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The feasibility of using Supercritical CO_2 cascade cycle in the combined cycle

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ABSTRACT

Electricity is a form of energy that expected to be increased and grown in most industrial sectors. The Gas turbine (GT) the equipment used to convert natural gas to electricity because it is cleaner than other fossils fuels. Using a stand-alone GT has many disadvantages like the low efficiency which about 30-40 % and most importantly is the waste of energy that goes through the exhausted gases. The bottom cycle is a solution to recover this energy. Commonly, heat recovery steam generation (HRSG) is used to utilize the waste energy but using HRSG is considered not efficient although it can increase the system efficiency to 60 %. The cycle used to recover the heat is called also a bottoming cycle. Besides, the requirement of the weight and space availability in mature onshore fields and offshore fields. The sCO_2 cycle is considered a promising alternative bottoming cycle due to the additional produced power and also its compactness. This study evaluates the cascade sCO_2 cycle used by (Kim et al., 2017) in comparison with the traditional steam Rankine cycle. To make this comparison fear, once through (one-stage) is combined with the bottoming cycle. The base case was remodelled by Aspen HYSYS software and was validated with the existing topping cycle a bottoming cycle studied by (Følgesvold et al., 2017). The thermodynamic analysis was evaluated to show the possible increase in the produced power. the results indicate that the cascade sCO_2 cycle produces higher power.

Keywords: *Chemical Engineering, Combined Cycle, SCO_2*

1. INTRODUCTION

The development of society in terms of sustainability and economic perspective is determined by the energy sector because of its significant impact. The dominated source of the energy is fossil fuels (Kim et al., 2017). The generated electricity sector, industrial and transportation and depend mainly on the fossil energy source (Energy. & Environmental Protection Agency, 2018). The electricity power share is considered mainly consumer see Fig. ***1***. Better utilization of energy and finding a more efficient and more friendly environment is an urgent need because of the pressure of the current environmental regulation. Another reason is the depletion of fossil energy sources over the last years. The variation of the oil price put more risk in the energy sector (Asif & Muneer, 2007).

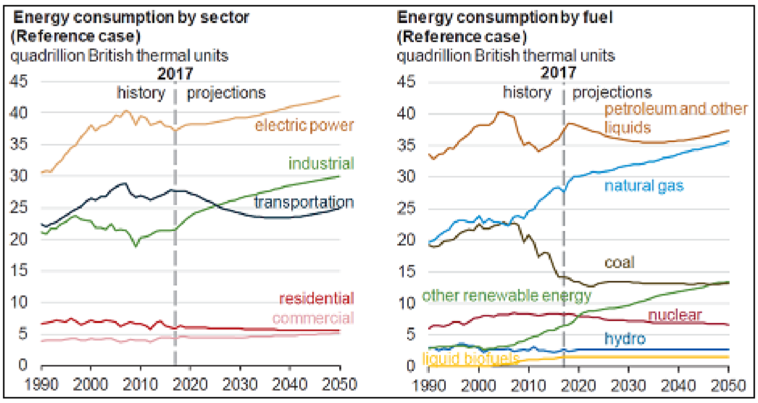

Fig. 1(A): energy consumption by sector (B) Energy consumption by fuel (Energy. & Environmental Protection Agency, 2018)

It can be seen that the main fuel used for electricity generation is Natural gas by the gas turbine. The efficiency is yet considered very low which about 40 % under optimal operation conditions and also the quantality of the waste energy (Moroz et al., 2015). To utilize this waste energy, a bottoming cycle is connected to the topping cycle (the gas turbine). A steam Rankine cycle is the traditional bottoming cycle. This combination called " combined cycle gas turbine" or " heat recovery steam generator (HRSG)". Water is the media of the Rankine cycle and the bottoming cycle consist of the following components. By using the HRSG, the overall efficiency of 61 % is achieved (about double of the gas turbine stand-alone thermal efficiency) (Smith, 2014). Different non-water working fluid have been investigated to produced more power form the bottoming cycle such as the Organic Rankine cycle (ORC) and Supercritical CO_2 (sCO_2) cycle. A higher generated power can be achieved and also can it can satisfy the constraints of the available space and weight in the offshore and mature onshore plants application. The properties of CO_2 make it a good alternative working fluid, such as low supercritical conditions of CO2 (7.38 MPa and 32 °C), the low cost, low toxicity and corrosively, no flammability, thermal stability and worldwide availability of CO2 for commercial distribution network (Nord & Bolland, 2013). The required compressor or pump for the sCO_2 is smaller die its high density (Dostál, 2005). The high density of sCO_2 lead to a smaller heat exchanger(s) size and a lower associated capital cost (Persichilli et al., 2012) see Fig. ***2***. This study aims to investigate the feasibility of using sCO2 cycle in combined cycle gas turbine (CCGT) to replace the traditional steam Rankine cycle (HRSG) and to investigate the sCO2 suitability for offshore plants in terms of generated power, heat recovery and compactness (Følgesvold et al., 2017). The sCO_2 cycle shows promising results in many applications of heat recovery from the Bottoming cycle (GT).

