A Novel Design for Plate Heat Exchangers in LNG Liquefaction Cycle

• M.R. Jafari Nasr and E. Mohammadi
Islamic Azad University, Science and Research Branch, Tehran-Iran.
(Email: nasrmrj@ripi.ir)

Received: Feb 14, 2017 / Accepted: Apr 12, 2017

Abstract

LNG production is an intense and complex process, in which the liquefaction accounts for more than 50% of costs. In recent years, design engineers have been made several attempts to optimize this process. The main objective was to increase the production yield and capacity, and minimize the costs. The most important process equipment in liquefaction stage is devoted to compact heat exchangers of Plate Fin or Spiral Wounded types. This article described the simulation of liquefaction cycle of Iran LNG project with triple mixed refrigerant to provide a new method for designing the plate heat exchanger used in this cycle; in addition, a simple method was introduced for selecting the best secondary surface based on the conceptual development of the volume performance index (VPI). The designed exchanger had the minimum surface area and volume. The reduction of required heat transfer surface area had a significant role in the reduction of investment capital cost in LNG production process.

The liquefaction cycle of Iran LNG was fully investigated in this article as an industrial case. According to the simulation, the cold and hot surface areas of the plate heat exchanger, used in the given process, are as large as 3001m^2 and 1933m^2 with the overall heat transfer coefficient of 425 W/m^{2}K°; whereas, designing this exchanger by developed rapid design algorithm (RDA) significantly can be reduced the required cold and hot surface areas by 5.2 and 3.3 times, respectively. The overall heat transfer coefficient was also increased by 2 times.

Keywords: Liquefied Natural Gas, Volume Performance Index, Rapid Design Algorithm

Introduction

Liquefied natural gas (LNG) is the best and healthiest fuel gas. LNG is an odorless, transparent, and non-toxic liquid with a specific weight of about 0.45 grams per cubic centimeter. It is produced with cooling and liquefying natural gas at around -160°C, under a pressure of about 1atm. The liquefaction of the natural gas decreases its volume by 600 times, which makes its transportation to more remote areas simpler and more cost-effective. Moreover, it produces low amount of combustion-produced pollutants and large amount of combustion-produced energy. LNG consists of a complex and costly process, in which liquefaction accounts for the 50% of the process costs. Several attempts have been made in recent years to improve the performance of this process and to reduce investment costs. In liquefaction stage, the plate fin heat exchanger (PFHE) and spiral wounded heat exchanger (SWGE) are the main elements. These exchangers are substituted for common shell-and-tube exchangers, as they are safer and more cost-effective.

The PFHE is a stack of alternating flat and corrugated (fin) plates. Flows exchange heat through pathways surrounded by corrugated sheets separated by flat plates. Fins are used as surfaces for secondary heat transfer and a mechanical barrier against intra-layer pressure. There are different types of fins that allow optimal design of these exchangers in terms of cost, weight, thermal efficiency, and/or pressure drop. The fin-plate heat exchangers are superior to other types of heat exchangers, due to having several advantages including small temperature difference between the cold-and-hot fluids, high thermal efficiency, large heat transfer surface area per unit of volume (almost $1000m^3$), low weight, and the feasibility of heat exchange between different flows [2].

Several attempts have been made by process design engineers in recent years to improve the performance of compact heat exchangers, especially for cooling, through developing modern designs. Among the most important causes are saving energy, cost, and equipment space which should be taken into account by design engineers before other things [3]. Therefore, they look for strategies to develop optimal designs, increase efficiency, and finally reduce costs.

In this article attempts were made to take a fresh step towards improving LNG production through optimization of the main element of LNG liquefaction cycle (compact plate-fin exchanger) with a modern design method. To this end, liquefaction cycle of Iran LNG project with pre-cooled propane along with the triple mixed refrigerant was simulated using Aspen HYSYS. This simulation was done to extract data of compact heat exchanger's inflows in this cycle, and to obtain the overall surface areas of the cold and hot sides, as well as the overall coefficient of heat transfer. Using this information and modern design method, the exchanger was optimally designed. This method is based on a thermo-hydraulic model (RDA) that shows the relationship of pressure drop, heat transfer coefficient, and exchanger volume. Given that the maximum pressure drop is considered as the objective of design in RDA, a smaller surface area is obtained. This reduction in the surface of exchanger has a major role in decreasing investment costs. In addition, a simple method is provided for the selection of the secondary surface of the exchanger based on VPI. Surfaces that produce smaller volumes will generate larger VPI. Therefore, surfaces with the largest VPI within the operational Reynolds ratio can be selected by the designer and then the given exchanger can be designed through coding in MATLAB. Finally, the overall heat transfer coefficient and the overall surface of the exchanger are obtained.

Performance Description of Liquefaction Cycle of LNG

The liquefaction of LNG or pre-cooled propane and a mixture of triple refrigerants (C3MR) (Fig.1) includes an initial pre-cooled stage that is done in the presence of almost pure propane (421). In this stage, the NG feed initially enters into two heat exchangers (E-104 and E-105), and leaves them at the temperature of 272°K. Then, it enters into the separator (V-103). After the separation of liquids and gases, its gaseous flow (V) enters into the heat exchanger (LNG-101). In the cooling stage, the gas temperature significantly decreases and reaches to 137.7°K with a mixture of methane, ethane, and propane (11) refrigerants. Natural gas is liquefied after leaving this exchanger. The parameters required before the initiation of simulation are presented in Table 1.

