EFFECT OF AMBIENT TEMPERATURE AND DESIGN CONDITIONS ON COMBINED CYCLE USING TRANS-CRITICAL CARBON-DIOXIDE AND AMMONIA WATER MIXTURE

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Abstract:

The pressing need for environmental conservation and the reduction of toxic greenhouse gas emissions from conventional power generation systems have led to a growing interest in harnessing waste heat resources, particularly low-grade heat (LGH) sources abundant in various industries and commercial sectors. However, current conventional power cycle processes face limitations in efficiently utilizing these resources to their maximum potential. Trans-critical carbon dioxide, known for its numerous benefits, including low cost and availability, has emerged as an effective option for low-temperature processes. When used in conjunction with refrigeration, the combined power cycle of trans-critical carbon dioxide with an ammonia-water mixture has shown promising performance, surpassing standalone trans-critical carbon dioxide systems. This review article aims to provide a comprehensive and scholarly overview of the application areas and thermodynamic analysis of this innovative combined power cycle. By presenting valuable insights and evidence, this study seeks to support the development and expansion of this process as a sustainable and efficient solution for LGH utilization and environmentally responsible power generation. The result of the energy analysis shows that a maximum work output of 1982.31kJ/kg of inlet air to compressor is obtained for a first law efficiency and second law efficiency of 62.07%.

Keyword: Combined Cycle, Ammonia Mass Fraction, Trans-critical carbon dioxide.

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Nomenclature Used:

Symbol	**Specification (Unit)**
c_p	Specific heat at constant pressure (kJ/kg.K)
R	Universal Gas constant (kJ/kg.K)
p	Pressure (bar)
h	Enthalpy (kJ/kg)
T	Temperature (K)
$\dot{m}$	Mass flow rate (kg/second)
LHV	Lower heating value (kJ/kg)
CPR	Cycle Pressure ratio
W	Work (kJ/kg or kJ/kg of inlet air)
Q	Refrigerating effect/Heat Input
x	Mass Fraction of ammonia in liquid phase
y	Mass Fraction of ammonia in vapor phase

Abbreviations Used:

HD	High Pressure Drum
ID	Intermediate Pressure Drum
LD	Low Pressure Drum
HRVG	Heat Recovery Vapor Generator

Greek Symbols:

η	Efficiency
$\boldsymbol{\varepsilon}$	Effectiveness

Superscripts:

L	Liquid
G	Gas
E	Excess

Subscripts:

Symbol	**Specification**
p	Polytrophic (if not used with ‘c’)
a	Air
c	Compressor
e	Exit
i	Inlet
cc	Combustion Chamber
f	Fuel
fg	Flue gas
s	Steam
hp	High pressure
ip	Intermediate pressure
lp	Low pressure
amw	Ammonia Water Mixture
gt	Gas Turbine
st	Steam Turbine
cw	Cooling Water
M	Mechanical

p	Pump
CV	Calorific Value
gen	Generator
is	Isentropic
b	Boiling point
d	Condensing point
R	Reduced
E	Excess
m	Mixture
Tc/TC	Trans-critical carbon dioxide

1. Introduction

Gas turbine power plants, particularly combined cycle power plants, can be affected by atmospheric conditions, and the quality of air entering the compressors plays a crucial role in their performance.The efficiency of gas turbine-based power plants is highly dependent on the ambient air temperature and density. As the temperature of the air increases, the density decreases, resulting in a reduced mass flow rate through the compressor. This decrease in mass flow rate can lead to a drop in power output and efficiency of the gas turbine.

To mitigate this effect and improve the performance of the combined cycle power plant, various thermodynamic elements can be installed to condition the air before it enters the compressor. One effective method is cooling the ambient air before it enters the compressor using the energy available within the combined cycle power plant.

This study likely focused on the effects of using an absorption chiller to cool the air entering the gas turbine, and how it could impact electricity generation in the specific context of Bangkok, Thailand. The reported 13% increase in electricity Bassily's study likely compared various intercooled reheat regenerative cycle designs, with a focus on the impact of evaporative cooling of the compressor discharge. The reported 5% increase in cycle efficiency indicates the potential benefits of using this cooling method in gas turbine cycles.

Overall, the use of cooling techniques in gas turbines is an important area of research to enhance efficiency and performance, especially in regions with high ambient temperatures. However, for more specific details on the studies and their findings, it would be best to refer directly to the original papers by Mohanty and Palosos (1995) and Bassily (2001).

