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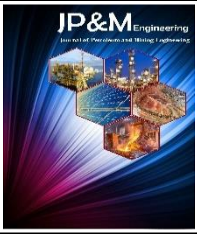

Optimum operating conditions for existing natural gas compression station

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Abstract

There has been continuous evolution of NG treatment processes to meet the ever–increasing demands of electricity generation and various manufacturing sectors. Our focal point is the booster compression station of an existing gas plant which consists of two centrifugal compressors. The objective of the presented work is to debottleneck both compressor trains to increase its capacity and to study the effect of the proposed modifications on the downstream dehydration package that using adsorption by silica gel fixed beds. The presented modifications are adding two slip steams to the gas/gas heat exchangers, the first one to be installed on tube side and the second one on shell side, each slip stream will accommodate 160 MMscf/d. These will give the compression station an important advantage as the capacity of each train will increase from 450 to 610 MMscf/d per train that means we can got a gain of 320 MMscf/d per both trains to be added to the national gas grid at Egypt. The adsorption performance changed after adding both slip streams where the operating parameters of the inlet gas have changed such as temperature and pressure. The simulations for both cases existing and modifying compression station were conducted using Aspen- HYSYS simulation program. The simulation results shows that fuel gas consumption of both compressors driver gas turbines has increased from 6880 to 10000 Kw that equal 45.0% equivalent to 2.25 MMscf/d (1.125 MMscf/d per each) compared to compressed gas capacity increment by 35.5 % that equal 320 MMscf/d. Adsorption calculation for hydration beds shows that pressure drop per each bed increased from 0.66 to 1.19 bar but still within limit (Max 1.25 bar) and adsorbed water for each bed has increased from 1260 to 2850 lb H2O that equal 1590 lb H_2O or 126 % while the bed should hold 6500 lb H_2O.

Keywords

Compressor; inlet temperature; compression efficiency; dehydration

Introduction

Natural gas continues to be favored as an environmentally attractive fuel compared with other hydrocarbon fuels. It is the fuel of choice for the electric power and industrial sectors in many of the world's regions, in part because of its lower carbon intensity compared with coal and oil, which makes it an attractive fuel source in countries. Compression of gases and vapors is an important operation in chemical and petrochemical plants. It is necessary to be able to specify the proper type of equipment by its characteristic performance; the compression step is conveniently identified for the process design engineer by the principal operation of the equipment such as Reciprocating; Centrifugal; Rotary displacement & Axial flow compression may be from below atmospheric as in a vacuum pump or above atmospheric as for the majority of process applications. [1]

In specifying a compressor it is necessary to choose the basic type, the number of stages of compression, and the horsepower required. A first approximation of the number of stages can be made by assuming a maximum compressor ratio per stage of 3.0 to 4.0 and choosing the number of stages. In order to do this the volume of gas, suction and discharge pressure, suction temperature, and gas specific gravity must be known. [2]

Once the required horsepower and number of stages are estimated, a choice of compressor type can be made from the considerations included earlier.

Inlet compression is too much important as it raise the gas arrival pressure from gas wells to the onshore terminal operating pressure.

Mahmoud Abdelwahab in 1981 [3] investigated the effect of fan inlet temperature disturbance on the stability of a turbo fan engine, an experimental investigation was conducted to determine the effects of steady-state and time-dependent fan inlet total temperature disturbances on the stability of turbofan engine. The compressor system response to fan inlet temperature transients with low rates and magnitudes was similar to the steady-state response in that stall began in the high-pressure compressor. At very high rise rates, Farhad Kazemzadeh in 1988 [4] focused on the effect of intercooler size and temperature on performance of two-stage rotary air compressor. The intercooler volume and cooling rate can affect the compressor performance .Larger volume will reduce the amplitude of pressure fluctuation and intercooler temperature will affect the average intercooler pressure and the inlet density of second stage. Venkateswarlu. K et al, in 1988 [5] studied the compressor performance optimization by selecting proper design variants where it is an essential part of any product design exercise.

