipment in year, i is annual discount rate; P is the pumping power; k_{el}, η and τ are price of electrical energy, the pump efficiency and hours of operation per year respectively.
Finally, $\rho_s = x\, \rho_{s,s} + (1-x)\, \rho_{s,l}$,where $\rho_{s,s}$ density of steam at shell side and x is steam quality.

Therefore, total cost, C_{total} :

$$C_{total} = C_{con} + C_{op} \quad 34$$

Mechanical Design:
For thick wall cylinder [4],

$$\sigma_t = \frac{p_i r_i^2 - p_o r_o^2 - (p_o - p_i)\, r_i^2\, r_o^2 / r^2}{r_o^2 - r_i^2} \quad 35$$

$$\sigma_r = \frac{p_i r_i^2 - p_o r_o^2 + (p_o - p_i)\, r_i^2\, r_o^2 / r^2}{r_o^2 - r_i^2} \quad 36$$

Here, $p_i \; and \; p_o$ are the internal and external pressure, $r_i \; and \; r_o$ are the inner and outer radius, and r is the radius of interest. Now von mises stress, σ' [4],

$$\sigma' = \sqrt{\sigma_x^2 - \sigma_x\, \sigma_y + \sigma_y^2 + 3\tau_{xy}^2} \quad 37$$

Some Other Formulas for Accessories Calculation:

$$V_d = 5\, \dot{m}_{steam} / \rho_{steam} \quad 38$$

Here, V_d is steam drum volume, where 5 is assumed as a safety factor as well as void fraction.

Now, longitudinal stress σ_L,

$$\sigma_L = \frac{x\, p_s\, D_d}{4\, t} \quad 39$$

Here, D_d is the diameter of the drum, p_s working pressure of shell, t is thickness of the drum and x is steam quality.

Using, the volume of the drum inner region V_d,

$$V_d = \pi\, D_d^2\, H_d \quad 40$$

Using the D_d and ID_s volume of riser V_r can be evaluated.

$$V_r = \frac{\pi(D_d^3 - ID_s^3)}{2^3 \tan\theta} \quad 41$$

$$H_r = \frac{(D_d - ID_s)}{2\ \tan\theta} \quad 42$$

Here, θ is the cone angle of the riser.
Finally, for annular region volume, V_a is:

$$V_a = \pi\ (D_a^2 - OD_s^2)\ L/2 \quad 43$$

Where, D_a is inner diameter of the annulus region, which is $D_a = OD_s + 2$ nozzle of shell inlet.

Calculation Procedure:

During calculation, many iterations are performed. The following figure illustrates segment of code which indicates the implementation of the iterative means the of selection of tube material and ID_t. Two database one for BWG: 18 standards other for list of materials with their yield strength are stored. Now, at certain k a list r_list is generated from r_i to r_o at 10 intervals with the given input conditions, (p_i *and* p_o) it is checked for which member of r_list von mises stress $\sigma_{v,r}{'}$ is maximum. Fiber at which $\sigma_{v,r}{'}$ is maximum is stored at σ_{max}. Now, this process is continued till for all r_i in BWG:18 database (k<= no of elements in BWG:18). Maximum stress in the entire process is taken as σ_{max}. Now, this σ_{max} is compared with S_y of all the materials in the database (list of materials with yield strength). Material for which n is maximum is chosen as tube material.

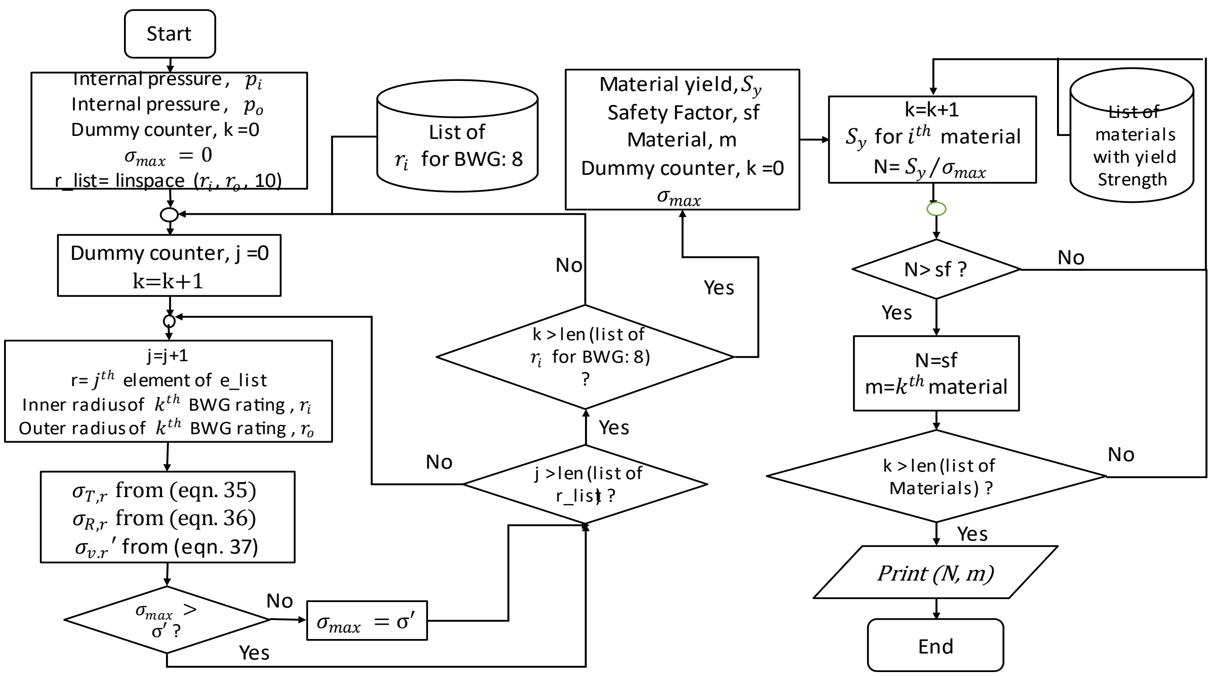


Fig. 2: Flowchart for the selection of tube materials and tube inner diameter

Now, almost similar approach is taken for selection of shell thickness, except in this case, 〖OD is unknown. So, using approximation of po ≈ 0, shell outer diameter and shell material is selected.

