Characteristics and performance of heat and mass flowrate in LPG recovery unit

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Abstract

Liquefied Petroleum Gas (LPG) is a common fuel in several sectors. It is a mixture of variable content, but it is primarily comprised of propane and butane. Like natural gas, LPG combustion can be considered lower emissions combustion. The present work aimed to study the heat and mass transfer of LPG recovery unit located in Alexandria Petroleum Company (APC). The study focused on the characteristics and performance of one of the main counter flow shell and tube heat exchangers in the LPG plant by studying the effect of changing the feed mass flow rate and the effect of changing inlet pressure on the effectiveness, Number of transfer unit, performance and pressure drop in shell side and tube side. The distillation column and the auxiliary heat exchangers were simulated by HYSYS program. By using NTU method, the simulation results showed that increasing the load of the heat exchanger will reduce the effectiveness and number of transfer unit of the heat exchanger. On the other hand, it will increase the performance of the heat exchanger and the pressure drop in shell side and tube side.

KEYWORDS: LPG, Shell and tube heat exchanger, HYSYS; effectiveness, NTU method.

Nomenclature

$\dot{m}$: Mass flow rate of fluid, *kg*/*s*
C_p:specific heat of fluid, *J*/ (*kg*·*K*)
T: temperature, ℃
i–inlet *o* – outlet
$T_{H,in}$: inlet temperature of the hot fluid °C
$T_{H,out}$: outlet temperature of the hot fluid °C
$T_{C,in}$: inlet temperature of the cold fluid °C
$T_{c,out}$: outlet temperature of the cold fluid°C
U: overall heat transfer coefficient, $W/(m^2 \cdot K)$
A:area of surface across heat transfer occurs, m^2
C_{min} :min (C_C, C_H)
Smaller of the two heat capacity rates (cold and hot) (*W*/*K*)
C_{max} :max (C_C, C_H)
– Larger of the two heat capacity rates (cold and hot) (*W*/*K*) ε :effectiveness

1. Introduction

Liquefied Petroleum Gas (LPG), also known as propane, is a non-renewable gaseous fossil fuel. LPG, a by-product of natural gas processing and oil refining, includes various mixtures of hydrocarbons. The term liquefied petroleum gas (LPG) describes hydrocarbon mixtures in which the main components are propane, butane, Iso-butane, propene, and butenes (butylenes). Most commonly this term is applied to mixtures of propane and butane. These components and mixtures thereof are gaseous at normal temperature and pressure but can be liquefied by cooling, compression, or a combination of both processes.

LPG is a low-carbon-emitting hydrocarbon fuel available in rural areas. Being a mixture of propane and butane, LPG emits less carbon per joule than pure butane but more carbon per joule than pure propane. As a low carbon and low-polluting fossil fuel, LPG is recognized by governments around the world for its contribution in improving the indoor air quality and reduced greenhouse emissions. LPG is widely available and can be used in various applications. It is also used alongside renewable technologies in decentralized electricity generation to help in reducing carbon emissions on a local level.

It can be produced in LPG unit by introducing light hydrocarbons (C1–C5) into different streams obtained from atmospheric crude distillation, plat former, hydro cracker, and other catalytic processing units. It is also produced in gas fractionation units charged by associated gas from oil reservoirs or gas from gas reservoirs. However, any economically LPG stream must be routed to the LPG recovery unit.

The composition of LPG varies significantly, and is influenced by several factors. Globally, approximately 60% of LPG comes from processing so-called conventional natural gas (Mustovic , 2011). The remainder is mainly produced during the refining of crude oil.

LPG primarily contains paraffins when sourced from natural gas. These include propane and the butanes, as well as smaller amounts of ethane and longer chain paraffins. LPG produced during oil refining can also contain olefins, particularly propylene (propene) and the butylenes (butenes), which originate from various processes. These olefins are sometimes separated and used as feedstocks for other processes (Gary et al., 2007). However, due to the similar volatilities of propane , propylene, butanes and butylenes, the separation of these constituents are relatively costly and often not undertaken (Sadeghbeigi, 2012; Lamia et al., 2007; Rege and Yang, 2002). As a result, refinery-sourced LPG can have significant olefin content, with up to 30% by volume reasonably common.

The production of gasoline also influences LPG's butane content. The economic value of Iso-butane is usually higher as an alkylation feedstock (Gary et al., 2007). This process produces a very desirable gasoline blending stock. Therefore, generally only surplus amounts Iso-butane are directed into LPG production streams. Similarly, N-butane can be blended directly into gasoline as a low-cost octane improver, or to regulate the vapor pressure of the gasoline pool. The availability of N-butane for LPG can therefore vary due to the seasonally and geographically varying Reid Vapor Pressure (RVP) requirements for gasoline.

Given these different sources and processing methods, the composition of LPG is highly variable. Whilst LPG content varies significantly across the globe, it is primarily a mixture of four species: propane, propylene, iso-butane and n-butane. Whilst the ethane content can be up to 10 (% vol) in some markets, it is a less significant component overall. The butylene content in LPG also appears to be less significant, but is not widely reported.