This paper describes the compressor simulation model and complete parameter design procedure for optimizing the performance of a 1.5 ton single cylinder reciprocating compressor. A computer model incorporating the thermodynamic process, gas flow dynamics, heat transfer- and valve dynamics was developed to simulate the performance for the given design variants. Eleven design variants were considered for optimizing the cooling capacity, power, EER (Energy Efficiency Ratio) and discharge superheat. Cortés et al in 2009 [6] discovered a way to optimize the operating conditions related to compressor performance, based on artificial neural network and the Nelder–Mead simplex optimization method is proposed. It inverts the neural network to find the optimum parameter value under given conditions (artificial neural network inverse, ANNI). Thamir and Ibrahim in 2011 [7] focused on Performance of a gas turbine where it mainly depends on the inlet air temperature. The power output of a gas turbine depends on the flow of mass through it. Brian in 2011 [8] investigated the Accurate measurement of turbine engine compressor inlet total temperature which is paramount for controlling engine speed and pressure ratio. Various methods exist for measuring compressor inlet total temperature on turbojet engines with hydro-mechanical control. One method involves the use of an ejector-diffuser system (eductor) to pull air from the engine inlet in order to measure the incoming total temperature. Gopinath1 and Navaneethakrishnanz in 2013 [9] studied the effect of inlet air temperature on the performance of a gas turbine. The power output of a gas turbine depends on the mass flow rate of air. Now days the atmosphere temperature is hot, so the air is less density, the mass flow rate will reduce and automatically the power output will reduce. A reduce of 1°C temperature of inlet air increases the power output by approximately 0.7 MW. To reduce the inlet air temperature by using the various techniques evaporative coolers, vapor compression chillers, and absorption chillers. Anoop and Onkar in 2014 [10] studied the effect of Compressor Inlet Temperature & Relative Humidity on Gas Turbine Cycle Performance, where Gas turbine cycle power plants are being extensively used for power generation across the world. The variation in temperature and humidity of atmospheric air significantly affects the plant performance of naturally aspirated compressors in gas turbine based installations. Ian and Andrew , in 2014 [11] explored experimentally the effect of inlet pressure and temperature on the total efficiency of a steady flow centrifugal compressor across a range of conditions in isolation of pulse flow effects and with negligible heat transfer. The results suggest that for any given corrected mass flow rate and total-total pressure ratio the isentropic efficiency is dependent on inlet conditions. The effect of a 40 K change in inlet temperature or a 0.5 bar change in inlet pressure results in up to 15 % point +/− 1% point change in total-total efficiency at low rotor speeds and up to 5 % points +/− 1% point at higher rotor speeds.

The objectives of the present study is to debottleneck the booster compression station in BG-Rashpetco gas plant (Burullus) to increase the compression station capacity that consists of two centrifugal compressors driven by gas turbines from 880 to 1200 MMscf/d with studying the effect of operating conditions changes on the downstream adsorption package.

Case study

BG-RASHPETCO is a joint venture company between EGAS (50%) & British Gas (25%) & PETRONAS (25%). By 2016 BG sold out its sharing percentage to SHELL. We have two separated gas plants Rosetta & Burullus; our focal point is Burullus gas plant. BG-RASHPETCO gas field located on the coast of the Mediterranean Sea in the Nile delta in Egypt adjacent to the east side of Abu-Qir, approximately 34 km east of Abu-Qir city and 50 km east of Alexandria, the site collectively referred to as IDKU. Burullus Gas Company, on behalf of the partners, has continued with the exploration and development of the concession. There are currently more than 50 production wells on the concession from the Scarab, Saffron, Simian, Sienna, and Sapphire, Sinbad, Sequoia and Serpent reservoirs. The overall WDDM gas production system is operated based upon the demands of the downstream ELNG plant and NTS (domestic market - GASCO). The Scarab and Saffron fields have been supplying gas for the Domestic market since March 2003. The Simian, Sienna and Sapphire have been supplying gas to ELNG Trains 1 and 2 since the facility was completed in 2005. The Scarab Serpent has provided the NTS in 2008. The Sequoia wells have provided ELNG by 2009. Booster compressors project (phase 5) has installed in 2010 to overcome the decline in gas arrival pressure from 80.0 to 55.0 barg. Main compression station installed by 2012 to overcome the decline in arrival gas pressure from 55.0 to 26 barg. The last set of gas wells came on production line by 2013/2014 where it called (phase 9

A&B) **Figure (1)** shows simply phase 7 installations. The concession has been developed by a series of subsea wells tied back to a Pipeline End Manifold (PLEM) located in 95m water depth approximately 70km offshore IDKU (the onshore receiving terminal).

Rosetta gas plant has two offshore production platforms Rosetta P1 and P2 while Burullus gas plant owns one facility offshore platform called Simian CP-1.

