Design of an Industrial Heat Exchanger Unit at the TEG Inlet of a Natural Gas Dehydration Plant

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ORIGINAL RESEARCH

Abstract -Gas processing industries globally grapple with water vapor in natural gas, causing issues like methane hydrate formation, caking, corrosion, and flow problems. Due to natural gas's pivotal role in energy and petrochemical production, the triethylene glycol (TEG) dehydration process proves vital for efficient water removal. Within this process, the heat exchanger at the TEG inlet to the contactor is crucial. It maintains optimal lean TEG absorption temperature and regulates sales gas temperature for pipeline transmission. Employing simulation tools like HYSYS, the analysis determined specific parameters such as an overall heat transfer coefficient of 140.4 kJ/h-m²-C, shell and tube side pressure drops of 34.47 kPa and 68.95 kPa respectively, a 5.027 m² heat exchanger area, shell volumes of 0.1955 m³ and 0.0608 m³ for the shell and tube sides respectively, and a positive heat duty of 1.727 x 10^4 kJ/hr. This positive duty signifies successful gas heating for standard transmission and maintaining lean TEG absorption temperature. Remarkably, the water composition in natural gas reduced from 0.005 mol% to 0.0002 mol% after the process simulation.

Keywords: Water Vapor; Heat transfer; Heat Exchanger Design; Dehydration; Aspen HYSYS.

1.INTRODUCTION

Efficient heat transfer is vital in industrial processes like the TEG (Triethylene Glycol) dehydration process, which relies on optimal heat transfer to maximize productivity and reduce energy costs. The TEG contactor is the heart of the TEG dehydration process plant, where TEG absorbs water vapour from natural gas streams (Wosu*et al*., 2023a). The performance of the TEG contactor directly depends on the temperature and flow rate of TEG entering the contactor. Hence, it is important to design and analyze the performance of the heat exchanger that heats TEG before it enters the contactor. This article presents the design and performance analysis of the heat exchanger at the TEG inlet to the contactor in a TEG dehydration process plant. The article aims to provide insights into the impact of various design parameters and operating conditions on heat transfer and the overall performance of the TEG dehydration process. Natural gas is a clean and environmentally benign source of energy for households and industries (Zhang, 2009; Wosu*et al*, 2023b) and a key feedstock for petrochemical production. Nigeria substantial gas reserves have significantly contributed to the country's growth and development. However, natural gas contains contaminants like water, which can cause problems during processing, storage and transmission.

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To address this, Nigeria has adopted dehydration technology, with triethylene glycol (TEG) being the preferred method (NCDB, 2004; Wosu*et al*, 2024). TEG can reduce the water content of natural gas to meet pipeline transmission transmission standards, making it a crucial process for the country's energy industry (Foss, 2004).

A critical component of the TEG dehydration plant is the heat exchanger, located at the TEG inlet to the contactor. For efficient and ideal dehydration, it is crucial to keep the temperature range of the lean TEG below 80^{0}C (Kidnay& William, 2006) and the temperature of the sales gas within 20-35^{0}C (Christensen, 2009). Therefore, the design and simulation of a TEG dehydration plant utilizing Aspen HYSYS as the design tool, as well as the delevoment of heat exchanger performance models based on the fundamental principle of material and energy balance, will be the main focus of this study.

Previous studies have highlighted the importance of removing water contaminants during TEG dehydration process to meet pipeline transmission standards. According to Mohammed *et al*.,(2014) water is one of the pollutants that must be eliminated during the TEG dehydration process to bring raw natural gas up to the standards needed for pipeline transmission. Natural gas processing is a sophisticated industrial operation. The heat exchanger, a crucial component of the TEG dehydration plant, must be sized or built for successful dehydration to function.A heat exchanger can be set up in a natural gas processing plant as a heat transfer device that exchanges heat between two or more fluids throughout a process, according to Bahman, (2017). Depending on the process conditions, a heat exchanger can be used to achieve both heating and cooling.

