Research Article

Thermodynamic analysis of the Allam cycle and its pressure sensitivity

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ABSTRACT

The Allam cycle is a novel system which operates the oxy-combustor at a high pressure under supercritical conditions, uses a single gas turbine, circulates carbon dioxide as the working fluid in a semiclosed-loop and utilizes low-pressure-ratio recuperated Brayton cycle. These properties of the Allam Cycle enable to reach high efficiencies. This system was simulated using Chemcad software for a methane feed flow rate of 1 kmol/s. The pressure at the inlet of turbine was taken as 285 bar. The net power generation rate was estimated considering the generation in the turbine and the usage in the compressors for circulating carbon dioxide and for oxygen and natural gas feeds. The calculations using energy balance for the process gave 388 MW net power output, whilst Chemcad software gave 392 MW. The thermal energy recovered by lowering the temperature of the flue gas before condensing its water content was also considered in the overall efficiency of the system. The net power cycle efficiency was determined as 48.89%. The second law analysis of the cycle was also made. Entropy generation rate, S_{gen}, was determined as 965.79 kW/K and exergy destroyed, $\psi_{destroyed}$, was found as 287.81 MW from entropy balance for the process. The exergy balance for the process was also made and the results were compared with the findings from the entropy balance. The second law efficiency of the process was found as 62.54%.

Working at high pressure, naturally, affects the material of construction and consequently the fixed capital investment as well as operating and maintenance costs. Therefore, in this work, a sensitivity analysis is also made to see the effect of pressure on power generation and efficiency. The sensitivity analysis was made using Chemcad software simulation. It was found that the optimum pressure range for operation of the system was between 250 and 350 bar.

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INTRODUCTION

Carbon dioxide is the principle greenhouse gas responsible from global warming. Reduction its emission to atmosphere is of great importance in sustainable development.

As the major source of carbon dioxide emission is due to power plants, emission control from power plant stacks has gained enormous attention. Addition of carbon dioxide capturing units to the conventional power plants becomes

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essential in addition to other filters used to minimize polluting toxic gases such as sulfur dioxide and nitric oxides as well as particulates. This brings additional cost in power generation. The additional CO_2 removal systems can increase the cost of electricity by 50% to 70%. High capital investment and operating costs, high parasitic energy requirements, and additional environmental concerns over scrubbing chemicals have combined to question the sustainability of CO_2 capture systems for fossil fuel power plants. Alternatives were then considered to develop economical means to mitigate carbon dioxide emission. Oxy-combustion is one of these alternatives which is based on using oxygen instead of air in combustors. This eliminates nitric oxide emission considerably when fossil fuel is used, flue gas cleaning becomes much easier and carbon dioxide emission control becomes quite feasible. Although, air separation unit is an additional cost, the reductions in flue gas handling cost validates its usability.

Oxy-combustion has been studied and applied to conventional coal-fired boilers, combined cycles, and high pressure steam-based systems [1, 2]. It is then realized that instead of using steam-based process, use of hot combustion gas at high temperatures directly in turbine eliminates the energy losses that steam-based cycles encounter due to the heat of vaporization and condensation, and improves the efficiency appreciably. Material of construction, on the other hand, limits the maximum temperature that a turbine can handle. This brings in the necessity of diluting combustion gas with carbon dioxide itself. The combustion of hydrocarbons gives carbon dioxide and water. Thus, the natural choice for dilution component

is either water or carbon dioxide. This choice affects the working fluid, i.e. either H_2O rich or CO_2 rich streams. Currently, CO_2 seems to be widely accepted as diluent. Water can be separated from the combustion gas very easily by condensing it. The remaining component in the combustion gas is essentially carbon dioxide which can effectively be recycled to the combustor to adjust the thermal and hydrodynamic conditions to the optimum values and limit the exit temperature of the combustion gas to acceptable values for the turbine. As high pressure and high temperature improve the efficiency of a turbine, the further step to reach high overall efficiency was proposed by Allam et al. [3, 4] to operate the process with carbon dioxide at supercritical conditions.

The Allam cycle is a novel system which was originally presented in Kyoto at GHGT-11 [2]. It operates the oxy-combustor at a high pressure under supercritical conditions, uses a single gas turbine, circulates carbon dioxide as the working fluid in a semiclosed-loop and utilizes low-pressure-ratio recuperated Brayton cycle. Of course, another advantage is that the Allam Cycle can run substantially water free [5, 6].

