



## Steam Alternatives for Natural Gas Combined Cycle with Piperazine

Aburime, E. I.

Department of Chemical Engineering,  
University of Benin, Ugbowo, Benin City, Nigeria

### ABSTRACT

Natural Gas Combined Cycle (NGCC) power plants are one of the largest anthropogenic point sources of atmospheric CO<sub>2</sub>. The amine solvent-based post-combustion CO<sub>2</sub> capture is regarded as a suitable option for retrofitting the natural gas combined cycle (NGCC) power plants. However, the energy requirement for solvent regeneration remains a major challenge, due to the amount of steam extracted from the power plant, which affects the power generated. The most promising near-term strategy for mitigating CO<sub>2</sub> emissions from these facilities is the post-combustion capture of CO<sub>2</sub> using the piperazine with subsequent geologic sequestration. In this work, the steam cycle for an NGCC power plant is modelled. The steam cycle is comprised of two triple-pressure heat recovery steam generators (HRSGs) and one steam turbine. The focus of this study is on the design of the triple and single-pressure HRSGs with combined and simple cycle gas turbines (CCGT & SCGT) respectively, which aims to define the required steam conditions for plant efficiency, fuel flexibility, and economic operations. Pinch temperature, heat duty, and conductance (UA) are the main design parameters. Results revealed that the steam turbine power drops by 41.7% due to steam extraction, while NGCC gross power output reduces from 424.6 MW to 363.3 MW without and with steam extraction respectively. More importantly, the total heat input for solvent regeneration and compressor is 129 MWt, while at 100% load, SCGT can provide steam at a pinch temperature and UA of 262 °K, and 2.2 MW/K respectively.

### ARTICLE INFO

Article History

Received: December, 2020

Received in revised form: February, 2021

Accepted: March, 2021

Published online: May, 2021

### KEYWORDS

Natural gas combined cycle, Steam,  
Piperazine, Pinch temperature

### INTRODUCTION

Conventional natural gas combined cycles (NGCC) will continue to be used as a means of reducing CO<sub>2</sub> emissions and increasing the electricity capacity of the power sector (NETL, 2010). The main restriction for deploying large-scale CO<sub>2</sub> capture systems integrated into power plants is that these processes reduce the net power output of the entire system. The reduction in the power output is because of the parasitic load of the capture plant. The parasitic load demand arises from the re-boiler steam requirements and drivers such as pumps, and compressors in the CO<sub>2</sub> capture process, resulting in an energy penalty. Therefore, continuous improvements are being made to improve the technical performance of power plants. Typically, the net efficiency of

NGCC without CO<sub>2</sub> capture range from a 55%-60% lower heating value (LHV) basis (Amrollahi et al., 2011). As a result of retrofitting the CO<sub>2</sub> capture plant using chemical solvents, the net efficiency decreases by 7%-10% (Peeters et al., 2007). Monoethanolamine (MEA), is the most widely used technology for post-combustion carbon capture from power plants (IPCC, 2014), but having several challenges such as low reaction rate, high resistance to degradation and oxidation etc. With a fast reaction rate, low volatility & viscosity, good energy performance, and resistance to thermal degradation and oxidation, aqueous piperazine (PZ) has been demonstrated as a benchmark solvent for amine scrubbing (Chen et al., 2014; Freeman and Rochelle, 2012; Rochelle et al., 2011). Although,

\*Corresponding author: Aburime, E. I. ✉ [iebuwa.osagie@uniben.edu](mailto:iebuwa.osagie@uniben.edu) ✉ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved



piperazine has a lower solubility window than MEA, it is usually mixed with other solvents such as 2-amino-2-methyl-1-propanol etc. to overcome this challenge.

The NGCC power plant comprises a gas turbine, a heat recovery steam generator (HRSG), and a steam turbine. Piperazine absorption of CO<sub>2</sub> from natural gas-derived flue gases on the commercial scale is technologically feasible. Due to the high prices of energy and the reduced fossil fuel resources, optimized power generation system is very important. This provides a better system for optimal energy consumption and system configuration (IEA, 2018).