According to Nivargiet *al*., (2005), removing contaminants from natural gas entails removing water, oil and condensate, separating natural gas liquids, and removing sulfur and carbon dioxide to fulfil pipeline specifications. Triethylene glycol (TEG), rather than ethylene glycol (EG), diethylene glycol (DEG), and tetraethylene glycol (TREG), was the dehydration agent of choice for natural gas because it was the easiest to regenerate to a concentration of 98–99.99% in an atmospheric stripper and had a low vapourization temperature, high boiling point, and decomposition temperature, as well as low capital and operating costs. They designed the TEG dehydration plant using the advanced process simulation software HYSYS.

Natural gas is a fossil fuel that is created from the dead, decomposing remains of plants and animals that are buried beneath the earth's crust under conditions of extreme heat, pressure, and the absence of air, according to Undiandeye*et al*., (2015). They claim that natural gas, which is used for heating, cooking, producing electricity, and as fuel for vehicles, is the third most widely used energy source in the world (Zimmerman & Zimmerman, 1995). It contains some impurities, such as water, which must be sufficiently removed to meet its specifications for pipeline transmission using TEG as an absorbent in the natural gas dehydration plant. Their study employed Shell Gbaran as a case study to compare various natural gas dehydration processes.

Heat exchangers are widely employed in sectors (Omoyi*et al*, 2024; Ojong*et al*, 2023) and plays a vital role in the processing of natural gas, according to Arturo *et al*., (2011) definition, they enable the transfer of heat between two fluids that are at different temperatures. The design and optimization of heat exchangers are crucial because they increase their competitiveness and enable process energy savings. Their study concentrated on the design or sizing of several exchanger types to ascertain the correlation between heat transfer and energy loss for a turbulent flow. Most literatures in the past and recent past have focussed on the application various technologies for effective natural gas dehydration as well as the design of the contactor and regeneration column of the TEG dehydration plant with no consideration of the design specification of the heat exchanger unit located at the TEG inlet to the contactor.

The design of heat exchangers just like reactors and other process equipment basically involves the application of the conservation principles of mass and energy for the equipment sizing such as determination of the heat duty as in the case of heat exchanger design (Wosu 2024a; Wordu&Wosu, 2019; Oba *et al*., 2024; Wosu 2024b)

2.MATERIALS AND METHODS

2.1 Materials

The materials needed in this research are the feed materials such as the characterized natural gas, TEG, data obtained from plant, HYSYS simulation as well as the calculated or derived data.

2.2 Methods

The research methodology is both quantitative and qualitative or analytical. The procedures involved in the research are;

i. Presentation of the characterized natural gas composition and HYSYS simulation operating condition

ii. Development of the TEG natural gas dehydration plant from HYSYS simulation

iii. Presentation of the various units and streams of the dehydration plant

iv. Development of the heat exchanger design models from the conservation principle of mass and energy balance.

2.2.1 Natural Gas Composition and HYSYS Simulation Operating Condition

The characterized natural gas composition and TEG dehydration process simulation data are presented in table 1

Table 1: Natural Gas Properties (Wosu*et al*, 2023 ;Wosu&Ezeh, 2024)

Components	Composition	Molar Mass (g/mol)
C_1	0.8939	16.00
C_2	0.0310	30.00
C_3	0.0148	44.10
i-C_4	0.0059	58.12
n-C_4	0.0030	58.12
n-C_5	0.0005	72.15
i-C_5	0.0010	72.15
H_2O	0.0050	18.00
N_2	0.0010	14.00
H_2S	0.0155	34.10
CO_2	0.0284	44.00
TEG	0.0000	150.154

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Total	1.0000	610.894
Operating Condition		
Pressure(kPa)	6205.2832	
Temperature (^0C)	29.4444	
Flow rate (kg/s)	768.6343	

2.2.2 TEG Natural Gas Dehydration Plant

The process flow diagram of the TEG natural gas dehydration plant obtained from HYSYS simulation is presented in figure 1.