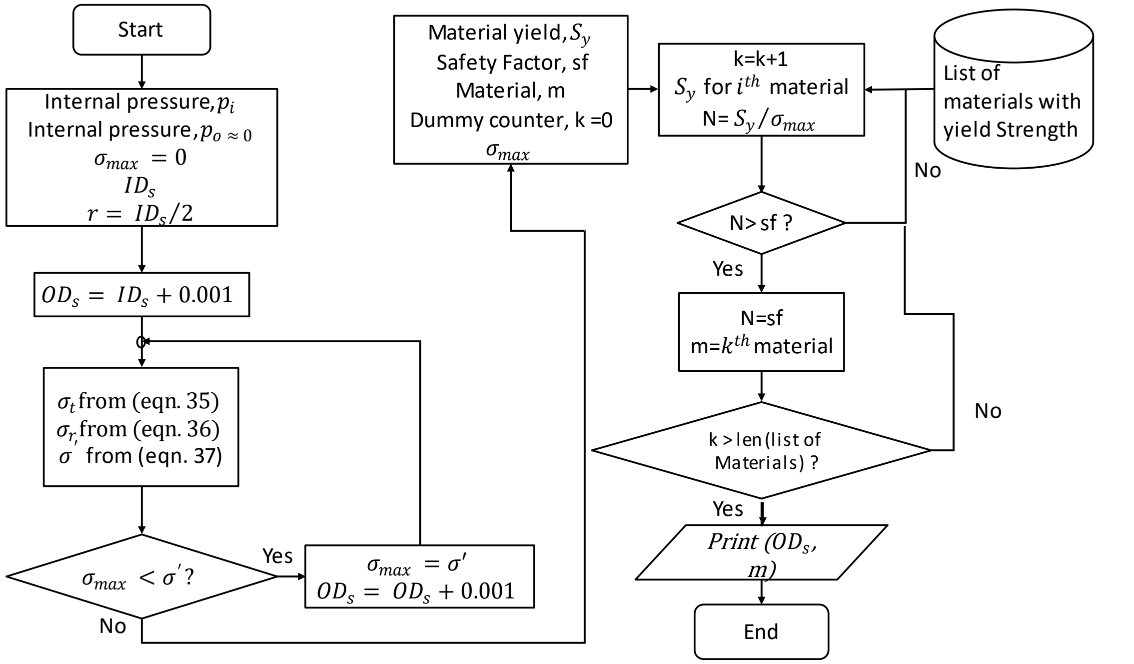


Fig. 3 Flowchart for Selection of shell outer diameter and shell material.

So, to sum up the entire process,

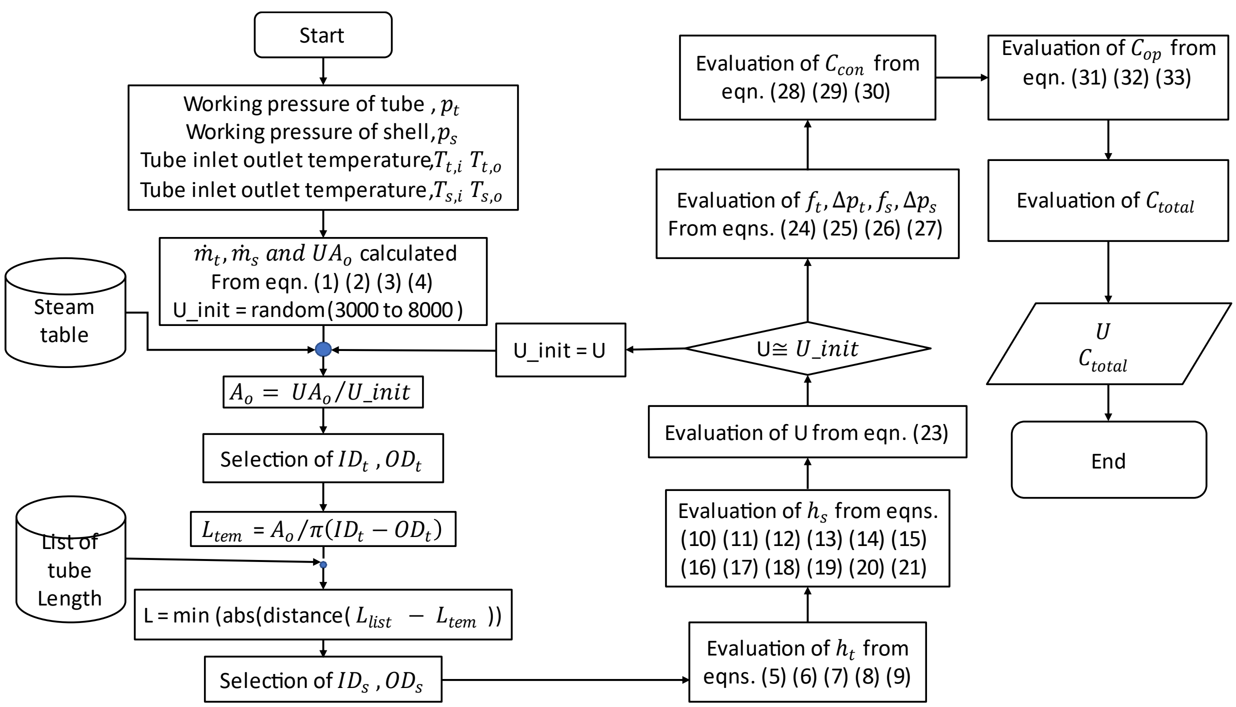


Fig. 4: Overall description of code.

Fig. 4 denotes the algorithm implemented in python. First the flow rates at sides are depicted. Afterwards, from heat balance and LMTD UA_o required for the flow is determined. Then, from standard range U_init is guessed. Using the guessed U_init A_o required is determined. Now, ID_t , OD_t are selected as described in fig. 2. Then, effective length of heat transfer L_{tem} is determined. Now, L is taken from the list of available tube length which is close to L_{tem} (in this case, minimum absolute distance algorithm is deployed). Afterwards, ID_s , OD_s are selected as shown in fig. 3. Now, h_t , h_s are determined to calculated U. Calculated U is then compared with U_init, if they are quite close, then the design is chosen, else the entire process is continued. Afterwards, from f_t, Δp_t, f_s, Δp_s, C_{op} is calculated. And from volumes, density and price per unit kg C_{con} is calculated. Finally, total cost of the project is reported.

Result and Discussion:

The procedure stated in fig. 4,
From,
$h_{t,o}$,Enthalpy of Water @ (300°c, 15MPa) = 1338.06 kJ/Kg
$h_{t,i}$,Enthalpy of Water @ (337°c, 15MPa) = 1568.84 kJ/Kg
$h_{s,o}$,Enthalpy of Water @ (285°c, 6.9MPa) = 2773.67 kJ/Kg
$h_{s,i}$,Enthalpy of Water @ (200°c, 6.9MPa) = 1854.59 kJ/Kg
(Steam Table)
$\dot{Q}$= 150 MW (given)
circulation rate= 3.7 [ ]
Calculated, $\dot{m}_t$ = 649.969 kg/m^3, $\dot{m}_{steam}$ = 78.162 kg/m^3 and $\dot{m}_s$= 289.2 kg/m^3
Steam quality, x = 0.27

Now, as given, $T_{t,i}$ = 337°c, $T_{t,o}$= 300°c, $T_{s,i}$= 200°c, $T_{s,o}$= 285°c, and F = 0.891,
Using, eqn. (4), UA_o = 2,293,600.9 W/K,

Table:1: Thermo physical properties of both side fluid

Properties	Tubeside	Shellside
Working Pressure	p_t: 15MPa	p_s: 6.9 MPa
Pr	1.116	0.858
ρ (kg/m^3)	682.7	Liquid: 741.25, gas: 36.02
K (W/m^2.K)	0.49	Liquid: 0.574
μ (Pa.s)	$7.23\times\ 10^{-5}$	Liquid: $9.16\times\ 10^{-5}$

In table: 1, thermo physical properties of both side fluids are reported.