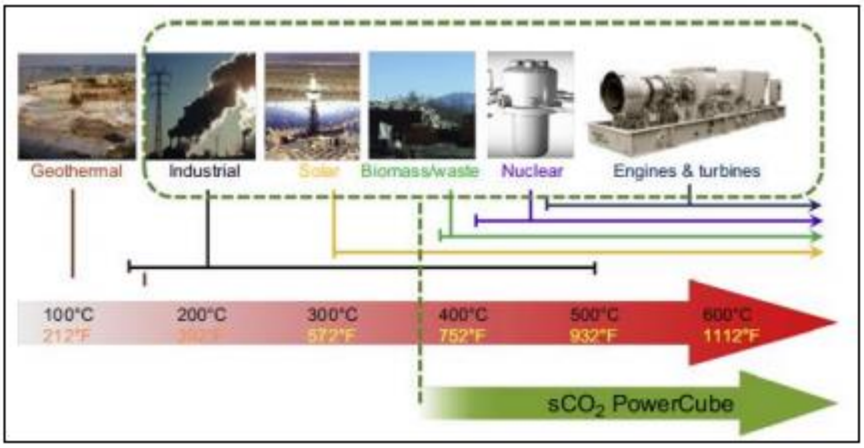

Fig. 2: Supercritical CO2 applications for various heat sources showing the general temperature(Musgrove & Wright, 2017)

2. METHODOLOGY

The comparison between using the sCO_2 as a bottoming cycle instead of the traditional steam Rankine cycle will be conducted by the simulation for both cycles by HYSYS as a combined topping cycle (SGT -800 gas turbine). The target is the calculation of the advantages of sCO_2 in terms of the total produced electricity and its compactness over the steam cycle. These two parameters are essential in the plants and sites where the lack of space like the offshore plans or a lack of space in mature onshore oil fields. Replacing the working fluid, water, with a different one can open a horizon to achieve an additional produced power and a smaller footprint. The methodology includes main assumptions, the simulation tool, the required input data for the topping and the bottoming cycles. The topping cycle is modelled By Aspen HYSYS and validated with performance data reported by SIEMENS Company. The Rankine steam cycle is validated with the cycle used by (Følgesvold et al., 2017).

To achieve the aim of the study, the using of sCO_2 cycle aa bottoming is based on the work of (Kim et al., 2017). The operation conditions were adjusted to be compatible with the proposed topping cycle which is SGT-800 gas turbine. The heat exchanger used in the combined cycles is shell and tube although it is not the typical type used for the sCO_2. The heat exchanger area is calculated to show its suitability to the application where it requires a smaller heat exchanger. The fluid package used for the topping cycle is Peng Robinson and for the steam cycle s ASME and the sCO_2 is (lee-Kessler-Plocker). Most of the calculations are conducted by Microsoft Excel.

3. ASSUMPTIONS

- The piping streams pressure drop, neglecting the kinetic and potential energies in the piping and the heat exchanger.
- Complete conversion is assumed to be 100 %.
- Total net produced power output (TNP) and the thermal efficiency of the combined cycle power plant were calculated. The following equations are employed by Aspen HYSYS.

$$Net\ Power_{\ topping\ cycle} = power_{Gas\ turbine} + power_{compressor} \quad \text{eq (1)}$$

$$Net\ Power_{\ bottomimg\ \ cycle} = power_{Gas\ turbine} + power_{compressoror} \quad \text{eq (2)}$$

$$Total\ net\ produced\ output(TNP)_{CCGT} = NP_{topping\ cycle} + NP_{bottoming\ cycle}$$

$$Q_{Fuel} = \ M_{Fuel} + LHV_{Feul}\ \ or = \ M_{Fuel} * CP * (T_{in} - T_{out})$$

Where:

$To_{pping\ cycle}$= Net produced topping cycle power (MW)