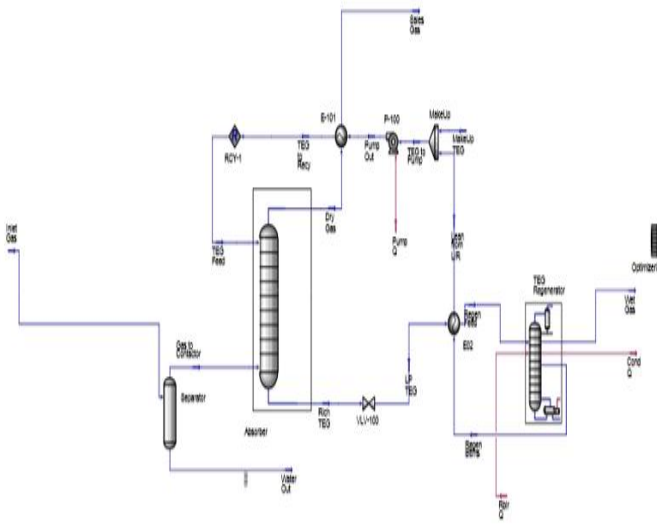

Figure 1: Process Flow Diagram of Natural Gas Dehydration Units

2.2.3 Dehydration Plant Units and Process Streams

The TEG natural gas dehydration plant units and process streams are presented in table 2 and 3.

Table 2: Equipment and the Units of Proposed/Modified Plant Design

Design Equipment	**Designation/Unit**
Separator	U01
Absorber	U02
Heat exchanger 1	U03
Regenerator/Distillation column	U04
Mixer	U05
Pump	U06
Heat exchanger 2	U07

Table 3: Streams Associated with Developed TEG Dehydration Plant

Streams	Name
S_1	Inlet gas
S_2	Water our
S_3	Gas to contactor
S_4	TEG feed
S_5	Dry gas
S_6	Sales gas
S_7	Rich TEG
S_8	Low pressure TEG
S_9	Regeneration feed
S_{10}	Wet gas
S_{11}	Regeneration bottom
S_{12}	Lean TEG L/R
S_{13}	Make-up TEG
S_{14}	TEG to pump
S_{15}	Pump out
S_{16}	TEG to recycle
S_{17}	Lean TEG recycle flow streams

2.2.4 Development of the Heat Exchanger Performance Models

Consider the schematic of the shell and tube heat exchanger unit [E101] of a TEG dehydration plant with input and output streams.

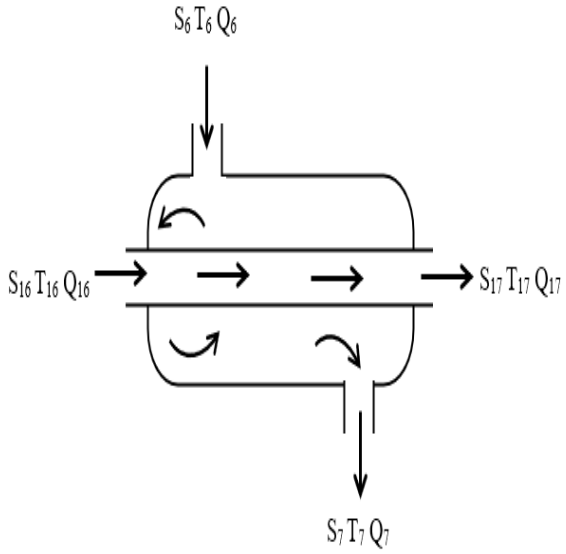

Figure 2: Schematic of Shell and Tube Heat Exchanger

For the shell and tube heat exchanger in figure 2, the design and energy balance models can be developed by apply the conservation principle of mass and energy as follows;