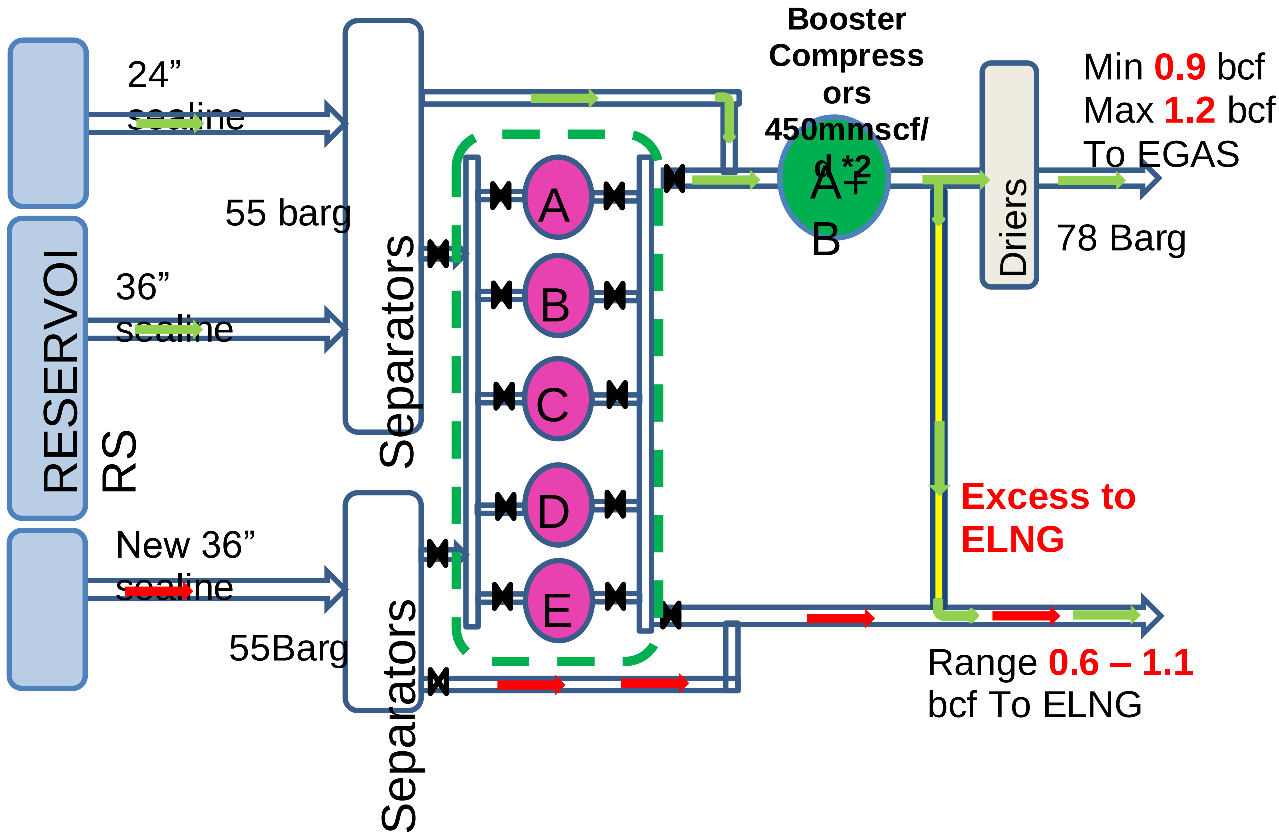

Figure 1 Figure 1 schematic graph shows simply installation of main compression station.

Booster compression station

Consists of two centrifugal compressors K-1050 A&B with capacity of 450 MMscf/d aligned to compress the produced gas to the conditioning skid to NTS. The compression station located between the HP separators & conditioning skid to raise pressure from 52.8 to 78 barg.

Each train begins with gas/gas heat exchanger with maximum capacity 500 MMscf/d where inlet gas passes through tube side with pressure 55 barg & temperature 19.0 Degree C to be exchanged with the hot compressed stream out from compressor, so the resulted stream temperature is 35 Degree C to the suction K.O drum V-1050 to guarantee the liquid droplets separation then to the compressor machine.

The outlet compressed gas with temperature 71.0 Degree C and pressure 78.0 barg to be cooled in the after cooler which is variable speed air cooler to cool the gas to 43.0 Degree C, the gas out of the air cooler go to the inlet gas/gas heat exchanger for further cooling and to heat the inlet gas stream to compressor, therefore it flows to the discharge K.O V-1051 to trap any condensed liquids and finally to the inlet of the condition trains. **Figure (2)** shows the existing status for booster compressor train.

