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Process simulation and thermodynamic analysis of a micro turbine with post-combustion CO_2 capture and exhaust gas recirculation

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Abstract

With the effects of the emissions from power plants causing global climate change, the trend towards lower emission systems such as natural gas power plant is increasing. In this paper a Turbec T100 micro gas turbine is studied. The system is assessed thermodynamically using a steady-state model; model results of its alteration with exhaust gas recirculation (EGR) are presented in this paper. The process simulation with EGR offers a useful assessment when integrated with post-combustion CO_2 capture. The EGR model results in the enrichment of the CO_2 which decrease the energy demand of the CO_2 capture system.

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Keywords: Micro gas turbine; Brayton cycle; Carbon dioxide; Exhaust gas recirculation; Carbon dioxide capture; Reboiler duty

1. Background

With the effects of the emissions from power plants which cause global climate change, the trend towards lower emission systems by reducing the carbon intensity is increasing. In order to reduce the worsening effect of carbon emissions in the atmosphere in terms of pollution, and climate change, then there should be a shift from dependence on conventional power generation systems towards sustainable systems along with the adoption of carbon capture and storage as a means for CO_2 mitigation [1, 2]. There is a driving shift towards natural gas power plants around the globe to meet the increasing power demand and gas turbines are a viable and secure option, both economically and environmentally, for power and heat generation. However, the integration of the power generation cycle with CO_2

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capture technologies poses a high electrical efficiency penalty which is a major barrier.

Nomenclature

ACP	amine capture plant
CFD	computational fluid dynamics
CHP	combined heat and power unit
c_p	molar specific heat capacity at constant pressure, [J/mol-K]
c_v	molar specific heat capacity at constant volume, [J/mol-K]
EGR	exhaust gas recirculation
MGT	micro gas turbine
PACT	pilot-scale advanced capture technology
Q	heat, [J]
r	pressure ratio
S	entropy, [J/mol-K]
T	temperature, [°C]
TOT	turbine outlet temperature, [°C]
UHC	unburned hydrocarbons
W	work, [J]
α	maximum to minimum temperature of the cycle
γ	c_p/c_v
η	efficiency, [%]

Subscripts

a	actual
c	compressor
e	electrical
i	ideal
t	turbine
th	thermal

Of the available options, a gas turbine with exhaust gas recirculation (EGR) is one of the promising technologies which are under discussion. The EGR cycle has various advantages in terms of the higher CO_2 concentration in the exhaust gas and a lower exhaust flow rate towards the capture system. Despite these advantages, technical problems remain regarding the maximum amount of the exhaust gas to be recycled, maintaining the required level of flame stability, and issues regarding unburned hydrocarbons (UHC) and CO. Elkady et al. [3] performed experiments on the DLN F-class turbine combustor and showed a stable operation with 35% EGR. Ditaranto et al. [4] and Evulet et al. [5] reported flame stability issues when O_2 concentration decreased to 14% at the combustor inlet, with higher UHC and CO. It was recommended that the O_2 concentration should be kept higher than 16% to have stable operation at 40% EGR with enrichment to 8% CO_2 at the outlet. Cameretti et al. [6] studied the EGR effect on the performance of a micro gas turbine (MGT) for different types of the fuels, such as natural gas, biogas, kerosene and bioethanol. They also showed EGR as a viable option for the reduction of NO_x through CFD modelling of the MGT [7]. Sipocz and Assadi [8] presented the integration of a post-combustion capture plant with a 400 MW combined cycle at 40 % EGR. Jonshagen et al. [9] developed an IPSE Pro model for a 300 MW GE 109 FB combined cycle at 40 % EGR and studied the effect of the EGR on the isentropic exponent and gas constant. A common conclusion may be drawn that the EGR may enhance the performance of the gas turbine when integrated with the carbon capture system [10-14]. The studies of the effect of the EGR on the CO_2 and O_2 in the flue gas have also included the impacts of EGR on the performance of the post-combustion capture plant. Based on techno-economic analyses, EGR may offer the potential and opportunity to reduce the cost for the CO_2 capture systems [10, 15-17]. Most of the reported literature is focused towards commercial combined cycle systems with higher CO_2 content in the base case as compared to the micro gas turbine which is much leaner. In the context of MGT with EGR, the literature reports only 40% and 50% EGR ratio [18, 19].