The mass balance models of the shell and tube heat exchanger can be developed as follows:

$$\begin{bmatrix} Rate\ of\ input\ of\ mass/ \\ material\ to\ the\ system \end{bmatrix} = \begin{bmatrix} Rate\ of\ output\ of\ mass/ \\ material\ from\ the\ system \end{bmatrix} \quad (1)$$

$$S_6 + S_{16} = S_7 + S_{17} \quad (2)$$

whereS_6 = Dry gas flow stream (kg/s)

S_7 = Sales gas stream (kg/s)

S_{16} = Lean TEG inflow stream (kg/s)

S_{17} = Lean TEG to recycle flow stream (kg/s)

Energy Balance Model of Heat Exchanger Unit

The energy balance of the shell and tube heat exchanger can be developed as follows:

$$\Delta Q_2 = Q_{17} - Q_{16} \quad (3)$$

$$Q_2 = S_{16} C_{P_{TEG}} (T_{17} - T_{16}) \quad (4)$$

$$Q_7 = \Delta Q_1 + Q_6 \quad (5)$$

$$\Delta Q_1 = S_6 C_{P_{Dry\ Gas}} (T_7 - T_6) \quad (8)$$

Heat Exchanger Design Models

The quantity of heat transfer in a heat exchanger is given by (Sinnott&Towler, 2009) as;

$$Q = UA\Delta T_m \quad (9)$$

Overall Coefficient based on the Outside Area of the Tube (U_0)

This is given as the reciprocal of the overall resistance to heat transfer, which is the sum of several individual resistances. It is mathematically given by (Sinnott&Towler, 2009) as;

$$\frac{1}{U_0} = \frac{1}{h_0} + \frac{1}{h_{od}} + \frac{d_0 ln\left(\frac{d_0}{d_i}\right)}{2K_w} + \frac{d_0}{d_i}\left(\frac{1}{h_{id}}\right) + \frac{d_0}{d_i}\left(\frac{1}{h_i}\right) \quad (10)$$

For the shell and tube design; the following design models are vital for the exchanger to be successfully designed.

$$\Delta T_m = F_t \Delta T_m \quad (11)$$

Temperatures of the fluid in the shell and tube as well as the quantity of shell and tube passes affect the correlation factor. The correlation is often a function of two dimensionless temperature ratios.

$$F_t = F(R, S) \quad (12)$$

Where,

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{T_{h_1} - T_{h_2}}{T_{C_2} - T_{C_1}} \quad (13)$$

$$S = \frac{t_2 - t_1}{T_1 - T_2} = \frac{T_{C_2} - T_{C_1}}{T_{h_1} - T_{h_2}} \quad (14)$$

where, R is given as Shell-side fluid flow rate times the fluid mean, specific heat; divided by the tube-side specific heat and S is a measure of the temperature efficiency of the exchanger

$$f_t = \frac{\sqrt{(R^2+1)\ln\left[\frac{(1-S)}{1-RS}\right]}}{(R+1)\ln\left[\frac{2-S\left[R+1-\sqrt{R^2+1}\right]}{2-S\left[R+1-\sqrt{R^2+1}\right]}\right]} \quad (15)$$

Alternatively, it can be obtained from a correlation of S and R for counter-current flow as given by (Sinnott&Towler, 2009) thus;

$$\Delta T_{lm} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left[\frac{(T_1 - t_2)}{(T_2 - t_1)}\right]} \quad (16)$$

For a co-current flow, it is given as;

$$\Delta T_{lm} = \frac{(T_1 - t_1) - (T_2 - t_2)}{\ln\left[\frac{(T_1 - t_1)}{(T_2 - t_2)}\right]} \quad (17)$$

Tube-Sheet Layout (Tube-Count)