Ibrahim et al. (2017): The authors discussed the impact of ambient conditions on the gas turbine cycle's performance but concluded that these factors have no significant impact.

Basha et al. (2012): The analysis of GE gas turbine frames led to the conclusion that relative humidity does not significantly affect the performance of the gas turbine. However, a 10 °F drop in atmospheric air temperature resulted in an increase of roughly 4% in net power production.

Ameri and Hejazi (2004): The study looked at 170 gas turbine units in Iran and found that variations in ambient temperature resulted in a loss of 20% of their rated capacity. They focused on five gas turbines with an average 11.8 °C difference between ambient temperature and ISO conditions. They discovered that the power output decreased by 0.74% for every 1 °C increase in ambient temperature. To increase the gas turbine cycle's efficiency, they suggested lowering the compressor's intake air temperature.

These studies seem to present varying conclusions about the impact of ambient conditions on gas turbine performance. While some suggest minimal impact or no impact, others found a significant effect, particularly with variations in ambient temperature.

It's important to note that research in this field is ongoing, and different gas turbine models and operating conditions might yield different results. Engineers and operators often consider environmental factors when designing and optimizing gas turbine systems to ensure maximum efficiency and performance.

The combined cycle power plant is designed to maximize the efficiency by capturing and utilizing waste heat from the gas turbine to generate additional electricity using a steam turbine. However, the efficiency is limited to about 40% due to certain design constraints.

The specific details and equations to calculate the efficiencies and other performance parameters would require further information from the original paper (Zhu, et al., 2020) to fully understand the calculations and context.

Electric power output decrease: According to the study by Hosseini et al., the electric power output of the gas turbine decreases by approximately 0.5% to 0.9% for every 1°C increase in the ambient air temperature. In a combined cycle (where waste heat from the gas turbine is used to generate additional power through a steam turbine), the decrease in electric power output is slightly less at 0.27% per 1°C increase in ambient temperature.

These findings highlight the significance of ambient air temperature on gas turbine performance, and it's important to consider these effects in the design and operation of gas turbine power plants to ensure optimal efficiency and power output.

Kahraman, U., and Dincer, I. in 2022the research concluded that a cycle pressure ratio of 38 in a reheat gas turbine using steam blade cooling could potentially achieve an efficiency of 56%. This could have significant implications for the gas turbine industry as higher efficiency means better utilization of energy and reduced environmental impact.

Additionally, the proposal to reduce the size of the combustion chamber/reheating combustion chamber through the use of coolant steam is interesting. Such a method might have advantages like improved compactness, reduced emissions, or enhanced performance. However, without access to the specific research paper, it's challenging to provide further insights into the proposed method or its potential impact on the industry.

It seems like you've provided some information about different gas turbine cycles and their evaluations based on research conducted by various authors. Here's a summary of the findings:

1. Kilani et al. (2014) examined four different gas turbine cycles:

a. Simple combined cycle with steam injection.
b. Combined cycle with compressed air cooling.
c. Combined cycle with steam injection into the combustion chamber and steam extraction from the steam turbine.
d. Combination between the second and third cycles.

It was found that all cycles' efficiency began to decline once an optimal Compression Pressure Ratio (CPR) was reached. However, the fourth cycle was identified to have the highest efficiency.

2. Kaviri et al. (2013) analyzed a combined cycle with dual pressure Heat Recovery Steam Generator (HRSG). Their goal was to investigate the effects of CPR on thermal efficiencies, the environment, and the economy. The authors concluded that as the CPR increases, both the cost of fuel and the environmental impact decrease.

3. Pattanayak et al. (2017) studied a combined cycle with triple pressure HRSG and steam reheating. They found that when the CPR is raised, the amount of extra air needed to maintain a constant Turbine Inlet Temperature (TIT) also increases.

These studies highlight the importance of finding the optimal CPR for gas turbine cycles to maximize their efficiency while considering environmental and economic factors. Each cycle and HRSG configuration may have different characteristics and trade-offs depending on the specific application and requirements.