Variations in the composition of LPG have a significant effect on its knock characteristics (Falkiner, 2003). However, LPG fuel standards usually do not define compositional limits. Instead, restrictions on a number of the fuel properties are generally specified. European automotive-grade LPG, for example, must comply with the requirements of the EN 589 standard (European Committee for Standardization, 2008). This standard specifies a minimum Motor octane number (MON) of 89.0, and a minimum fuel vapor pressure that varies between 275 kPa and 950 kPa throughout the year, to account for seasonal variations. The latter requirement is set at a national level, and therefore may still vary from region to region during a given period.

Hosseini et al. (2007) obtained Experimentally the heat transfer coefficient and pressure drop on the shell side of a shell-and-tube heat exchanger for three different types of copper tubes (smooth, corrugated and with micro-fins). Also, experimental data has been compared with theoretical data available. Additionally Xie et al. (2007) Carried out an experimental system for investigation on performance of shell-and-tube heat exchangers, and limited experimental data is obtained. Further more José et al. 2009 Presented an approach based on genetic algorithms for optimum design of shell and tube heat exchanger and for optimization major geometric parameters.

In the study of Hong et al.(2009) he found that effectiveness-NTU approach is adopted to predict the thermodynamic behavior of the heat exchanger for the Joule-Thomson refrigerator. The study showed the influences of mass flow rate and the supply pressure on the effectiveness of heat exchanger. Also Patel et al. (2010) Explores the use of a no traditional optimization technique; called particles warm optimization (PSO), for design optimization of shell-and-tube heat exchangers from economic view point. Murugesan et al.(2012) studied the Effect of Mass Flow Rate on the Enhanced Heat Transfer Characteristics in A Corrugated Plate Type Heat Exchanger. Also Asadi et al (2013) studied Effects of mass flow rate in terms of pressure drop and heat transfer characteristics in a plate-fin heat exchanger with Wavy fin. Finally Kulkarni et al.(2014) analyzed the heat transfer coefficient and pressure drops in shell and tube heat exchangers for different mass flow rates and inlet and outlet temperatures, using Kern, Bell and Bell Del aware methods.

Accordingly, Increase LPG production process efficiency is an important issue to be considered. This can be done by improving the efficiency of the heat exchangers and studying the factors affecting the performance of the heat exchangers. Therefore, the present work aimed to study the heat and mass transfer of an LPG recovery unit located in Alexandria Petroleum Company (APC) in Egypt. The study focused on the characteristics and performance of one of the main counter flow shell and tube heat exchangers in the LPG plant by studying the effect of changing the feed mass flow rate (unit load) and the effect of changing inlet pressure on the effectiveness, Number of transfer unit, performance and pressure drop in shell side and tube side.

2 Process description and modeling

The objective of the process shown in Fig. 1 is to recover LPG, with a fixed mass fraction of methane, from the feed crude oil whose characteristics are shown in Table 1. As shown in the simulation process flow diagram in Fig 1. simulated in Aspen Hysys software, the operation in the process is done by preheating the crude oil in a preheat heat exchanger (E315) from 47°C to 102 °C and then is directed to the fired heater (E316) to be heated to 143°C before flowing to main fractionation tower (C306). The stream produced from the bottom of the tower (STAB. NAPHTA), is directed to heat exchanger E315 to take the advantage of its high temperature (288.8°C) in order to heat the feed crude oil reaching a temperature of 188 °C and then further cooled by the water cooler W319 to 47°C before being sent as a second product from the process (Naphta pro.). On the other hand, the liquefied petroleum gas exits from the top of the tower C306 with temperature 134°C and passes by Air cooler for being cooled to 70°C. After that the flow is further cooled by water coolers through two branches towards vessel. The first branch includes two water coolers (W318A and W318B), and the second branch includes four heat exchangers (W322A, W322B, W322C and W322D). In the vessel (V-101), the stream after being cooled in water coolers to 38°C is separated into vapor (To dry vessel) and liquid (Stream 15). Stream 15 is then branched to reflux stream (Back to tower) and LPG stream which is the main product. Pressure drop in heat exchangers and distillation column is neglected for simplicity. The characteristics of the products are illustrated in Table 1.