$P_{Gas\ Turbine}$ = gas turbine output Power (MW)
$P_{Compressor}$ = compressor required power (MW)
$NP_{Bottoming\ Turbine}$ = Net Power bottoming (MW)
$P_{bottoming\ turbine}$ = Power output steam cycle (MW)
$P_{bottoming\ compressor\ or\ pump}$ = power required for pump or compressor (MW)
Q_{Fuel} = Combustion heat supplied by the fuel (MW)
M_{Fuel} = Fuel mass flowrate (Kg/s)
LHV_{Fuel} = Fuel lower heat value (MJ/KG)
CP_{Fuel} = Mass heat capacity ($Kg.m^2 / K*s^2$)

- *The heat recovery of the waste heat is calculated according to equation (6):*

$$Heat\ Recovery\ (\eta_{HR}) = \frac{the\ recovery\ heat}{The\ available\ heat}\ x100$$

- *The thermal efficiency of the cycle is calculated according to equation (7):*

$$Thermal\ efficiency = \frac{The\ net\ Power\ produced}{The\ recovery\ heat}$$

- *System efficiency is calculated according to the following equation (8)*

$$System\ efficiency\ (\eta_{sys}) = \frac{the\ net\ produced\ power}{The\ avialable\ heat}\ X\ 100$$

- *The adiabatic efficiencies of the pump (or compressor) and the expander for the steam cycle are 88% and 92% respectively.*

4. THE STEAM RANKINE CYCLE

The combined cycle is consisting of an SGT-800 gas turbine and HRSG. The reason behind choosing the single-stage because it is smaller in term of the footprint in comparison with the dual and tri-stage HRSG leading to the difficulty of using in an application where no space is available. The data which is extracted for the real performance report of SGT-800 gas turbine detailed in Table 1. For the topping cycle (Gas turbine) the simulated model by Aspen HYSYS is validated. The model is detailed in

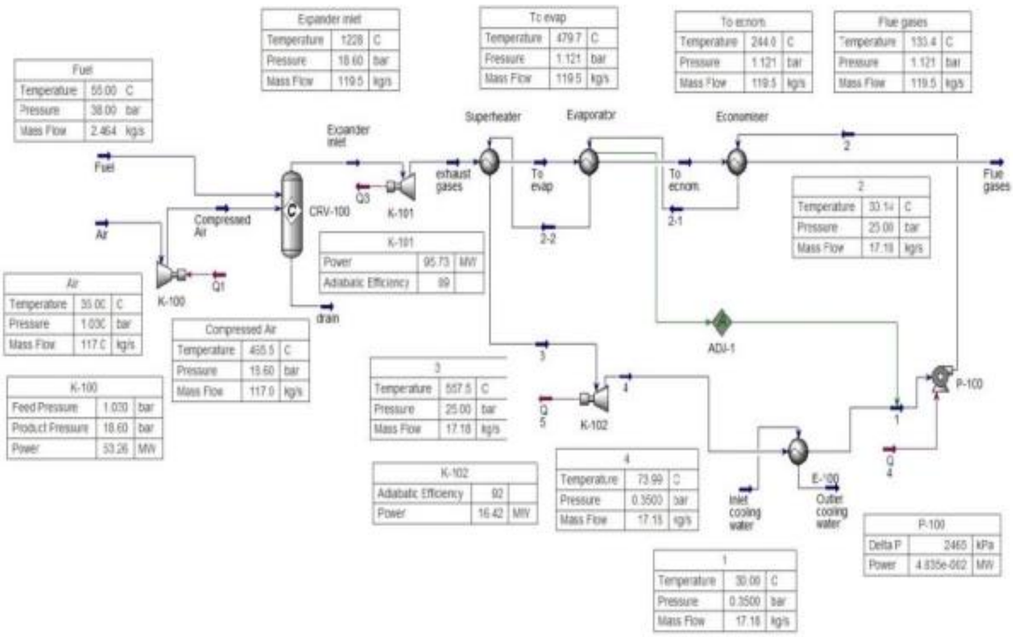

Fig. *3*

Table 1: The obtained data for the gas turbine from Aspen HYSYS simulation

Compressor efficiency [%]	90.76	Net produced power [MW]	43.87
Compressor out let temperature [°C]	465.5	Combustor GT temperature [°C]	123
Compressor consumed power [MW]	52.16	Combustor outlet flow rate [kg/s]	117
Expander produced power [MW]	96.03	Combustor outlet pressure [bar]	1.121
Expander polytrophic efficiency	86.33		