ALLAM CYCLE DESCRIPTION

The basic unit operations of the Allam Cycle using natural gas fuel are shown in Figure 1. The cycle operates with a single turbine with an inlet pressure of about 300 bar and a pressure ratio of 10. Compressed gas fuel, pure oxygen provided by an Air Separation Unit (ASU), and a hot CO_2 diluent recycle stream at about 300 bar are fed to the combustor.
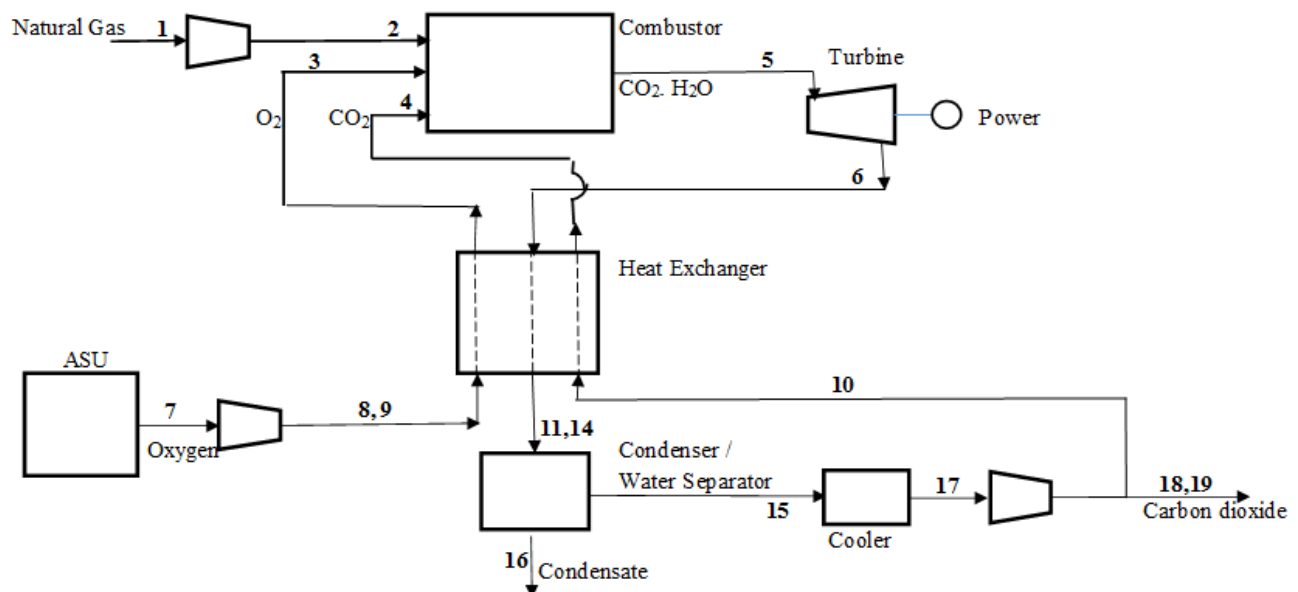

Figure 1. Natural gas allam cycle.

The exhaust stream from the burner is expanded through a turbine to about 30 bar, which drops over 700° C. After the turbine, the exhaust flow is cooled in a recuperator, which transfers the heat from the hot exhaust stream to the high pressure CO_2 recycling stream. This recycle stream of CO_2 serves as a diluting agent in the combustion chamber so that the temperature of the exhaust gas from the combustor is lowered to an acceptable level between 1150–1200°C before its entry to the turbine. The turbine exhaust gas is further cooled close to ambient temperature and combustion water is separated. The remaining component in the turbine exhaust gas is thus essentially only carbon dioxide. The amount equivalent to stoichiometric yield from combustion is removed from the system for sequestration. The remaining part is then recompressed to a pressure of about 300 bar and sent to the recuperator to increase its temperature to about 700°C before introduced to the combustor. The net export of CO_2 is about 5% of the total recycling flow, i.e. most of the process inventory is recirculated [6, 7].