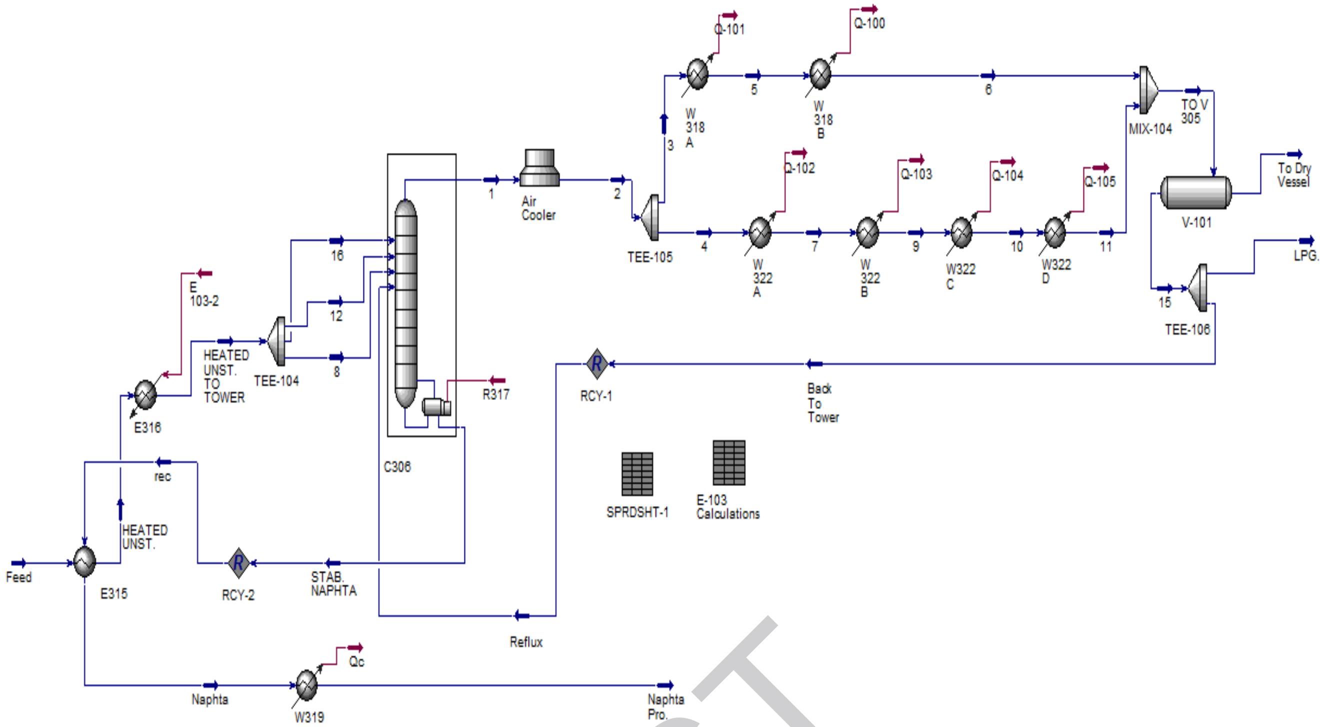

Figure 1 Principal process flow diagram for the studied LPG process.

Table 1 Feed crude oil, LPGD product and Naphtha product streams.

	Feed crude oil	LPG product	Naphtha product
Main properties			
Pressure [Kpa]	1600	1373	1389
Temperature [°C]	47	38	47
Molar flow [Ton/h]	25	13.9	11.1
Composition [mass fraction]			
Methane	0.0010	0.0066	0
Ethane	0.0027	0.0094	0
Propane	0.0684	0.1632	0
i-butane	0.0948	0.1716	0
n-butane	0.3335	0.6036	0
i-Pentane	0	0	0
n-Pentane	0	0	0
n-Hexane	0	0	0
C5+	0.4996	0.0457	1
H_2O	0	0	0

2.2 Case study

The Hysys simulation was applied to predict the thermodynamic behavior of the heat exchanger E315. The study shows the effect of changing mass flow rate under different feed inlet pressure keeping the design dimensions constant on the heat exchanger performance, shell side pressure drop, tube side pressure drop, Effectiveness and NTU. The heat exchanger specifications are illustrated in Table 2. In this study, the inlet heat exchanger mass flow rate is changed from 10000 Kg/h to 25,000 Kg/h at different inlet pressure ranged from 14to22 bar.

Table 2 Specifications of the studied heat exchanger E315.

Specifications	
Tag number	E315
Flow rate [Ton/h]	25000
Heat duty [kW]	940
Area [m^2]	40
Tube side	
No. of tubes	88
Length [mm]	6000
Inlet temperature [°C]	47
Outlet temperature [°C]	102
Pressure [kPa]	1600
Pressure loss	(0) neglected
Shell side	
Inner diameter (mm)	484
Inlet temperature [°C]	289
Outlet temperature [°C]	188
Pressure [kPa]	1389
Pressure loss	(0) neglected

The (NTU) method is developed to simplify a number of heat exchanger design problems. The heat exchanger effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate if there were infinite surface area. The heat exchanger effectiveness depends upon whether the hot fluid or cold fluid is the minimum fluid, that has the smaller capacity coefficient C = m˚ C_p. If the cold fluid is the minimum fluid then the effectiveness is defined as:

$$\varepsilon = \frac{C_{max}(T_{H,in}-T_{H,out})}{C_{min}(T_{H,in}-T_{C,in})} \quad (1)$$

Otherwise, if the hot fluid is the minimum fluid, then the effectiveness is defined as:

$$\varepsilon = \frac{C_{max}(T_{C,out} - T_{C,in})}{C_{min}(T_{H,in} - T_{C,in})} \quad (2)$$