Table 1. Specifications of Feed in Iran LNG Project

	Vaper Phase	Liquid Phase
Vaper phase fraction	1	-
Temperature	318 °K	-
Pressure	9150 kPa	-
Mass density	69/03 kg/m³	-

Feed	Mass Velocity(kg/s)
CO_2	0/02084
N_2	14/31458
CH_4	145/42181
C_2H_6	14/77288
C_3H_8	0/79788
i C_4H_{10}	0/01407
n C_4H_{10}	0/00701
Total	175/34929

Table 2. Physical and process specifications of hot fluid (V) to PFHE in liquefaction cycle

	Vaper Phase	Liquid Phase
Vaper phase fraction	1	-
Temperature	282.28 °K	-
Pressure	7809 kPa	-
Mass density	82/58 kg/m³	-

Feed	Mass Velocity(kg/s)
CO_2	0/02084
N_2	14/31458
CH_4	145/42181
C_2H_6	14/77288
C_3H_8	0/79788
i C_4H_{10}	0/01407
n C_4H_{10}	0/00701
Total	175/34929

Table 3. Physical and process specifications of cold fluid (11) to PFHE in liquefaction cycle

	Vaper Phase	Liquid Phase
Vaper phase fraction	0.2764	0.7236
Temperature	132.7 °K	-
Pressure	214 kPa	-
Mass density	19 kg/m³	-

Feed	Mass Velocity(kg/s)
CH_4	1225/47
C_2H_6	626/44
C_3H_8	1837/36
Total	3689/33

In this simulation, the selection of the equation of state is very important. The Peng-Robinson equation of state is commonly used for the mixture of light hydrocarbons such as natural gas. Since our simulation feed is a natural gas that consists of methane, ethane, propane and butane (three light hydrocarbons), the Peng-Robinson Equation was selected.

In general, this liquefaction process consists of three heat exchangers, one compact heat exchanger and six compressors. This process also accounts for the majority of investment costs. In this article, the compact heat exchanger (LNG-101) and reduction of investment costs through its usage are considered.

This simulation was done using Aspen HYSYS to extract the process and physical specifications of the compact heat exchanger inflows (11 and V flows) and to calculate the surface of LNG-101. This information was used for designing the exchanger.

Modified Volume Performance Index (VPI)

To start the design, the selection of surfaces, which provide the smallest units of weight and volume with high performance, is the first choice in design. If one of the limitations was violated in the first design, frequent stages in the selection of different surfaces with lower functional features should be taken. A number of indices have been provided by researchers to select the best secondary surface (fin) within the operational Reynolds ratio. This index is indeed the modified version of the index $\left(VPI = \frac{(\frac{St^3}{f})^{\frac{1}{2}}}{d_h}\right)$ previously provided by Polley et al. [8]. In the original index, the effects of inferences from other thermal resistances on the heat transfer coefficient inside the tubes were not well taken into consideration; whereas, the effect of fouling coefficient and other thermal resistances were considered in the modified version of the index [6]: $VPI = (\frac{1}{1+R_{opp}h_t})^{1/5} \cdot \sqrt{\frac{St^3}{f}}$

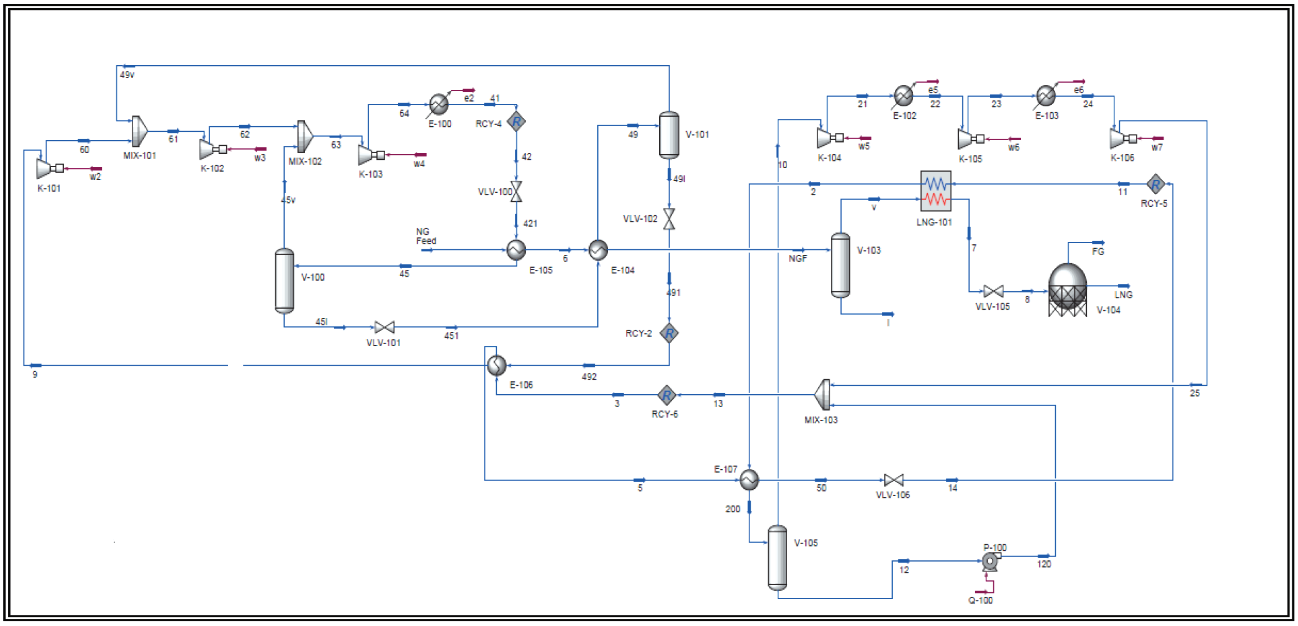

Fig 1. Liquefaction cycle or pre-cooled propane, and a mixture of refrigerants per unit of natural gas

