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Design Approach of Shell and Tube Vaporizer for LNG Regasification

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Abstract

Natural Gas as a fossil fuel has been an emerging source of renewable energy in the last decade. It is converted to liquefied form (LNG) by keeping in a cryogenic state and transported via pressure vessels. We have considered the use of shell and tube heat exchanger which is appropriate for high pressure applications. Turbulent heat transfer conditions are utilised in the design calculations. The design of TEMA (BJ21M) type heat exchanger was modelled with the consideration that, tube side fluid is LNG and two different shell side fluids are ethylene and propylene glycol water. Design and comparison was carried out in relevance to cases of with and without twisted tape turbulators. Heat Transfer Research Inc. (HTRI) software was used to perform the thermal design. It was found that heat exchangers with twisted tape turbulators operating with L/D ratio of 18 gave better heat transfer co-efficient on both shell and tube side compared to heat exchangers without turbulators. Further with optimization of tube side parameters and tube length, it was noted that the heat transfer rate increased in both shell and tube side considerably for both the cases of ethylene and propylene glycol water on shell side. For the given design constraints, it was seen that the working of the vaporiser with ethylene glycol water and turbulators in shell side was efficient and better compared to propylene glycol water and turbulators in shell side. The authors have aimed at reducing the space and cost constraints of regasification equipment. The software results are in good agreement with analytical and numerical outputs.

Keywords: *Thermal design, LNG Regasification, Shell and Tube Heat Exchanger, HTRI, Turbulators*.

1. Introduction

Natural gas can be referred as a hydrocarbon gas mixture comprising predominantly of methane, and containing variable extents of other higher alkanes along with minute amounts of carbon dioxide, nitrogen and hydrogen sulphide. It assumes popularity due to its low environmental impact. The low density characteristic makes natural gas challenging for storage and transportation. The feasible way is to liquefy it in a liquefaction plant where it is super cooled to -162°C. This operation leads to a reduction in volume by more than 600 times, which makes it efficiently practical to store and transport. LNG is vaporized for further use at receiving terminals by superheating the high-pressure LNG. The vaporization process is accomplished by means of shell and tube heat exchanger. The liquefaction process eliminates the odour from the gas which necessitates that the gas must be odorized before exiting the vaporizer. Figure 1 depicts a flow diagram of the regasification process. The intention behind selecting a shell and tube heat exchanger is that it can provide enhanced heat transfer, wide variation of pressure and pressure drops, inexpensive accommodation of thermal stresses, easy cleaning and repair, considerable flexibility concerning materials of construction to manage corrosion and other concerns.

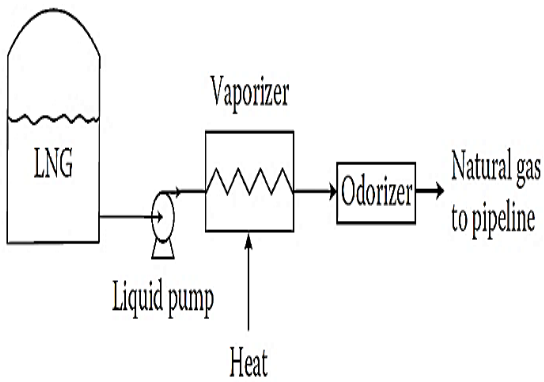

Figure 1.Simplified flow diagram of a typical LNG regasification system.

Allocation of the fluid is done based on the considerations shown in the Table 1.

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Table 1: Parameters for allocation of fluid

PARAMETERS	DESCRIPTION
Corrosion	Tube side is selected to carry more corrosive fluid.
Fouling	Tube side is designated to carry more fouling fluid.
Fluid Temperature	Hotter fluid in the tube side if temperatures are high enough that require special alloys.
Operating Pressure	Tubes are comparatively cheaper than shells for high pressure conditions.
Viscosity	High viscous material in the shell side, if it can provide turbulence. The value of Critical Reynolds number for turbulent flow in the shell ranges around 2300. We can consider transferring the fluid to the tubes if the shell fails to pronounce desired turbulence.
Flow rate	Economical design of the vessel demands that the shell should accommodate the fluids with lowest flow-rate.
Pressure drop	It is necessary that the shell side should put up with the fluid having lowest acceptable pressure drop.

The structure of the heat exchanger is made on the basis TEMA standards which are shown in Figure 2. The figure represents the different types of stationary head, shell and rear head type. Configuration of the shell and tube heat exchanger assembly is based on different combination of stationary head, shell and rear head types.