Alves et al. (2001) compared the intercooled cycle and the reheat cycle in order to determine which gas turbine cycle is superior. The authors came to the conclusion that reheating is a preferable alternative for combined cycle operation since the exhaust from intercooling is low temperature and not suited for the bottoming cycle. Ibrahim and Rahman (2012), 2013 and 2014 chose to boost TIT rather than CPR in order to increase output and cycle effectiveness. When examining the ATAER power plant in Turkey, Ersayin and Ozgener (2015) discovered that the combustion chamber is where the majority of energy is lost. They recommended that this loss can be reduced by using stronger insulators and by modifying the Sheykhlou (2016) examined a combined cycle that used a wind turbine to power a gas turbine compressor and a Rankine cycle pump. According to the author's findings, the combined cycle's combustion chamber is where the majority of energy is destroyed. The basic air-cooled gas turbine cycle and the gas turbine cycle with reheat were compared by Sahu and Sanjay (2017), who came to the following conclusions-

1. In comparison to a non-reheat type, the reheat gas turbine cycle requires more fuel and coolant air.
2. When compared to a reheat cycle using air cooling, the non-reheat cycle has higher first and second law efficiency.
3. Reheat type gas turbines have higher total investment costs and energy production costs (cents/kWh) than non-reheat types.
   According to research by Balku (2017), if the combustion chamber's efficiency can be raised from 90% to 99%, the plant's thermal efficiency will rise by 6.37%, or 135.5 MW additional thermal energy will be accessible.

2. System Description

The ambient air at state 1, after being cooled in the mechanical chiller up to state 2 enters the compressors. The compressor compressed the air; resulting at state 3 the Pressure &Temperature of the air is increased.This high Pressure &Temperature air enters in the combustion chamber (CC) where the combustion of the fuel take place. Due to combustion of fuel the flue gases are formed. This flue gases having high Pressure &Temperature expand in turbine state 4” resulting shaft work is found.

After expanding the flue gases in turbine, it discharge from turbine and the resulting CO2 is stocked in stock chamber state 6”.As the CO2 gases stocked, the ammonia-water vapor fluid expand in Ammonia Water Turbine from high pressure drum (state 7”).Resulting shaft work is obtained.After expanding it discharge in Reheat Heat Exchanger at state 8’ where the Pressure &Temperature of the Ammonia Water is decreased as compare state 7”.

After discharging from RHE from state 8 to state 9” Pressure &Temperature is increased. And then it passed through absorber 9 to 10.The absorber absorbs the unwanted gases let as NOx, CO, etc.With the help pump (P3), the Ammonia-Water pumped in a heat exchanger from 10 to 11.In heat exchanger the Pressure &Temperature is increased of the Ammonia-W ater (11 to 12).The separator (S) separates the rich and lean mixture (12 to 13).The rich mixture enters in feed heater where Pressure &Temperature is increased (state 13 to 14).After it, it enters in condenser at state 14.

In condenser “a” indicates cooling water in and “b” indicate hot water out.The vapor form of ammonia-water is converted then to liquid form.With the help of pump (P2) the liquefied Ammonia-Water pumped in feed heater state 15 to 16.The feed heater raises the Pressure &Temperature of liquefied Ammonia-Water and it change in saturated liquid form (state 16 to 17).After passing Feed Heater the saturated liquefied Ammonia-Water enters in heat recovery steam generator (HRSG) where it changes its phase in vapor form.

And then discharge in high pressure drum.After passing High Pressure Drum it enters separator and then it enters in heat exchanger (HE)[state 18 to 19].In Heat Exchanger the heat exchange takes place between high temperature pressurized vapor Ammonia-Water and low pressure temperature vapor Ammonia-Water.At state 20, the high pressurized Ammonia Water the Pressure and Temperature is decreased. And then it passed in a throttle device Where a high pressure drop is take place.Resulting the enthalpy of Ammonia-Water vapor is decreased (state 20 to 21) and then it enters again in absorber.In absorber there is heat exchange take place between state 9 Ammonia-Water and state 22 Ammonia-Water.

Resulting the state 22Ammonia-Water, the Pressure &Temperature is increased due to heat absorption from state 9 Ammonia-Water.

The CO2 from intermediate pressure drum is expand in CO2 turbine and then it discharges in mixture M (state 24 to 25).As the CO2 discharge in mixture M the CO2 from low pressure drum is enter in mixture ‘M’ (state 33).Resulting the temperature and Pressure of discharge CO2 is increased and then it expand in another CO2 turbine.And then it discharge inRHE, where Pressure &Temperature is decreased of CO2 due to heat exchange between ambient air and CO2 (state 27 to 28).And then it enters in condenser where the vapor form of CO2 is changed in liquid form of CO2.

In condenser state ‘a’ indicates cooling water in and state ‘b’ indicates heat water out.After condensing of CO2the condensate extraction pump (CEP) pumped the CO2liquid in pump (P4) when it change in saturated liquid form.With the help of pump (P4) it pumped in RHVG where it change in vapor form (state 31 to 32) in low pressure drum.And thus the process again starts from 1.