The Stanton number and fraction factor are functions of Reynolds; therefore, the VPI is also a function of Reynolds number. It has been shown that the higher rate of this index implies greater compaction of the exchanger and selection of a surface with higher performance

[6]. We can draw the VPI diagram can be plotted for different surfaces based on Reynolds to be capable of selecting the best surface quickly. The VPIs of different surfaces of various types are drawn in fig.2 and 3. These surfaces have been based on the give data by Keyes and London [7]. In this demonstration the surfaces with the best performance are shown for every range of Reynolds. In this way, the smallest volume of exchanger is obtained by making sure of having access to complete pressure drop for the flow and the selection of the most effective surface. For the first case, a great need for the development of a design method that allows maximum use of pressure drop is felt; whereas, the second case allows the selection of surfaces with the best performance for a certain Reynolds number.

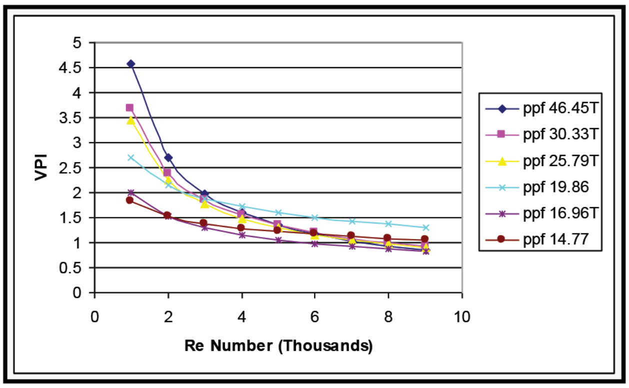

Fig 2. VPI based on Reynolds number for Plain Fin [5]

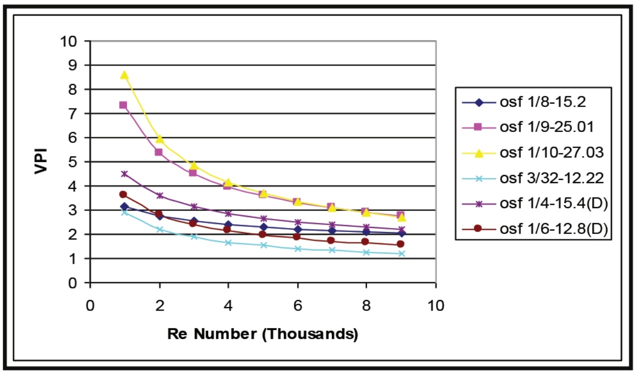

Fig 3. VPI based on Reynolds number for Offset Strip Fin [5]

Thermo-Hydraulic Model

The design model, which is based on the maximum use of pressure drop and development of thermo-hydraulic model, is used to design the compact plate-fin exchangers [12]. This model of pressure drop correlates a certain flow to the overall volume of the exchanger and the heat transfer coefficient of the same side of it. The performance of heat transfer for different compact surfaces is correlated by Reynolds number as follows:

$$j = a\,Re^{-b} \tag{1}$$

Where, J is known as Colburn factor and defined as follows:

$$j = St.\,Pr^{\frac{2}{3}} \tag{2}$$

The Prandtl and Stanton numbers are defined as follows:

$$Pr = \frac{C_p \mu}{k} \tag{3}$$

$$St = \frac{hA_C}{mC_P} \tag{4}$$

For the plane heat exchanger, the Reynolds number is defined as a function of hydraulic diameter of the surface:

$$Re = \frac{md_h}{\mu A_C} \tag{5}$$

Equations 4 and 5 are combined and its solution for the heat transfer coefficient (h) is as follows:

$$h = K_h \left(\frac{1}{A_C}\right)^{1-b} \tag{6}$$

where A_c is the free surface and K_h is defined as follows:

$$K_h = \frac{a m^{1-b} \mu^{b} C_P}{d_h^{b}\, Pr^{\frac{2}{3}}} \tag{7}$$

The Equation 6 presents the heat transfer coefficient as a function of physical properties, mass flow rate, and free surface of flow.

It can be used to obtain a relationship that correlates pressure drop to free surface of the flow and physical properties. The term that expresses pressure drop in the heat exchanger is as follows:

$$\Delta P = \frac{f}{2\rho} \cdot \frac{A}{A_C} \cdot \frac{m^2}{A_{C_2}} \tag{8}$$

For the majority of secondary surfaces, the values of friction factor (f) can be correlated with the Reynolds number ranging from 500 to 10,000 as follows:

$$f = x\,Re^{-y} \tag{9}$$

where x and y are constant values. The overall heat transfer surface area of one side of the exchange is defined as a function of overall volume of the exchanger through following equation:

$$\hat{A} = \alpha V_T \tag{10}$$

where α is called "geometric parameter", which is indeed the ratio of the overall surface area of one side of the exchanger to overall volume of it. The combination of Equations 5 and 8-10 is obtained as follows:

$$\Delta P = K_P V_T \left(\frac{1}{A_C}\right)^{3-y} \tag{11}$$

Where,

$$K_P = \frac{x m^{2-y} \mu^{y} \alpha}{2\rho d_h^{y}} \tag{12}$$

By taking AC from Equation 2 and placing it in Equation 7, we have:

$$\Delta P = \frac{K_P}{K_h^{z}} V_T h^{z} \tag{13}$$

Where,

$$z = \frac{3-y}{1-b} \tag{14}$$

Equation 13 represents a thermo-hydraulic model that correlates flow pressure drop to the overall volume of the exchanger and heat transfer coefficient of the flow.