SIMULATION

The Allam Cycle is simulated using Chemcad software in this work (Figure 2). Recently, power generation using natural gas has attracted more attention as it has lower specific CO2 emissions than coal fired power generation [8]. Therefore, methane was chosen as the fuel and its flow rate was taken as 1 kmol/s as a typical value for such a case study. The combustor was operated adiabatically at 285 bar. Methane anf oxygen feeds as well as recycling CO_2 were compressed to this pressure before introduced to the combustor. For one mole of CO_2 leaving the process, 27 moles of CO_2 were recycled as diluent so that CO_2 concentration at the combustor exit was 90 mole-% and the temperature of the combustion gases at the turbine entrance was 1170°C. The pressure at the turbine exit was 24 bar and the pressure ratio for the turbine was 8.38. The temperature at the turbine exit was 799°C. The recuparator was used to heat the O_2 feed and recycling CO_2 streams to 720°C before
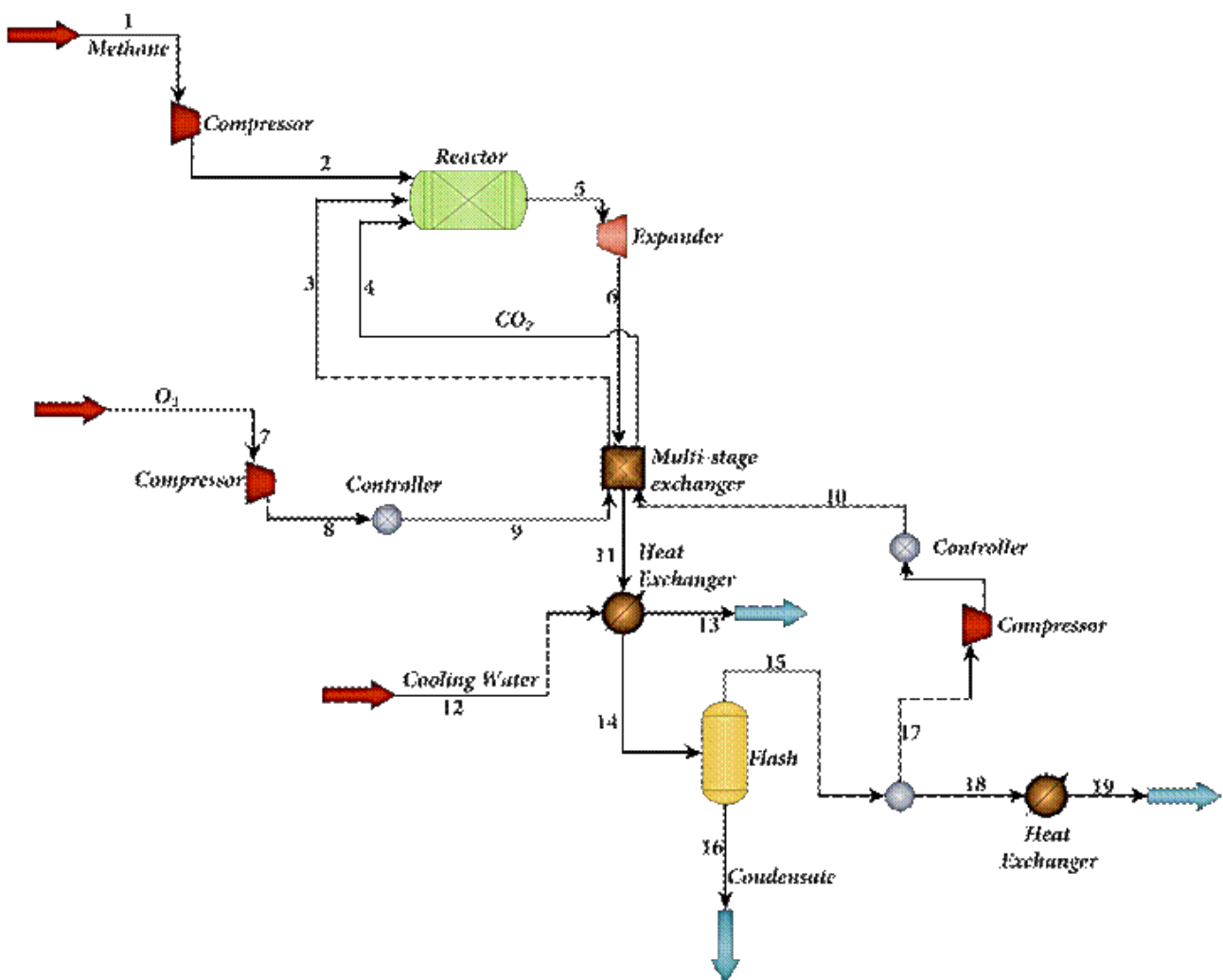

Figure 2. Simulation of the allam cycle.