So the heat transfer rate as:

$$Q = \varepsilon C_{min}\left(T_{H,in} - T_{C,in}\right) \quad (3)$$

Also the value of NTU is defined as:

$$\boldsymbol{NTU} = \frac{\mathbf{UA}}{\boldsymbol{C_{min}}} \quad (4)$$

It is a simple matter to solve a heat exchanger problem when

$$\varepsilon = f\,(NTU, C_r) \quad (5)$$

Where

$$C_r = \frac{C_{min}}{C_{max}} \quad (6)$$

Numerous expressions have been obtained which relate the heat exchanger effectiveness to the number of transfer units. The hand-out summarizes a number of these solutions and the special cases which may be derived from them. For convenience the $\varepsilon - \mathrm{NTU}$ relationships are given for a simple double pipe heat exchanger for parallel flow which is the direction considered in the studied heat exchanger E315. The equations for parallel flow can be seen as follows:

$$\varepsilon = \frac{1 - \exp[-\mathrm{NTU}(1 + C_r)]}{1 + C_r} \quad (7)$$

Or

$$\mathrm{NTU} = \frac{-\ln[1 - \varepsilon(1 + C_r)]}{1 + C_r} \quad (8)$$

3. Results and discussion

The simulation results of the case study performed on the heat exchanger E315 is discussed in this section. Five parameters are studied for the heat exchanger. They are effectiveness, Number of transfer unit, heat exchanger performance and pressure drop in shell side and tube side with respect to changing the feed mass flowrate.

3.1 Effectiveness

Fig. 2 shows the variation of the effectiveness of the heat exchanger (E315) with different mass flow rates at different inlet pressures. It is noted that the effectiveness of the heat exchanger decreases as the mass flow rate increases. According to Eq. 1, it was be noted that the effectiveness depends on the mass flow rate, inlet and outlet temperature and capacity coefficient for both tube and shell sides. Since the feed stream is a cold stream, which has the minimum capacity coefficient compared to the hot stream with the maximum capacity coefficient, increasing the flow rate leads to increasing in the C_{min} which is in the denominator resulting in lower effectiveness.

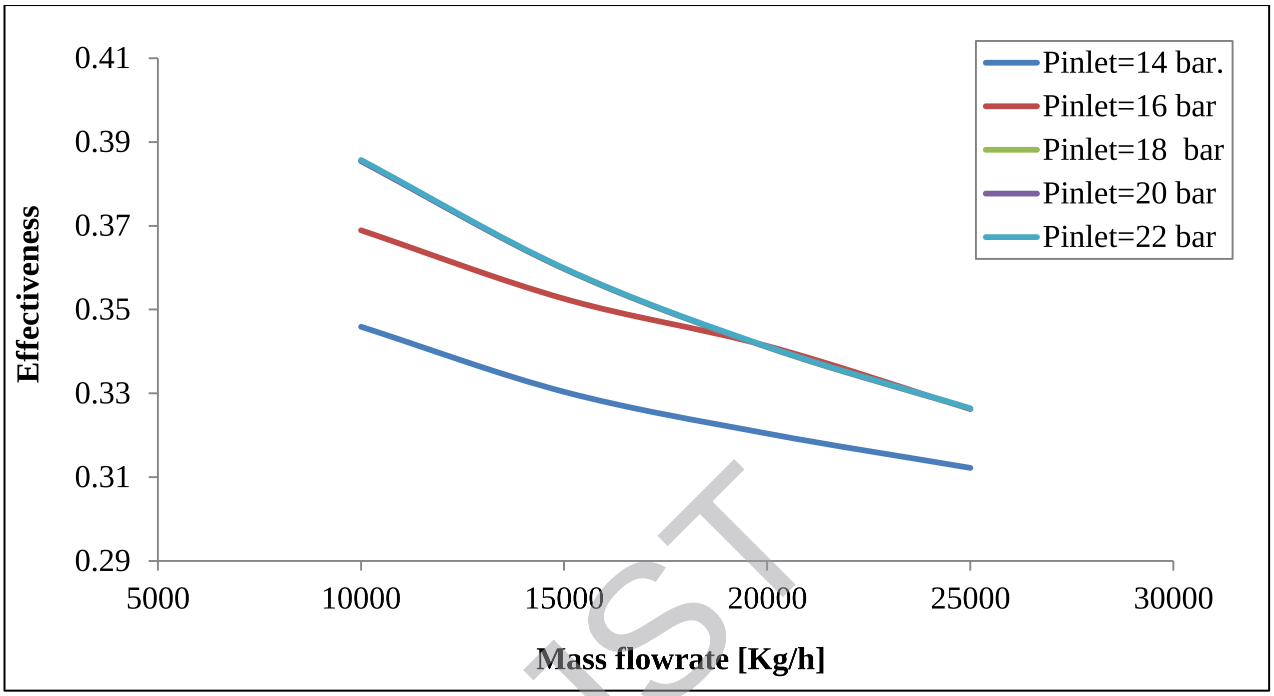

Figure 2 The effect of changing feed mass flowrate on the effectiveness of heat exchanger E315.