In this work, the steam cycle is modelled using Aspen Plus® V10. Temperatures, boiler turbine pressures, extraction pressures, pump

isentropic efficiency, are used as design parameters. The combined and simple cycle gas turbine (CCGT & SCGT) units are the sources of steam supply required to produce electricity. Steam requirements and heat input for the commercial piperazine capture plant from two NGCC units are obtained. Lastly, the most feasible steam extraction points for the NGCC power plant is determined.

## METHODOLOGY

### NGCC Power Plant

A 465 MW NGCC power plant is used to explore the performance of the steam cycle. Table 1 shows the flue gas composition from one (1) gas turbine.

**Table 1:** Flue gas composition for 1 gas turbine (NETL, 2010)

Components	Composition (mol/mol)
Ar	0.0089
CO <sub>2</sub>	0.0381
H <sub>2</sub> O	0.0845
N <sub>2</sub>	0.7438
O <sub>2</sub>	0.1247
Flue gas flowrate (kg/s)	399.2

### Steam Cycle

The steam cycle is composed of two sections: the HRSG and the steam turbine as shown in Figure 1. The HRSG recovers the energy from the hot flue gases to produce steam at three (3) pressure levels (triple pressure HRSG) namely: high-pressure (HP), intermediate-pressure (IP), and low-pressure (LP) steams. The HP steam is sent to the HP turbine. Steam exhausted from the HP turbine is known as the cold reheat steam. The cold reheat steam mixes

with IP steam produced and the mixture is admitted to a re-heater. After reheating, the steam produced is known as the hot reheat steam, and this is sent to the IP turbine. Lastly, the LP steam produced is sent to the LP turbine. Steam is expanded to produce electricity and condensed, the condensate is pumped back to the HRSG. The feed water is pumped into the HRSG with the respective pumps, and the flue gas out of the HRSG goes up to the stack.

\*Corresponding author: Aburime, I. S. ✉ [iebuwa.osagie@uniben.edu](mailto:iebuwa.osagie@uniben.edu) ✉ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved

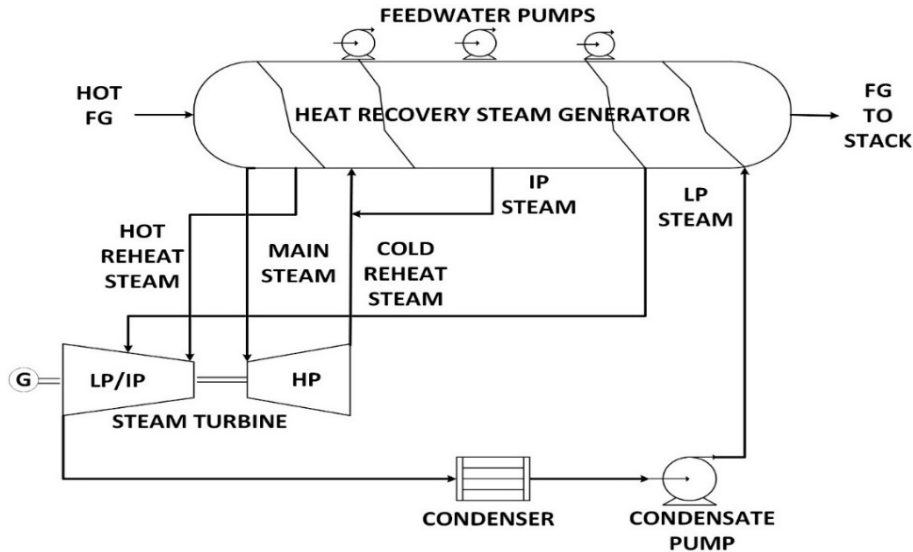


Figure 1: Steam cycle

### Heat Recovery Steam Generator (HRSG) Design and Specifications

The HRSG is configured with HP, IP, and LP steam drums, super heater, re-heater, and economizer sections as shown in Figure 2. The flue gases leaving the economizer are further cooled with cold combustion air in a preheater. The preheater heats the feed water to a minimum temperature to avoid cold water from entering the economizer. This helps avoid condensation from taking place which leads to corrosion of parts. Thus, the preheater reduces the loss of furnace and the exhaust gas temperature, improves the boiler thermal efficiency, and strengthens the heat transfer.