3. Thermodynamic Modeling

The governing equations for various thermodynamic components of the combined cycle power plant under consideration are as follows:

3.1 Gas Model:

Ambient air entering the refrigeration heat exchanger of the low pressure turbine of bottoming cycle of combined cycle power plant is assumed to be at 1.01325 bar. Natural gas is taken to be fuel for the proposed combined cycle [19] with specific heat of combustion gases, given by

$$c_p = a + bT + cT^2 + dT^3 + \dots\dots\dots\dots\dots \quad (1)$$

Where a,b,c,d, etc. are the coefficients of polynomial with their values taken from the work of Toulounkain and Tadash [20]. With the use of equation (1) enthalpy of gas can be calculated as

$$h = \int_{T_a}^{T} c_p\,(T)dT \quad (2)$$

All non-reacting gases are arbitrarily assigned zero thermodynamic enthalpy, entropy and availability at the ambient pressure of 1.01325 bar.

3.2 Compressor Model:

An axial flow compressor is used in the assumed power plant and inefficiencies are taken care through the concept of polytrophic efficiency [21], by using following expression

$$\frac{dT}{T} = \left[\frac{R}{\eta_{pc}.c_{pa}}\right]\frac{dp}{p} \quad (3)$$

and the corresponding work required by both the compressors is given by:

$$W_c = \left[\dot{m}_a\left(h_{a,i} - h_{a,e}\right)\right]_{lp,c} + \left[\dot{m}_a\left(h_{a,i} - h_{a,e}\right)\right]_{hp,c} \quad (4)$$

Compressor work is estimated as under,

$$W_c = \dot{m}_a[h_{c,e} - h_{c,i}] \quad (5)$$

3.3 Combustion Chamber Model:

In the considered combined cycle the preheating of fuel is considered. Irreversibility in the combustion chamber [22] is taken care by combustion efficiency. It is proposed to preheat the fuel of combustion chamber and fuel of reheat combustion chamber by the coolant which is being discharged from low pressure gas turbine. Value of specific heat of natural gas is taken form works of J.M.Campbell [23].

Mass and energy balance across combustion chamber and reheat combustion chamber for a given turbine inlet temperature provide the fuel required in combustion chamber/reheat combustion chamber

$$\dot{m}_{cc,e} = [\dot{m}_i + \dot{m}_f] \quad (6)$$

$$\dot{m}_f.(LHV.\eta_{cc} + \Delta T_f.c_{pf}) = [\dot{m}_{cc,e}.h_{cc,e} - \dot{m}_i.h_{cc,i}] \quad (7)$$

3.4 HRVG Model

HRVG is critical thermodynamic element in the considered configuration of combined cycle power plant. Performance of HRVG leads to increased performance of combined cycle power plant [24]. Mass and energy balance across heat recovery vapor generator ~~is~~ shown in fig.2 is detailed below

Mass of steam generated at high pressure drum of HRVG is given by

$$m_{fg}.\left(c_{pfg9}.t_{fg9} - c_{pfg2'}.t_{fg2'}\right) = m_{s,hp}.[(h_{11} - h_{25}) + bd(h_{25} - h_a)] \quad (8)$$

Mass of steam generated at intermediate pressure drum of HRVG is given by

$$m_{fg}.\left(c_{pfg3'}.t_{fg3'} - c_{pfg5'}.t_{fg5'}\right) = m_{s,hp}.(h_a - h_{24}) + m_{s,ip}.(h_{30} - h_{24}) + m_{s,ip}.(h_{30} - h_b) + bd(h_{29} - h_b) \quad (9)$$