The Bundle Diameter (D_b) depends on both the number of tubes (N_t) and Number of tube passes:

$$N_{t,t} = K_1 \left(\frac{D_{b,t}}{d_0}\right)^{n_1} \quad (18)$$

$$D_{b,t} = d_0 \left(\frac{N_t}{K_1}\right)^{\frac{1}{n_1}} \quad (19)$$

where,

K_1andn_1 are constants and depend on the triangular or square pitch with the corresponding number of passes

$$P_t = 1.25\, d_o \quad (20)$$

Tube-Side Heat-Transfer Coefficient Determination

The heat-transfer coefficient can also be related as follows:

$$h_i = F(N_u, P_r, R_e, \mu) \quad (21)$$

$$N_u = \frac{h_i d_e}{k_F} \quad (22)$$

But,

$$d_e = \frac{4A_F}{P_w} = d_i \text{for tubes} \quad (23)$$

Nusselt Number (N_u) for Turbulent Flow

Heatdata for turbulent flow inside conduits of uniform cross-section is given by (Sieder& Tate, 1936) as;

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$$N_u = C{R_e}^a{P_r}^b\left(\frac{\mu}{\mu_w}\right)^C \quad (24)$$

where C Constant value which is 0.021 for gases, 0.023 for non-viscous liquids and 0.027 for viscous liquids (Sinnott&Towler, 2009).

R_e = Renold's number, and is mathematically given as:

$$R_e = \frac{\rho U_t d_c}{\mu} \quad (25)$$

Alternatively,

$$R_e = \frac{G_t d_c}{\mu} \quad (26)$$

$$P_r = \frac{C_p \mu}{K_f} \quad (27)$$

NusseltNumber (N_u) for Laminar Flow

The film heat-transfer coefficient for a small natural convection effect where Renold's number of about 2000, the flow in the pipe will be Laminar and is given as;

$$N_u = 1.86(R_e P_r)^{0.33}\left(\frac{d_e}{L}\right)^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (28)$$

$$N_u = \frac{h_i d_e}{K_f} \quad (29)$$

Heat Transfer Factor (j_h) Determination

$$j_h = S_t{P_r}^{0.67}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \quad (30)$$

J_h = Heat transfer factor for Laminar and turbulent flow which can be obtained alternatively using R_e

It correlates with R_e as follows:

$$\frac{h_i d_i}{K_f} = j_h R_e {P_r}^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (31)$$

And can also be expressed as:

$$j_H = N_u {P_r}^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \quad (32)$$

where,

$$j_H = j_h R_e \quad (33)$$

Alternatively, the inside coefficient for water (h_i) can be obtained according to (Sinnott & Towler, 2009) as;

$$h_i = \frac{4200(1.35+0.02t){U_t}^{0.8}}{{d_i}^{0.2}} \quad (34)$$

Viscosity Correlation Factor Determination

This is normally significant for viscous liquids and it requires an estimate of the wall temperature by first calculating the coefficient without the correction and using the relationship below to estimate the wall temperature.

$$h_i(t_w - t) = U(T - t) \quad (35)$$

Tube-Side Pressure Drop (ΔP) Determination

The two sources of pressure loss on the tube side of a shell and tube exchanger are

i) Friction loss in the tubes
ii) Losses due to the sudden contraction and expansion as well as flow reversal that the fluid experiences in flow through the tube arrangement

The basic equation of pressure drop for isothermal flow in pipes (constant temperature) is:

$$\Delta P = 8j_f\left(\frac{L^1}{d_i}\right)\frac{\varphi {U_t}^2}{2} \quad (36)$$

Since the flow in a heat exchanger is non-isothermal, this can be accounted for by introducing the change in physical properties with temperature as follows:

$$\Delta P = 8j_f\left(\frac{L^1}{d_i}\right)\frac{\rho {U_t}^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-m} \quad (37)$$

Where,

$m = 0.25$ for Laminar flow, $R_e < 2{,}100$ or

0.14 for turbulent flow, $R_e < 2{,}100$

The tube-side pressure drop can be determined by combining a factor recommended by(Sinnott&Towler, 2009)

$$\Delta P_t = N_p\left[8j_f\left(\frac{L^1}{d_i}\right)\left(\frac{\mu}{\mu_w}\right)^{-m} + 2.5\right]\frac{\rho {U_t}^2}{2} \quad (38)$$