The economizer is designed not to handle steam because it can lead to steaming that

is detrimental to the equipment. As a result, the feed water is heated up to an approach temperature close to saturation in the economizer. Thus, a little amount called sub cooling takes in the economizer. Boiling takes place in the evaporator and saturated vapor produced. In the steam drum, saturated steam is separated from the steam/water mixture. Saturated steam rises through the drum as feed water enters the drum from the economizer. The separated water from the steam/water mixture is returned to the economizer to undergo further heating, while the separated vapor is sent to the super heater which increases steam above saturation temperature to produce superheated steam.

\*Corresponding author: Aburime, I. S. ✉ [iebuwa.osagie@uniben.edu](mailto:iebuwa.osagie@uniben.edu) ✉ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved

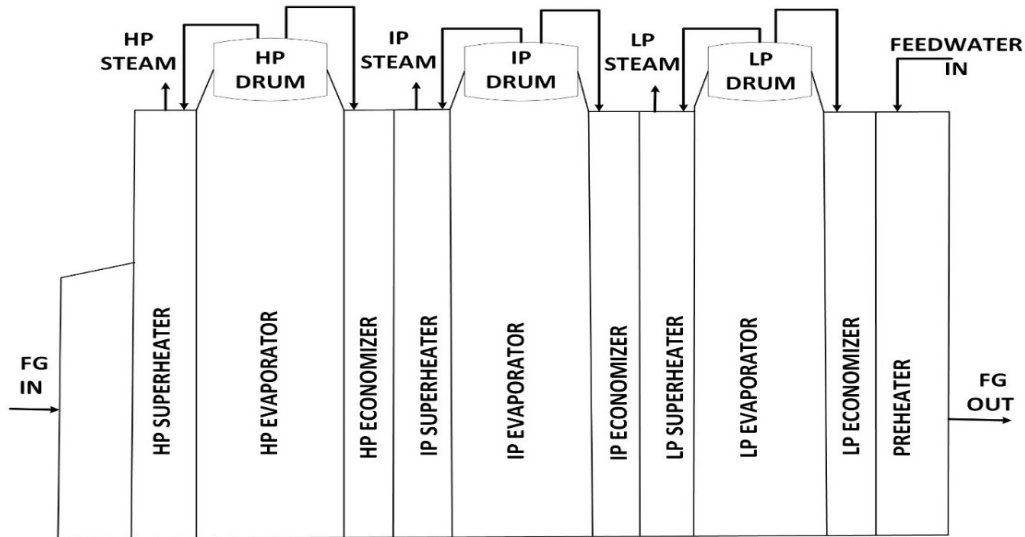


Figure 2: Triple-pressure HRSG

The design variables are shown in Table 2 for one (1) triple-pressure HRSG. The design specifications are pinch and approach point temperatures, superheat temperatures,

pressures, and steam mass flows for the HP, IP, and LP turbines. Flue gas (FG) in and out is at 0.03 bar, and 871 °K.

Table 2: Design specifications for 1 triple-pressure HRSG (NETL, 2010)

Input parameters	Value
HP SPH (bar)	107.3
HP SPH (klb/h)	389.7
HP SPH (°K)	807
HP pinch (°K)	260
IP SPH (bar)	27.6
IP SPH (klb/h)	65.2
IP SPH (°K)	591
IP pinch (°K)	275
LP SPH (bar)	5.6
LP SPH (klb/h)	32.2
LP SPH (°K)	506
LP pinch (°K)	264
Cold reheat (bar)	25.8
Cold reheat (klb/h)	387.1
Reheat (°K)	805
Number of heat exchangers	11

SPH – Superheater

### Steam Turbine Design and Specifications

The steam/water sides are modelled using the steam tables. The HP, IP, and LP steam turbines are characterized by design isentropic

efficiencies of 83.6%, 89.7%, and 91.1% (NETL, 2012) respectively. The steam cycle condenser has a pressure of 0.08 bar, with a saturation

\*Corresponding author: Aburime, I. S. ✉ [iebuwa.osagie@uniben.edu](mailto:iebuwa.osagie@uniben.edu) ✉ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved