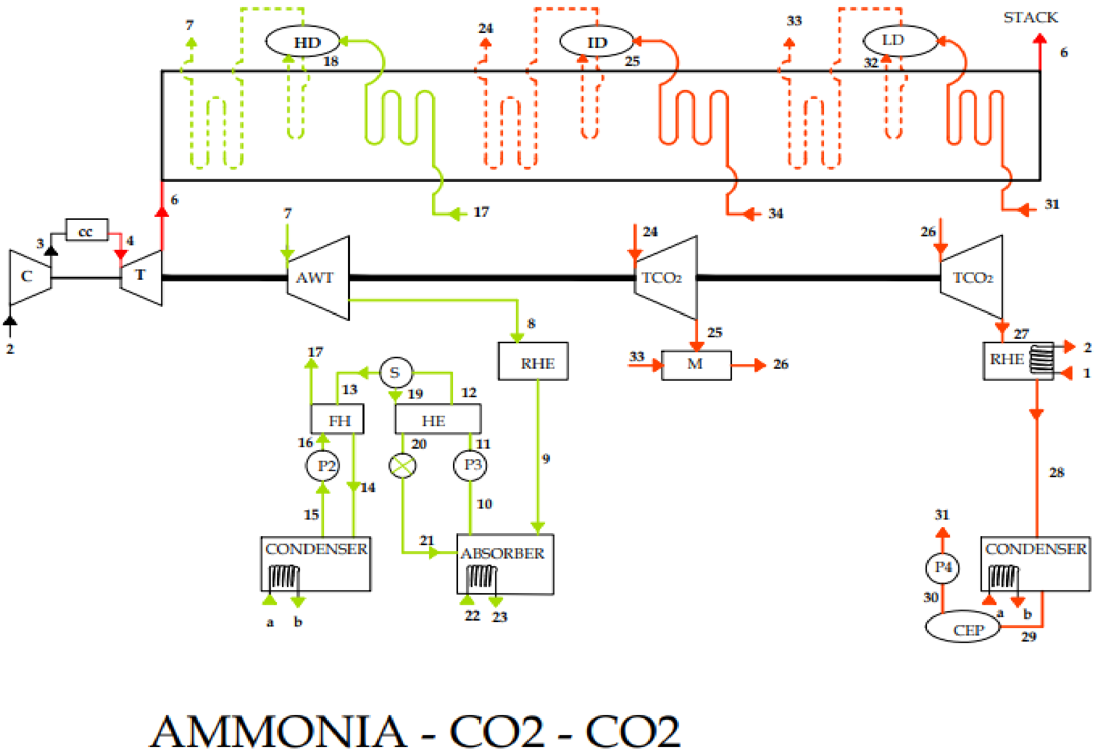

Fig.1 Schematic of combined cycle using ammonia water mixture and trans-critical carbon dioxide

Enthalpy of flue gas at pinch point of low pressure drum is given by [25]

$$\frac{m_{fg}}{m_{amw}} = \frac{h_{6'} - h_{7'}}{h_{46} - h_{44}} \quad (10)$$

$$\text{Since,} t_{45} = t_{7'} - PP \quad (11)$$

Enthalpy of ammonia water mixture is calculated from following equation

$$m_{fg}.\left(c_{pfg5'}.t_{fg5'} - c_{pfg7'}.t_{fg7'}\right) = m_{s,lp}.(h_b - h_{28}) + \left(m_{amw}.h_{46} - m_{s,lp}.h_{28}\right) + m_{amw}.(h_{46} - h_{45})$$

(12)

Temperature of stack is calculated as;

$$m_{fg}.\left(c_{pfg7'}.t_{fg7'} - c_{pfg10}.t_{fg10}\right) = m_{amw}.(h_{45} - h_{46}) \quad (13)$$