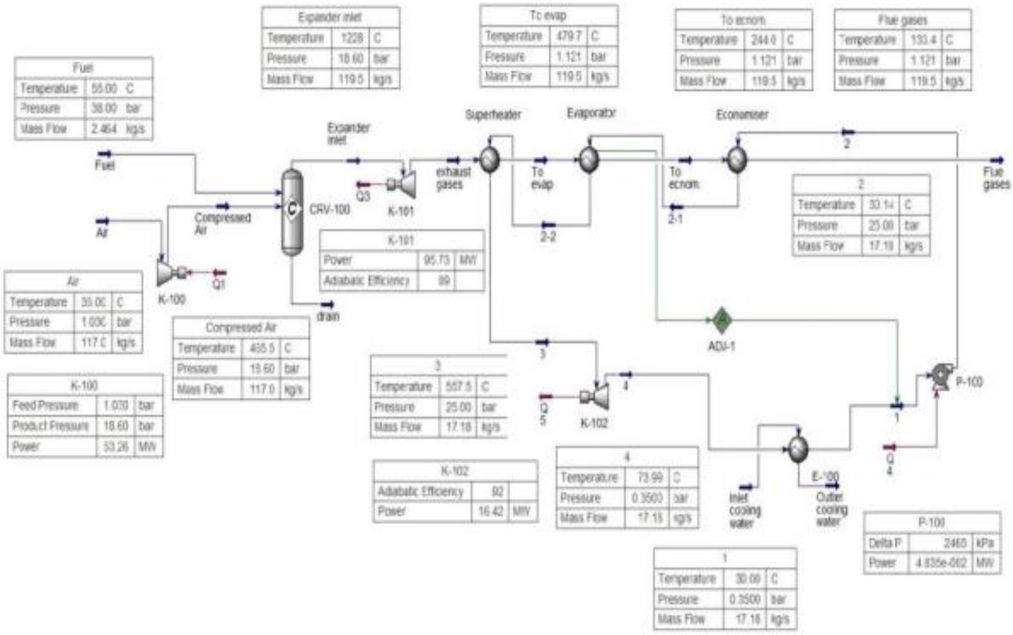

Fig. 3: The developed model for the steam Rankine cycle

5. THE ONCE-THROUGH STEAM GENERATOR

OSGT (once-through steam generator) is the HRSG used as a bottoming cycle similar to the model used by (Følgesvold et al., 2017) to satisfy the requirements of the weight and space as discussed by (Nord & Bolland, 2013). the components of the bottoming steam cycle are heat exchanger (HE), separators, expanders, mixers, tees and pumps. The bottoming cycle is extracting heat from the topping cycle through a series of heat exchangers which are economizer, evaporator and superheated HE. The heat exchanger in this research is shell and tube type as the exhausted gasses passes through the tube side to minimize the heat losses and the water through the shell side. The heat of the exhausted gases will raise the temperature of the water and convert it to steam. The steam will be liquified in a cooler to satisfy the requirements of the pump's operating conditions. The outlet temperature of the flue gases is higher than its dew point. The specifications of the bottoming cycle are detailed in