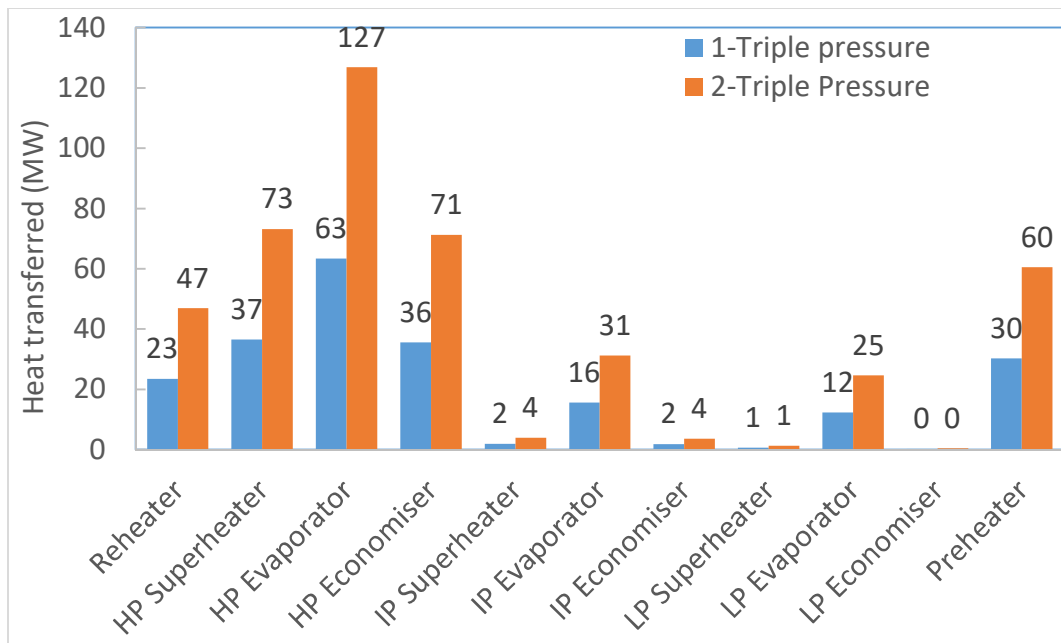


Figure 4: Heat duty performance for 1 triple-pressure HRSG

#### Steam Extraction Results

The heat input required for solvent regeneration from 2 gas turbines unit, is obtained as 118.9 MW. Steam is extracted from both cold and hot reheat streams (before and after re-heater). The steam required for the piperazine capture process is tapped from the HP-IP crossover while the condensate from the stripper is sent back to the HRSG. Results show that extraction from the cold reheat stream has little or effect on the steam turbine. As a result, the feasible extraction point is the hot reheat steam (804 °K, 23.4 bar). Also, steam to be extracted is calculated as 452.2 klb/h

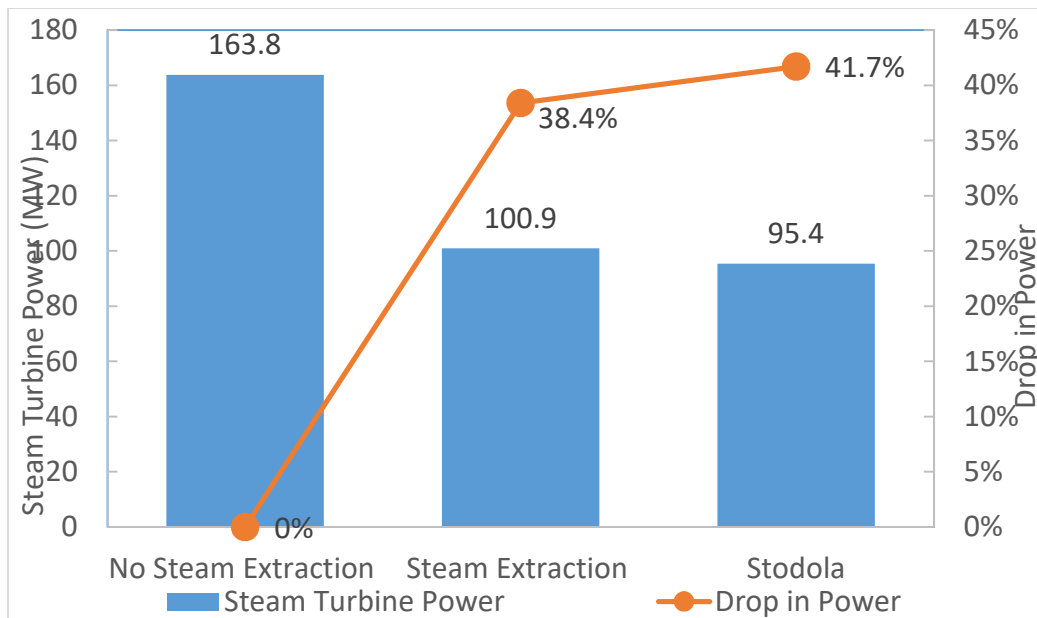
Furthermore, in order to account for pressure losses as a result of steam extraction,