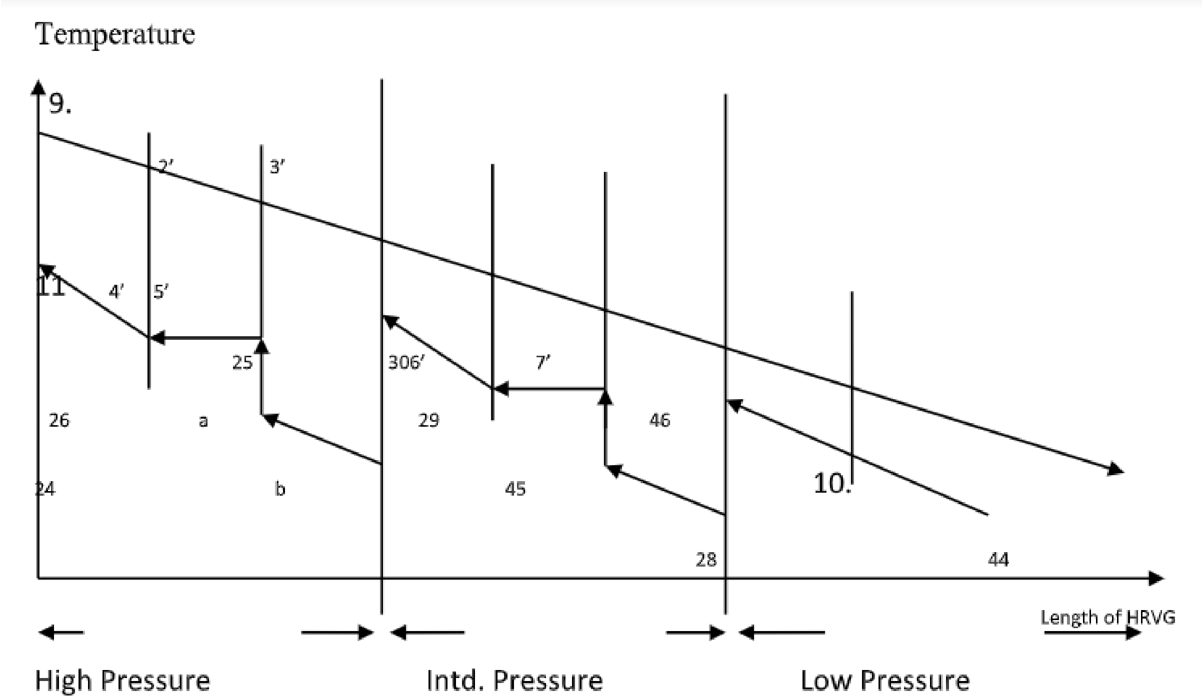

Fig.2.Temperature profile across the length of HRVG.

3.5 gas turbine model:

The expansion process is assumed to be polytropic and the temperature at any stage in the turbine is given by

$$\frac{dT_{fg}}{T_{fg}} = \left[\frac{p+dp}{p}\right]^{\frac{R.\eta_{pgt}}{c_{pa}}} - 1 \quad (14)$$

And the corresponding gas turbine work is given by

$$W_{gt} = \left[m_{fg,i}\left(h_{fg,i} - h_{fg,e}\right)\right]_{hpgt} \quad (15)$$

3.6 Trans-critical carbon dioxide and Ammonia Water mixture turbine Model:

Here it is assumed that the dissociation of ammonia does not take place in the binary mixture. Considering the isentropic efficiency of steam turbine and ammonia-water mixture turbine, the gross turbine work output is given bythefollowingequation

$$W_{st} = \left[\dot{m}_{s,hp}.\eta_{ishp}.\left(h_{s,i} - h_{s,e}\right)\right] + \left[\dot{m}_{s,lp}\eta_{islp}\left(h_{s,i} - h_{s,e}\right)\right] + \left[\dot{m}_{amw,hp}\eta_{islp}\left(h_{amw,i} - h_{amw,e}\right)\right] \quad (16)$$

3.7 Condenser Model:

The condenser of the combined cycle power plant is externally water cooled condenser and shared by intermediate steam turbine and low pressure turbine. Mass and energy balance as given below gives the mass flow required for cooling purpose.

$$\dot{m}_{cw}(\dot{h}_{cw,e} - \dot{h}_{cw,i}) = \dot{m}_{s,lp}.\left(h_{s,i} - h_{water,e}\right) + \dot{m}_{amw,lp}.\left(h_{amw,i} - h_{amw,e}\right) \quad (17)$$

3.8 Pump model:

Pump efficiency is introduced to take care of all the inefficiency of the pump.

$$w_{pump,actual} = \frac{w_{is,pump}}{\eta_{is,pump}} \quad (18)$$

3.9 Performance Parameters:

Performance parameters of theconsideredcombined cycle power plant are as follows:

$$W_{gt,net} = W_{gt} - \frac{W_c}{\eta_M} \quad (19)$$

$$W_{st,net} = W_{st}.\eta_{M,st} - \frac{W_p}{\eta_p} \quad (20)$$

$$W_{combined\ cycle} = [W_{gt,net} + W_{awt,net} + W_{t-CO2,net}].\eta_{gen.} \quad (21)$$

$$COP = \frac{\text{Cooling produced}}{\text{Potential work lost}}[35] \quad (22)$$

4. Results and Discussion

Based on thermodynamic modeling the results are obtained and presented to exhibit the variation of the work produced, first law efficiencywith respect to cycle pressure ratio, turbine inlet temperature, ammonia mass fraction and the ambient temperature.

Fig.3 depicts the variation of the work produced in the combined cycle with varying cycle pressure ratio, turbine inlet temperature and ammonia mass fraction and at different ambient temperature. Considering the variations occurring at an ambient temperature of 30°C and keeping the cycle pressure ratio and turbine inlet temperature constant as the ammonia mass fraction is increased work produced by the combined cycle decreases because specific heat of the mixture decreases due to which enthalpy of the mixture and hence the work produced decreases. Due to change in ammonia mass fraction no change in work output of trans critical carbon dioxide is observed.