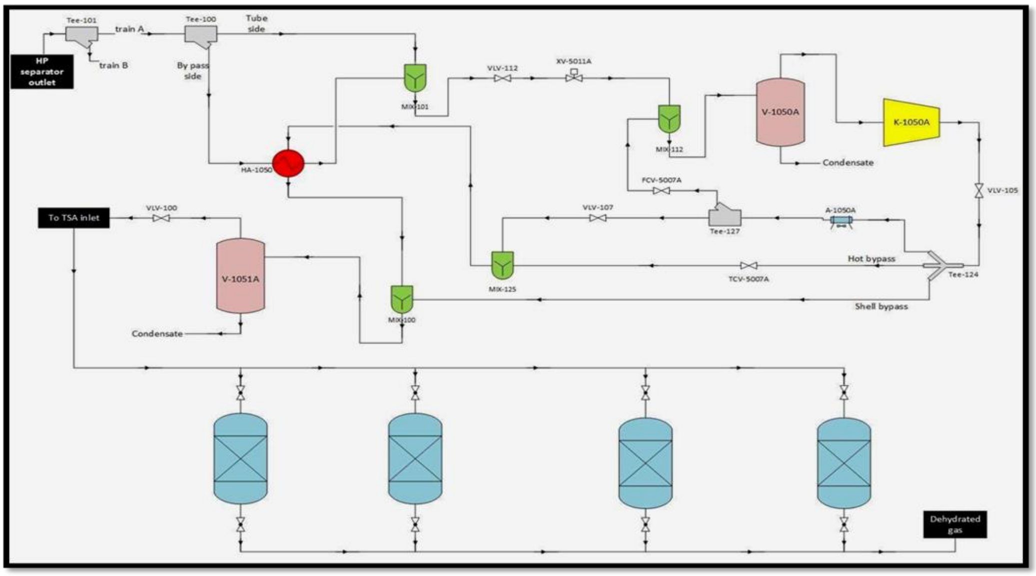

Figure 2 Existing case for Booster compressor train.

Results and discussion

Aspen-HYSYS is a process modeling tool for steady-state simulation, design, performance monitoring, optimization and business planning for chemicals, specialty chemicals and petrochemicals, the version 7.2 had been used in our case applying Peng- Robinson equation of state.

Since the booster compression station capacity is limited to 880 MMscf/d where the restriction found in two issues:

- The Gas /Gas heat exchanger H-1050 capacity located in the entrance of each compressor train is limited to 450 MMSCF/D.
- The adsorption package operating conditions interruption.
- The plan is to build up two cases for one compressor train using simulation software as follows:

Existing case: shows the current operating conditions and flow rates with the mentioned bottlenecks.

Proposed case: shows the compressor train associated with the proposed modifications in two steps. The first step is installing two proposed major modifications; adding slip stream on the gas/gas heat exchanger tube bundle side that accommodate around 160 MMscf/d.; adding slip stream on the gas/gas heat exchanger shell side bypassing the compressor after cooler that accommodate the same amount of gas 160 MMscf/d. The second step is studying the effect of the proposed modifications on dehydration package. Our dehydration package using adsorption technology with silica gel fixed beds located downstream the mentioned compression station and it has major rule to get the standard specification for the provided natural gas to the national gas grid "GASCO". The agreed specifications for burullus gas plant natural gas are 0.0 Degree C as water dew point & +5.0 Degree C as HC dew point.

The effect of the two slip streams that installed on the compressor gas /gas inlet heat exchanger will be studied as the inlet operating conditions of the adsorption beds gas feed expected to be changed specially the inlet temperature and pressure, in addition to the mentioned increment in gas flow.

Existing case model validation: (For one compressor train K-1050)

To validate our model (we can walk through the built up simulation model step by step), the feed gas composition **Table (1)** that delivered by HP separators has to be entered properly. Where it comes out from HP separators fed to train A inlet gas / gas heat exchanger tube side (H-1050 A) through (TEE-101), this exchanger is shell & tube horizontal exchanger. The rated flow found 445 MMSCFD that validated according to the pressure drop across the heat exchanger 0.13 barg (tube side) and 0.3 barg (shell side). The gas inlet tube side pressure & temperature is 19 Degree C & 57 barg while the expected outlet tube side is 30.5 degree C & pressure 56.5 barg matched with the tube side pressure drop.