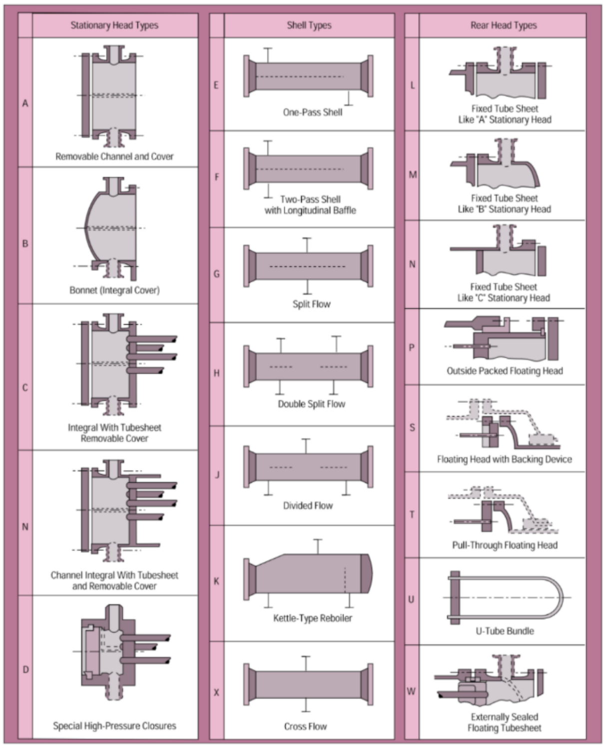

Figure 2: TEMA configurations of Heat Exchangers© 1988 by Tubular Exchanger Manufacturers Association [26]

This paper discusses about the thermal design and optimization of the shell and tube heat exchanger for LNG gasification. Shell and tube heat exchanger with and without turbulators were considered for the present design and study. Figure 3 shows the appearance of the heat exchangers with the incorporation of turbulators.

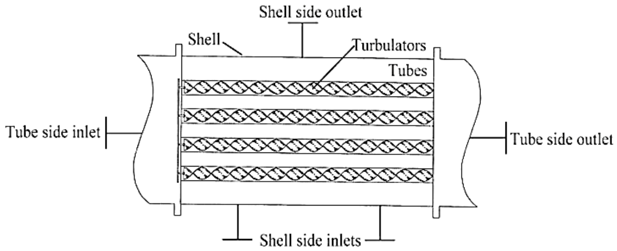

Figure 3: Schematics of Heat Exchanger with Turbulators

2. LITERATURE REVIEW

Shinji Egashira [1] explained clearly the concept of LNG vaporization in his paper. The author showed the different trends in receiving terminal, structures and features of LNG vaporizer for primary and secondary receiving terminals, and different types of vaporizers, such as floating storage re-gasification unit, open Rack Vaporizers, intermediate fluid vaporizers, submerged combustion vaporizers, and intermediate fluid vaporizers with air heat source..etc, along with their features, development of vaporizers with air heat-source and the future development of the LNG vaporizers.

Michelle Michot Foss [4] published a paper, where the author briefly explains about the LNG industry and growing role LNG may play in the energy future. The paper briefly informs us about the LNG safety and security, the role of LNG in developing countries natural Gas supply and demand, and it also addresses details on LNG operations and market place.

Brian Eisentrout, Steve Winter Corn and Barbara Weber [5] presented a paper where a study was conducted on six systems to provide an evaluation of each system in conjunction with the conditions that might influence its viability. The re-gasification systems evaluated in this study are fixed heaters with intermediate fluid, submerged combustion vaporizers, sea water vaporizers, heating towers with intermediate fluids, gas turbine generators with WHR, and steam turbine generator cycle.

Patel et al. [6] mentioned the guidelines necessary to choose a LNG vaporization design appropriate for present day terminals according to the site's climatic conditions. Conventionally, base load re-gasification terminals are categorised into two types: 70% utilities are in the form of Open Rack Vaporizer (ORV), 25% see the use of Submerged Combustion Vaporizer (SCV) and 5% use Intermediate Fluid Vaporizer (IFV). This paper showcases the results of LNG vaporization Screening Study for LNG re-gasification facilities located in warm climate and cold climate regions of the world.

Favi and Olt Livorno [7] gave a clear concept of FSRU's on their paper. The FSRU's are the LNG receiving terminals and the special feature in this type is that they are placed offshore and in event of disaster, loss to public can be evaded. They also gave a detailed outline of construction of the LNG tanks and Terminals and also the measures for pitching and rolling of the hull, as it works on offshore.

Rajiv Mukerjee [8] elucidates on the theory of thermal design which involves the topics on STHE components and their description according to their construction and service. He provides an overall idea to carry out an optimum design of heat exchangers. The paper also describes the fundamental principles related to thermal design of heat exchangers and effective use of software tools. He also briefly explains the components, classification, design data, configurations, and optimization in design etc.

Yusuf Ali Kara and Ozbilen Guraras [9] composed a computerized approach to achieve a pilot design of shell and tube heat exchanger with single phase fluid both shell and tube side. They stated that the present approach can provide a methodical variation in exchanger parameters. The model defines the overall dimensions of the shell, the tube bundle and optimum heat transfer surface area. They also explained that if minimum shell side pressure drop is considered as a criterion for optimum design then the presence of cold fluid in shell side is advantageous than hot fluid as shell as it causes lower shell side pressure drop and has small heat transfer area requirement. In general, it is advisable to have the stream with lower mass flow rate on shell side because of the baffle spacing.

Andre L.H. Costa, Eduardo M. Queiroz [10] have presented an optimization technique based on the minimization of the thermal surface are for a certain service. The proposed algorithm deals with the concept of tube count table search. It is necessary to consider discrete decision variables for optimisation. They have considered the inclusion of important additional constraints which were overlooked earlier in order to estimate the solution to the design practice. They attained minimum computational costs owing to the use of variable bounds, feasibility tests and fathoming procedures.