Now to generate the iteration U_init is picked as 1910 W/ m^2.K (as suggested range is 1500 W/ m^2.K to 6000 W/ m^2.K)[3].

Therefore, A_o = 1200.7 m^2

Now, as suggested in fig. 2 ID_r , OD_t is chosen with its material. In this regard, OD_t = ¾ in = 0.019m (BWG:18) as suggested in [6].

So, allowable ID_r = [0.01224, 0.01295, 0.01351, 0.01422, 0.01483, 0.01539, 0.01575, 0.01610, 0.01656, 0.01727].

During, material selection some common tube materials used in steam generators with their thermal conductivity, density, Elasticity modulus, yield strength and Cost are:

Table. 2: List of tube material with mechanical properties [7] and cost [9]

material	Grade	K_w (W/ m^2.K)	ρ (kg/m^3)	E (GPa)	$S_{y,t}$ (MPa)	Cost ($/kg)
Stainless Steel	SS316	17.9	7990	193	290	3.5
Ni- Alloy	Monel400	21.8	8830	173	276	30
	Inconel600	14.8	8420	207	176	24
	Incoloy800	16.5	7940	208	275	35

Now, from process described in fig. 2 selected material for tube is with ID_t = 0.01727 m, with von mises stress, σ' = 84.441MPa, and safety factor, n = 1.6.
Afterwards using eqn. (6)
L_{tem} = 6.25 m
Now,
[10] allowable L (m)= [1.83, 2.44, 3.66, 4.88, 6.1, 7.32, 8.54, 9.76,10.98, 12.2, 15.25, 18.3]
Therefore L = 6.1 m (as it is closed to L_{tem})
N_t = 3500 (assumed) [11]
P_R = 1.33 (30° triangular-arrangement) (assumed)
No. of pass = 2 (convention)

Now, for OD_s and shell material, process in fig. 3 is followed. Now, in this regard, ID_s = 2.36m (from eqn. 10) and OD_s is also unknown. Therefore, it is determined iteratively (fig. 3). Now, for determination OD_s thick wall cylinder with $p_o \approx 0$ is considered. This is due to, p_i = 6.9 MPa (working pressure of shell side fluid) and p_o = .101MPa. Therefore, it can be neglected to simplify calculation.
For shell material Carbon Steel is preferred [12], which is reported in table: 3.

Table. 3: List of shell material with mechanical properties [13] and cost [14]

material	**Grade**	**ρ (kg/m^3)**	**$S_{y,t}$ (MPa)**	**Cost ($/kg)**
Carbon Steel	A515 Gr. 70	7800	265	0.8

OD_s = 2.46m, with von mises stress, σ' = 164.2001 MPa
Finally, in this study cross baffle is used where central baffle distance B = L/4 = 0.76 m (assumed). Now, All the dimensions are known.

Now all the dimensions and working conditions of both fluids. Therefore, using equations (6) to (10),
h_t = 11, 385.28 W/m^2.K

Again, using equations (11) to (21),
h_s = 7798.9 W/m^2.K
Therefore, from eqn. (22),
U= 1927 W/m^2.K
In this regard, $R_{o,f}$ = 0.00009 K.m^2/W and $R_{i,f}$ = 0.00009 K.m^2/W
Clearly, U and U_init don't match therefore, taking U_init = U; the process is continued, till, U $\approx$ U_init.
So, after 12 iterations, it is observed that U doesnot change and that much, and the performance parameter values stays exactly same.
Therefore, after 12 iterations, observed performance parameters are reported in table:4:

Table. 4: performance parameters of the deisgn

Performance Parameters	**Tubeside**	**Shellside**
U (W/m^2.K)	1927.6	
Heat transfer co efficient (W/m^2.K)	11,385.28	7798.9
Reynolds number	189335	106201.4

With, evaluated design parameters,

Table. 5: Suggested design parameters

Design Parameters	**Values**
ID_t	0.01727m (SS- 316)
OD_t	0.019m
P_t / OD_t	1.33(triangular)
ID_s	2.36m (A515 Gr. 70)
OD_s	2.46m
N_t	3500 (assumed)
L	6.1m
B	0.76m (assumed)
No. of Baffle rod	3

During, iteration the performance parameter values does not vary that much this is due to discretization introduced in design parameters to considering available standards.

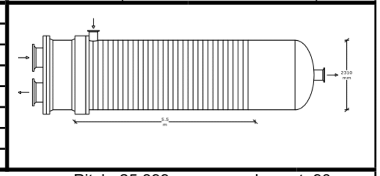
Now, this obtained results are used as inputs in HTRI software. The output results are compared with the evaluated design parameters

HEAT EXCHANGER SPECIFICATION SHEET			Page 1 SI Units
		Job No.	
Customer		Reference No.	
Address		Proposal No.	
Plant Location		Date	12/18/2020 Rev
Service of Unit		Item No.	
Size	2310.00 x 5499.86 mm Type CEU Vert.	Connected In	1 Parallel 1 Series
Surf/Unit (Gross/Eff)	2107.99 / 1995.76 m2 Shell/Unit 1	Surf/Shell (Gross/Eff)	2107.99 / 1995.76 m2

PERFORMANCE OF ONE UNIT					
Fluid Allocation		Shell Side		Tube Side	
Fluid Name		Water		Water	
Fluid Quantity, Total	kg/hr	280801		2339900	
Vapor (In/Out)			280801		
Liquid		280801		2339900	2339900
Steam			280801		
Water		280801		2339900	2339900
Noncondensables					
Temperature (In/Out)	C	200.00	285.00	337.00	300.00
Specific Gravity		0.8691		0.6254	0.7259
Viscosity	mN-s/m2	0.1356	0.0189	0.0724	0.0883
Molecular Weight, Vapor					
Molecular Weight, Noncondensables					
Specific Heat	kJ/kg-C	4.4639	5.2484	7.5810	5.4764
Thermal Conductivity	W/m-C	0.6682	0.0637	0.4727	0.5588
Latent Heat	kJ/kg	1511.59	1513.14		
Inlet Pressure	kPa	6900.10		15000.2	
Velocity	m/s	0.40		1.55	
Pressure Drop, Allow/Calc	kPa		24.080		20.194
Fouling Resistance (min)	m2-K/W	0.000090		0.000090	
Heat Exchanged W	150304301	MTD (Corrected)	35.7 C		
Transfer Rate, Service	2106.49 W/m2-K	Clean	3581.43 W/m2-K	Actual	2115.29 W/m2-K