Area for Cross-Flow Determination (A_S)

Experimental based work on commercial exchangers with standard tolerance which gives a reasonable satisfactory prediction of the heat transfer coefficient for standard designs using figures and data given by Kern (1950) and Ludwig (2021) shows that area of cross-flow (A_S) for the hypothetical row of tubes at the shell equator is given by:

$$A_S = \frac{(P_t - d_0)D_S l_B}{P_t} \quad (39)$$

Shell-Side Mass Velocity (G_S) Determination

$$G_S = \frac{W_S}{A_S} \quad (40)$$

Shell-Side Linear Velocity (U_S) Determination

$$U_S = \frac{G_S}{\rho} \quad (41)$$

Shell-Side Equivalent Diameter (d_c)

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This is also called Hydraulic Diameter and can be determined for

i) Square pitch arrangement
ii) Equilateral triangular pitch arrangement

$$d_c = \frac{1.27}{d_0}(P_t{}^2 - 0.785\, d_0{}^2) \quad (42)$$

For equilateral triangular pitch arrangement

$$d_c = \frac{1.10}{d_0}(P_t{}^2 - 0.917\, d_0{}^2) \quad (43)$$

Shell-Side Reynold's Number (R_e)

$$R_e = \frac{G_s d_e}{\mu} \quad (44)$$

Alternatively,

$$R_e = \frac{U_s d_e \rho}{\mu} \quad (45)$$

Shell-Side Heat Transfer Coefficient (h_s)

This can be obtained from the relationship

$$N_u = \frac{h_s d_e}{K_f} = j_h R_e P_r{}^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (46)$$

$$h_s = \frac{j_h R_e K_f P_r{}^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14}}{d_e} \quad (47)$$

Shell-side pressure drop (ΔP_s)

$$\Delta P_s = 8jf\left(\frac{D_s}{d_e}\right)\left(\frac{L}{L_b}\right)\frac{\rho U_s^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \quad (48)$$

The term (L/L_b) is the number of times the flow crosses the tube bundle $(N_b + 1)$

3 RESULTS AND DISCUSSION

The results of the heat exchanger unit at the TEG inlet to the contactor for mass, energy, composition and sizing or specification are presented and discussed in Tables 4, 5 6 and 7 below:

Table 4: Material Balance Results for Heat Exchanger 2 in the TEG Plant Process Design

Heat exchanger 2 Streams	Inflow(S_{15}) Pump Out	Inflow (S_6) Dry Gas	Outflow (S_{16}) Sales Gas	Outflow (S_{15}) TEG to Recycle
Molar Flow (Kgmol/S)	0.00101	41.47777	41.47777	0.00101
Mass Flow (Kg/S)	0.14221	765.22666	765.22666	0.00101

Table 4 is the mass balance summary of heat exchanger 2 in the process flow diagram of the TEG dehydration plant design. This heat exchanger unit exhibits two input, and two output characteristics and also validates the principle of conservation of materials where input streams equals output streams by implication, $S_{15} + S_5 = S_{16} + S_6$. This unit is utilized to ensure that the temperature of the dry gas (sales gas) is suitable for pipeline transmission, storage and distribution which prevents the formation of methane hydrate, sludge and other temperature or energy-related problems that may arise during transmission, storage and distribution of sales gas.