Stodola's ellipse law (Cooke, 1985) given in Equation 1 is used to account for the pressure drop across the turbines. Better pressure profiles are obtained using Stodola as compared to Aspen design calculations. The results are given in Figure 5.

$$\frac{m_1}{m_1^o} = \sqrt{\frac{T_1^o}{T_1}} \sqrt{\frac{p_1^2 - p_2^2}{p_1^{o2} - p_2^{o2}}}$$

1

Where  $m_1$ ,  $T_1$ , and  $P_1$  are mass flowrate, temperature, and pressure, respectively at off-design conditions, and  $m_1^o$ ,  $T_1^o$ , and  $P_1^o$  are at design conditions. Subscripts 1 and 2 are inlet and outlet conditions.  $P_1$  is the unknown pressure.

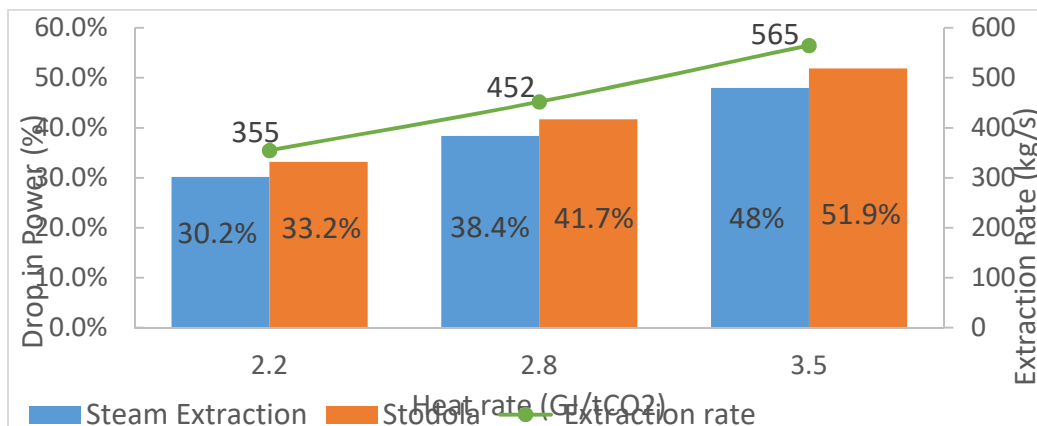


**Figure 5:** Effect of steam turbine power on steam extraction

Figure 5 shows the effect of the steam turbine power due to steam extraction. Results show that extracting 452.2 klb/h of steam, steam turbine power reduces from 163 MW to 100.9 MW (38.4%), while, applying Stodola, power output further reduces to 95.4 MW, which is a 41.7% drop in power. Also shown is the variation of heat rate and the steam to be extracted is obtained as shown in Figures 6 and 7. As the heat rate increases from 2.2 to 2.8 to 3.5 GJ/tCO<sub>2</sub>, steam to be extracted increases from 355 to 452 to 565 klb/h respectively. Thus, as heat rate increases by

a factor of 1.25, the extraction rate also increases by the same factor.

Results show that with a higher heat rate, more steam is required for extraction, and this has the highest effect on the drop in steam turbine power, producing the lowest power of 78.8 MW. On the other hand, the lower the heat rate, the less steam is required and the lowest drop in power output obtained, giving rise to the highest power of 114.4 MW. Also, power drops by 1.1% due to Stodola (Osagie et al., 2018) and the same trend for all heat rates.