CONSTRUCTION OF ONE SHELL				
			Shell Side	Tube Side
Design/Test Pressure		kPaG	6798.78 /	14898.9 /
Design Temperature		C	337.00	337.00
No Passes per Shell			1	2
Corrosion Allowance		mm		
Connections	In	mm	1 @ 258.877	1 @ 641.351
Size &	Out	mm	1 @ 336.551	1 @ 641.351
Rating	Intermediate		@	@

Sketch (Bundle/Nozzle Orientation)



Tube No. 2819U	OD 19.050 mm	Thk(Avg) 1.245 mm	Length 5.500 m Pitch 25.399 mm Layout 90
Tube Type	Plain		Material INCONEL (76 NI, 16 CR, 8 FE)
Shell	ID 2310.00 mm	OD mm	Shell Cover
Channel or Bonnet			Channel Cover
Tubesheet-Stationary			Tubesheet-Floating
Floating Head Cover			Impingement Plate None
Baffles-Cross	Type RODBAFFLE	%Cut (Diam)	Spacing(c/c) 152.400 Inlet mm
Baffles-Long		Seal Type	
Supports-Tube		U-Bend	Type
Bypass Seal Arrangement		Tube-Tubesheet Joint	
Expansion Joint		Type	
Rho-V2-Inlet Nozzle	2527.99 kg/m-s2	Bundle Entrance	Bundle Exit kg/m-s2
Gaskets-Shell Side		Tube Side	
-Floating Head			
Code Requirements			TEMA Class
Weight/Shell 68263.9		Filled with Water 106265	Bundle 32156.6 kg
Remarks:			
			Reprinted with Permission (v6)

Fig 5: HTRI report with TEMA specifications

The HTRI report has some discrepancy with code calculation. For U, h_s, h_t variation is not that much. But in terms of design parameters the variation is moderate. Mainly, HTRI suggested another form of result.

Another reason behind this variation is the software HTRI used effective mean temperature difference opposed to corrected LMTD to calculate the required area for design. This heavily affected the end results of designs.

Now, from suggested design cost is calculated,
For calculating operating cost,

Table 6: Values of parameter for cost calculation

Parameter	Value
k_{el}	0.106 \$/unit (BD-industry standard)
η	0.85

$\boldsymbol{\tau}$	8760 hrs/yr
ny	1yrs
i	0
$\Delta \boldsymbol{p_s}$	26.62 (KPa)
$\Delta \boldsymbol{p_t}$	29.509(KPa)

Therefore, C_{op} = 50,485.7 $

For, construction cost = 123, 639 $ (evaluated as suggested in fig.4)
Where,

Table 7: cost for construction of tube and shell

Tube	31801.496 $
Shell	30018.107 $

Where price per unit kg for tube is taken from table [2] and table [3] for shell.
Now, total cost = 174124.7 $

Now, for annular region, steam drum and riser region,
Since, $\dot{m}_{steam}$ and ρ_{steam} at the outlet is known, therefore,
V_d = 32.549 m^3

Now, from longitudinal stress,
σ_L= S_y / n
Where, the material is considered same as shell (A515 Gr. 70) to reduce construction difficulties safety factor, n=7 (assumed). From (eqn. 38), D_d can be calculated, which is,
D_d = 2.83 m
In this regards, thickness of the drum, t= 0.1m (exactly as shell).
Now, from (eqn. 40),
H_d = 1.28m

Now, using D_d and D_s in (eqn. 41 & 42),
V_r = 8.01 m^3
H_r = 0.508 m
θ= 25° in this regard .

Finally, For annular fall down region,
D_a = 2.968 m (since, nozzle at shell inlet = 0.254 m)
Thickness of annular = 0.0805 m (for calculation process in fig. 3 is followed)
Therefore, V_a = 52.84 m^3, with von mises stress = 132.331 MPa and safety factor, n= 2.

Finally, for separator, Curtis & Wright Steam separator is preferred, which is shown in the following figure:

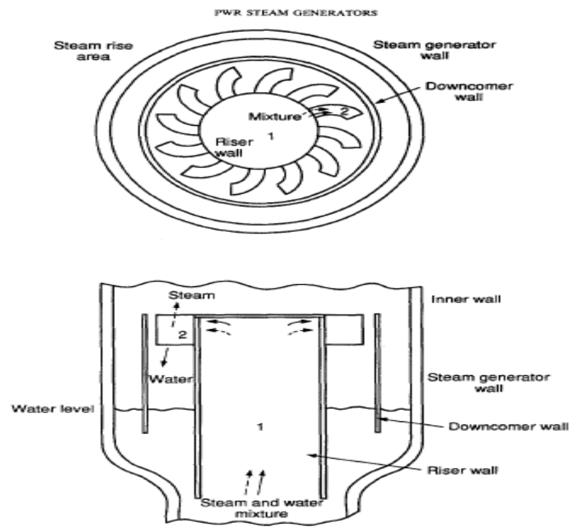
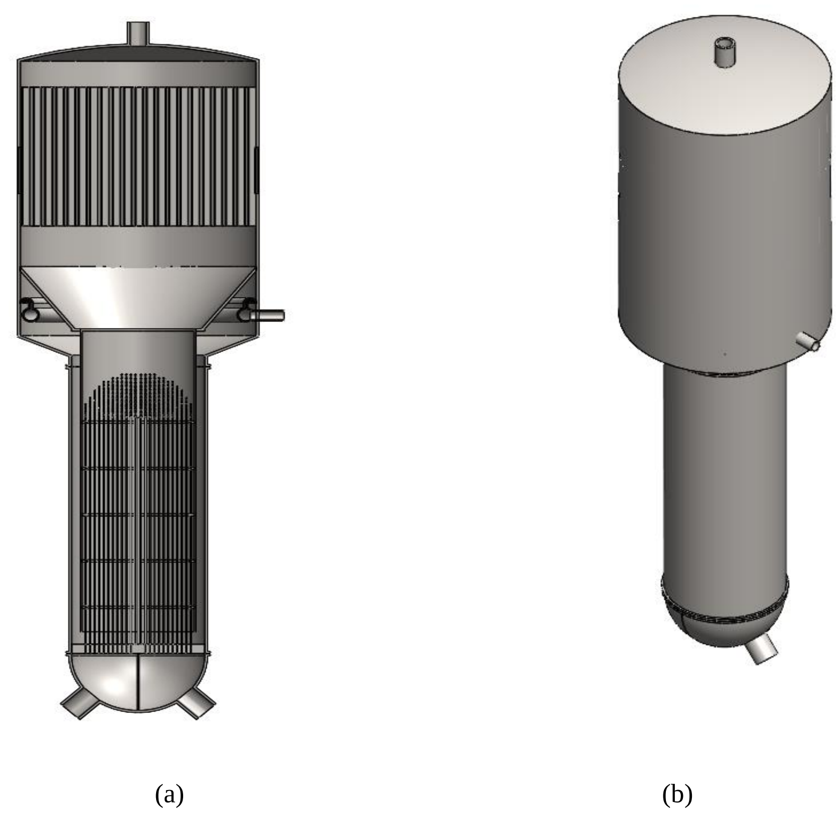