Equations Required in Volume Design

The main equation for heat transfer design is as follows:

$$Q = UAF\Delta T_{LM} \tag{15}$$

After combination of it with the overall heat transfer coefficient, following equation is obtained:

$$A_1 = \frac{Q}{F\Delta T_{LM}}\left[\frac{1}{\eta_1}\left(\frac{1}{h_1} + R_1\right) + \frac{1}{\eta_2}\left(\frac{A_1}{A_2} \cdot \frac{1}{h_2} + R_2\right)\right] \tag{16}$$

where A_2 and A_1 represent the overall surface of heat transfer at the Sides 1 and 2, receptively.

The overall volume of the exchanger and overall heat transfer surface of one side of the exchanger are correlated according to Equation 10. After the placement of A in the Sides 1 and 2 of the Equation 10, we have:

$$V_T = \frac{Q}{F\Delta T_{LM}}\left[\frac{1}{\eta_1 \alpha_1}\left(\frac{1}{h_1} + R_1\right) + \frac{1}{\eta_2 \alpha_2}\left(\frac{1}{h_2} + R_2\right)\right] \tag{17}$$

Equation 17 presents the overall volume of the exchanger as a function of heat duty, surface geometry, and heat transfer coefficients.

The heat efficiency of the surface of both sides of the exchanger is defined as:

$$\eta = 1 + f_s \left\{ \frac{\tanh\left[\left(\frac{2h}{k\tau}\right)^{\frac{1}{2}} \left(\frac{\delta}{2}\right)\right]}{\left(\frac{2h}{k\tau}\right)^{\frac{1}{2}} \left(\frac{\delta}{2}\right)} - 1 \right\} \tag{18}$$

The cross flow plate-fin heat exchangers function in a way that the flow pathways are independent in the cross flow; in addition, the maximum pressure drop becomes possible in both sides of the exchanger. The solution of the pressure drop equation for each flow determines the overall length and width of the exchanger along the flow. For a certain volume and front surface, access to the desired number of pathways or unique suitable dimensions is possible via manipulation of the width and height of the exchanger.

Rapid Design Algorithm (RDA)

Rapid design algorithm of heat exchangers is indeed the simplest method of direct design of heat exchangers. Given that the calculation of heat transfer coefficient and pressure drop are important elements of design, an allowable pressure drop is usually defined in exchanger design. If the pressure drop exceeds the normal level, a vibration will be made by the system; whereas, if this pressure drop is far lower than the threshold allowed by the design, heat transfer coefficient will be reduced. Thus, we have:

$$\Delta P_{\text{Hot Stream}} < \Delta P_{\text{Hot stream / allowable}} \tag{19}$$

$$\Delta P_{\text{Cold Stream}} < \Delta P_{\text{Cold stream / allowable}} \tag{20}$$

Taking the allowable pressure drop into consideration as the system drop seems dangerous and illogical. This technique is a procedure that generates an acceptable result in the optimization of the exchanger.

The general design technique is as follows: For hot stream (HS), we have:

$$h_{HS} = f(V_{HS}) \tag{21}$$

$$\Delta P_{HS} = f'(V_{HS}) \tag{22}$$

We remove V_{HS} from the equations:

$$\Delta P_{HS} = F(h_{HS}) \tag{23}$$

The same is true for the cold stream:

$$h_{CS} = f(V_{CS}) \tag{24}$$

$$\Delta P_{CS} = f'(h_{CS}) \tag{25}$$

After the removal V_{CS}, we have:

$$\Delta P_{CS} = F'(h_{CS}) \tag{26}$$

According to the owing equations:

$$\begin{cases} Q = UAF_T\Delta T_{LM} \\ \dfrac{1}{U} = \dfrac{1}{h_{HS}} + \dfrac{1}{h_{CS}} + R_D \\ \Delta P_{HS} = F(h_{HS}) \\ \Delta P_{CS} = F'(h_{CS}) \end{cases} \tag{27}$$

A nonlinear A-based equation is obtained, which is solved through Newton-Raphson method.

Using the briefly described thermo-hydraulic model and RDA method, an algorithm was developed for designing plate-fin heat exchanger in LNG liquefaction cycle.

Rapid Design Algorithm of Plate Heat Exchanger

According to RDA and the thermo-hydraulic model, the design algorithm of cross flow plate-fin heat exchanger is as follows:

1. The physical properties of the cold and hot fluids include thermal capacity of thermal conductivity (C_p)(K), density (P), viscosity (μ), and fouling resistance (R) are considered as inputs.
2. Operational parameters including input and output temperature of cold and hot fluids, mass flow rate (m°), and pressure drop (ΔP) are considered as inputs.
3. Type of the secondary surface for the cold and hot fluids, thickness of separating plate (a), and thermal conductivity of the separating plate (K) are considered as inputs.
4. Values of α_C, α_h for the cold and hot flows are computed via $\alpha_h = \frac{b_h\beta_h}{b_h + b_c + 2a}$ and $\alpha_C = \frac{b_c\beta_c}{b_h + b_C + 2a}$; where, b and β are surface-related geometric parameters that are determined with the selection of the surface type; and α is the ratio of overall transfer surface of one side of the exchanger to the overall volume of the exchanger, extracted from Keyes-London's book.
5. K_h and K_p values are computed according to the Formulas 7 and 12 for the hot and cold sides.
6. The h values for the hot and cold sides are

calculated via Formula 13 and based on V_T (overall volume of the exchanger).