**Figure 6:** Effect of heat rate on steam extraction

\*Corresponding author: Aburime, I. S. ✉ [iebuwa.osagie@uniben.edu](mailto:iebuwa.osagie@uniben.edu) ✉ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved



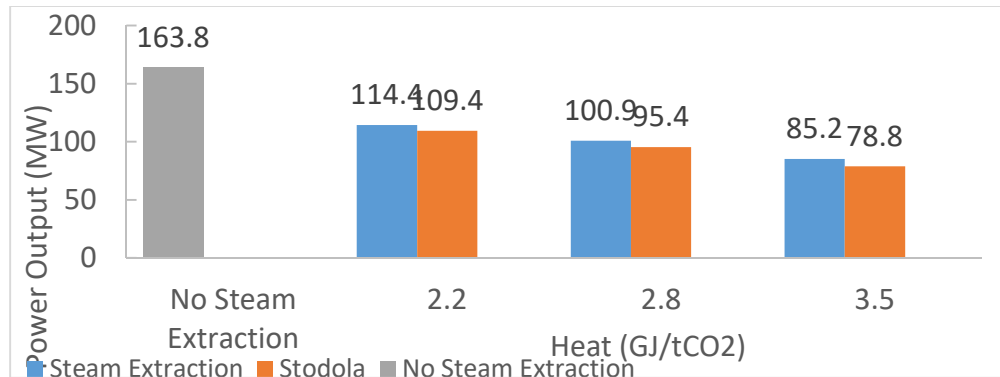


Figure 7: Effect of heat rate on steam turbine power

### Compressor Work

CO<sub>2</sub> leaves the stripper at 3.9 bar and is then compressed to 151.7 bar in the compression section. The compression system includes four compression stages, and the last stage is a pump. Each compression stage is equipped with intercoolers to reduce the stream temperature to 322 °K. Also, the presence of a knockout drum reduces the power requirements by removing water from the CO<sub>2</sub> stream and decreasing the volumetric flow rate through the compressor stages. Results showed that the total compressor work is 10.1 MW.

### Design of Single-Pressure HRSG

In order to achieve high levels of operational flexibility, simple cycles are recommended because the NGCCs have a less flexible operation. Their flexibility is restricted by the steam cycle and balance of the plant (IEA, 2018). Design and operational changes to the HRSG can improve overall flexibility.

In this process, steam is generated from the SCGT. Also, the transfer surfaces used are the super heater, evaporator, and economizer with no preheater to reduce cost. In other to model the process, some assumptions are made, and the process is modelled at off-design conditions given in Table 3.

### Assumptions for HRSG

1. HRSG is unfired
2. Inlet flue gas conditions (flow rate, temperature, and pressure) are fixed
3. Pinch and approach points are selected
4. Temperature and pressure for super heater are fixed
5. Both water and gas side are steady state
6. Negligible heat loss from the heat exchangers
7. Saturated liquid is at the condenser outlet

Unfired HRSGs are purchased more due to their low first cost. The pinch and approach points for unfired HRSGs are usually in the range of 283-288 °K. In the case where specifically less steam is desired, a larger pinch temperature may be used. Thus, pinch and approach points selected in the unfired mode will enable steam temperature to be maintained under fired mode which can be controlled using an attemperator. Also, low heat losses ensure the efficient performance of the system and reduce energy consumption.

Table 3: Conditions for single-pressure HRSG

IP-parameters	Design	Off-design
Hot reheat temperature (°K)	804	561
Hot reheat (bar)	23.4	22.1
Hot reheat (klb/h)	904.7	558.6

Corresponding author: [Abubakar, T. S.](mailto:Abubakar.T.Usaigie@uniben.edu) ⇨ [Abubakar.T.Usaigie@uniben.edu](mailto:Abubakar.T.Usaigie@uniben.edu) ⇨ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved





### Different Cases for Single-Pressure HRSG

At full load (100%), different simulation Cases as shown in Table 4 at off-design conditions are run to achieve the same heat input

required for solvent regeneration at 2.8 GJ/tCO<sub>2</sub> (118.8 MW) combined with compressor work (10.1 MW), giving a total of 129 MWt.