Than, Lin and Mon [11] presented the design process for an oil cooler in relevance to shell and tube heat exchanger. Their aim was to achieve a high heat transfer rate without exceeding the allowable pressure drop. They used numerical and software tools to serve their purpose. The spoke of limitations occurring in the program and how to eliminate them.

G. Hima Bharati [12] has gone through different types of vaporizer which is suitable for climatic conditions of India, and also explains the details of different types of vaporizers used by different organizations and which they are going to install on the proposed site.

Cong Dinh, Joseph Cho and Jay Jang [13] proposed novel re-gasification methods that use Multi-Temperature Level (MTL) air heaters to achieve less economy and also to be environment friendly. This paper also describes how LNG can be heated and vaporized using cold Heat Transfer Fluid (HTF).

The various literatures available on the topic suggest that sufficient amount of work has been carried out in relation to Open Rack Vaporizers, Floating storage regasification unit, Submerged combustion Vaporizer, and Intermediate fluid vaporizer. It is observed that regasification techniques through shell and tube heat exchanger have been given inadequate outlook. With the increase in demand of natural gas it is quite essential that better methodologies should be employed to carry out regasification process. Hence it is an area which can be explored and valuable amount of research can be conducted. This project is an initiative to understand and work on the above area so that scientific benefit can be achieved.

3. METHODOLOGY

The design procedure is initiated with an analytical approach by implementing the **Kern method** which is described below:

- **Step 1:** Initially, we have to find out the required thermo-physical properties of hot and cold fluids at the caloric temperature or arithmetic mean temperature. Calculation of these properties at the caloric temperature is essential if the variation of viscosity with temperature is large. The comprehensive approach can be derived from "Process Heat" Transfer by Kern.
- **Step 2:** Perform energy balance and find out the heat duty (Q) of the exchanger.
- **Step 3:** Assume a reasonable value of overall heat transfer coefficient ($U_{o,assm}$). The value of $U_{o,assm}$ with respect to the process hot and cold fluids can be taken from the Heat and Mass Transfer Data hand books.
- **Step 4:** Decide tentative number of shell and tube passes (N_p). Determine the LMTD and the correction factor F_T. F_T normally should be greater than 0.75 for the steady operation of the exchangers. Otherwise it is required to increase the number of passes to obtain higher F_T values.
- **Step 5:** Calculate heat transfer area (*A*) required:

$$A = \frac{Q}{U_o,assm.LMTD.F_T} \quad (1)$$

- **Step 6:** Select tube material, decide the tube diameter (ID= d_i, OD = d_o), its wall thickness (in terms of BWG or SWG) and tube length (L). Calculate the number of tubes (Nt) required to provide the heat transfer area (A):

$$Nt = \frac{A}{\pi d_o L} \quad (2)$$

Calculate tube side fluid velocity, $u = \frac{4m\left(\frac{N_p}{N_t}\right)}{\pi \rho d_i^2}$ (3)

If $u < 1\ m/s$, fix N.0p so that, $Re = \frac{4m\left(\frac{N_p}{N_t}\right)}{\pi d_i \mu} \geq 10^4$ (4)

where, m, ρ and μ are mass flow rate, density and viscosity of tube side fluid. However, this is subject to allowable pressure drop in the tube side of the heat exchanger.

- **Step 7:** Decide type of shell and tube exchanger (fixed tube sheet, U-tube etc.). Select the tube pitch (P_T), determine inside shell diameter (D_s) that can accommodate the calculated number of tubes (N_t). Use the standard tube counts table for this purpose. Tube counts are available in standard text books.
- **Step 8:** Assign fluid to shell side or tube side. Select the type of baffle (segmental, doughnut etc.), its size (i.e. percentage cut, 25% baffles are widely used), spacing (B) and number. The baffle spacing is usually chosen within 0.2 D_s to D_s.
- **Step 9:** Determine the tube side film heat transfer coefficient (h_i) using the suitable form of Sieder-Tate equation in laminar and turbulent flow regimes. Estimate the shell-side film heat transfer coefficient (h_o) from:

Nu = 0.36 (Re) 0.55(Pr) 0.33(μ / μw)0.14 (5)

ho = (Nu x K) / De (6)

You may consider, $\frac{\mu}{\mu_w} = 1.0$

Select the outside tube (shell side) dirt factor (R_{do}) and inside tube (tube side) dirt factor(R_{di}). Calculate overall heat transfer coefficient ($U_{o,cal}$) based on the outside tube area (you may neglect the tube-wall resistance) including dirt factors:

$$U_{o,cal} = \left[\frac{1}{h} + R_{do} + \frac{A_o}{A_i}\left(\frac{d_o - d_i}{2k_w}\right) + \frac{A_o}{A_i}\left(\frac{1}{h_i}\right) + \frac{A_o}{A_i}R_{di}\right]^{-1} \quad (7)$$

- **Step 10:** If, $0 < \frac{U_{o,cal} - U_{o,assm}}{U_{o,assm}} < 30\%$, go the next **step 11**. Otherwise go to **step 5**, calculate heat transfer area (A) required using $U_{o,cal}$ and repeat the calculations starting from step 5. If the calculated shell side heat transfer coefficient (h_o) is too low, assume closer baffle spacing (B) close to 0.2 Ds and recalculate shell side heat transfer coefficient. However, this is subject to allowable pressure drop across the heat exchanger.