In spite of the work on EGR at the large scale and its integration with the CO_2 capture system, more work needs to be done to better understand the EGR effects on the MGT system, as well as its integration with the amine capture plant. Due to the limited literature found in this field, an extensive study needs to be performed in this regard. Therefore, this paper aims to determine and assess the impact of the EGR ratio on the micro gas turbine. The emphasis has been on analyzing the results generated from process simulation and assessing the thermodynamics of the system to identify the optimized operating configuration.

2. Thermodynamic Analysis and Model Development

2.1. Thermodynamics of the Ideal Cycle

In the simplest gas turbine, the compressed air absorbs heat in the combustion chamber, which expands in the turbine before going to the atmosphere. The ideal Brayton cycle consists of four steady state steps, including two isentropic steps for compression and expansion; while there are two constant pressure steps for heat addition and rejection. In a simple gas turbine, the exhaust gas from the turbine is at a sufficiently high temperature such that energy would be wasted if it were simply vented to the atmosphere. Therefore, it is used to raise the temperature of the compressed air before the combustor in order to reduce the specific fuel consumption. This is usually achieved by employing a heat exchanger, in the form of a recuperator/regenerator, to preheat the compressor air by the hot exhaust of the turbine. Even after this, the gas turbine exhaust still has sufficient energy from which heat can be extracted and used as a means of producing steam, or as a heat source for the heating and chilling requirements. The maximum amount of the extractable energy depends on the dew point of the sulfur compounds; these are acidic in nature, and it is best to avoid condensation of them in order to reduce corrosion problems in the heat extractors and exhaust ducts. The thermodynamic details of the Brayton cycle are summarized in Figure 1. They consist of the isentropic compression of the air from step 1 to step 2, the preheating of the air from step 2 to 3, the heat addition to the air at constant pressure in the combustor from step 3 to 4, the isentropic expansion in the turbine from step 4 to 5, the heat extraction step is shown from 6 to 7, and the heat rejection at constant pressure from step 7 to back step 1, which completes the closed cycle.

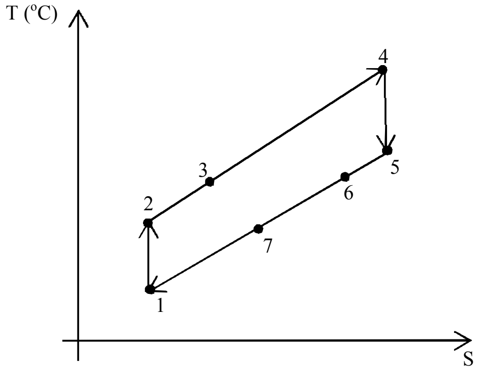


Figure 1: A typical T-S diagram for the combined heat and power cycle.

2.2. Thermodynamics of the Real Cycle

The real gas turbine cycle is an ‘open’ cycle rather than a ‘closed’ cycle, and the composition of the working fluid changes during its movement through the cycle due to the combustion reactions in the combustion chamber.

The compression and expansion processes are irreversibly adiabatic, and therefore this results in an increase in the entropy, and more compression work is required to overcome the fluid friction. The performance of the real open cycle is expressed or judged by knowing the compressor inlet temperature, turbine inlet temperature and fuel consumption by incorporating the fuel/air ratio and the combustion efficiency [20, 21]. Therefore the losses must be incorporated in the components of the real gas turbine cycle in order to judge the actual performance of the system. The thermodynamic analysis of the compressor and the turbine, in the context of the real cycle and incorporating the isentropic compressor and turbine efficiency, will yield a performance that is close to the actual one [20]. The isentropic compressor efficiency is given by [20, 22, 23]:

$$\eta_c = \frac{W_{c\_i}}{W_{c\_a}} = \frac{T_{2\_i} - T_1}{T_2 - T_1} \tag{1}$$