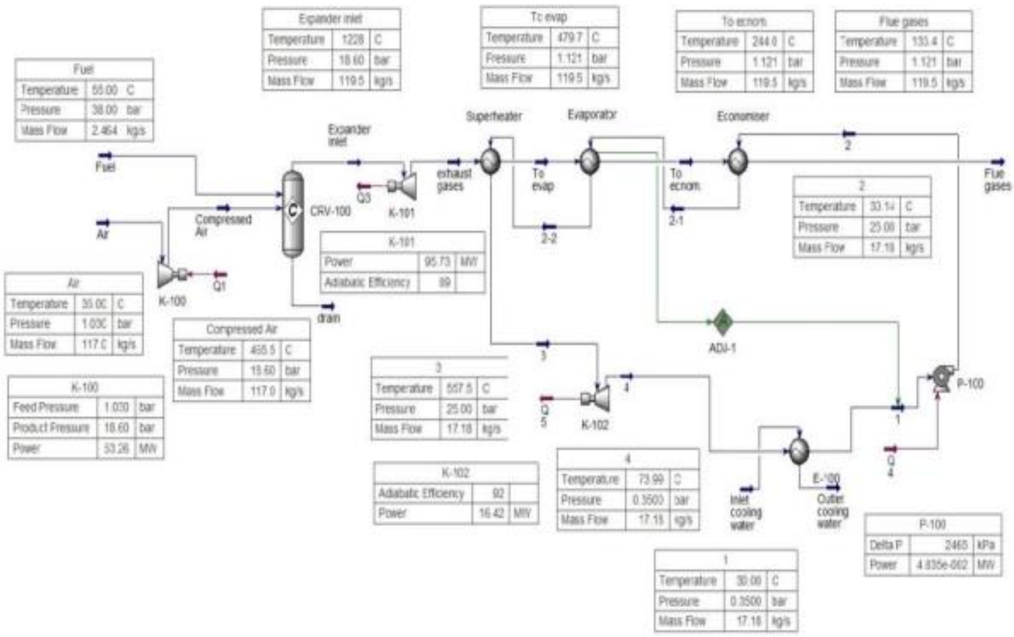

Fig**.** ***3*** and Table 2: The optimised parameters of the steam Rankine cycle used by *(Følgesvold et al., 2017)*

Table 2: The optimised parameters of the steam Rankine cycle used by (Følgesvold et al., 2017)

Pump Adiabatic efficiency [%]	88	Expander Adiabatic efficiency [%]	92
Pump inlet flowrate [kg/s]	147.18	Turbine inlet temperature	557
Pump inlet temperature [°C]	30	Turbine inlet pressure [bar]	25
Pump inlet let pressure[bar]	0.35	Turbine outlet pressure[bar]	0.35
Pump outlet temperature [°C]	30.11	Turbine outlet temperature [°C]	73.99

Pump outlet pressure [bar]	25	Turbine produced power [MW]	16.42
The pump requires power [MW]	0.3456	HRSG Min. temp. [°C]	20
Inlet cooling water temp. [°C]	10	Total produced power [MW]	16
Cooler Min. Delta temperature [°C]	20		

6. THE SCO₂ BOTTOMING CYCLE

Space and weight are considered problematic issue in some applications. Different sCO_2 configurations were investigated by (Crespi et al., 2017). (Kim et al., 2017)studied three configurations but the crucial factor in choosing which one is suitable is the simplicity. In this research, the cascade sCO_2 cycle is chosen. The heat source is the exhausting gases from the Gas Turbine (SGT-800) which as the same as the source used in the steam Rankine cycle. The same parameter will be used in the cascade sCO_2 cycle and it will be remodelled by Aspen HYSYS using Lee-Kesler-Plocker as fluid package property. The same analysis is conducted on the steam cycle in the previous section.

7. THE CASCADE SCO2 RANKINE CYCLE CONFIGURATION

This cycle is considered as two sCO_2 cycles sharing the same cooler and expander. It consists of the following components: expander, cooler, two intermediate heat exchangers (low temperature and high temperature) and a single pump. The splitting begins directing after the pump into two streams. The splitting is used to increase the utilization by maintaining the minimum approach temperature across the cycle heat exchangers. More details are mentioned in Table 1 and Fig. ***1***.

Table 3: The data of the cascade sCO2 cycle modelled by Aspen HYSYS

CO_2 pump flowrate [kg/s]	190	Inlet cooling water temp. [°C]	10
pump Inlet pressure [bar]	57.3	Inlet cooling water pressure [bar]	0.7
pump outlet pressure[bar]	230	Inlet cooling water flowrate [kg/s]	852.1
Pump Inlet temp. [°C]	20	Min. temp. approach [°C]	20
LT loop		HT loop	
The CO_2 flowrate [kg/s]	69.2	The CO_2 flowrate [kg/s]	121.4
Min. temp. approach Recuperator [°C]	5.4	Min. temp. approach Recuperator [°C]	22
Min. temp. the approach of LT IHE [°C]	30	Min. temp. the approach of HT IHE [°C]	30
HT Turbine inlet temp [°C]	75	LT Turbine inlet temp. [°C]	508