- **Step 11:** Calculate % overdesign. Overdesign represents extra surface area provided beyond that required to compensate for fouling. Typical value of 10% or less is acceptable.

$$\% \text{ Overdesign} = \frac{A - A_{reqd}}{A_{reqd}} \times 100 \quad (8)$$

A = design area of heat transfer in the exchanger; A_{reqd} = required heat transfer area.

- **Step 12:** Calculate the tube-side pressure drop (ΔP_T): (i) pressure drop in the straight section of the tube (frictional loss) (ΔP_t) and (ii) return loss (ΔP_{rt}) due to change of direction of fluid in a "multi-pass exchanger".

Total tube side pressure drop: $\Delta PT = \Delta P_t + \Delta P_{rt}$ (9)

- **Step 13:** Calculate shell side pressure drop (ΔP_S): (i) pressure drop for flow across the tube bundle (frictional loss) (ΔP_s) and (ii) return loss (ΔP_{rs}) due to change of direction of fluid.

Total shell side pressure drop: $\Delta PS = \Delta P_s + \Delta P_{rs}$ (10)

If the tube-side pressure drop exceeds the allowable pressure drop for the process system, decrease the number of tube passes or increase number of tubes per pass. Go back to **step 6** and repeat the calculations steps. If the shell-side pressure drop exceeds the allowable pressure drop, go back to **step 7** and repeat the calculations steps.

The thermal design of shell and tube heat exchanger for LNG re-gasification is carried out using Heat Transfer Research Inc. (HTRI) software. Based upon the design requirement as shown below, shell and tube heat exchanger with two inlets and one outlet on shell side and single tube pass (BJ21M) was designed as per TEMA standard which is shown in Figure 4. Design requirement involves:

- Maximum length of heat exchanger should not exceed 15 meters.
- Mounting of the vaporizer should be vertical.
- Two inlets and one outlet should be on shell side.
- Heat exchanger tube side should have single pass and
- Maximum weight of heat exchanger should not exceed 60,000 kgs.

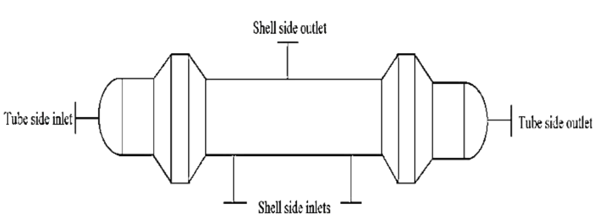

Figure 4: TEMA BJ21M Heat Exchanger with One Tube Pass© 1988 by Tubular Exchanger Manufacturers Association [26]

Design pressure of a heat exchanger is the gage pressure at the top of the vessel. This pressure is used to determine the minimum wall thickness of the various pressure parts. The IS: 4503 specifies that the design pressure should at least 5% greater than the maximum allowable working pressure. Usually a 10% higher value is used. The design temperature is used to determine the minimum wall thickness of various parts of the exchanger for a specified design pressure. It is normally 10ºC greater than the maximum allowable temperature. All materials used construction of shell and heat exchangers for pressure parts must have the appropriate specification as given in IS: 4503 Appendix C. The materials of construction should be compatible with process fluids and others parts of materials and should be cost effective.

Based on design requirements, the working methodology involves assessment of input parameters which are suitably given to the software to initiate processing as given in Table 2.

Table 2: Input Parameter Datasheet

LNG VAPORIZER DATA SHEET					
		SHELL SIDE		TUBE SIDE	
		INLET	OUTLET	INLET	OUTLET
Fluid Name		WATER ETHLENE GLYCOL/WATER PROPYLENE GLYCOL		LNG	
Total Fluid	kg/hr.	1439648	1439648	109664	109664
Total Liquid	kg/hr.	1439648	1439648	109664	-
Total Vapour	kg/hr.	-	-	-	109664
Temperature	°c	16	2	-150	6
Inlet Pressure	bar	1.5	-	84.6	-
Allow. Pressure Drop	bar	0.7		2	
Design Pressure	bar	10		145	
Design Temperature	°c	65		65	
Fouling Factor	m²°c/w	0.00035		0.00018	
Heat Exchanged	kw	20.142			
Material of Construction	Stainless Steel				

The nominal diameter (outside diameter in millimetres rounded is to the nearest integer) of the heat exchanger is specified in IS: 2844-1964 in case of shells manufactured from flat sheet. The following diameters (in mm) should be preferably used in the case of cylindrical pipe shell: 159, 219, 267, 324, 368, 419, 457, 508, 558.8, 609.6, 660.4, 711.2,762, 812.8, 863.6, 914.4 and 1016. The minimum shell thickness should be decided in compliance with the nominal shell diameter including the corrosion allowance as specified by IS: 4503.