Fig. 6: Curtis & Wright Steam separator

So, the entire design is shown in the following figure,



(a) (b)

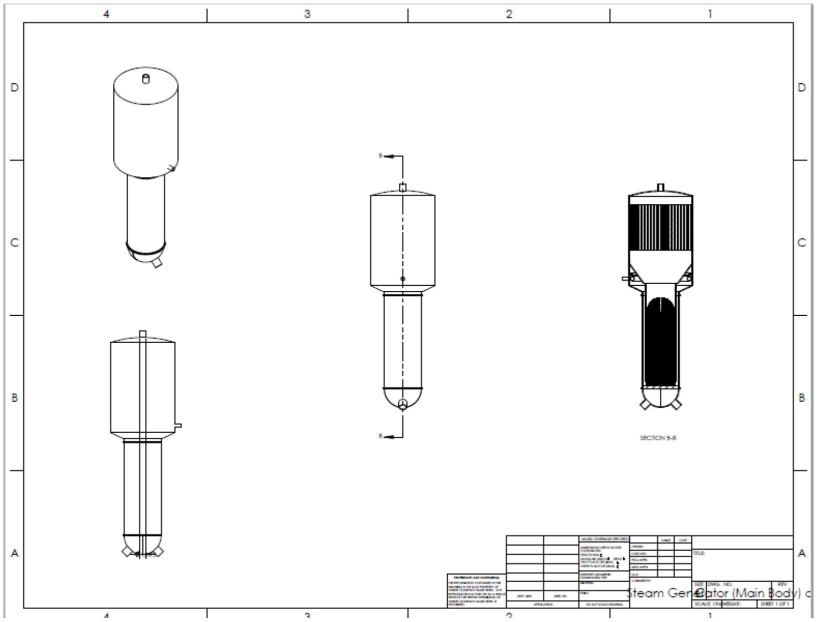


Fig 7(a) sectional view (b) iso view (c): CAD model of the proposed design.

Now parametric study has been done, on the model to evaluate its performance as well as its total cost. Main focus in parametric study is given to design parameters which were assumed forehand. Also, constraints are from the given case study, are also taken in consideration in this regard.

First, Number of tubes is considered. During design suggestion it was strictly assumed 3500 due to lack of ways to calculate it. Therefore, now it varied and observed with U which is reported in Fig: 7.

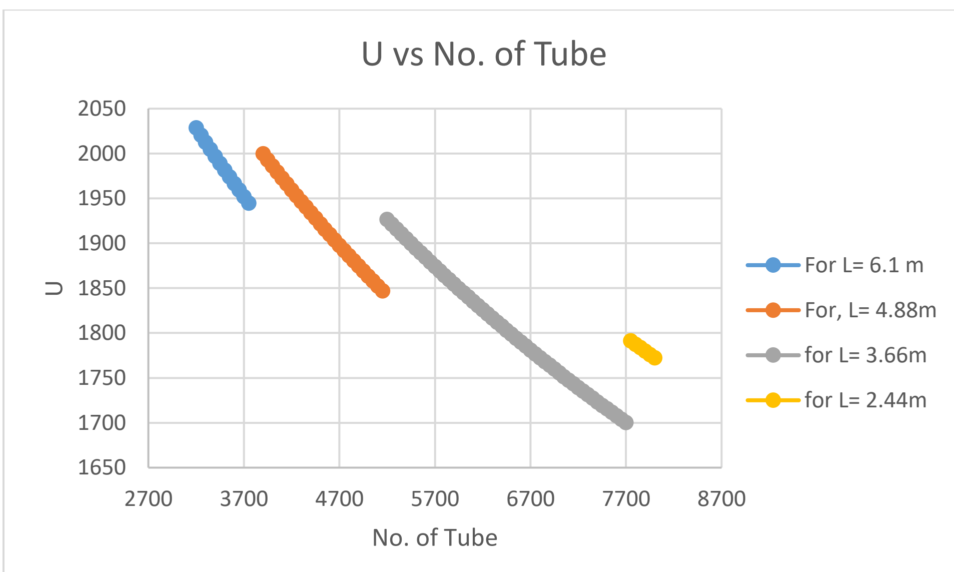


Fig. 7: U vs Number of tubes (for constant effective heat transfer length, L)

This downward trend can easily be described, in eqn. 4, $\dot{Q}, F,\ LMTD$ are constant as the can be evaluated from given case study, so, only varying terms are $U\ and\ A_o$ so, A_o is proportional to both L and N_t so for fixed L, U is inversely proportional to N_t as shown in plot 7.