The isentropic turbine efficiency is given by [20, 22, 23]:

$$\eta_t = \frac{W_{t\_a}}{W_{t\_i}} = \frac{T_4 - T_5}{T_{4\_i} - T_5} \tag{2}$$

Using the above definitions, the actual work and the actual efficiency of the turbine cycle is given by [20]:

$$\text{Cycle Work} = W_a = W_{t\_a} - W_{c\_a} \tag{3}$$

$$W_a = c_p T_1 \left[\eta_t \frac{T_4}{T_1} \left(1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}} \right) - \frac{1}{\eta_c} \left(r^{\frac{\gamma-1}{\gamma}} - 1 \right) \right] \tag{4}$$

$$\eta_a = \frac{\eta_t \frac{T_4}{T_1} \left(1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}} \right) - \frac{1}{\eta_c} \left(r^{\frac{\gamma-1}{\gamma}} - 1 \right)}{\alpha - \frac{1}{\eta_c} \left(r^{\frac{\gamma-1}{\gamma}} - 1 \right) - 1} \tag{5}$$

The overall actual cycle efficiency is mainly dependent on the pressure ratio, maximum and minimum temperature in the cycle and the component isentropic efficiencies. When the pressure ratio is increased, the isentropic compressor efficiency tends to decrease and the isentropic turbine efficiency tends to increase for which the whole system is designed, and this results in an increase in the overall cycle efficiency.

The temperature entropy diagram of the MGT is illustrated in Figure 2. The compressor is truly isentropic in the ideal case, indicated by the red dashed line, while in an actual system there are loses in the compressor and this is shown by the black line from point 1 to 2 in Figure 2. The process from point 2 to point 3 is the preheating of the compressed air which corresponds to the saving of the heat through recuperating hot exhaust gases from the outlet of the turbine from point 5 to 6. The combustion process causes an increase in the heat content of the gases indicated from point 3 to 4, which drives the turbine from point 4 to 5 and this results in an increase in the entropy as indicated by the black line in contrast to the ideal red dashed line. The section between point 6 and 7 shows the heat recovered in the gas-liquid heat exchanger in combined heat and power (CHP) mode. The section from point 5 to 7 indicates the heat recovery section either through the air pre-heating in the recuperator which results in the increase in the electrical efficiency or the heat recovery in terms of thermal energy. W_{in} represents the power requirement of the compressor and W_{out} indicates the power produced by the turbine to run the compressor or the generator.

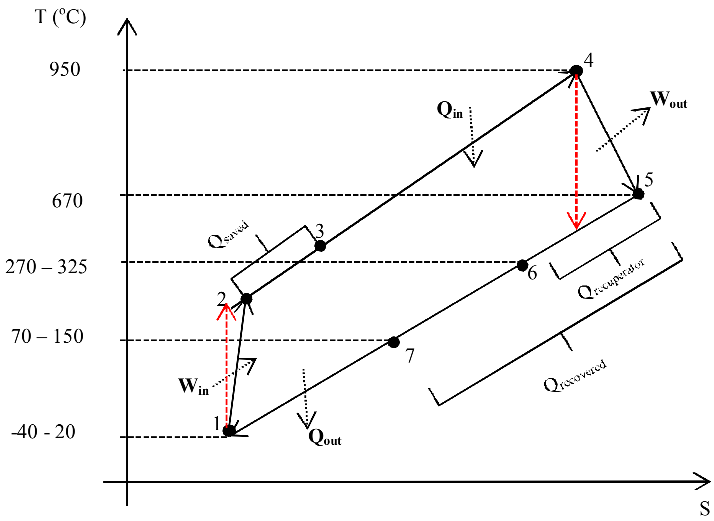


Figure 2: A typical T-S diagram for an actual micro gas turbine.