The software yields corresponding output values, which help us in generalizing the design. Optimization was carried out by iterative process till the obtained design suits the given requirements and process. Further the design process was carried out using turbulators as tube inserts for increasing the heat transfer rate in both shell and tube side operating fluids with an L/D ratio of 18 thus increasing the performance of the complete heat exchanger system. A comparative study between shell side operating fluid as ethylene glycol water and out which gives us an initial design of the vaporizer. It was found that the initial design did not fulfil the design requirements, hence continuous iteration process in HTRI software was carried out using the initial values until a satisfactory optimum design of the shell and tube heat exchanger is made to give maximum heat transfer rate on both shell and tube side. The analytical results were validated with the results obtained from HTRI software. Propylene glycol water was carried out for both the cases of with and without turbulators. The HTRI software was operated in Design mode and the above values (Table 1) were filled in respective areas. Table 3 contains the values which are required to be checked in outputs.

Table 3: Parameters to be checked in outputs

Sl. No.	Properties	Units	Range
1	Over Design	%	6 to 8
2	Pressure Drop Liquids (Shell Side , Tube Side)	kg/cm^2	< 0.7
3	Pressure Drop High Viscous Liquid (Shell Side , Tube Side)	kg/cm^2	> 0.7
4	Pressure Drop Gas side (Shell Side , Tube Side)	kg/cm^2	0.05 to 0.2
5	Tube Velocity	m/s	1 to 3
6	Baffle Cut Single Segmental	%	15 to 45
7	Baffle Cut Double Segmental	%	25-35
8	Baffle Spacing	mm	shell id/5 to shell id
9	A Stream Flow Fraction On Shell Side	%	< 10
10	B Stream Flow Fraction On Shell Side	%	> 40
11	C Stream Flow Fraction On Shell Side	%	< 10
12	E Stream Flow Fraction On Shell Side	%	< 15
13	F Stream Flow Fraction On Shell Side	%	< 10
14	Rho V2 Shell Side (Inlet , Outlet)	$kg/m\ s^2$	< 4500
15	Rho V2 Tube Side (Inlet , Outlet)	$kg/m\ s^2$	< 8938
16	Rho V2 Bundle (Entry , Exit)	$kg/m\ s^2$	< 5953
17	Ratio Of Cross Velocity To Window Velocity Ntiw (NoTubes In Window) Baffles	≤ 3	
18	Ratio Of Cross Velocity To Window Velocity (Others)	1 to 1.2	
19	Bundle Weight	kg	≤ 20000
20	Bundle Dia	mm	≤ 1500
21	Tube Length	m	4 to 12

The selection of material is done according to the ASME standards (ASME Section VIII, Div. 1 & 2). The ASME standard offers material sustainability for different temperatures and pressures. For this case, we selected stainless steel, because it can withstand the temperature and pressure constraints and it is comparatively cheaper. Once again the program is executed and the output values are checked for small variations. The parameters are re-iterated till optimum values are obtained.

4. RESULTS AND DISCUSSION

The design of Shell and Tube Vaporizer is attempted through manual process by implementing D. Q. Kern Method. The solution converges after 3 iterations. This study contains the output of final iteration.

4.1. Given data:

Hot fluid inlet temperature (T_1) = 16°C
Hot fluid outlet temperature (T_2) = 2°C
Cold fluid inlet temperature (t_1) = -150°C
Cold fluid outlet temperature (t_2) = 6°C
Fouling factor of hot fluid (R_h) = 0.00035 (for ethylene glycol water)
Fouling factor of cold fluid (R_c) = 0.00018 (for LNG)
P_{inlet} (for hot fluid) = 1.5 bar
P_{inlet} (for cold fluid) = 84.5 bar
Δp_{max} (for hot fluid) = 0.7 bar
Δp_{max} (for cold fluid) = 2 bar
Mass flow rate of hot fluid (m_h) = 66.66 kg/s Mass flow rate of cold fluid (m_c) = 10.154 kg/s

4.1.1. LMTD Calculation

ΔT_{lmtd} = 43.38°C
R = 0.08974
S = 0.93975
F_T = 0.93

4.1.2. Energy Balance

Assuming no heat loss to the surroundings,
$Q_h = m_h \times C_p \times \Delta T$ = 3353.466 kW

4.1.3. Calculation of heat transfer area and tube numbers

The iteration is started assuming 1 shell and 1 tube pass shell and tube exchanger with following dimensions and considerations.
Fixed tube sheet
19.05 mm OD tubes (d_o) on 25.4 mm square pitch (P_T)
Outer diameter of tube = 19.05 mm
Tube length (L_t) =8534 mm
Tube ID (d_i) = 14.85 mm
Fluid arrangement = LNG is placed in tube side.
Assuming overall heat transfer coefficient value as $U_{o,assm}$ = 217 $W/m^2°C$
A = 382.1202 m^2
Nt = 1502.3
The next standard number of tubes is 1560 and corresponding shell ID is 1.16 m

4.1.4. Calculation of Heat transfer coefficient of Shell side and Tube side

A_s = 0.0783 m^2
G_s = 851.3410 kg/m^2-s
D_e = 0.0241 m
R_e = 7404.6

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$P_r = 22.3452$

$N_u = 155.3411$

$h_o = 2.8719\ W/m^2°C$

In a similar way we have to find out tube side Heat Transfer Coefficient,

$A_{tp} = 0.2702m^2$

$G_s = 37.5811\ kg/m^2\text{-}s$

$R_e = 4493.4$

$P_r = 2.117$

$h_{o,l} = 334.6041\ W/m^2°C$

Similarly for vapour,$h_{o,v}$ =319.9942 W/m2°C

We have to calculate 2 phase heat transfer coefficient using Kandlikar correlation.