Table 1 Feed composition

Component	Mole %
H_2O	0.09
CO_2	0.25
Nitrogen	0.09
Methane	97.28
Ethane	1.92
Propane	0.16
i-Butane	0.06
n-Butane	0.04
i-Pentane	0.02
n-Pentane	0.01
n-Hexane	0.03
n-Heptane	0.01
n-Octane	0.01
n-nonane	0.01
Total	100.00

Heat exchanger Shell side should accommodate the same flow 445 MMSCFD with inlet temperature 43.0 Degree C & pressure approximately equal to compressor discharge pressure 78 barg. The expected shell side outlet temperature found 32 Degree C & pressure according to the shell side pressure drop in data sheet. The heated outlet gas (stream no 53) passing through (VLV-112) then (XV-5011) which is the compressor sequence start up valve gas then combine with the anti-surge stream (stream no 03) together into mixer (MIX-112) resulted in (stream no 37) & went through the pipe segment (pipe 101), the outlet stream (stream 28) entering suction scrubber (V-1050 A) to ensure that the gas is free from any liquid droplets. Gas out from the Scrubber trough (stream no 142) then pass through the compressor suction pipe segment (pipe 102) then to be fed to the booster compressor (K-1050 A). The compressed gas (stream no 50) is flowing through the discharge pipe segment (pipe 103) then manual isolation valve (VLV-105). Centrifugal compressor receives the gas with pressure around 55.0 barg to raise it to 78.0 barg, this compressor can handle up to 610 MMSCFD at summer conditions 30 Degree C and speed 100% as per manufacturer compressor curve, see **Figure (3)**

The compressor capacity is restricted due to the limitation in the upstream shell and tube gas/gas heat exchanger. So the maximum compressor train throughput is 450 MMSCFD. The compressor after cooler is responsible for cooling the compressed gas close to suction gas temperature but in our case the compressed gas came out with 71.0 Degree C therefore cooled till around 43.5 Degree C which is not matched to the compressor suction gas temperature so the designer has utilized the gas/gas exchanger to cool the outlet gas as a second stage cooler to be close to 25.0 Degree C, (this is from the designer point of view). Dehydration package that positioned downstream the compression station to fulfill the natural gas dew point specifications delivered to the national grid. This specification is 0.0 Degree C (water dew point at 70.0 Barg) & +5.0 Degree C (HC dew point). The results obtained are in **Table (2)**.

Figure 3 the booster compressor performance curve at Tin = 30 C.

Table 2 Existing case model results for one compression train

Conditions	Existing case (440 MMscf/d)			Existing case (610 MMscf/d) Without modification		
Item	Temp (C)	Press. (Barg)	Flow (MMscf/d)	Temp (C)	Press. (Barg)	Flow (MMscf/d)
Tube side inlet	19	57	439.2	19	57	439.2
Tube side outlet	37.49	56.86	439.2	35.65	56.75	439.2
Shell side inlet	43.11	79.5	439.9	43.21	79.49	439.9
Shell side outlet	25.28	78.68	439.9	26.72	77.98	439.9
Tube side bypass	--	--	0	--	--	0
Shell side bypass	--	--	0	--	--	0
Compressor inlet	35.49	53.49	440	33.87	53.49	610
Adiabatic head	6034	--	--	5992		

Polytropic head	6077	--	--	6060		
Adiabatic efficiency	86.76	--	--	80.28		
Polytropic efficiency	87.29	--	--	81.19		
Power consumption (KW)	6882	--	--	1.023*104		
Speed (rpm)	8772	--	--	9365		
Temperature swing Adsorption (TSA) Feed	25.28	78.68	439.8	26.72	77.98	609.8

Proposed modified case (for one compressor train K-1050)

in this case, the quantity of natural gas produced was 610 MMscf/d per train as we added two slip streams around the inlet gas / gas heat exchanger tube bundles side and shell side with around 12" diameter that can accommodate around 160 MMscf/d. The maximum gas flow rate for the inlet gas/ gas heat exchanger has increased to 610 MMscf/d which is near to the optimum capacity of the compressor machine .After installing the proposed modifications the dehydration capacity has increased to 1200 MMscf/d but still controlled by maximum pressure drop on each bed (max 1.25 bar.) We observed that the gas inlet temperature has increased to 95 degree F (35 Degree C) with gas flow rate 190 to 192 MMscf/d. This increment has affected the pressure drop as it increased from 0.66 to 1.19 bar with also increment in superficial velocity from 29 to 40 feet/min. The regeneration gas flow rate has increased to 52.6 MMscf/d which is matched to design gas flow rate. The adsorption bed can adsorb around 6600 lb. H_2O as a maximum capacity while the quantity of adsorbed water calculated and found 2850 lb.H_2O. From previous we conclude that after increasing the flow rate on adsorption bed to reach to the maximum pressure drop, the quantity of adsorbed bed increased by 1600 lb.H_2O.