2.3. Micro Gas Turbine

In this paper, a Turbec T100 micro gas turbine is studied, which is a combined heat and power unit. It is a $100kW_e$ micro gas turbine consisting of a centrifugal compressor, radial turbine and high speed generator, which all are mounted on the same shaft [24]. In the combustor, natural gas is burnt and the hot exhaust gases are expanded through the turbine. The recuperator is used to preheat the compressed air using these hot exhaust gases before injecting it to the combustor, thus increasing the efficiency of the turbine. The outlet of the gas side of the recuperator still has sufficient energy to generate ~150 kW_{th} of heat through the gas-water heat exchanger. The MGT components are shown in Figure 3.

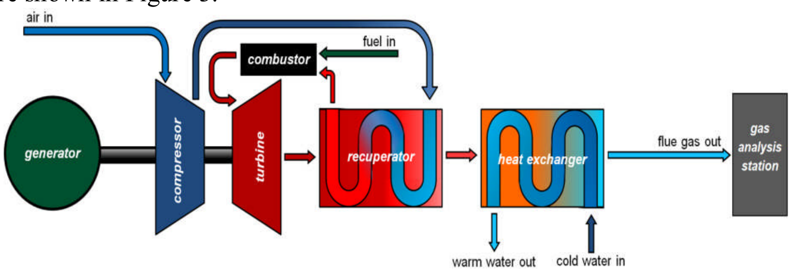


Figure 3: Schematic of micro gas turbine.

The MGT along with pilot scale amine capture plant are located at the UKCCSRC National Pilot-scale Advanced Capture Technology (PACT) facility in Sheffield, UK [25]. The MGT with the recuperator and the gas-water heat exchanger for heat and power generation is modelled and simulated using Aspen HYSYS® V8.4. The major components of the model include the compressor, turbine, reactor/combustor, two heat exchangers, mixer and splitter. The chemical equilibrium in the combustor is estimated through the minimization of the total Gibbs energy.

The thermodynamic property package selected in the model is the Peng-Robinson equation of state. The number of species involved in the simulation is approximately 20 and they include minor species, such as carbon monoxide and sulphur dioxide emissions, as well as nitrogen-based species, such as nitrogen dioxide, nitrous oxide and nitric oxide. The steady-state base case model was developed, and then the simulations were performed for various power outputs (50-100 kW_e) to represent different operational modes. The simulated turbine outlet temperature and the combustion gas pollutant concentration values showed a reasonable consistency with the experimental results obtained at the PACT Core Facility.

The base case model was modeled for the power output of 100kW_e at ISO conditions [26] with the constraint of turbine inlet temperature (TIT) and turbine outlet temperature (TOT) within the limits of 950°C and 650°C, respectively. The composition of the natural gas components used for the model development is listed in the Table 1. The results obtained indicate that the MGT model can accurately model the performance of the micro gas turbine.

Table 1. Natural gas composition.

Component	Mole fraction
CH_4	0.906
C_2H_6	0.051
C_3H_8	0.013
i-C_4H_{10}	0.002
n-C_4H_{10}	0.002
N_2	0.011
CO_2	0.014

2.4. Exhaust gas recirculation

The outlet concentration of the CO_2 from the MGT is much leaner than an industrial or commercial-scale gas turbine, and will cause a performance penalty when integrated with the carbon capture plant due to energy demand in the carbon capture plant which decreases the net output from the system. The CO_2 from an industrial or commercial scale gas turbine is in the range of 3.8 to 4.4 mol% [12-14, 17] while it is only in the range of 1.6 to 1.8 mol% from MGT. Different options are investigated in the literature to offset this penalty [10, 13]. In the MGT model, the exhaust gas is split: for one part the water is condensed and removed, and the rest is re-circulated to the MGT inlet. The additional equipment includes the splitter to divide the exhaust gas into two streams, a condenser for cooling and drying the recycle stream to the required level, and booster fan for the recirculation from the condensing pressure back to ambient at the compressor inlet. The amount of the exhaust gas recirculated is defined by the following relation:

$$\text{EGR ratio} = \frac{\text{Volume flow of reciculated exhaust gas}}{\text{Volume flow of exhaust gas}} \tag{6}$$