Moreover, it can be observed from Fig. 2 that the change in pressure of the feed flowrate (cold stream) effects on the capacity coefficient of the stream $C_{min.}$ So, decreasing the pressure increases the capacity coefficient of the cold stream which leads to a reduction in the effectiveness. From the curve it is noted that the effectiveness at 14 bar is significantly lower than the effectiveness when the pressure is above 20 bar. Also it can be observed that the effectiveness is almost the same when the inlet pressure is above 18 bar. It is found that changing in pressure has a small effect on Effectiveness as decreasing the pressure from 22 bar to 14 bar (8 bar change) decreases the effectiveness by 0.03 and this effectiveness slightly decreases as the flow rate increases.

3.2 Number of heat transfer (NTU)

Fig. 3 shows the variation of the NTU of the heat exchanger (E315) with different mass flow rates at different inlet pressures. It is noted that the NTU of the heat exchanger decreases as the mass flow rate increases.

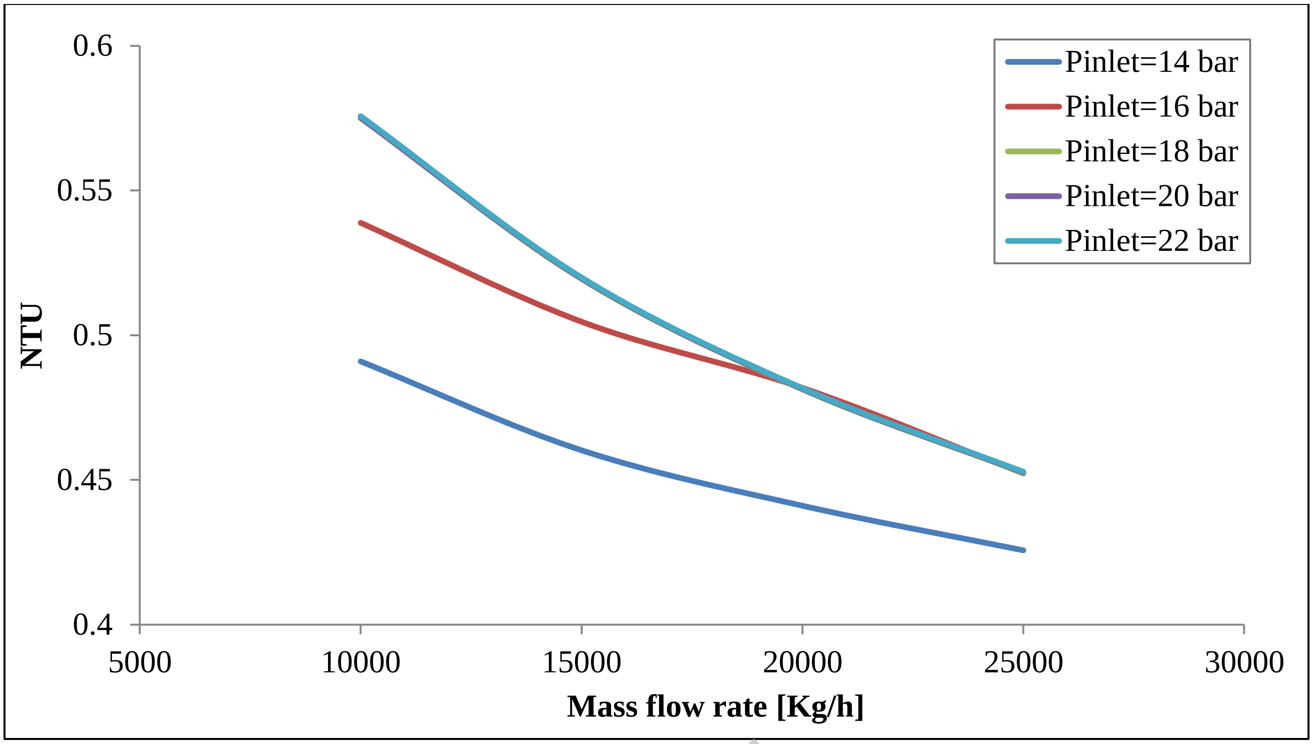

Figure 3 The effect of changing feed mass flowrate on the NTU of heat exchanger E315.

According to Eq. 8, it is notified that the NTU depends on the mass flow rate, inlet and outlet temperature and capacity coefficient for both tube and shell sides. Since there is a proportional relationship between the effectiveness and NTU, and increasing the mass flow rate from the previous study raises the effectiveness which leads to an increase in the NTU as well. Fig. 3 has been plotted according to Eq. 8 as $C_r < 1$ ($C_{min}/C_{max} < 1$). The equation shows that NTU depends on the effectiveness and the ratio between the minimum capacity coefficient and maximum capacity coefficient ($C_r = C_{min}/C_{max}$). Also it can be observed that the change in pressure of the feed flow rate (cold stream) effects on the capacity coefficient of the stream C_{min} so that, decreasing the pressure increases the capacity coefficient of the cold stream which leads to a reduction in the NTU.