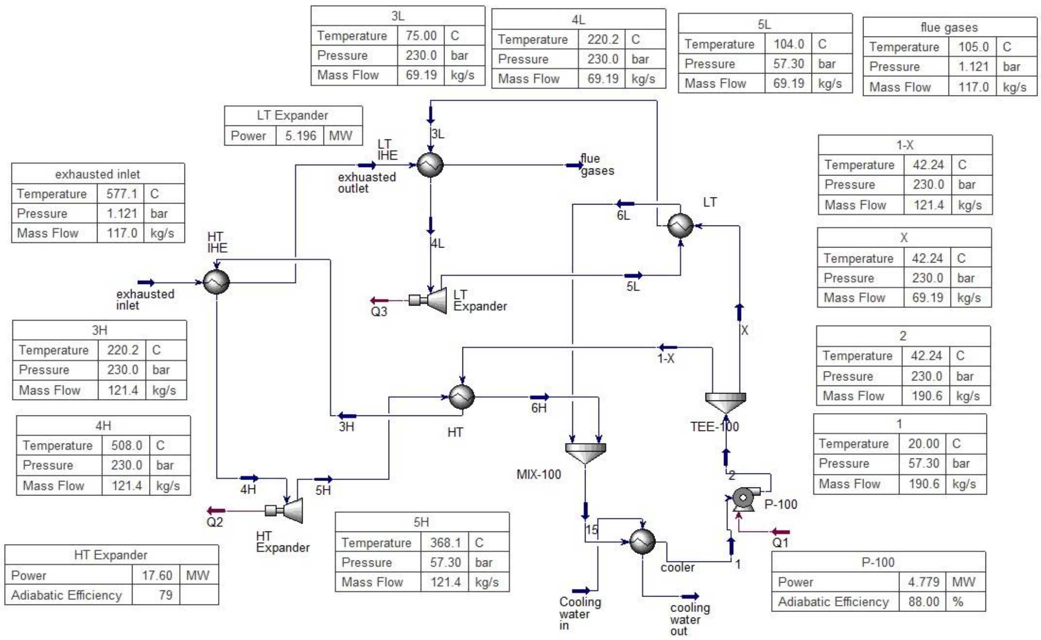

Fig. 4: the cascade sCO_2 cycle modelled by Aspen HYSYS

8. RESULTS AND DISCUSSION

In this section, a discussion about the thermodynamic analysis for the sCO_2 cycle and the steam Rankine cycle including the system efficiency, thermal efficiency, heat recovery and the net produced power. Besides, the required area which is represented by the heat exchanger area is discussed to show the advantage of using the sCO_2 cycle over the traditional ones.

The flue gases temperature is 133 °C at 1.121 bar. The flow rate of the water in the bottoming cycle is 17.18 kg/s. This quantity was adjusted to achieve the minimum approach in the evaporator. The duty of the pump which is required to increase the pressure of the water to 25 bar is 4.4665 x10^{-3}. The economizer will increase the temperature of the water to 223.9 °C and will be superheated to 224.9 °C and finally to 557.5 °C. The total net produced power is 16 MW.

The outlet temperature of the pump is 230 bar and 42.42 °C with flowrate 190.6 Kg/s which was determined by Aspen HYSYS according to the input data. The pressure of the CO_2 is reached to a supercritical state at 230 bar. The power consumed by the pump is 4.778 MW. The sCO_2 is then divided into two loops, the split ratio is (0.6366/0.3634) in the LT and HT respectively. The generated power in the HT loop expander is 17.59 MW and it is 5.2 MW in the LT Expander. The total net produced power is 18.012 MW as detailed in Fig. ***4***.

The heat recovery of the sCO2 cycle is calculated according to equation (6) explained in the assumption section and the thermal efficiency according to equation (7) and the system efficiency according to equation (8). The heat recovery of the cycle from the exhaust gas is equal to the summation of the recovered hat in HT IHE and LT IHE. The Q of HT loop = 117.1 * 1.187 *(577.1-250.2) = 45.438 MW. The Q of HT loop = 117.1 * 1.112 *(250.2- 250.2) = 18.59 MW. The total recovered heat = 45.438 +18.59 =64028 KW = 64.028 Mw and the heat recovery (%) = (64.028/ 71.488) * 100 = 89.61 %. The thermal efficiency = 18.012 /64.028 =28.02%. The system efficiency is 25.2 %.

9. CONCLUSION

Using sCO_2 cycles for offshore plant application instead of the traditional steam Rankine cycle is the aim of this study and also has a smaller physical footprint, higher heat recovery and higher thermal efficiency. The simulated model by Aspen HYSYS for the cascade sCO2 cycle to be used as bottoming cycle achieve more utilization for the waste heat from the exhaust gases for a topping cycle represented by a commercial SGT-800 gas turbine (topping cycle). The cascade sCO_2 cycles were compared with a simulated base case model for a once-through steam generator (OSGT) by Aspen HYSYS V8.4. The net produced power from the cascade sCO_2 is 18.21 which more than the traditional OSGT (16.4 MW).

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