$h_{TP} = 443.1122\ W/m^2°C$

$h_i = 365.9035\ W/m^2°C$

4.1.5. Overall Heat transfer coefficient calculation

$U_{o,cal} = 217.7135\ W/m^2°C$

The above results bear similar results when run through MATLAB program. Figure 5 depicts a small part of the MATLAB program used for the calculations.

```
% program for Design of SNT Vaporizer by Kern Method%
clc;
fprintf('Given Data ');
Mh    = input('Enter the hot fluid mass flow rate in kg/s: ');
Mc    = input('Enter the cold fluid mass flow rate in kg/s: ');          % mass flow rate in kg/s
th1   = input('Enter the hot fluid inlet temperature in degrees: ');    % in degrees
th0   = input('Enter the hot fluid outlet temperature in degrees: ');   % in degrees
tc1   = input('Enter the cold fluid inlet temperature in degrees: ');   % in degrees
tc0   = input('Enter the cold fluid outlet temperature in degrees: ');  % in degrees
rfh   = 0.00035                   % Fouling resistance of hot fluid in m^2/kw
rfc   = 0.00018                   % Fouling resistance of cold fluid in m^2/kw
Np    =1                          % No of passes
Rhoh   = 1064.5                   % Density of hot fluid in kg/m^3
Rhocl  = 463.47                   % Density of cold fluid at inlet in kg/m^3
Rhocv  = 99.925                   % Density of cold fluid at outlet in kg/m^3
Vish   = 0.0027675                % Viscosity of hot fluid in kg/m-s
Viscl  = 0.0001242                % Viscosity in cold fluid inlet in kg/m-s
Viscv  = 0.0000136                % Viscosity in cold fluid outlet in kg/m-s
Cph    = 3593                     % Specific heat of hot fluid in J/kg-C
Cpcl   = 2945.6                   % Specific heat of cold fluid inlet in J/kg-C
Cpcv   = 3437.1                   % Specific heat of cold fluid outlet in J/kg-C
Kh     = 0.445                    % Thermal conductivity of hot fluid in W/m-C
Kcl    = 0.1728                   % Thermal conductivity of cold fluid inlet in W/m-C
Kcv    = 0.0431                   % Thermal conductivity of cold fluid outlet in W/m-C
Tw1    = (tc0-tc1)
Tw2    = (th1-th0)
Q      =Mh*Cph*Tw2
DelT   =((th1-tc1)-(tc0-th0))/log((th1-tc1)/(tc0-th0))
Uas    =input('Enter the value of Assumed Overall Heat Transfer Coefficient: ');
```

Figure 5: MATLAB Code for Kern Method

Figure6 shows the results obtained from HTRI software after several iterations carried out to optimize the performance of the heat exchanger. The values obtained includes shell inner diameter (ID), number of tubes, tube outer diameter (OD), thickness, pitch and length to obtain maximum heat transfer from the heat exchanger. The maximum heat transfer rate in both the shell and tube side are also obtained from the HTRI software result for the optimized design as mentioned earlier.

Ethylene Glycol Water As Shell Side Fluid (Case Type 1)									
Description	Shell ID (Mm)	No. Of Tubes	Tube OD (mm)	Tube Thickness (mm)	Tube Pitch (mm)	Tube Length (mm)	Tube Layout Angle	Shell Side h (w/m2k)	**Tube Side h (w/m2k)**
Without Turbulators	1200	1640	19.05	2.11	25.4	8534	90°	2324.5	**350.61**
With Turbulators	1100	1380	19.05	2.11	25.4	7315	90°	2585.6	**711.83**
Propylene Glycol Water As Shell Side Fluid (Case Type 2)									
Description	Shell ID (Mm)	No. Of Tubes	Tube OD (mm)	Tube Thickness (mm)	Tube Pitch (mm)	Tube Length (mm)	Tube Layout Angle	Shell Side h (w/m2k)	**Tube Side h (w/m2k)**
Without Turbulators	1200	1640	19.05	2.11	25.4	8700	90°	1432.3	**356.58**
With Turbulators	**1100**	**1380**	**19.05**	**2.11**	**25.4**	**8000**	**90°**	**1524.5**	**588.9**

Figure 6: HTRI results

From Figure 6, it is clear that the heat transfer rate in both shell and tube side is maximum with the usage of turbulators compared to without turbulators for the given design constraints. This is because in the former case, the inlet fluid enters with high pressure and goes out randomly in plain tubes i.e. without turbulators thus resulting in low heat transfer rate. In the case of latter with the use of turbulators, the velocity of the fluid flow gets reduced and creates turbulence so that the heat transfer rate increases thus increasing the efficiency of the heat exchanger. Various comparisons with respect to shell side fluid operation with and without turbulators for the optimization of the results and maximum heat transfer rate are discussed below.

4.2. Comparison of Shell ID:

Figure7 shows the comparison of shell inner diameter (ID) values obtained from HTRI software between the case type of with and without turbulators. It is notable that the shell ID has been considerably decreased from 1200mm to 1100mm in the case type of with turbulators which is cost effective.