Compressor K-1050: The compressor will be directly affected with the modifications mentioned above by increasing the flow from 440 to 610 MMscf/d we find the compressor inlet temperature has reduced from 35.49degree C to 30.34 degree C that resulted in optimizing in compressor power consumption from $1.023*10^4$ Kw to 1.00*e4 Kw. Ref. **Table (3)**.

Heat exchanger pressure drop: For our case, the pressure drop in inlet gas/gas heat exchanger H-1050 is taken 0.13 barg for tube side and 0.30 barg for shell side. The simulation model for the proposed modified case is shown in **Figure (4)**.

Table 3 Proposed case model results for one compression train

Conditions	**Existing case (440 MMscf/d)**			**Existing case (610 MMscf/d) Without modification**		
Item	Temp (C)	Press. (Barg)	Flow (MMscf/d)	Temp (C)	Press. (Barg)	Flow (MMscf/d)
Tube side inlet	19	57	439.2	19	57	439.2
Tube side outlet	37.49	56.86	439.2	32.2	56.85	439.2
Shell side inlet	43.11	79.5	439.9	43.11	79.5	440
Shell side outlet	25.28	78.68	439.9	25.29	78.68	440
Tube side bypass	--	--	0	--	--	170.8
Shell side bypass	--	--	0	--	--	170.0
Compressor inlet	35.49	53.49	440	30.37	53.49	610
Adiabatic head	6034	--	--	5897		
Polytropic head	6077	--	--	5962		
Adiabatic efficiency	86.76	--	--	80.8		
Polytropic efficiency	87.29	--	--	81.7		
Power consumption (KW)	6882	--	--	$1.00*10^4$		
Speed (rpm)	8772	--	--	9262		
TSA Feed	25.28	78.68	439.8	35.7	78.68	610