3.3 Performance of heat exchanger (UA)

Fig. 4 shows the variation of the performance of the heat exchanger with respect to feed mass flow rate at different inlet pressures. The Performance of the heat exchanger (UA) increases as the mass flow rate the heat exchanger increases. As the mass flow rate increases, it leads to increase the area of the heat transfer of the heat exchanger.

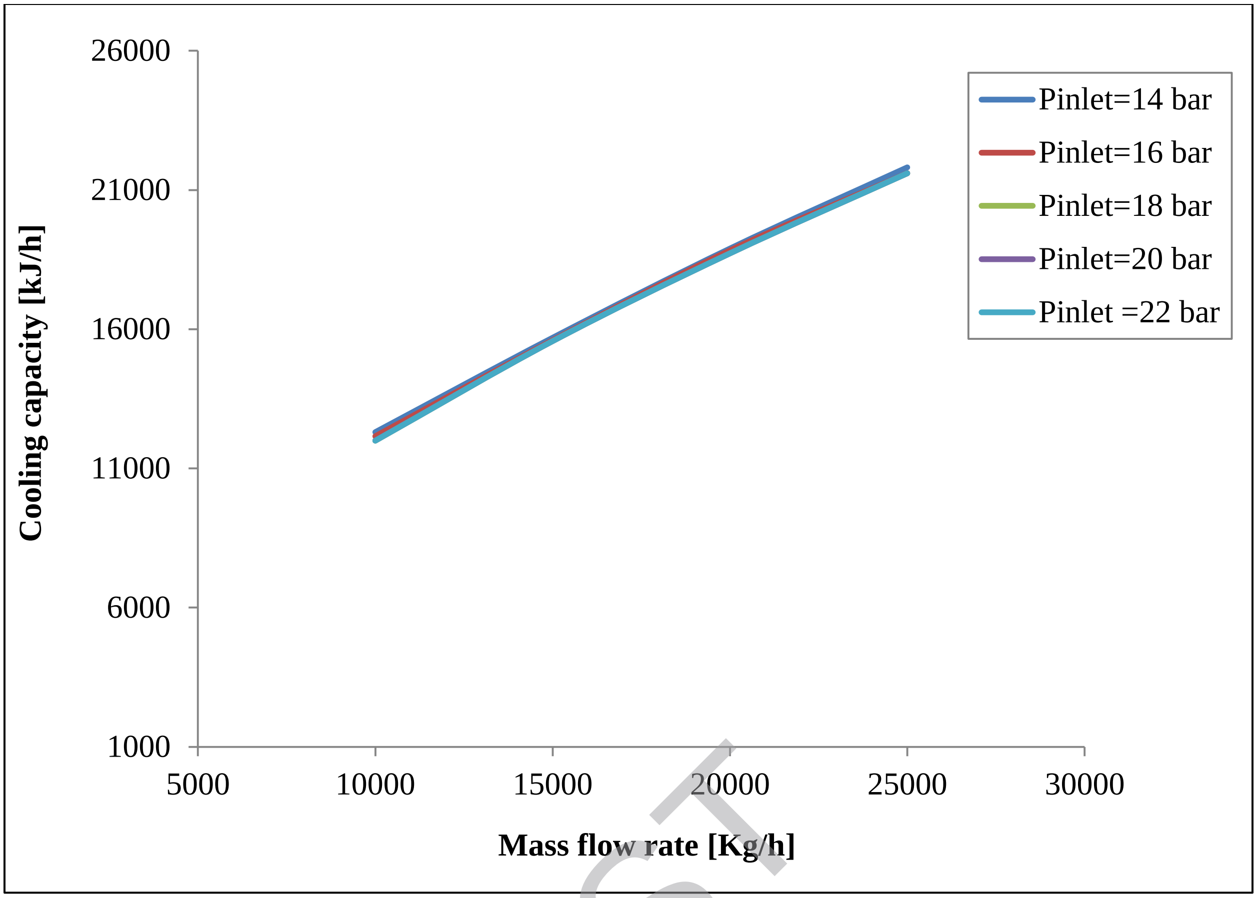

Figure 4 The effect of changing feed mass flowrate on the the performance (UA) of heat exchanger E315.

3.4 Pressure drop in shell side

Fig. 4 shows the variation of pressure drop in shell side of the heat exchanger with changing in the feed mass flow rate at different inlet pressures. It is obvious that the pressure drop increases as the mass flow rate increases.as the mass flow rate increases in the heat exchanger, the friction losses also increases which leads to higher friction losses. It is notified that while operating the study at range of pressure from 14 bar to 22 bar, there is no change in pressure drop detected.