7. The η (thermal efficiency of surface) values for the hot and cold sides are computed using Formula 18, based on h which is a function of V_T.

8. Values of $\alpha_C, \alpha_h, \eta_C, \eta_h, h_C, h_H$ are placed in Equation 17. The obtained equation is a nonlinear equation based on V_T, which is solved using Newton-Raphson method. In this way, V_T (i.e. the overall volume of the exchanger) is obtained.

9. $R_C, R_h, h_C, h_h, \eta_C, \eta_h, V_T$, A_C, A_h are produced as the outputs of the computer program.

A Case Study Designing Plate Heat Exchanger

The introduced algorithm was used for the design of compact plate-fin heat exchanger in the liquefaction cycle of Iran LNG project. Simulation-extracted information of inflows was used as input of the algorithm programing. Moreover, results related to the cold and hot sides and the overall heat transfer coefficient obtained from the simulated exchanger were compared with the results obtained from rapid algorithm program. The post-design results of the exchanger were acceptable.

Information related to the operational conditions and physical properties of the problem, and data related to the surfaces used in the exchanger are summarized in Table 4.

In this design, the maximum allowable pressure drop has been considered for the

Table 4. Process information and physical properties for design of the simulated cross flow exchanger

	Cold stream(1)	Hot stream(2)
Process Information		
Mass Flow Rate $(\frac{kg}{s})$	3688.888	175.361
Allowable Pressure Drop (Pa)	15000	140000
Inlet Temp. $(^\circ K)$	132.7	282.277
Outlet Temp. $(^\circ K)$	135.3	137.7
Physical Properties Information		
Density (kg/m^3)	19	85.58
Mass Heat Capacity $(\frac{J}{kg\,^\circ K})$	2120	2924
Thermal Conductivity $(\frac{W}{m\,^\circ K})$	0.134752	0.04066
Viscosity (kg/m.s)	0.0278698	0.001383
Surface Information		
Type of Surface	$ofs\frac{1}{10}-27-03$	$ofs\frac{1}{10}-27-03$
Heat Transfer Coefficient (a)	0.5231	0.5231
Heat Transfer Exponent (b)	0.5042	0.5042
Friction Factor Coefficient (x)	1.5369	1.5369
Friction Factor Exponent (y)	0.4648	0.4648
Thermal Conductivity of the Fin $(\frac{W}{m\,^\circ K})$	220	220
Plate Thickness (m)	0.0003	0.0003

Table 5. Design of the simulated cross flow heat exchange (comparison of results)

	Simulation	Program
Re (Cold=1)	-	2469
Re (Hot=2)	-	102090
Efficiency(Cold=1)	-	0.56
Efficiency(Hot=2)	-	0.53
Total Surface(Cold) (m^2)	3001	575
Total Surface(Hot) (m^2)	1933	575
Heat Transfer Coefficient(Hot) $(\frac{W}{m^2.^\circ K})$	-	3980
Heat Transfer Coefficient(Cold) $(\frac{W}{m^2.^\circ K})$	-	4545
Total Volume (m^3)	-	0.46
Overall Heat Transfer Coefficient* Surface (W/hr-°C)	2.3×10^6	4.84×10^5

cold and hot sides. This maximum pressure drop maximizes the rate and thus increases Reynolds. According to the Nusselt Equation, an increase in Reynolds produces the maximum heat transfer coefficient for the hot and cold surfaces; therefore, the overall coefficient of heat transfer is maximized. According to Q=UAFAT$_{LM}$, despite the maximum rate of overall heat transfer coefficient and fixed temperature difference between the cold and hot sides, the minimum surface area is obtained. Surfaces obtained via programing with secondary surface of $\mathrm{ofs}\frac{1}{10}-27-03$ for both cold and hot sides are 575m^2, which decreased by 2.5 and 3.3 times as compared to the surfaces areas of 3001m^2 and 1933m^2 for the cold and hot sides in the simulation, respectively. This surface area reduction was highly desirable as it significantly decreased investment costs of developing such types of exchangers.

Results Using VPI in Design of Plate Heat Exchanger

Using the modified VPI and drawn diagrams, surfaces with maximum VPI within the range of operational Reynolds were selected. The given exchanger was designed using the computer program, and the overall volume and surface area of it were obtained. A comparison was provided in Tables 5 and 6 between the surface obtained from designing cross flow LNG exchanger using VPI and the surface obtained from the simulated exchanger.

Between different secondary surfaces (Table 5), ppf46.45T for the hot side and $\mathrm{ofs}\frac{1}{10}-27-03$ for the cold side presented the minimum surface areas (422m^2 for the hot side and 556m^2 for the cold side). The overall heat transfer coefficient 785 (W/m^{2}K) increased by 1.8 times. Reduction by 1.1 and 5.1 times was observed in the hot side as compared to the hot side (1933m^2) and the cold side (3001m^2) in the simulated exchanger, respectively. After comparison of the obtained surfaces in Tables 5 and 6, $\mathrm{ofs}\frac{1}{10}-27-03$ was selected for the hot and cold sides. This is because it presented the minimum surface area for the given exchanger. For example, the overall surface area of the heat transfer obtained for $\mathrm{lpf}\frac{1}{2}-6.06$ and $\mathrm{ofs}\frac{1}{10}-27-03$, ppf 46.45T under a certain pressure drop, is presented in figure 4. Also, figures 5 and 6 compared the heat transfer levels obtained for the cold and hot sides of these three different secondary surfaces with the heat transfer levels of the cold and hot surfaces in the simulated exchanger.