Table 1. Properties

Reactants	MW	LHV, kJ/kmol	HHV, kJ/kmol	h_{fo}, kJ/kmol	g_{fo}, kJ/kmol	s_o, kJ/kmol × K	Exo-ch, kJ/kmol	C_p, kJ/kmol × K
CH_4	16	802300	890400	−74873	−50768	186.251	824348	35.6155
O_2	32			0		205.148	3951	29.408
CO_2	44			−393522	−394389	213.795	14176	37.136
H_2O (liq)	18			−285830	−237141	69.95	45	75.24

fed to the combustor. The temperature of the combustion gas leaving the recuperator was 353°C. In order to utilize its energy, a second heat exchanger was used to produce superheated steam at 7 bar and 233°C at the rate of 175 kg/s. The superheated steam thus produced can either be used as thermal energy elsewhere or can be used in a steam turbine to produce electricity. The later case makes the Allam Cycle a semi-closed oxy-combustion combined cycle. Water condensed in the second heat exchanger is then separated at 60°C and ca 34 bar. One kmol CO_2 per second was taken away as the product of the process from the vapor stream at ca 34 bar after cooled down to 55°C. The remaining part of CO_2 was recycled.

RESULTS AND DISCUSSION

Table 1 summarizes the properties of the species involved in the process, Table 2 the ambient conditions and Table 3 the overall material balance for the process.

Performance
Energy Balance

Steady state energy balance for an open flow system with chemical reaction (combustion) is

$$\left[\left(H + E_K + E_P\right)m\right]_{in} - \left[\left(H + E_K + E_P\right)m\right]_{out}$$
$$+ Q + W_s + \left\{\begin{array}{c} Energy\ generated \\ with\ chemical\ reaction \end{array}\right\} = 0 \quad (1)$$

It should be noted that Q is considered positive when added to the system and W is taken as positive when work is done on the system. Neglecting the changes in kinetic and potential energies between inlet and outlet of the system,

$$\left[Hm\right]_{in} - \left[Hm\right]_{out} + Q + W_s + \left\{\begin{array}{c} Energy\ generated \\ with\ chemical\ reaction \end{array}\right\} = 0 \quad (2)$$

For methane (M) combustion,

$$\frac{Energy\ generated}{with\ chemical\ reaction} = m_{M,in}\left(-\Delta H_{r,M}\right) \quad (3)$$

Table 2. Ambient conditions

Ambient pressure	1 bar
Ambient dry bulb temperature (summer)	25°C
Cooling water temperature	25°C

Table 3. Overall material balance for the process

	kmol/s		kg/s	
	Input	Output	Input	Output
Methane	1.000	0.000	16.043	0.000
Oxygen	2.000	0.000	63.997	0.000
Carbon Dioxide	0.000	1.000	0.000	44.010
Water	9.714	11.714	175.000	211.030
Total	12.714	12.714	255.040	255.040

Therefore, energy equation becomes

$$\left[H_m\right]_{in} - \left[Hm\right]_{out} + Q + W_s + m_{M,in}\left(-\Delta H_{r,M}\right) = 0 \quad (4)$$

Heat generated with combustion can be evaluated using the lower heating value (LHV) of methane.

$$-\Delta H_{r,M} = -\Delta H_{r,M}^o = 50\frac{MJ}{kg} = 802.3\,MJ\,/\,kmol \quad (5)$$

Overall enthalpy balance for the process is summarized in Table 4.

Neglecting the heat effects in the compressors and turbine, for 1 kmol/s (16.043 kg/s) methane feed rate, the energy equation can be written numerically as

$$[-427.705 - (-13.828)] - 0.288 + W_{s,net,theo} + 802.3 = 0 \quad (6)$$

$$-W_{s,net,theo} = 388.135\,MW \quad (7)$$

This result is in very good agreement with 392.17 MW obtained from the simulation made by Chemcad software (Table 5). The power requirements of the compressors used in the cycle and the power generated with the turbine are also calculated with Chemcad software. The efficiencies for the turbine and all the compressors were assumed as 95%. The results are tabulated in Table 5.