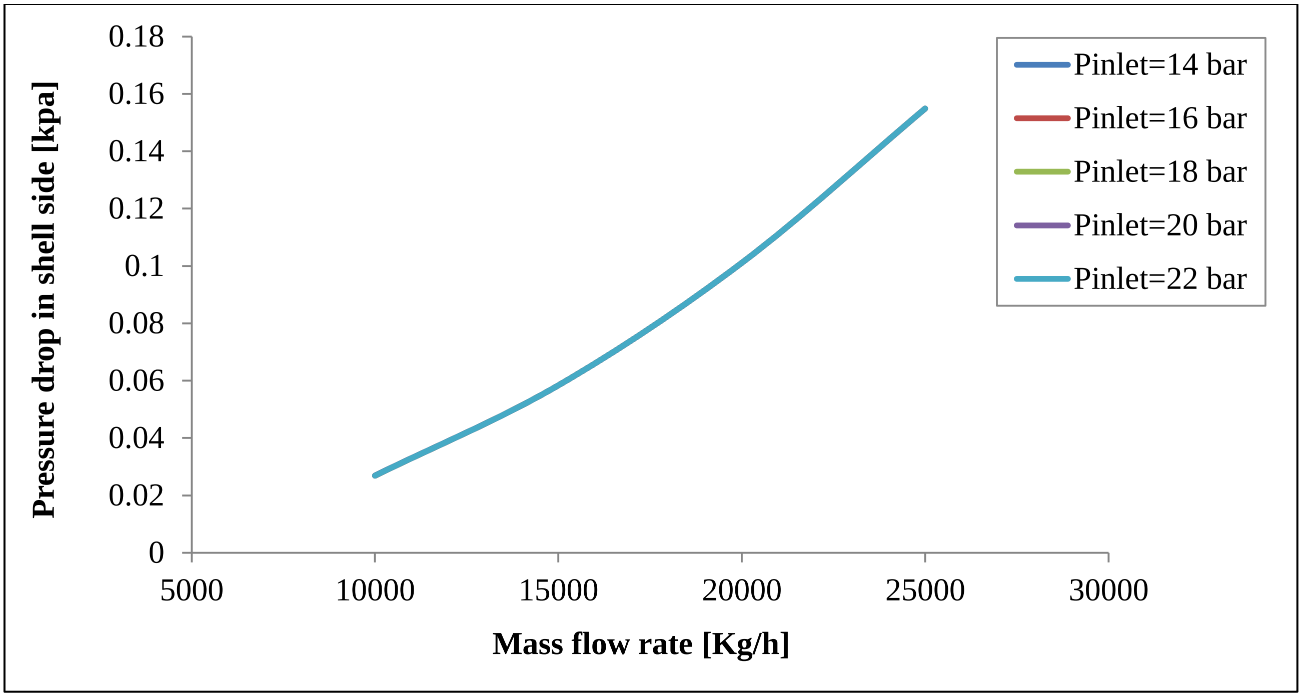

Figure 5 The effect of changing feed mass flowrate on the pressure drop in shell side of heat exchanger E315.

3.5 Pressure drop in tube side

Fig. 6 shows the variation of pressure drop in tube side of the heat exchanger with changing in the feed mass flow rate. It is obvious that the pressure drop increases as the mass flow rate increases.as the mass flow rate increases in the heat exchanger, the friction losses also increases which leads to higher friction losses. It is notified that while operating the study at range of pressure from 14 bar to 22 bar, there is a slight change in comparison with shell side so, changing in inlet pressure effects on pressure drop in tube side.