In figures 7 and 8, the overall volume and heat transfer coefficient of the simulated and designed exchangers were compared. According to the figures, the overall volume of the designed exchanger decreased by 5.4 times; in addition, the overall heat transfer coefficient increased by 2 times.

Conclusion

The design procedure developed in this article shows that this method is well capable

Table 6. Comparison of Novel Designed Heat Exchanger with the Simulated Exchanger

No. of HS	No. of CS	Re (H)	Re (C)	V(m^3)	Hot Surface	Cold Surface	Heat transfer .surfac coefficiente	Heat transfer coefficient
					1933	3001	2.3×10^6	425
ppf46.45T	$ofs\frac{1}{10}-27.03$	107160	2629	0.61	1620	587	1.2723×10^6	785
ppf11.11	$lpf\frac{3}{8}-6.06$	366140	9524	1.041	605	450	6.2656×10^5	1034
ppf46.45T	ppf30.33T	109260	3512	0.66	1553	830	8.8344×10^5	568
ppf46.45T	$lpf\frac{1}{4}-11.11$	79040	7363	1.014	3138	356	3.825×10^6	1218
$ofs\frac{1}{10}-27.03$	$lpf\frac{3}{8}-11.1$	80255	7793	08.6	1058	379	1.1948×10^6	1128
ppf11.11	$lpf\frac{3}{8}-11.1$	421360	6729	0.81	422	556	3.8310×10^5	906
ppf46.45T	$lpf\frac{3}{8}-6.06$	65549	10473	1.5	4711	346	5.8723×10^6	1246
ppf46.45T	$wpf11.5-\frac{3}{8}w$	69567	6521	1.35	4140	461	3.9966×10^6	965
ppf30.33T	$wpf11.5-\frac{3}{8}w$	148160	6353	1.32	2366	493	2.1649×10^6	914
ppf25.79T	ppf30.33T	203480	3272	08.2	981	969	4.8956×10^5	498

Table 7. Comparison of Novel Designed Heat Exchanger with Simulated Heat Exchanger (Equal Surfaces)

No. of HS	No. of CS	Re (H)	Re (C)	V(m^3)	Hot Surface	Cold Surface	Heat transfer .surfac coefficiente	Heat transfer coefficient
-	-	-	-	6.04	1933	3001	23.10^6	425
$ofs\frac{1}{10}-27-03$	$ofs\frac{1}{10}-27-03$	102090	2649	0.46	575	575	484150	842
ppf46.45T	ppf46.45T	123040	1732	0.54	1201	1201	484003	403
ppf30.33T	ppf30.33T	224480	3301	0.7	951	951	483108	508
ppf25.79T	ppf25.79T	198290	3144	0.75	1039	1039	483135	465
$lpf\frac{1}{2}-6.06$	$lpf\frac{1}{2}-6.06$	294120	9933	1.12	473	473	482933	1021
$wpf11.5-\frac{3}{8}w$	$wpf11.5-\frac{3}{8}w$	201450	5662	1.16	664	664	483392	728

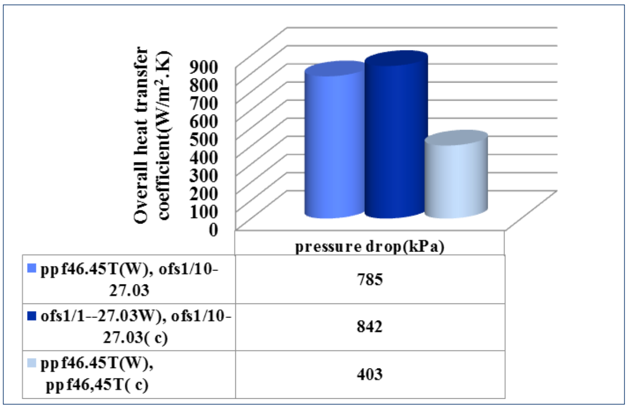

Fig 4. Comparison of the overall surface of heat transfer with different secondary surfaces under a certain pressure drop

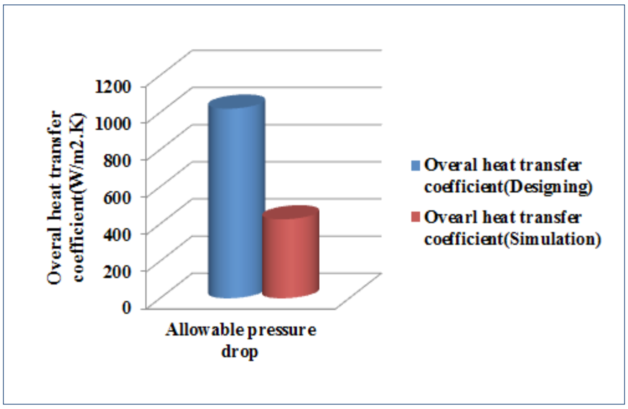

Fig 7. Comparison of volume of the simulated exchanger with that of the designed exchanger