Other parametric studies are taken from HTRI,

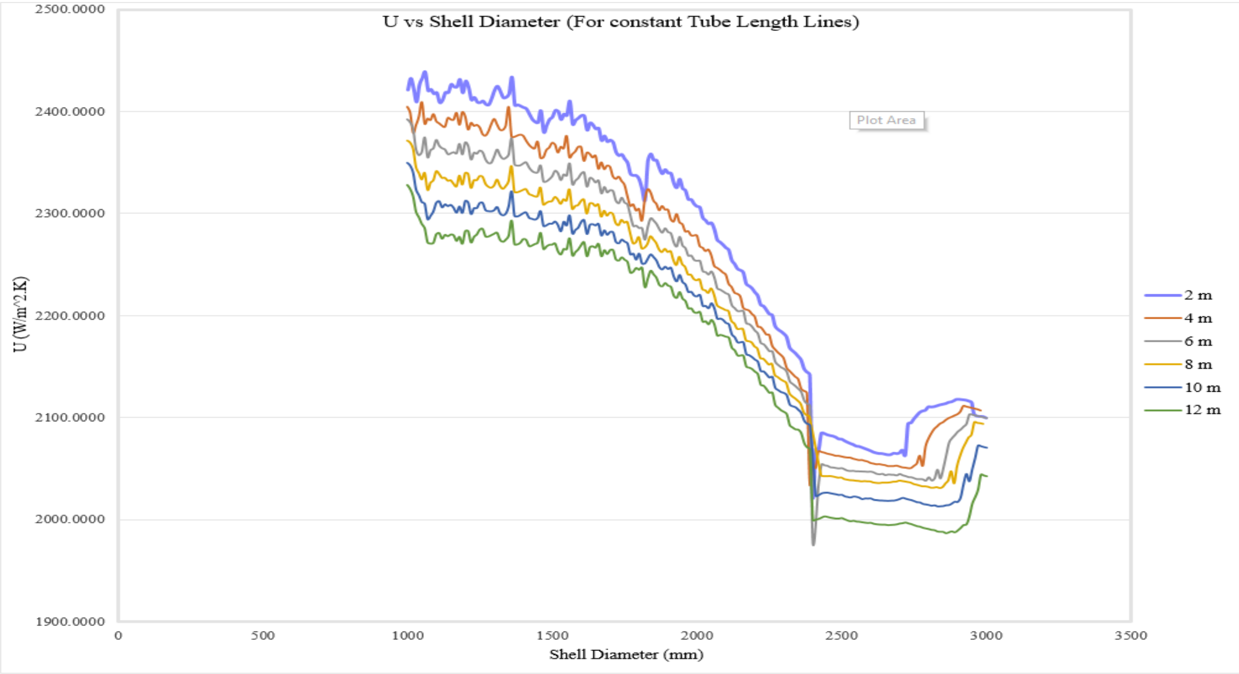


Fig 8: U vs Shell Diameter

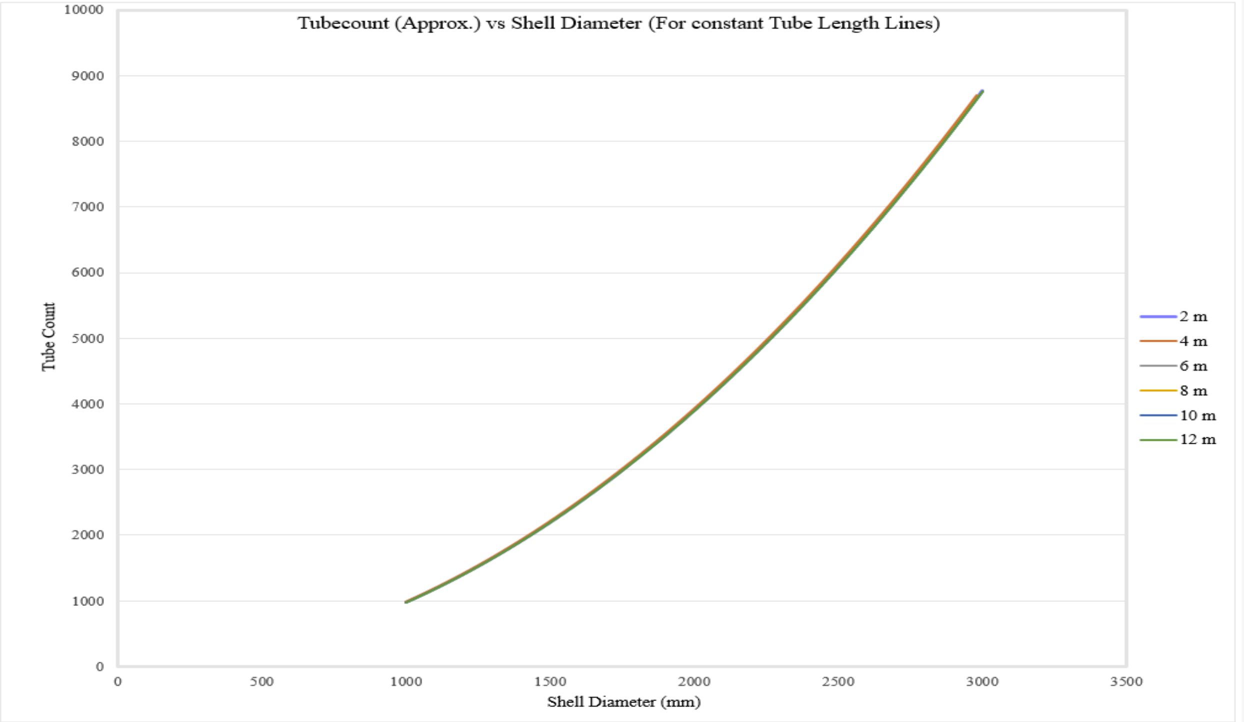


Fig 9: No. of tube vs Shell diameter.

There two figures can easily be elaborated from eqn. 10. N_t is proportional to ID_s^2 which is shown in Fig. 9 and stated earlier, for fixed length U is inversely proportional to N_t so, the curves in fig 8 follows the same trend. For varying length doesnot effect that much in Nt vs ID_s Fig 9. Some wiggles are due to change in other parameter.