**Table 4:** Single-pressure HRSG Cases at full load

IP steam conditions			
Superheated temperature (560.9 °K), pressure (22.1 bar), feed water in (433.2 °K)			
	Case 1	Case 2	Case 3
Pinch temperature (°K)	261	278	355
UA (MW/K)	2.2	1.4	0.6
Heat duty (MW)	171	164	129
Steam rate (klb/h)	669.1	629.9	558.6
Stack temperature (°K)	490	509	588

Table 4 shows different Cases, where the gas rate is kept constant, and the steam rate is varied. Firstly, as pinch temperature increases, heat duty reduces, this is because the steam rate is reduced. More importantly, at a pinch of 355 °K (Case 3), the same heat input combined with work (129 MWt) is achieved. Also, at a higher pinch temperature, a higher stack temperature is required. The heat transfer coefficient combined with the area (UA), also called the conductance, reduces as pinch temperature increases, this implies that a smaller surface area and heat exchanger required, and this leads to larger driving forces, which brings about savings in capital cost.

Three heat exchangers; super heater, evaporator, and economizer are used for the design. Feed water into the HRSG is at 433 °K, and saturation temperature occurs at 490 °K,

while superheated steam is produced at 561 °K. Also shown is the hot flue gas which is cooled from 871 °K to 582 °K (stack temperature). Results show that the heat transferred is 9.7 MW, 119.5 MW, and 0.1 MW in the IP super heater, IP evaporator, and IP economizer, respectively. Thus, the least temperature gradient is observed in the IP economizer.

Furthermore, a reduction in load increases the bandwidth of their operation, which in turn increases the plant flexibility. Also, part-load operations lead to a reduction in fuel efficiency, which brings about improvements and operations benefits. Thus, fuel flexibility helps to prepare for future emissions regulations. Part-load operations for Case 3 are explored as shown in Table 5. For these Cases, the steam rate is kept constant, while the gas rate is varied.

**Table 5:** Single-pressure HRSG Cases at part-load

IP steam conditions				
Superheated temperature (561 °K), pressure (320 psia), mass flow (558.6 klb/h)				
Feed water in (433 °K)				
Load (%)	100	80	70	50
Pinch temperature (°K)	355	283	256	-
UA (MW/K)	0.6	1.0	2.8	-
Heat duty (MW)	129	129	121	-
Steam rate (klb/h)	558.6	558.6	558.6	-
Stack temperature (°K)	588	514	488	-

As shown in Table 5, pinch temperature reduces with load, this is because more heat is transferred and a lower driving force obtained, this

also shows that more energy can be recovered from the flue gases at a lower load. Hence, a lower stack temperature is obtained at lower loads.

\*Corresponding author: Aburime, I. S. ✉ [iebuwa.osagie@uniben.edu](mailto:iebuwa.osagie@uniben.edu) ✉ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved

Also, a higher UA is obtained at a lower pinch, this implies an increased cost of the heat exchangers with a lower load. Most importantly, the same heat input combined with compressor work (129 MWt) can only be achieved at 80%, while at 70% load and below the required heat duty cannot be achieved. In addition, the 70% load plant has a lower heat capacity, so the temperature rise increases when moving from 80% to 70% load.

These analyses provide the design basis for the single-pressure HRSG and serve as operational guidelines in building the Mustang Station power plant. It also gives useful information for investors.

### Alternate Configuration (Dual-pressure HRSG)

An alternate configuration is a dual-pressure HRSG from the SCGT. Dual pressure systems are widely used, because of higher efficiencies than single-pressure systems and a lower investment cost than triple-pressure HRSGs. The HRSG comprises a series of six (6) heat exchangers: IP super heater, evaporator, economizer, LP super heater, evaporator, and economizer. The design specifications are given in Table 6, and the feed water is at 433 °K.