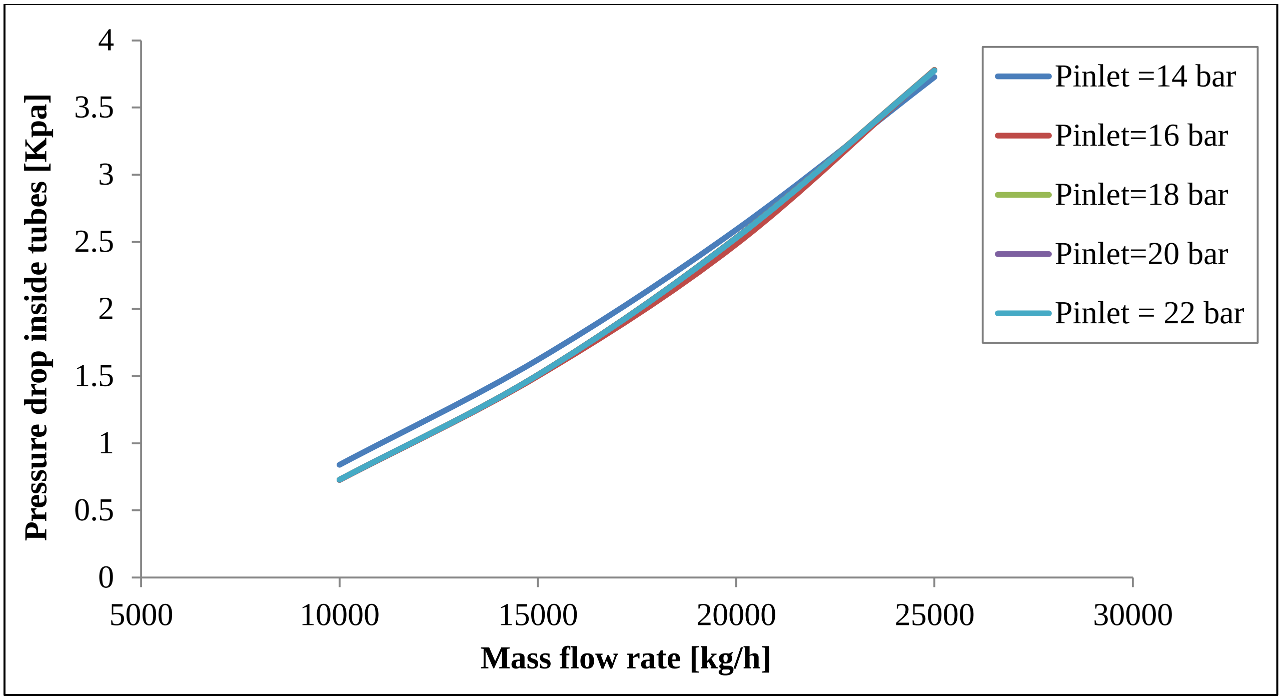

Figure 6 The effect of changing feed mass flowrate on the pressure drop in tube side of heat exchanger E315.

4. CONCLUSION

The purpose of this thesis is to study the effect of changing the feed mass flow rate on atmospheric distillation unit at Alexandria Petroleum Company (A.P.C) with the aid of Aspen Hysys software to simulate the process of LPG Production. In addition, the heat exchanger (E315) has been selected to study its performance and characteristics: effectiveness, number of transfer unit, performance of the heat exchanger and pressure drop in shell side and tube side. The results show that increasing the mass flow rate (unit load) entering the heat exchanger (E315) decreases the effectiveness(ε) and number of transfer unit (NTU) of the heat exchanger while it increases the performance of the heat exchanger (UA),and the pressure loss in both shell side and tube side.

References

1. Horst Baur,ed(1996).Automotive handbook (4th ed.) Stuttgart : Robert Bosch Gmbh. ISBN0837603331.
2. F. Mustovic. Autogas Propulsion Systems for Automobiles. IBC Engineering and Publishing,Sarajevo, Bosnia and Herzegovina, 2011. ISBN 978-9958-9173-2-5.
3. J.H. Gary, G.E. Handwerk, and M.J. Kaiser. Petroleum Refining. CRC Press, Boca Raton, FL,fifth edition, 2007. ISBN 0-8493-7038-8
4. R. Sadeghbeigi. Fluid Catalytic Cracking Handbook. Butterworth-Heinemann, third edition,2012. ISBN 978-0-12-386965-4. Oxford, UK.
5. V.K. Patel, R.V. Rao, "Design optimization of shell and tube heat exchanger using particle swarm optimization technique", Applied Thermal Engineering 30 (2010) 1417-1425.
6. R. Hosseini, A. Hosseini-Ghaffar, M. Soltani, "Experimental determination of shell side heat transfer coefficient and pressure drop for an oil cooler shell and tube heat exchanger with three different tube bundles", Applied Thermal Engineering 27 (2007) 1001–1008.
7. G.N. Xie, Q.W. Wang , M. Zeng, L.Q. Luo, "Heat transfer analysis for shell and tube heat exchanger with experimental data by artificial neural networks approach", Applied Thermal Engineering 27 (2007) 1096–1104.
8. José M. Ponce-Ortega, Medardo Serna-González, Arturo Jiménez-Gutiérrez, "Use of genetic algorithms for the optimal design of shell and tube heat exchanger", Applied Thermal Engineering 29 (2009) 203–209.
9. R.J. Falkiner. Liquefied Petroleum Gas. In: Fuels and Lubricants Handbook, pages 31–59. ASTMInternational, West Conshohocken, PA, 2003. ISBN 9780803120969.
10. Murugesan M.P. and Balasubramanian R., "The Effect of Mass Flow Rate on the Enhanced Heat Transfer Charactristics in A Corrugated Plate Type Heat Exchanger" ,Research Journal of Engineering Sciences ISSN 2278 – 9472 Vol. 1(6), 22-26, December (2012).
11. Yong-Ju Hong, Seong-Je Park, and Young-Don Choi,Crycoolers 15,edited by S.D. Miller and R.G Ross Jr,international cryocooler Conference,inc Boulder,CO, 2009.
12. Shweta Y Kulkarni , Jagadish S B, Manjunath M B, "ANALYSIS COMPARING PERFORMANCE OF A CONVENTIONAL SHELL AND TUBE HEAT EXCHANGER USING KERN, BELL AND BELL DE LAWARE METHOD", IJRET: International Journal of Research in Engineering and Technology eISSN: 2319-1163 | pISSN: 2321-7308