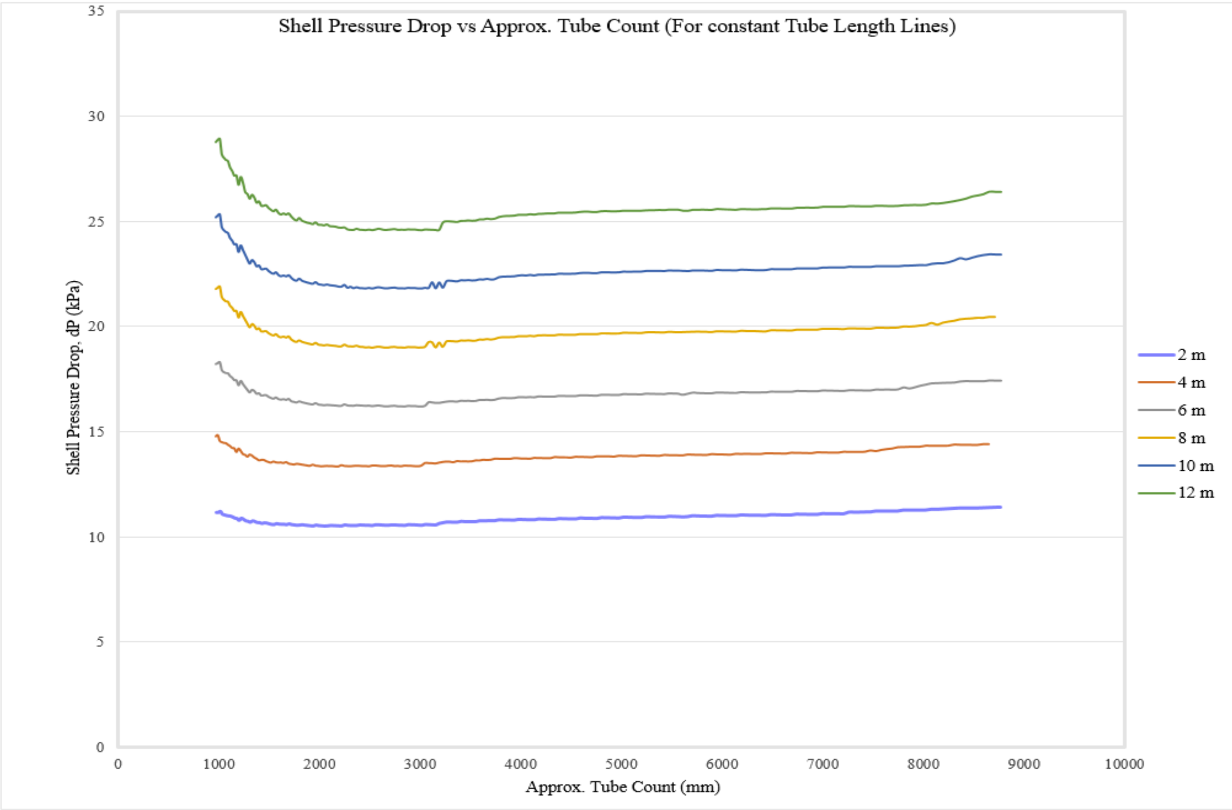


Fig 10: Shell Pressure drop vs No. of tube

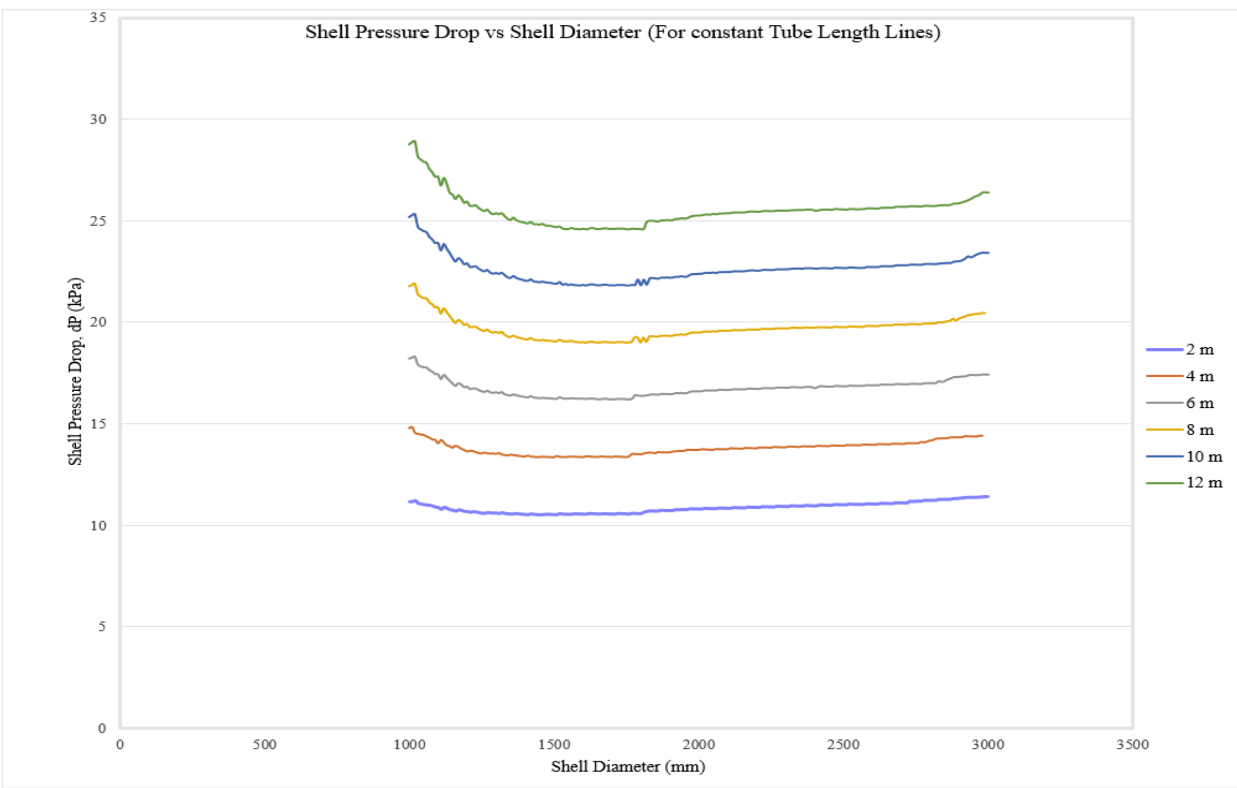


Fig 11: pressure drop vs shell diameter

Shell side pressure drop is independent of Shell diameter and No. of tubes (from fig 10 & 11) for fixed length. This is due to varying shell diameter and No. of tube effect other design parameters in order to preserve the constraint in problem statement the trendline follow almost constant pattern as shown in fig 10 & 11. But for changing length a discrete jump is observed, this is due to for increasing length friction factor increases.

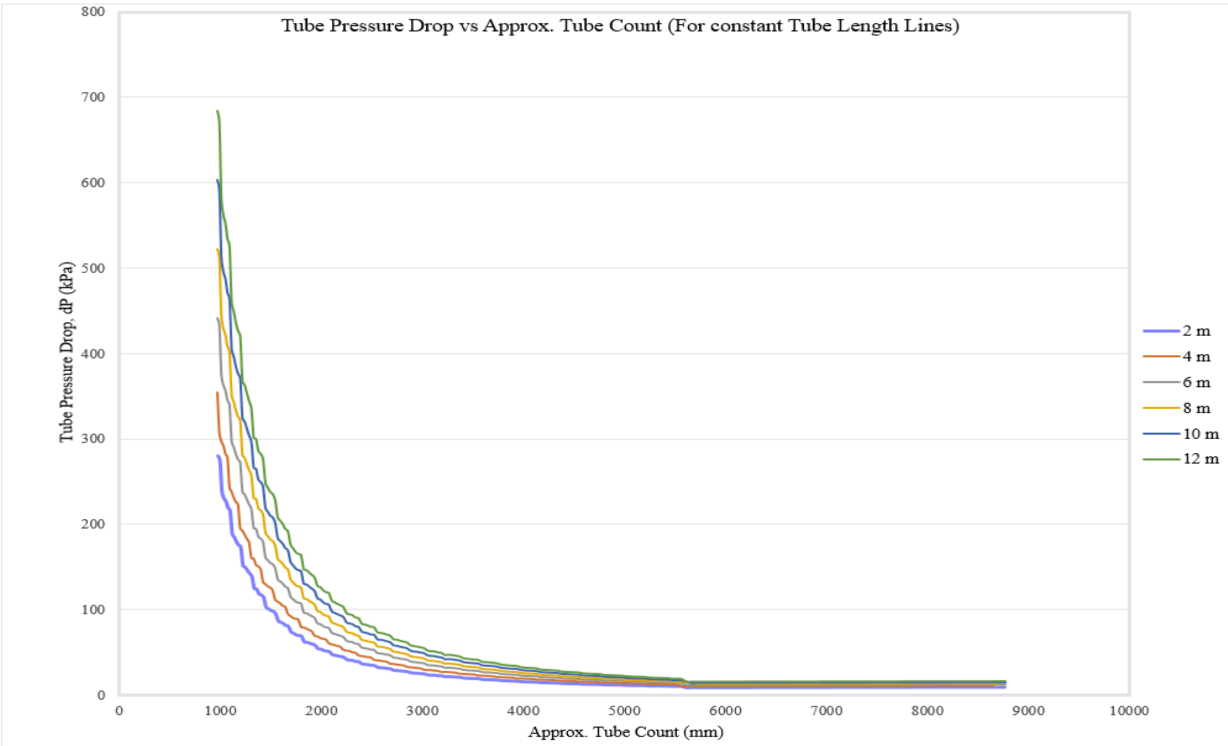


Fig 12: Tube pressure drop vs No. of tubes

Fig 12 states pressure drop in tube side decreases for with increasing this is due to for fixed length Re at tube side decreases for increasing tube number so, friction factor decreases. For varying, length the scenario is obvious. For increasing tube number length reduces so pressure drop decreases due less major loss.