Table 4. Overall enthalpy balance for the process (Enthalpy = 0 for gaseous state at T_{ref} = 25°C)

Inlet streams	Enthalpy flow, MW	Outlet streams	Enthalpy flow, MW
Methane	0	Carbon dioxide	1.114
Oxygen	0	Condensate	−82.718
Cooling water	−427.705	Steam	67.776
Total	−427.705		−13.828
Net enthalpy flow		−413.877	
		Total cooling (Heat recovered from carbon dioxide output stream)	−0.288

Table 5. Input and output power for the process

Input power	Power, MW	Output power	Power, MW
Methane compressor	6.42	Gas turbine	643.73
Oxygen compressor	47.81		
Recycle carbon dioxide compressor	197.33		
Total	251.56		643.73
Net power output		392.17	

Using net power generation and basing on lower heating value (LHV) of methane, the energy efficiency of the cycle can be calculated as

$$\eta_{net} = \frac{Net\, W_s}{m_{M,in}\left(-\Delta H_{r,M}\right)}100 = \frac{392.17}{(16.043)(50)}100 = 48.89\,\% \quad (8)$$

This result agrees very well with 47% reported by Laumb et al. [9] and is close to 54.8% found by Scaccabarozzi et al. [10] employing simulation by Aspen-Plus.

The second heat exchanger used after the recuperator definitely improves overall performance.

$$\text{Thermal energy production as steam =}\\ [64.946 - (-427.35)] = 492.3\, \text{MW} \quad (9)$$

One may note that this is as important as the output power obtained from gas turbine.

Second Law-Entropy Analysis

Second law analysis of the Allam cycle was also made. For gases going from (P_1, T_1, V_1) state to (P_2, T_2, V_2) state, entropy change is given as

$$\Delta S = C_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \quad (10)$$

Where,
S = kJ/kmol · K
C_p = kJ/kmol · K
R = 8.314 kJ/kmol · K

For liquids going from (T_1) state to (T_2) state, entropy change is

$$\Delta S = C_p \ln\frac{T_2}{T_1} \quad (11)$$

These equations were used to make necessary corrections for the entropy values taken from tables. The entropy generated can be calculated by the following expression, which can easily be obtained from the entropy balance for a flow process at steady state [11, 12];

$$S_{gen} = S_{outlet} - S_{inlet} - \frac{Q}{T_{surr}} \geq 0 \quad (12)$$

and the entropy destroyed is given as

$$\psi_{destroyed} = W\text{rev} - W_{act} = T_o S_{gen} \quad (13)$$

The results are summarized in Table 6.

$\psi_{destroyed}$ = 288 MW, in fact, is not lost completely in the process, but 252 MW is used up by the compressors. The rest is due to the heat evolved in compressor and other heat losses.

Exergy analysis

Neglecting potential and kinetic energy terms, the flow thermomechanical exergy, or stream thermomechanical exergy, is defined as

$$\psi^{tm} = (h - h_0) - T_0(s - s_0) \quad (14)$$

Chemical exergy of a gas stream consisting of n number of components is defined as

$$\psi^{ch} = \sum_{k=1}^{n} x_k \psi_k^{0,ch} + RT_o \sum_{k=1}^{n} x_k \ln x_k \quad (15)$$

Total exergy is the sum of thermomechanical and chemical exergies.

$$\psi = \psi^{tm} + \psi^{ch} \quad (16)$$

$$\psi = (h - h_0) - T_0(s - s_0) + \sum_{k=1}^{n} x_k \psi_k^{0,ch} + RT_o \sum_{k=1}^{n} x_k \ln x_k \quad (17)$$

For streams consisting of one pure component;

$$\psi^{ch} = \psi^{0,ch} \quad (18)$$

$$\psi = (h - h_0) - T_0(s - s_0) + \psi^{0,ch} \quad (19)$$

Table 6. Entropy balance for the process* (T_{ref} = 25°C, P_{ref} = 1 bar)

Inlet				Outlet			
	N,	**s,**	**N s, kW/K**		**N,**	**s,**	**N s, kW/K**
	kmol/s	**kJ/kmol · K**			**kmol/s**	**kJ/kmol · K**	
Chemical Process							
Methane	1	155.582	155.582	Carbon dioxide	1	188.039	188.039
Oxygen	2	196.014	392.028	Condensate	2	78.305	156.610
Utilities							
Cooling water	9.7222	6.534	63.53	Steam	9.7222	126.65	1231.32
Total S, kW/K		611.14				1575.96	
$S_{out} - S_{in}$, kW/K				964.82			
Q/T_{surr}, kW/K				−288/298 = −0.97 (negligible)			
S_{gen}, kW/K				965.79			
$\psi_{destroyed}$, MW				(298)(965.79)/1000 = 287.81 MW/kmol CH_4			

*Water and steam properties are taken from steam tables. Properties of other species are taken from "Yüncü, H., Ekserji Analizi, 2010 [13]".