Due to the consistency and robustness of the steady state model developed for the base case MGT cycle, it is further extended to include the exhaust gas recirculation mode, to study the effect of enrichment with carbon dioxide. The model is evaluated in EGR mode and the flue gas shows carbon dioxide enrichment with a decreasing trend in the oxygen concentration in the flue gas. The simulation studies provided detailed information on the impact of varying the EGR ratio on the gas turbine output and CO_2 capture plant efficiency.

2.5. Amine capture plant

The MGT cycle can be integrated with the amine capture plant (ACP) available at the PACT Core Facility [25]. This integration will yield a better insight into the performance of the post-combustion capture plant when integrated with the gas fired power generation system. It uses monoethanolamine as the solvent and is designed to remove 1 ton/day of CO_2. It consists of an absorber, stripper, water wash column, cross heat exchanger, solvent tanks for spent and fresh solvents, reboiler and condenser for the stripper column. Also, the ACP is equipped with a flue gas desulphurization unit. After desulphurization, the flue gas enters the bottom of the absorber in which it is contacted with the amine solution. The treated gas from the absorber top enters the water wash section to remove traces of the amine solvent carried by the treated gas before being released to the atmosphere. The rich amine solution containing CO_2 then passes through the cross heat exchanger into the stripper column for solvent regeneration, releasing a high concentration CO_2 stream at the top of the stripper. Further, the regenerated solvent after make-up is recirculated back to the top of the absorber.

The amine capture plant model was developed in Aspen HYSYS® V8.4 by employing the new Acid Gas property package along with the packed column. The Acid Gas property package uses rate-based calculations to give better results compared to the previous Amine package. It is observed that the major penalty in efficiency is caused by the reboiler duty of the stripper. The energy demand for the CO_2 capture decreases with an increase in the concentration of the CO_2 in the flue gas, which results in a higher partial pressure of the CO_2 to improve the driving force and lead to a lower reboiler duty. Also, since the mass flow of the flue gas decreases due to the EGR equipped MGT cycle, this reduces the energy penalty caused by the ACP due to the lower pumping requirements for the solvent. The model of the MGT with EGR integrated with ACP is shown in the Figure 4.

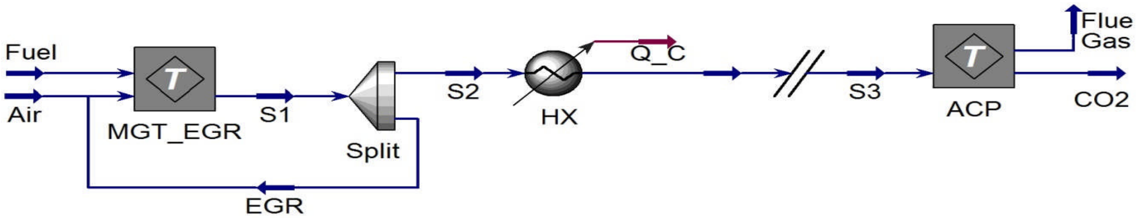


Figure 4: Schematic of the MGT with EGR model integrated with ACP model.

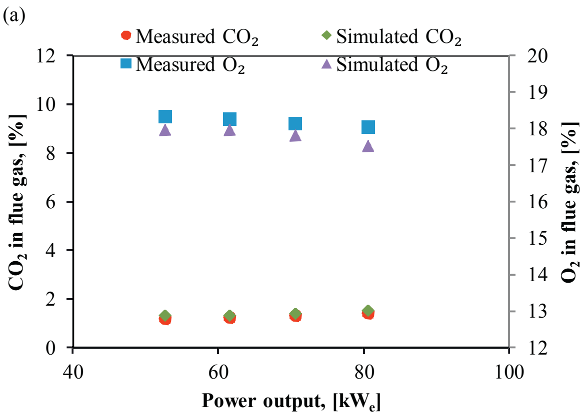
3. Results and Discussion

The base case model is developed for an electric power output of 100 kW_e, with a thermal output of 153.3 kW_{th}. The base case model is developed at ISO conditions [26] and it is in quite good agreement with the reference data [24] and the errors are minimum. Table 2 lists the base case performance of the MGT cycle at ISO condition.