Now, some parametric study for cost is also considered,

Fig 13: Operational cost vs Tube pressure drop

This linear trend in fig 13 is obvious. Operational cost is linearly proportional to pumping power and pumping power is proportional to tube side pressure drop eqn.32 & 33.

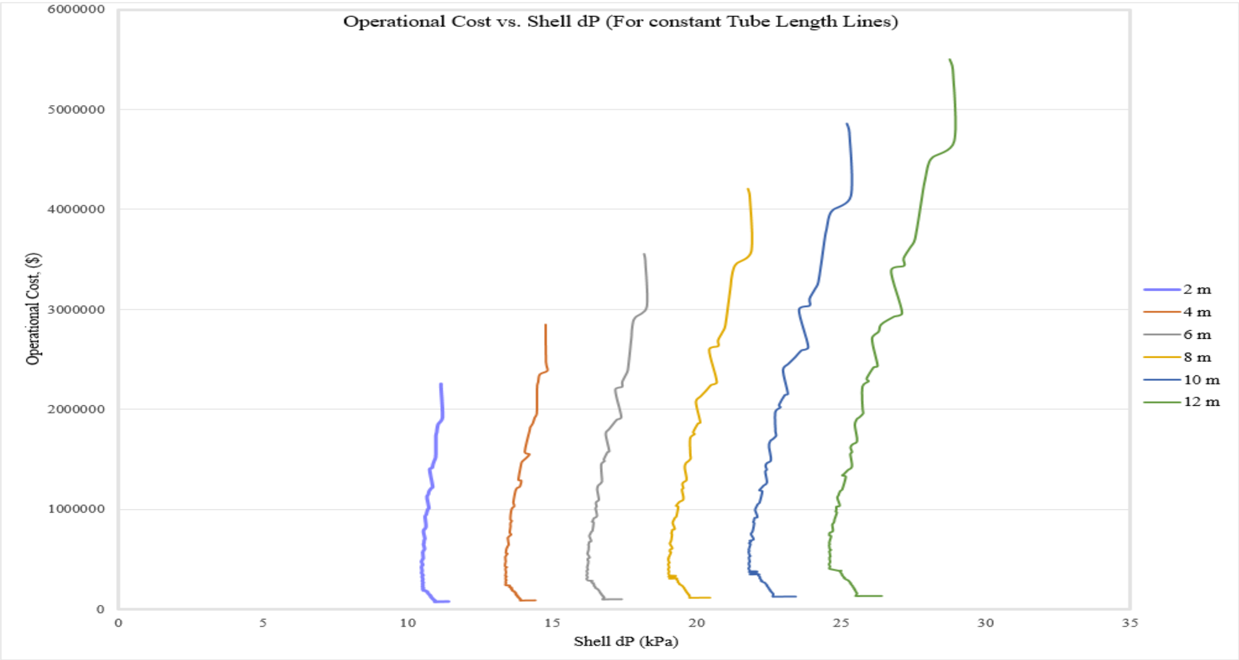


Fig 14: Operational cost vs shell pressure drop

This trend in fig 14 is also obvious from eqn. 32 & 33.

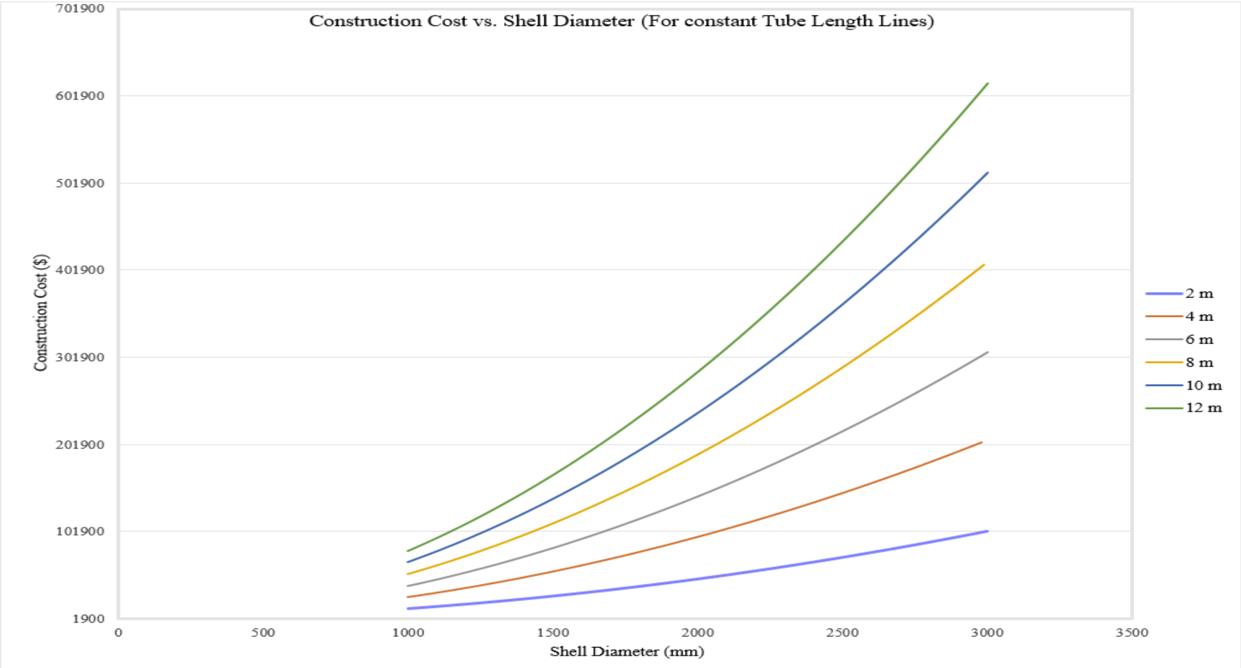


Fig 15: Construction Cost vs shell diameter

Since, Construction cost is proportional to shell volume which is proportional to ID_s^2 , fig 15 reports a parabolic upwards trend.

Conclusion:

So, in this entire report, a steam generator model is proposed to meet the requirement of the given case study, for this purpose thermo hydraulic analysis, of the thermo-fluid as well as mechanical design constraints are taken into account. Using python code and HTRI simulation software, the entire analysis is performed. The suggested design parameters as well as its performance parameter values are reported. Variation of the results with reasoning is reported. The design suggested by HTRI software is more accurate to use since it took several factors into account. Afterwards cost evaluation of the project is estimated. then, design of accessories are also reported. Finally parametric study is also performed and to show other forms of solutions.

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