Whereas if the cycles pressure ratio or the turbine inlet temperature is increased keeping the ammonia mass fraction constant an increase in work produced is obtained.

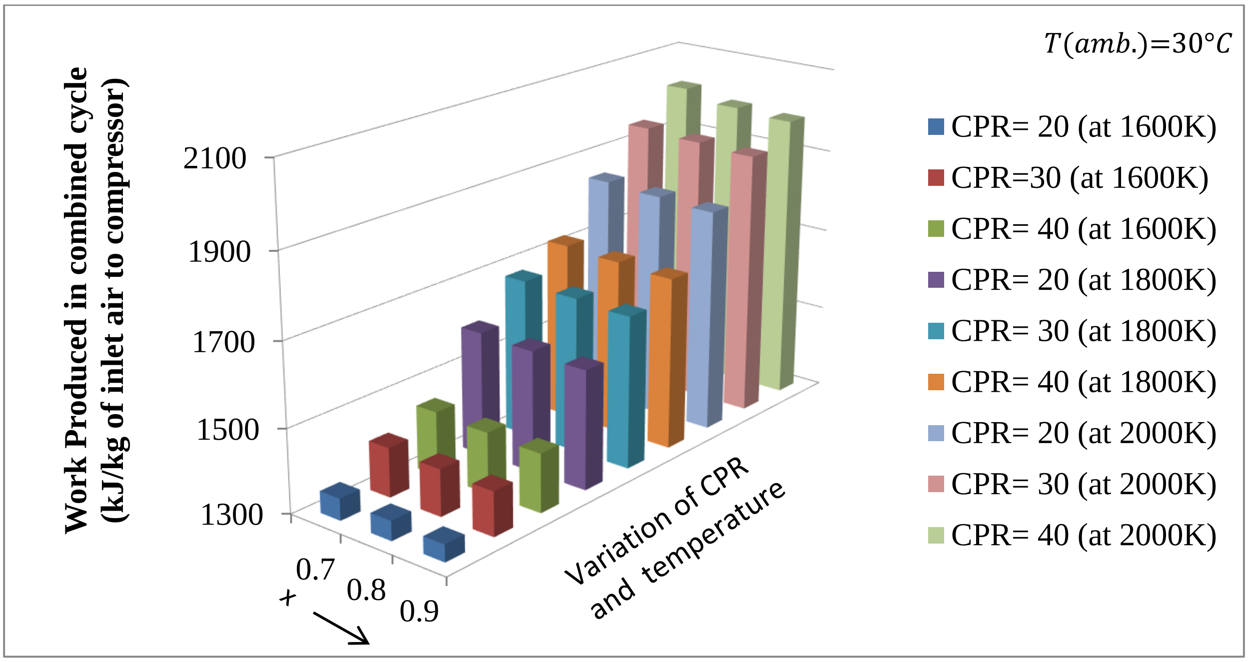

Fig.3 (a) Variation in work produced of the combined cycle with varying cycle pressure ratio, turbine inlet temperature and ammonia mass fraction at ambient temperature of 30°C.