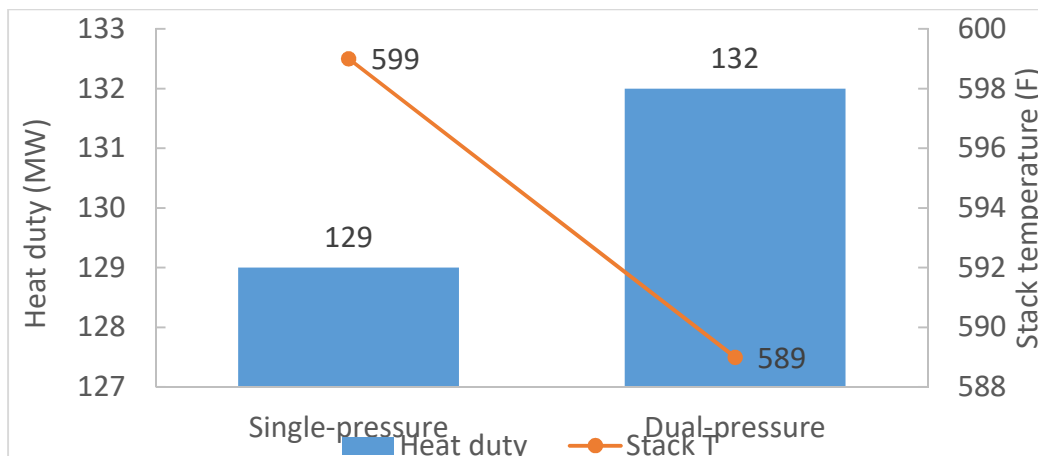
**Table 6:** Dual-pressure HRSG Case

Parameters	IP steam	LP steam
Superheated temperature (°K)	561	506
Pressure (bar)	22.1	5.6
Flow (klb/h)	558.6	32.1
Pinch temperature (°K)	355	355

Using the same pinch temperature, the performance for single and dual-pressure HRSGs is compared for a 100% load plant as shown in Figure 8. Results showed that the single-pressure cycle recovers less heat, this is because the stack temperature remains high, while dual-pressure recovers more heat resulting in lower stack temperature.

This implies that the cost of HRSG will be the same for both single and dual (the same UA of 0.6 MW/K), but more energy recovered from

the flue gas for the dual than that of the single, and also fewer heat transfer surfaces used as compared to the triple-pressure HRSG. However, the dual-pressure HRSG has lots of degrees of freedom for design to achieve the heat input required for 2 NGCCs, such as maximum LP steam from 2 HRSGs can be explored, IP steam can be reduced, etc. As a result, the dual-pressure HRSG will be designed based on demand and specific needs from investors.



**Figure 8:** Effect of heat duty and stack temperature on single and dual-pressure HRSGs

\*Corresponding author: Aburime, I. S. ✉ [iebuwa.osagie@uniben.edu](mailto:iebuwa.osagie@uniben.edu) ✉ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved



## CONCLUSION

This study has provided two different steam supply options for the NGCC power plant. Results showed that the total heat input for solvent regeneration and compressor is 129 MW, and the most feasible extraction point is the HP/IP crossover in the steam turbine. Also, steam turbine power dropped by 41.7% due to steam extraction, while NGCC gross power output reduced from 424.6 MW to 363.3 MW without and with steam extraction respectively, for a 90% CO<sub>2</sub> removal. Furthermore, the single-pressure HRSG with SCGT can provide steam at 100% load, designed at a pinch temperature of 355 °K and UA of 0.6 MW/K, and at 80% load, a pinch temperature of 283 °K and UA of 1.0 MW/K are required.

Future work focuses on the evaluation of the steam cycle with gas treatment. This is because humidity and condensation can cause water to accumulate in the fuel system. The gas treatment removes water from fuel to improve gas quality and helps extend the life of engine parts. Secondly, supplementary firing (duct burning) is required for the single IP-pressure to reduce exit gas temperature from HRSG (only fired mode is assumed). Duct burning increases the temperature of the steam generated to produce additional power. It helps compensate for the loss of power due to steam extraction and seasonal temperature variation. Thirdly, further evaluation of the dual-pressure HRSG is required.

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\*Corresponding author: Aburime, I. S. ✉ [iebuwa.osagie@uniben.edu](mailto:iebuwa.osagie@uniben.edu) ✉ Department of Chemical Engineering, University of Benin, Ugbowo, Benin City, Nigeria. © 2019 Faculty of Technology Education, ATBU Bauchi. All rights reserved