Table 2. Base case performance at ISO [26] condition.

Parameter	Value
Electrical power output (kW_e)	100
Thermal output (kW_{th})	153.3
Electrical efficiency (%)	32.1
Pressure ratio	4.5
Turbine inlet temperature (°C)	945
Turbine outlet temperature (°C)	644
CO_2 molar composition in flue gas (%)	1.6
O_2 molar composition in flue gas (%)	17.3

To check the robustness of the model, the electrical power output is varied from 50 to 80 kW_e to represent different operational modes and the simulations are performed for each case. There is good consistency in the experimental and simulated values for each power output. The measured and simulated CO_2 and O_2 molar composition in the flue gas at different power outputs are illustrated in Figure 5 (a), while the measured and simulated TOT are shown in Figure 5 (b). The validation indicates that the MGT cycle model is sufficiently robust to estimate the system performance with high accuracy. Since the flue gas from the MGT is very lean, it will increase the penalty caused by the carbon capture system. In order to avoid this penalty, the CO_2 concentration in the flue gas could be enhanced, and one option is EGR.



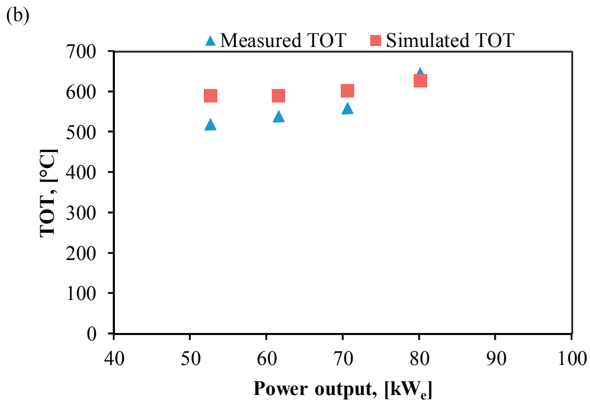


Figure 5. Measured versus simulated data for (a) CO_2 and O_2 molar composition in flue gas; and (b) TOT.

3.1. Effect of EGR ratio

EGR is a preferable option in order to decrease the penalty caused by the carbon capture system, as it increases the CO_2 concentration and reduces the mass flow of the flue gas entering the absorber column. The reduced mass flow towards the amine capture plant will allow the absorber and stripper columns to be of smaller sizes, that is, a reduced capital investment. In addition, the increased CO_2 concentration will reduce the reboiler duty, thus reducing the operating cost.

Table 3. Thermodynamic properties of the fluid at different locations in the MGT and EGR cycles.

	Mass density	Heat capacity	$\gamma = c_p/c_v$
Units	kg/m^3	kJ/mol K	-
		MGT Cycle	
Compressor inlet	1.21	29.1	1.403
Compressor outlet	3.18	30.4	1.381
Turbine inlet	1.27	34.6	1.317
Turbine outlet	0.376	33.2	1.335
		EGR Cycle at 55% EGR ratio	
Compressor inlet	1.15	29.5	1.397
Compressor outlet	3.05	30.9	1.373
Turbine inlet	1.26	35.2	1.310
Turbine outlet	0.372	33.7	1.328