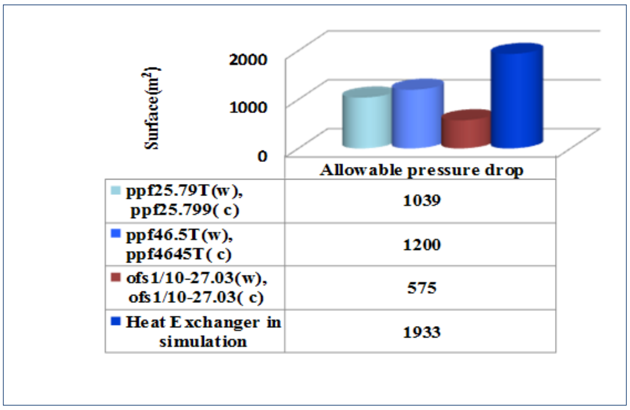

Fig 5. Comparison of heat transfer surface of the cold side of the simulated exchanger with the surface obtained after the optimal design

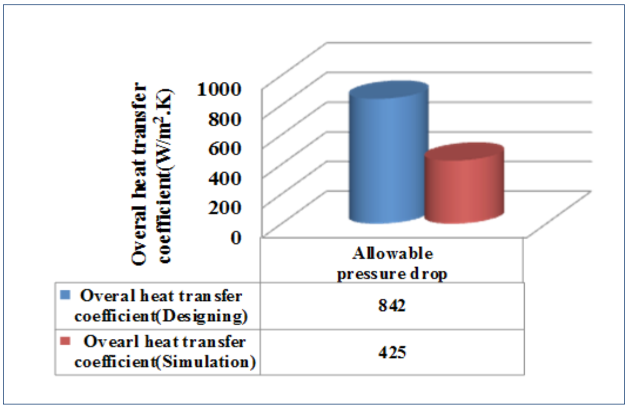

Fig 8. Comparison of the overall heat transfer coefficient of the simulated exchangerwith that of the designed exchanger

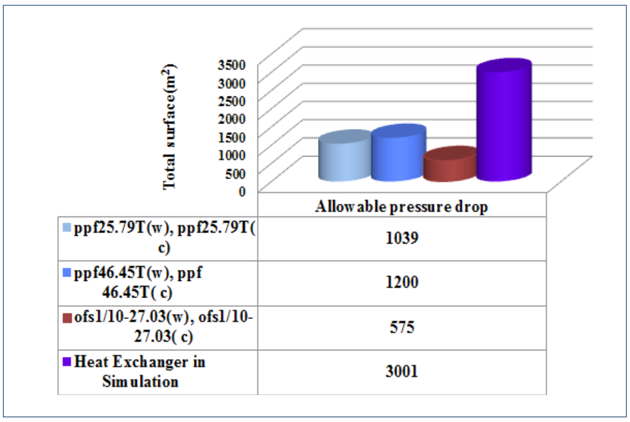

Fig 6. Comparison of the heat transfer surface of the hot side of the simulated exchanger with the surface obtained after the optimal design

of developing an optimal design of plate fin heat exchanger in terms of the surface area, volume, and overall heat transfer coefficient in Iran LNG project. In design of this exchanger, predetermination of the specification of surface involved in thermal process in both sides of the exchanger is needed. The rational objective of the design is an exchanger with the minimum surface area within the block dimensions. This goal is achieved by ensuring about having full access to allowable pressure drop, and selecting a heat transfer surface with high efficiency. A correlation can be made between simultaneous design and selection of surface using VPI curves.

The provided design algorithm has been used for the case of Iran LNG project. Results obtained from designing the plate heat exchanger with the computer program show a good consistency with the results obtained from the simulated exchanger in the LNG cycle. Moreover, those fins were selected that provided surfaces of several times smaller than those of the simulated exchanger, using performance index. The cold and hot surfaces of the designed exchanger were 5.2 and 3.3 times smaller than those of the simulated exchanger; in addition, its overall heat transfer coefficient increased by 2 times. Given this significant reduction in surface area, a reduction will be achieved in investment costs in LNG process.

Symbols

A: Overall heat transfer surface area (m^2)
A_c: Free surface of flow (m^2)
a: Correlation coefficient of heat transfer
b: Correlation power of heat transfer
C_p: Thermal capacity of fluid $(\frac{J}{kg.^{\circ}K})$
d_h: Hydraulic diameter (m)
f: Friction coefficient of fluid in tube, friction factor
f_s: Secondary surface to overall heat transfer surface ratio
h: Heat transfer coefficient $(\frac{W}{m^2.^{\circ}K})$
J: Colburn heat factor
K_h: Dimensional constant
K_p: Dimensional constant
K: Thermal conductivity $(\frac{W}{m^2.^{\circ}K})$
M: Mass flow rate (kg/s)
Pr: Prandtl number (-)
ΔP: Pressure drop (kPa)
Q: Thermal load (J)
R: Fouling resistance $(\frac{m^2.^{\circ}K}{W})$
St: Stanton number (-)
ΔT_{LM}: Logarithmic mean temperature difference $(^{\circ}K)$
ΔT: The temperature difference between two ends of the flow $(^{\circ}K)$
Re: Reynolds number
U: Overall heat transfer coefficient $(\frac{W}{m^2.^{\circ}K})$
V: Exchanger volume (m^3)
VPI: Volume performance index
x: Correlation coefficient of friction factor
y: Correlation power of friction factor

Greek letters

α: Overall heat transfer surface of one side of the exchanger to overall volume of the exchanger $(\frac{m^2}{m^3})$

β: Overall heat transfer surface of one side of the exchanger to overall volume of the exchanger $(\frac{m^2}{m^3})$