Chemical exergies are not needed to be considered for utilities which do not undergo any chemical change in a process. It is enough to consider thermomechanical exergies for utilities. The thermomechanical flow exergy for unit molar flow of the utilities, i.e liquid water and steam, were calculated by the following expressions as recommended by Yüncü [13, 14] and the properties are taken from thermodynamic tables;

$$\psi^{tm} = h_i(T,P) - h_o(T_o,P_o) - T_o[s(T,P) - s_o(T_o,P_o)] \quad (20)$$

For liquid water at 7 bar and 25°C,

$$\psi^{tm} = (111 - 105) - 298[0.363 - 0.3673]$$
$$= 7.45 \frac{kJ}{kg} \equiv 0.134 \, MJ / kmol \quad (21)$$

For steam at 7 bar and 233°C,

$$\psi^{tm} = (2918 - 105) - 298[7.0363 - 0.3673]$$
$$= 825.64 \frac{kJ}{kg} \equiv 14.861 \, MJ / kmol \quad (22)$$

The exergy balance for a flow process at steady state can be written as

$$\sum_{in} m\psi + \sum\left(1 - \frac{T_0}{T}\right)Q = \sum_{out} m\psi - W_s + \psi_{destroyed} \quad (23)$$

One may note that the difference between the exergy flows of feed streams and products gives the ideal work produced for a reversible work-producing process.

$$W_{ideal} = \sum_{in} m\psi - \sum_{out} m\psi \quad (24)$$

$\psi_{destroyed}$ is also referred to as irreversibility rate or lost work (W_{lost}) and may also be shown by I.

Thus, the equivalent expression of the exergy balance for work producing process can also be written as

$$\left|W_{ideal}\right| + \sum\left(1 - \frac{T_0}{T}\right)Q - \left|W_s\right| = W_{lost} \quad (25)$$

Stream exergies for input and output streams of Allam cycle are summarized in Table 7.

In the process, stream exergy should decrease. Indeed,

$$\sum_{out} m\psi - \sum_{in} m\psi = 79.839 - 848.137 \cong -768.3 \, MW \quad (26)$$

This shows that the process conforms to the thermodynamic rules and, furthermore, ideal work that can be produced is

$$W_{ideal} = 768.3 \, MW \quad (27)$$

Exergy destroyed calculated from the exergy balance is different from the one calculated from $\psi_{destroyed} = T_o S_{gen}$. The difference is about 30%. This may be because of the difficulties and uncertainties involved in estimation of parameters used in exergy calculations.

The uncertainties involved can be shown by a simple calculation. If flow exergies of utilities (cooling water and steam) are not included in the exergy balance but instead the exergy generated by the heat exchanger used to generate steam from cooling water is included as $\left[\left(1 - \frac{T_0}{T}\right)Q\right]$, the choice for T will have a profound effect on the final result.

Table 7. Flow exergies for the process

Inlet streams	N, kmol/s	ψ, MJ/kmol	$N\psi$, MW	Outlet streams	N, kmol/s	ψ, MJ/kmol	$N\psi$, MW
Methane	1	833.487	833.487	Carbon dioxide	1	22.965	22.965
Oxygen	2	6.673	13.346	Condensate	2	–43.804	–87.608
Cooling water	9.7222	0.134	1.304	Steam	9.7222	14.861	144.482
$\sum_{in} m\psi$, MW		848.137		$\sum_{out} m\psi$, MW		79.839	
$\left(1-\dfrac{T_0}{T}\right)Q$, MW	$\left(1-\dfrac{298}{60+273}\right)(-0.288)=-0.0302$			$-W_s$, MW	(Calculated from energy balance) 388.135		
				$\psi_{destroyed}$, MW	(Calculated from the balance) **380.13**		
TOTAL, MW		848.107		TOTAL, MW		848.107	