The performance of the EGR cycle is evaluated by varying the EGR ratio from 0 to 80%. It is observed that with an increase in the EGR ratio, the CO_2 content in the flue gas increases while the O_2 decreases, as shown in Figure 6 (a). This results in O_2 starvation at the combustor inlet at higher EGR ratios as shown in the Figure 6 (b). It is found in the literature that for efficient and complete combustion, the O_2 composition at the combustor inlet should be 16 mol% or higher [3-5]. If the molar composition of the O_2 at the combustor inlet falls below 16% then such O_2 starved air will negatively affect the combustion efficiency, stability, kinetics and emissions. Although the O_2 concentration at the combustor inlet can be dropped to as low as 14 mol%, it is not recommended because the unburnt hydrocarbons and CO emissions are much higher [4]. Therefore, following the limit of 16 mol% O_2 concentration, from Figure 6 (b) EGR ratios of ≤ 0.55 are recommended, and this results in the CO_2 enrichment from 1.6 mol% in the base case MGT cycle (no recycle) to 3.7 mol% in the EGR cycle. There is a small decrease in efficiency from 32.1% in the MGT cycle to 29% at 55% EGR ratio, due to the blower power in recirculating the exhaust gas from the condenser pressure back to ambient, and due to the changes in the fluid thermodynamic properties which may affect the compressor and turbine operation. The changes in thermodynamic properties of the fluid are listed in Table 3, and although small changes, along with the blower power, will affect the efficiency of the EGR cycle.

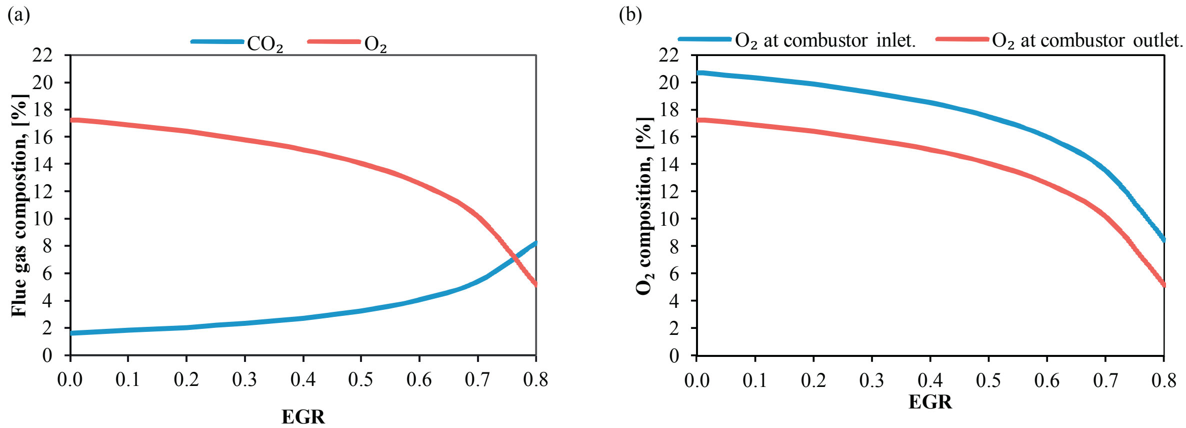


Figure 6. Effect of EGR ratio on the (a) CO_2 and O_2 molar composition in flue gas; and (b) O_2 molar composition at the combustor inlet and outlet.

4. Conclusion

The thermodynamic and simulation studies provide detailed information on the MGT performance along with the impact of varying the EGR ratio on the gas turbine output and CO_2 capture plant efficiency. The results of the recuperated micro gas turbine steady state model and its modification into EGR mode provide a useful assessment of the system. The detailed thermodynamic analysis, modelling and simulation results are an accurate demonstration of the micro gas turbine with exhaust gas recirculation.

The thermodynamics of the recuperated combined heat and power unit provides useful details for the system behavior during modelling. The validated model of the MGT cycle indicates that the MGT model can calculate the performance of the system with high accuracy and an EGR ratio of 55% can enhance the CO_2 content in the flue gas from 1.6 mol% to 3.7 mol%. The results show a considerable improvement due to the presence of an EGR cycle and its integration with the CO_2 capture plant with reduced penalty and cost. It is concluded that the MGT cycle with EGR will be a useful integration with the amine capture plant with a smaller penalty.

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