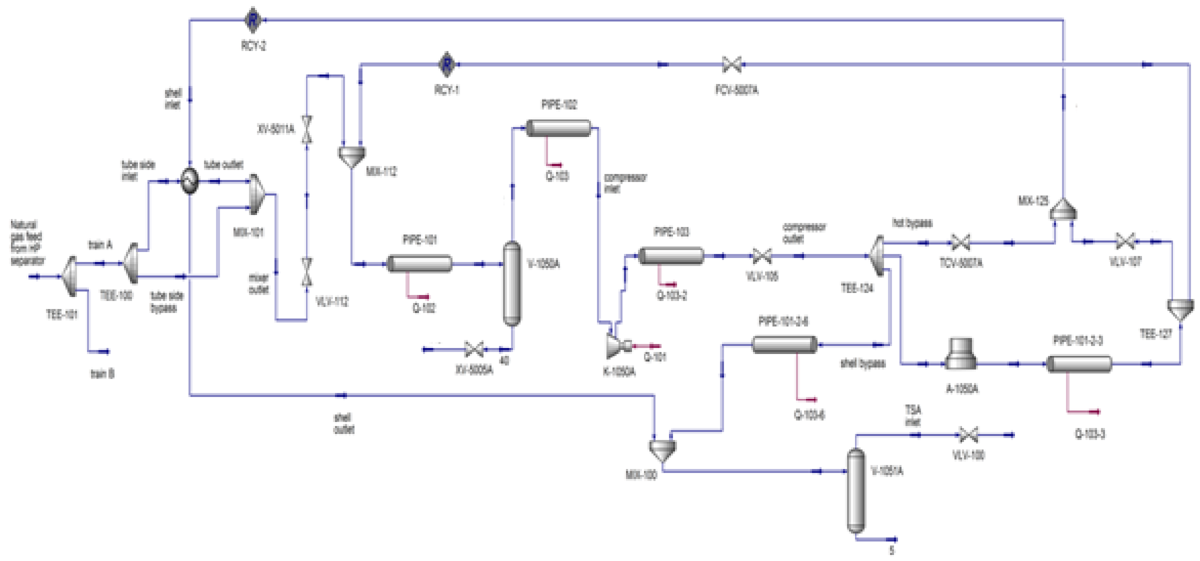

Figure 4 simulation model for the proposed modified case

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Dehydration Bed Calculation

The second part of our work is to study the effect of increasing the gas flow from 140 to 190 MMscf/d on the adsorption beds, we have to review our dehydration calculations (pressure drop across the adsorption beds, maximum adsorbed water capacity & regeneration gas flow rate) for the Existing and proposed modifications cases. The dehydration package in burullus gas plant consist of two trains A and B, each train includes four (4) beds, the dehydration cycle states that 3 beds in adsorption mode and the fourth bed in regeneration mode. The dehydration bed cycle is 6.0 hours = 360 minutes (270 minutes adsorption and 90 minutes regeneration). The regeneration cycle is 30 minutes for heating, 46 minutes for cooling and 14 minutes for valves switching.

By calculating the gas density from the next equation:
(Gas M.wt * P) / (Z * R* T) Therefore,
Density = 3.7279 lb/ft^3

By calculating the actual gas flow rate:
Va = Pa*Va*Ts / Ps*Ta*Z Therefore,
Va = 1706986.3 ft3/day = 1185.4 ft^3/min

By calculating the superficial velocity (Vs):
Vs = Va (ft3/min) / Bed cross sectional area (ft^2) Therefore,
Vs = 29.5 ft/min (30 – 35 Max)
Maximum limits (25-30) [13].

Calculation of pressure drop across the adsorption beds:

The following equation for pressure drop [13]
(DP / L) (Psi/ft) = B*U*Vs + C * gas density * (Vs) ^2
L: Bed height in (ft), U: gas viscosity in (cp)
B: constant for beads (1/8 in) = 0.0560,
C: constant for beads (1/8 in) = 0.0000889
Vs: superficial velocity (ft/min)
Therefore, (DP/L) = 0.311, DP = 9.70 psia.
DP = 0.66 bar (Max 1.25) advantica standard [14]

Dehydration calculations: (For existing case)

The gas flow rate per each train is 450 MMscf/d. (144 MMscf/d/each adsorption bed) considering the regeneration gas flow rate is 42.3 MMscf/d.
Gas Temperature inlet (T in) = 25.35 C = 77.576 F
Gas pressure inlet (P in) = 77.65 barg = 1140 psia
Gas M.wt = 16.56
Adsorption beds information are as follows:
Mass of adsorption beads is 24837 kg = 54755.7 lbs.
Volume of bed is 35.44 m3 = 1251.6 ft3
Adsorbent density 54755.7 / 1251.6 = 43.75 lb/ft3
Bed diameter = 7.152 ft
By calculating the cross sectional area of the bed:
A = 3.14 * (D^2) / 4 = 3.14 * (7.152) ^2 / 4 = 40.156 ft2.
Bed height 1251.6 / 40.156 = 31.17 ft

Bed Capacity Calculations:

The bed water content @ 75 F is around 13 lb H2O/ 100 lb. sieve.

Ct (temp correction factor @ bed inlet temperature) = 0.98 [13].

Therefore,

Adsorption bed water content is 13.0 * 0.98 = 12.74 lb H2O/ 100 lb sieve

Length of math transfer zone L (MTZ) = 2.5 + (0.025 * Vs)...

Therefore,

Length of MTZ = 3.24 ft (represent 10.4% of bed height)

By assuming that the MTZ hold 50% of the equilibrium loading, therefore the bed should hold water content as follow:
(12.74 / 100) * 54755.75 * ((1- 0.13) + 0.13*0.5)) = 6613.5 lb. water mecketta chart (15)

Therefore, it equals 35 lb H_2O / MMscf

By calculating the water adsorbed during the bed dehydration cycle =

Corrected water content * gas flow rate * (bed cycle time/24)

Therefore, it equals

35 * 144 * (6 / 24) = 1260 lb. H_2O

Regeneration calculation:

Regeneration gas inlet temperature (T rg) = 288 C = 550 F [13].

Regeneration gas inlet temperature (T rg) corrected = 550 – 50 = 500 F.

Feed Gas inlet temp (T in) = 80 F.