Table 8. Flow exergies for the process-alternative

Inlet streams	N, kmol/s	ψ, MJ/kmol	$N\psi$, MW	Outlet streams	N, kmol/s	ψ, MJ/kmol	$N\psi$, MW
Methane	1	833.487	833.487	Carbon dioxide	1	22.965	22.965
Oxygen	2	6.673	13.346	Condensate	2	–43.804	–87.608
$\sum_{in} m\psi$, MW		846.833		$\sum_{out} m\psi$, MW		–64.643	
$\left(1-\dfrac{T_0}{T}\right)Q$, MW	$\left(1-\dfrac{298}{60+273}\right)(-0.288)=-0.0302$			$-W_s$, MW	(Calculated from energy balance) 388.135		
$\left(1-\dfrac{T_0}{T}\right)Q_{HE}$, MW	$\left(1-\dfrac{298}{\dfrac{353+60}{2}+273}\right)(-495.481)=-187.549$			$\psi_{destroyed}$, MW	(Calculated from the balance) **335.76**		
TOTAL, MW	659.25			TOTAL, MW		659.25	

T is the temperature of the system boundary at the point of heat exchange. The process stream enters the heat exchanger at 353ºC (626 K) and leaves at 60ºC (333 K). If arithmetic average is taken, T would be 479.5 K. If geometric average is taken, T would be 456.6 K. If logarithmic mean is taken, T would be 464.2 K.

Noting, from enthalpy balance,

$$Q_{HE} = 67.776 - (-427.705) = 495.481 \; MW \quad (28)$$

and taking the arithmetic average, the exergy balance would be as follows;

$\psi_{destroyed}$ is 335.76 MW in this case. If geometric mean is used $\psi_{destroyed}$ will be 351.21 MW and if logarithmic mean is used will be 345.91 MW. These differences in calculations of $\psi_{destroyed}$ from exergy balance leads us to the conclusion that $\psi_{destroyed}$, estimated from entropy generation, seems to be more dependable. Using this value, the second law efficiency can be calculated from [11–16]

$$\eta_{II} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy destroyed}}{\text{Exergy supplied}} \quad (29)$$

$$\eta_{II} = 1 - \frac{\psi_{destroyed}}{\sum_{in} m\psi - \sum_{out} m\psi} \quad (30)$$

Exergy supplied is,

$$\left[\sum_{in} m\psi - \sum_{out} m\psi\right] = 848.137 - 79.839 \cong 768.3 \, MW \quad (31)$$

$$\eta_{II} = 1 - \frac{287.81}{768.3} = 0.6254 \equiv 62.54\% \quad (32)$$

The second law efficiency thus calculated is greater than 48.89% energy efficiency of the cycle as expected. This is in good agreement with similar values reported in literature [7, 16].

Sensitivity to Pressure

Critical temperature and critical pressure for carbon dioxide are 31.1ºC and 73.9 bar, respectively. As a whole, the Allam cycle operates in trans-critical regime. But the
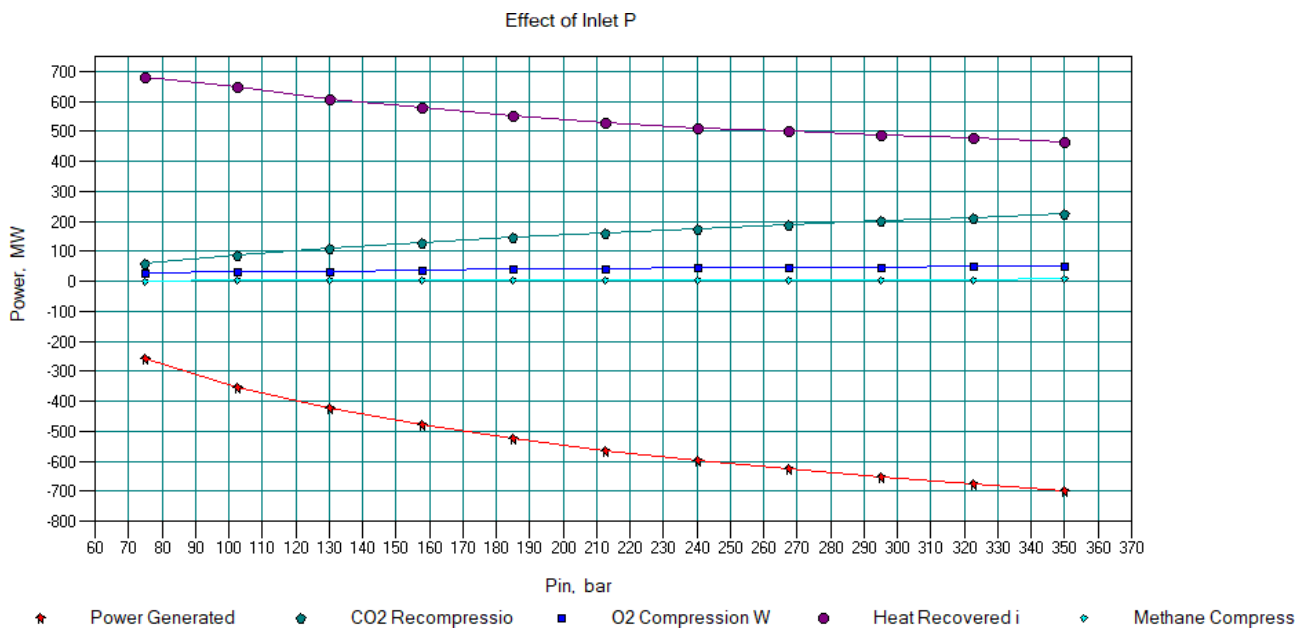
Figure 3. Sensitivity of power generated to pressure.