Table 5: Energy Balance Result of Heat Exchanger 2 in the TEG Plant Process Design

Heat Exchanger 2 Streams	Inflow (S_{15}) Pump Out	Inflow (S_5) Dry Gas
Temperature (⁰C)	60.1437	29.6829
Pressure (kPa)	6274.2308	6205.2832
Heat Flow (KJ/S)	-7.75×10^2	-3.56×10^6

Table 5, shows the energy balance results of heat exchanger2 in the process flow diagram. This unit exhibitsa two-input and two-output system. The dry gas and sales gas streams help regulate the condition of temperature and pressure to a standard required for pipeline transmission, storage and distribution. From the table, the sales gas temperature is 29.5249⁰C which satisfies the standard temperature range of 20⁰C to 35⁰C for transmission of natural gas (Manning & Thompson, 1991) while the pressure of sales gas is 6170.8094kPa which also satisfies the pressure condition for sales gas transmission. Another advantage of this unit is that it does not require additional heat through the pump. The sufficient heat in the pump out stream is reduced and conserved by the heat exchanger from 60.1437⁰C to 48.8889⁰C to suit the lean TEG inlet to the contactor.

Table 6, shows the component balance results of heat exchanger2 at the TEG inlet to the contactor. The two-input and two-output system whose aim is to place the dry gas properties (temperature and pressure) in a condition suitable for pipeline transmission (sales gas) to prevent flow problems as a result of methane hydrate formation, sludge as well as preventing corrosion during transmission and storage.

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Table 6: Composition Balance of Natural Gas Component in Heat Exchanger 2 (Unit 07)

Composition (Mole Fraction)				
Components	**Inlet Stream (S_{15}) Pump Out**	**Inlet Stream (S_5) Dry Gas**	**Outlet Stream (S_6) Sales Gas**	**Outlet Stream (S_{16}) TEG to Recycle**
N_2	0.0000	0.0010	0.0010	0.0000
CO_2	0.0000	0.0285	0.0285	0.0000
H_2S	0.0000	0.0156	0.0156	0.0000
C_1	0.0000	0.8980	0.8980	0.0000
C_2	0.0000	0.0311	0.0311	0.0000
C_3	0.0000	0.0149	0.0149	0.0000
$i-C_4$	0.0000	0.0059	0.0059	0.0000
$n-C_4$	0.0000	0.0030	0.0030	0.0000
$i-C_5$	0.0000	0.0010	0.0010	0.0000
$n-C_5$	0.0000	0.0005	0.0005	0.0000
TEG	0.9250	0.0000	0.0000	0.9250
H_2O	0.0750	0.0005	0.0005	0.0750

In this unit, there is no observable changes in natural gas composition but changes or variation in material and energy balance occurs.

Table 7: Results of Sizing/Design of Heat Exchanger 2 at the TEG Inlet to the Contactor in the TEG Process Flow Diagram

Plant Unit	Design/Sizing
Heat Exchanger 2	
Overall heat transfer coefficient (kj/h-m^2-C)	140.4
Shell side pressure drop (kPa)	34.47
Tube side pressure drop (kPa)	68.95
Heat exchanger area (m^2)	5.027
Shell volume per shell (m^3)	0.1955
Tube volume per shell (m^3)	1.608×10^{-2}
Shell diameter (mm)	530
Tube pitch (mm)	50
Shell fouling (C-h-m^2/kj)	0.000
Number of tubes per shell	80
Tube layout angle (degree)	30
Baffle cut (% Area)	20
Tube outside diameter (mm)	20
Tube inside diameter (mm)	16
Tube thickness (mm)	2.000
Tube length (m)	1.000
Tube fouling (C-h-m^2/kJ)	0.000
Tube thermal conductivity (w/m-k)	45
Heat duty (kJ/hr)	1.727×10^4

Table 7 shows the design or rating of the heat exchanger equipment/unit at the TEG inlet to the contactor which plays an important role in cooling the temperature of the lean TEG for effective absorption and also regulating the temperature of the sales gas within acceptable limits for pipeline transmission with a heat duty of 1.727×10^4kj/hr.