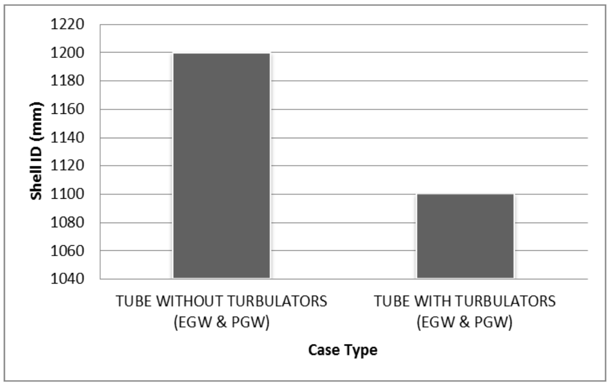

Figure 7: Comparison of shell ID

4.3. Comparison of Tube Length and Number of Tubes:

With the use of turbulators as tube inserts with constant shell and tube outside diameter (OD), pitch and thickness, it is seen that the heat transfer rate on both shell and tube side is considerably increasing. Further the length of the tubes and tube count decrease substantially even for high heat transfer rate in both shell and tube side. The comparison of the tube length and the number of tubes for different case types is shown in Figure8. Therefore, with the use of turbulators as tube inserts, increase in heat transfer rate and decrease in weight and cost of the heat exchanger is achieved.

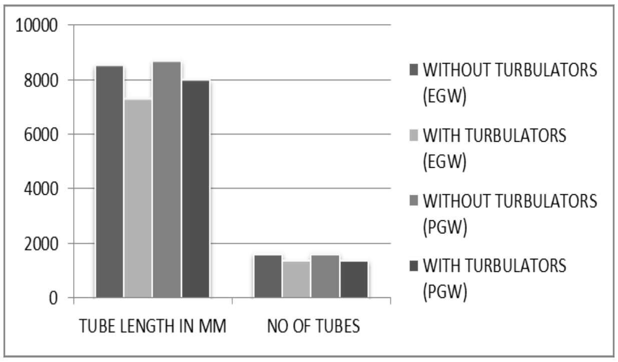

Figure 8: Comparison of Tube length and Number of Tubes

4.4. Comparison of Shell Side Heat Transfer Rate:

Figure9 shows the comparison of shell side heat transfer rate for both the case types of with and without turbulators using ethylene and propylene glycol water mixture on shell side. It is observed that for both the case types, the heat transfer rate increases with the usage of turbulators with the decrease in tube length and tube count as discussed earlier.

4.5. Comparison of Tube Side Heat Transfer Rate:

The comparison of tube side heat transfer rate with and without turbulators for both ethylene and propylene glycol water case types is shown in Figure10. As in the case of shell side heat transfer rate, it was found that the heat transfer rate in tube side also increased considerably with the use of turbulators**.**

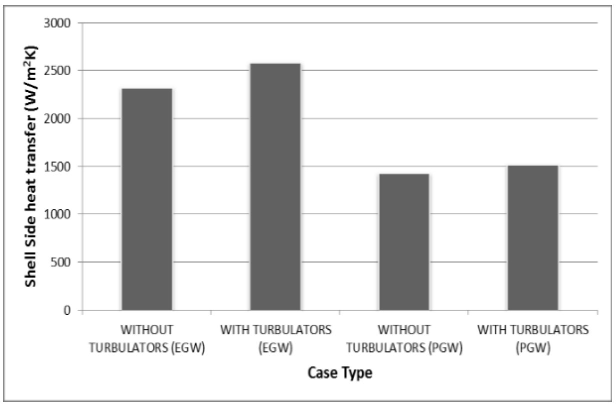

Figure 9: Comparison of Shell Side Heat Transfer Rate

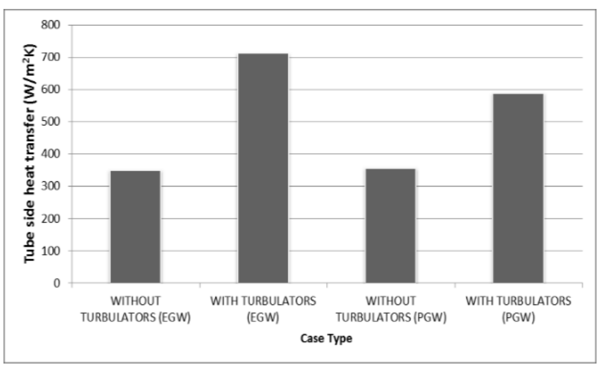

Figure 10: Comparison of Tube Side Heat Transfer Rate

Therefore, from the above comparisons, it is evident that the use of turbulators plays an important role in the heat transfer rate in both shell and tube side. The shell ID has considerably reduced with the use of turbulators. Considering tube length for the case types of ethylene glycol water and propylene glycol water as shell side fluid, the tube length has decreased from 8534 mm to 7315 mm in the former and 8700 mm to 8000 mm with the use of turbulators. The number of tubes has also decreased from 1640 to 1380 in both the case types with the use of turbulators. The shell and tube side heat transfer rate has also increased at a higher rate with the use of turbulators as shown in Table 2 for both the case types. Thus it is also evident that the use of ethylene glycol water as shell side fluid has much better performance compared to the use of propylene glycol water as tube side fluid with and without turbulators.