combustor and the turbine entrance is at supercritical region, i.e. pressure needs to be above 73.9 bar and temperature must be kept above 31.1°C. Therefore, in this work, a sensitivity analysis is also made to see the effect of pressure on power generation. The sensitivity analysis was made using Chemcad software simulation by changing the pressure at inlet of the turbine from 75 to 350 bar (Figure 3). Higher the pressure at inlet of the turbine, higher is the power generated. This is clearly shown in the figure. But, the total power requirement by the compressors of inlet streams, i.e. methane and oxygen, and recycling carbon dioxide also increases. The net power is, however, not very much affected in the range between ca 250 and 350 bar. It is recommended to run the Allam cycle with a single turbine that has an inlet pressure of approximately 300 bar and a pressure ratio of 10 [5]. The sensitivity analysis results given in Figure 3 confirm this and suggest that the optimum pressure range for operation of the system can be between ca 250 and 350 bar. Higher pressure would increase fixed capital investment as well as operating and maintenance costs very much.

It should be noted that the gas turbine is the heart of the cycle and dictates tailoring the rest of the process. Commercial gas turbines run safely at temperatures as high as 1300°C. The highest inlet pressure achieved to date is around 310 bar for safe operation [3, 17]. These limits as well as the sensitivity analysis results were considered in this work to choose the pressure at turbine inlet as 285 bar and at outlet as 34 bar, and the carbon dioxide recycling rate such that the temperature at turbine outlet is 1170°C.

CONCLUSIONS

Simulation and thermodynamic analysis of the Allam Cycle were made for a typical feed rate of 1 kmol/s of methane as the fuel. Net power output was found as 388 MW. The net power cycle efficiency was determined as 49%. The second law analysis yielded entropy generation rate as 966 W/K and 288 MW exergy destroyed. The second law efficiency was 62.5%. Sensitivity analysis for power generation to pressure indicated that pressure between 250–350 bar at the turbine entrance would be suitable. Recycling rate of CO_2 is very important to affect the temperature of combustion gases and 90% CO_2 in reactants' mixture is reasonable to limit the temperature at turbine entry to about 1170°C. Furthermore, addition of a second heat exchanger after the recuperator improves overall performance and total energy recovery.

NOMENCLATURE

C_p	Heat capacity [kJ/kmol×K]
E_k	Kinetic energy [kJ/kg]
E_p	Potential energy [kJ/kg]
H	Enthalpy [kJ/kg]
$(-\Delta H_r)$	Heat of reaction [kJ/kg]
m	Mass flow rate, [kg/s]
Q	Heat added to the system, [kW]
R	Gas constant, 8.314 [kJ/kmol×K]
s	Entropy [kJ/kmol-K]
T	Temperature [K]
W	Work [kW]

h	Energy efficiency[–]
h_{II}	Second law efficiency [–]
ψ	Exergy [kJ/kmol]
ψ^{ch}	Chemical exergy [kJ/kmol]
ψ^{th}	Thermomechanical exergy [kJ/kmol]

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

No new data were created in this study. The published publication includes all graphics collected or developed during the study.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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