By calculating the regeneration gas flow rate:

Reg. gas flow rate (m) = q / (Cp * Ɵ * (T.rg – Tin))
Where:

m: regeneration gas flow rate - lb/hr / Cp: specific heat – btu /lb F.

q: total quantity of heat required to remove the adsorbed water.

Ɵ: (time of heating / time of regeneration) * bed cycle time.

T.rg: Regen gas temperature. / T in: gas inlet temp.

First step: calculating the heat required to desorbed water

q w = 1800 * quantity of desorbed water

Note: 1800 BTU/lb H2O is a constant [13].

Therefore, (1800 (BTU/lb) * 1260 lb. of desorbed water) = 2268 MBTU

Second step: calculating the heat required to heat the adsorbent

q .si = 0.24 (BTU/ lb. F) * lbs. of sieves * (T.rg corrected – Tin).

= 0.24 * 54755.7 * (500.4 - 80) = 5524.6 MBTU

Third step, by calculating the heat required to heat the vessel steel

q .st = 0.12(BTU/lb. F) * lbs of steel * (T.rg corrected – Tin).

= 0.12 * 75000* (500.4 - 80) = 3783.6 MBTU

Finally, we can conclude:

Q total = 2.5 (qw + qsi + qst) * 1.1

Therefore,

Qt = 2.5 (**2268 + 5524.6 + 3783.6**)*1.1 = 31835 BTU

By calculating the Time allocated for heating = 30 / 90 = 0.3333

Ɵ = 0.3333 * 6 = 2 hrs.

Cp / Cv = 1.298 and Cv = 0.57 btu/lb. F ……. referred to existing case simulation

Therefore,

Cp = 0.57 * 1.298 = 0.7398 BTU/lb. F

By applying in equation (15), therefore

m = [(31835 / 0.7398 * 2 * (550 – 80))*1000] = 45735.4 lb/hr

Regeneration gas flow (M) = 49.46 MMscf/d @ 1140 psia and 550 F

Dehydration calculations (For proposed case)

As mentioned before, the targeted gas flow rate per train is 610 MMscf/d. (each adsorption bed will treat 190 MMscf/d), therefore we will proceed in the same calculations as the existing case.

The results for existing and proposed modification are shown in **Table (4)**.

Table 3 Dehydration bed calculation

Condition	Existing case 440MMscf/d	Modified case 610 MMscf/d
Gas temperature inlet Tin °F	77.56 (25.55 °C)	95.57 (35.35 °C)
Gas compressibility factor	0.89	0.9
Bed height (ft)	31.17	31.17
Pressure drop (bar)	0.66*	1.09*
Bed hold water content lb.H2O	6613.5	6313.6
Water adsorbed during dehydration lb.H2O	1260	2850
Regeneration flow rate lb/hr	45735.4 (49.5 MMscf/d @ 1440 psia, 550 °F)	53379.1 (60.3 MMscf/d @ 1440 psia, 550 °F)

Conclusions

Our thesis aimed to install two major modifications to both booster compressors trains resulted in capacity increment from 880 to 1200 MMscf/d for both trains (Max 610 MMscf/d per each train), the modifications focused on debottlenecking the inlet gas/ gas heat exchanger via adding two slip streams 12" in dia. That accommodates 160 MMscfd. The maximum gas flow rate for the inlet gas/ gas heat exchanger has increased from 450 to 600 MMscfd per train which is near to the optimum compressor operating capacity. After installing the two slip streams around the gas / gas heat exchanger tube &shell sides, The gas flow rate increased from 145 to 190 MMscfd per each dehydration bed resulted in higher pressure drop from 0.66 to 1.19 bar. The adsorption beds gas inlet temperature observed increased from 77.0 to 95.0 F in addition to increasing in superficial velocity from 29 to 40 feet/min. The regeneration gas flow rate has increased from 42.3 to 52.6 MMscfd, this is considered much closed to design regeneration gas flow rate. The adsorption bed capacity had been checked & calculated for both cases the existing and the proposed as it is critical point, found that it can adsorb water around 6600 lb. H_2O as a maximum. Quantity of adsorbed water has calculated found 2850 lb.H_2O. That means increasing the flow rate on each bed to reach the maximum pressure drop resulted in an increment of adsorbed water in bed by 1600 lb.H_2O. Finally, we utilized Burullus gas plant existing facilities to increase the gas throughput to national grid GASCO by 320 MMscfd.

this means providing different industries factories all over Egypt with the necessary fuel gas to increase production & profits and also increasing the electricity generation as a strategic target to our country.

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