4 CONCLUSION

The design of the natural gas TEG dehydration plant was performed using the advanced process simulation tool HYSYS and the performance model of the heat exchanger unit at the TEG inlet to the contactor was developed from the first principle of mass and energy balance. The results of the heat exchanger mass balance, energy balance, composition balance and design/sizing of the equipment are summarized in Tables 4 to 7. The discussions of the results were in agreement with the objectives of this article.Furthermore, the design and performance analysis of a heat exchanger at the TEG inlet to the contactor is critical to the success of a TEG dehydration process plant. A well-designed heat exchanger can maintain optimal performance by providing efficient heat transfer into the TEG mixture, improving the separation of water from the hydrocarbons. The performance analysis of the heat exchanger indicated that optimizing design parameters such as heat transfer area, fluid flow rate, and operating conditions like inlet/outlet temperatures and TEG flow rate, can significantly improve heat transfer rates. Such improvements will lead to an overall increase in TEG dehydration process efficiency, reducing energy consumption, and cost of production. Future research can focus on enhancing heat exchanger design

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parameters and examining the effects of different design configurations for optimal plant performance.

Nomenclature

Symbol	Definition	Unit
ΔQ	Heat change	Kw
Q_6	Heat of dry gas	Kw
Q_7	Heat of sales gas	Kw
T_6	Dry gas temperature	K
T_7	Sales gas temperature	K
$C_{P,TEG}$	Specific heat capacity of TEG	KJ/kgK
$C_{p,\ dry\ gas}$	Specific heat capacity of dry gas	KJ/kgK
Q	Heat transfer per unit time	Watt
U	Overall heat transfer coefficient	$W/m^{2}\,^{0}C$
A	Heat transfer area	m^2
ΔTm	Mean temperature difference	^{0}C
U_o	Overall heat transfer coefficient based on outside area of tube	$W/m^{2}\,^{0}C$
h_0	Outside fluid film coefficient	$W/m^{2}\,^{0}C$
h_i	Inside fluid film coefficient	$W/m^{2}\,^{0}C$
h_{id}	Inside dirt coefficient	$W/m^{2}\,^{0}C$
K_w	Thermal conductivity of the tube wall material	$W/m^{2}\,^{0}C$
d_i	Tube inside diameter	M
d_0	Tube outside diameter	M
$\Delta Tlm.$	Log mean temperature difference	^{0}C
F_t	Temperature correction factor	Dimensionless
T_{h1}	Inlet temperature of hot fluid	K
T_{h2}	Outlet temperature of hot fluid	K
t_{c1}	Cold fluid inlet temperature	K
t_{c2}	Cold fluid outlet temperature	K
R	Correction factor	Dimensionless
S	Correction factor	Dimensionless
P_t	Tube pitch	mm
d_e	Equivalent or hydraulic mean diameter	m
K_f	Fluid thermal conductivity	$W/m^{2}\,^{0}C$
A_f	Cross-sectional area for flow	m^2
P_w	Wetted perimeter	m
C	Constant value	Dimensionless
N_u	Nusselt number	Dimensionless
P_r	Prandtl number	Dimensionless
R_e	Renold's number	Dimensionless
$\rho.$	Density of fluid	Kg/m^3
U_t	Fluid velocity	m/s
$\mu.$	Fluid viscosity at the bulk fluid temperature	Ns/m^2
G_t	Mass flow rate per area	Kg/m^2s
$\alpha.$	Index for Renold's number (0.8)	Dimensionless
C_p	Fluid specific heat	$J/Kg\ ^{0}C$
B	Index for Prandtl number (0.33)	Dimensionless
C	Index for viscosity factor (0.14)	Dimensionless
μ_w	Fluid viscosity at the wall	Ns/m^2
L	Length of tubes	M
J_h	Heat transfer factor for larminar and turbulent flow	Dimensionless
T	Tube inside bulk temperature (mean)	K
t_w	Estimated wall temperature (mean)	K
T	Shell side bulk temperature (mean)	K
L^1	Effective pipe length	M

ΔP_t	Tube side pressure drop	N/m^2
N_p	Number of tube-side passes	Dimensionless
D_s	Shell side diameter	M
L_B	Baffle spacing	M
W_s	Fluid flow rate on the shell side	Kg/s
N_b	Number of baffles	Dimensionless

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