5. CONCLUSION

Thermal design of TEMA (BJ21M) type shell and tube heat exchanger was carried out for LNG regasification as per given design constraints. LNG is used as a tube side fluid due to its high pressure and ethylene and propylene glycol water is used as a shell side fluid. Design and optimization of this heat exchanger was carried out using HTRI software. Further the case types of without and with turbulators was considered for performance analysis. The major highlights are as follows:

1. The use of turbulators increases the performance of the heat exchanger and decreases the number of tubes and tube length thus making it cost effective.
2. The performance of the heat exchanger with the use of ethylene glycol water was considerably efficient compared to propylene glycol water as shell side fluid.
3. The heat transfer rate at shell and tube side of heat exchanger consisting of ethylene glycol water with turbulators was much better than that of propylene glycol water with turbulators thus increasing the overall performance of the heat exchanger.
4. Thus for the given design constraints, shell and tube heat exchanger operating with ethylene glycol water as shell side fluid with tabulators is much efficient and preferred compared to heat exchanger with propylene glycol water with turbulators.

REFERENCES

[1] Egashira, S. LNG Vaporizer for LNG Re-gasification Terminal. *Kobelco Tech. Rev.*, 32, 64-69 (2013).

[2] Kim, H., & Lee, J. Design and Construction of LNG Regasification Vessel. *Proceedings of GASTECH*(2005).

[3] Kawamoto, H. Natural gas regasification technologies. *Coast Guard Journal of Safety & Security at Sea, Proceedings of the Marine Safety & Security Council*, 65 (4) (2008).

[4] Michelle Michot Foss. Offshore LNG Terminals.*Centre for Energy Economics*, Texas University (2006).

[5] Eisentrout, B., Wintercorn, S., & Weber, B. Study focuses on six LNG regasification systems. *LNG journal*, 21-22 (2006)

[6] Patel, D., Mak, J., Rivera, D., & Angtuaco, J. LNG vaporizer selection based on site ambient conditions. *Proceedings of the LNG*, 17, 16-19 (2013).

[7] Favi. And OLT Livorno,FSRU's: An innovative solution for oil and gas industry. *Convegni Tematici ATI-2012*, Sesto San Giovanni (MI) (2012).

[8] Mukharji, R. Effective design of shell and tube heat exchanger. *American Institute of Chemical Engineering*(1988).

[9] Kara, Y. A., & Güraras, Ö. A computer program for designing of shell-and-tube heat exchangers. *Applied Thermal Engineering*, 24(13), 1797-1805 (2004).

[10] Costa, A. L., & Queiroz, E. M. Design optimization of shell-and-tube heat exchangers. *Applied Thermal Engineering*, 28 (14), 1798-1805 (2008).

[11] Than, S. T. M., Lin, K. A., & Mon, M. S. Heat exchanger design. *World Academy of Science, Engineering and Technology*, *46*, 604-611(2008).

[12] G. Hima Bharathi.Comparitive Study of LNG Vaporizers in India. *International Oil and Gas Conference, Petrotech*, New Delhi (2012).

[13] Cong Dinh, Joseph Cho, Jay Yang, *Cost Effective LNG Regasification with Multi-Temperature Level Air Heaters.* SK Engineering and Construction co., Texas, USA.

[14] Kakac, S., Liu, H., & Pramuanjaroenkij, A. *Heat exchangers: selection, rating, and thermal design*. CRC Press (2012).

[15] Perry, R. H., & Green, D. W. *Perry's chemical engineers' handbook*. McGraw-Hill Professional(1999).

[16] Kandlikar, S. G. A general correlation for saturated two-phase flow boiling heat transfer inside horizontal and vertical tubes. *ASME J. Heat Transfer*, 112(1), 219-228 (1990).

[17] Shah, M. M. Chart correlation for saturated boiling heat transfer: equations and further study. *ASHRAE Trans.;(United States)*, 88 (CONF-820112) (1982).

[18] Bell, K. J., & Mueller, A. C. Condensation Heat Transfer and Condenser Design. *AI Ch. E. Today's Series*(1971).

[19] Butterworth, D. Condensers and their design. In *Two-Phase Flow Heat Exchangers* Springer Netherlands (pp. 779-828) (1988).

[20] Breber, G. E. O. R. G. E. Condenser design with pure vapor and mixture of vapors. *Heat Transfer Equipment Design*, 477(1988).

[21] Indian Standard (IS: 4503-1967): *Specification for Shell and Tube Type Heat Exchangers*, BIS (2007).

[22] Sinnott, R. K. *Chemical engineering design: SI Edition*. Elsevier (2009).

[23] Kern, D. Q. *Process heat transfer*. Tata McGraw-Hill Education (1950).

[24] Dutta, B. K. *Heat transfer: principles and applications*. PHI Learning Pvt. Ltd.(2000).

[25] Couper, J. R., Penney, W. R., & Fair, J. R. *Chemical process equipment revised 2E: selection and design*. Gulf Professional Publishing (2009).

[26] R. Brogan, SHELL AND TUBE HEAT EXCHANGERS, in: A-to-Z Guide to Thermodynamics, Heat and Mass Transfer, and Fluids Engineering, Begellhouse, n.d. doi:10.1615/AtoZ.s.shell_and_tube_heat_exchangers.