δ: Distance between surfaces (m)
η: Fin efficiency
μ: Viscosity $(\frac{kg}{ms})$
ρ: Density $(\frac{kg}{m^3})$
σ: Ratio of free surface of flow to surface of flow in one side of the exchanger
σ: Fin thickness (m)

Footnotes

T: Total
C: Cold
H: Hot

References

1. Baek, S., Hwang, L., Jeong, C., "Novel design of LNG (liquefied natural gas liquefaction process", Energy convection and management, 52(8-9), 2807-2814, (2011).
2. Cesar, P., Julio,"A review on heat exchanger thermal hydraulic models for cryogenic applications", Cryogenics, 51(7), 366-379, (2011).
3. Fossas, Joan, S., «Modeling of multi stream LNG heat exchanger«, A thesis for degree of M.Sc., Norwegian University of science and technology, 8-10, (2011).
4. Fredheim AO, H.R, "Possibilities of cost reductions in base-load". EUROGAS96. Proceedings from the European applied research conference on natural gas, 1, (1996).
5. Jafari Nasr, M. R., Kalantari, A., "Choose surface &design of plate fin heat exchanger by RDA method" RIPI, 10, (2005).
6. Jafari Nasr, M. R., Zoghi, A. T., Khakdaman, H. R., "Performance evaluation of heat transfer enhancement in heat exchanger design«, RIPI, 43, (2002).
7. Kays, W. M., London, A. L., "Compact heat exchanger", McGraw Hill, (1984).
8. Nunez, M., Polley, G. T., Reyes, E., Munoz, A., »Surface selection and design of plate-fin heat exchangers«, Applied thermal engineering, 19- 917- 931, (1999).
9. Mohammadi, E., "Design and optimization of compact heat exchanger in LNG production unit" A Dissertation submitted to Chem. Eng. Dept. of Azad University, 87-89, (2013).
10. Saffari, H., »Modeling and optimization of C3MR LNG plant efficiency by change ff mixed refrigerants' components«, LNG research laboratory of mechanical engineering, Iran university of Science and Technology, 2-4, (2011).
11. Sanggyu, L., Ngyyen V. D., "Design and optimization of natural gas liquefaction and recovery process for offshore floating liquefied natural gas plant«, School of chemical engineering, Yeungnam University, Gyeongsan, 712-749, South Korea, (2012).
12. Walker, V., Wilkie, D., "The wider application of roughened surface as developed for advanced gas cooled reactors, Symp. on high pressure gas as a heat transport medium, IMechE, London, 26, (1967).
13. Wang, L., Sunden, M., "Optimal design of plate heat exchangers with and without pressure drop specifications", Applied thermal engineering, 23-295-311, (2003).
14. Wang, M., Zhang, J., "Optimal design and operation C3MR refrigeration system for natural gas liquefaction", Computer & Chem. Eng., 36, 48-95, (2012).

یک طراحی نوین برای مبدل های حرارتی صفحه ای در چرخه مایع سازی تولید LNG

- الهام محمدی
- *محمدرضا جعفری نصر

گروه مهندسی شیمی، دانشکده فنی و مهندسی، واحد علوم و تحقیقات، دانشگاه آزاد اسلامی، تهران، ایران

*(ایمیل نویسنده مسئول: drnasr50@yahoo.ca)

چکیـــــده

تولید ال ان جی(LNG)، فرآیند فشرده و پیچیده ای است که بیش از نیمی از هزینه های آن مربوط به بخش مایع سازی است. در سالهای اخیر مهندسان طراح، تلاش های زیادی جهت بهینه سازی این فرآیند نموده اندکه هدف عمده آنها تلاش در افزایش بازده، بالابردن ظرفیت تولید و به حداقل رساندن هزینه ها بوده است. مهمترین تجهیزات فرآیندی در بخش مایع سازی، مبدل های حرارتی فشرده از نوع صفحه ای یا حلزونی هستند. این مقاله ضمن تشریح شبیه سازی سیکل مایع سازی پروژه ایران ال ان جی با مبرد مخلوط سه گانه به ارائه روشی نوین جهت طراحی مبدل حرارتی صفحه ای به کار رفته در این سیکل پرداخته است. همچنین روشی ساده، برای انتخاب بهترین سطح ثانویه بر اساس توسعه مفهوم شاخص عملکرد حجم(VPI) معرفی گردیده است. مبدل طراحی شده، کمترین سطح و حجم را بدست داده و کاهش سطح انتقال حرارت در آن نقش عمده ای بر کاهش هزینه های سرمایه گذاری فرآیند تولید ال ان جی داشته است. سیکل مایع سازی ایران ال ان جی به بعنوان یک مورد صنعتی در این مطالعه بطورکامل در نظر گرفته شده است. شبیه سازی، سطوح حرارتی سمت سرد و گرم مبدل صفحه ای به کار رفته درفرایند مورد نظر را به ترتیب $۳۰۰۱\ m^2$ و $۱۹۳۳\ m^2$ و ضریب کلی انتقال حرارت $425\ W/m^2 K^\circ$ نشان میدهد. در حالیکه با انجام طراحی این مبدل و توسعه الگوریتم طراحی سریع(RDA)، کاهش قابل ملاحظه ای در سطوح سرد و گرم لازم بترتیب به میزان ۵/۲ و ۳/۳ برابر و افزایش ضریب کلی انتقال حرارت ۲ برابر حاصل شده است.

کلمات کلیدی: گاز طبیعی مایع شده، شاخص عملکرد حجم، الگوریتم طراحی سریع