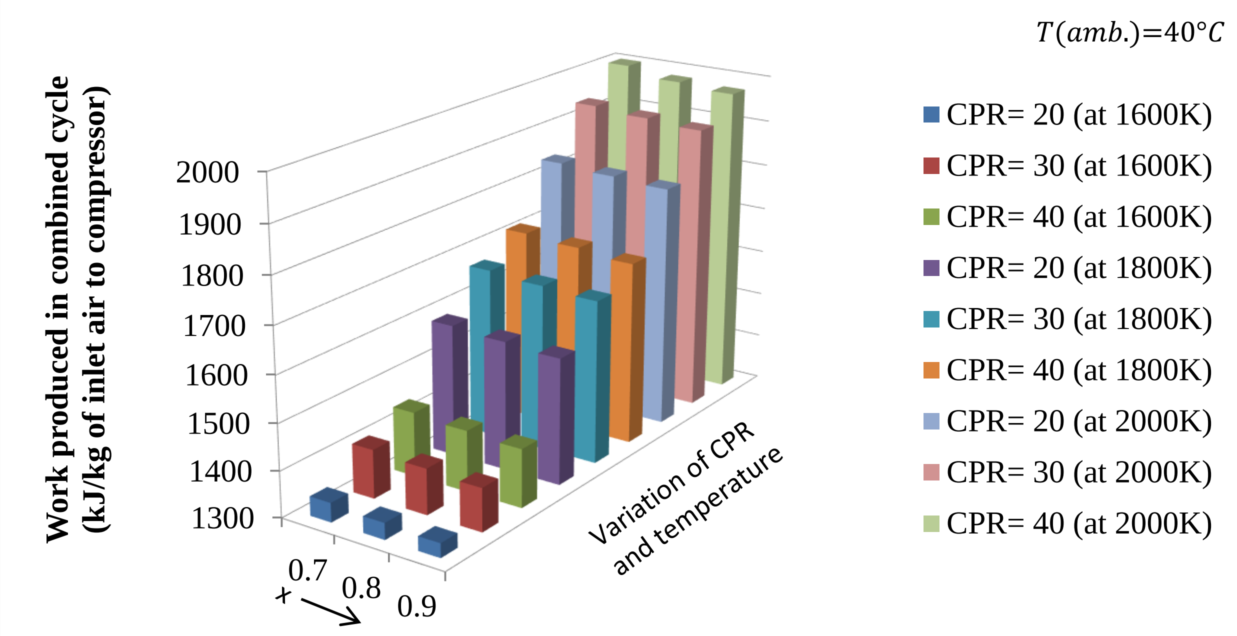

Fig.3(b) Variation in work produced of the combined cycle with varying cycle pressure ratio, turbine inlet temperature and ammonia mass fraction at ambient temperature of 40°C.

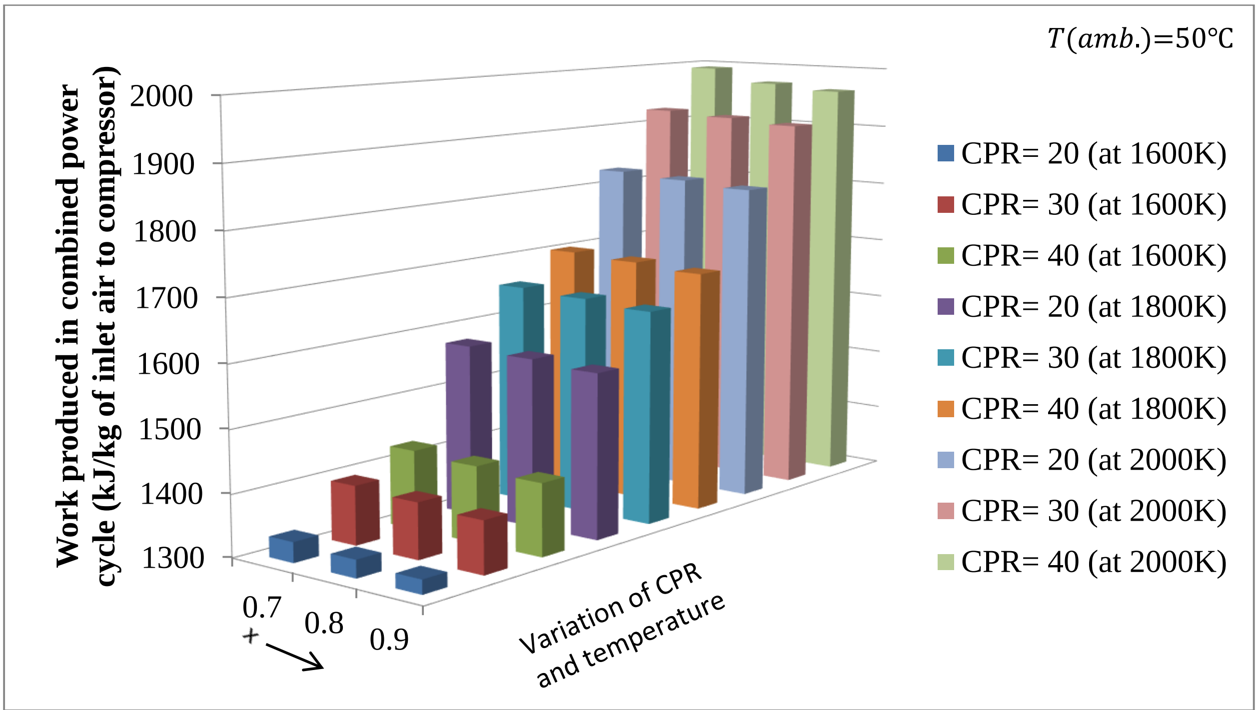

Fig.3(c) Variation in work produced of the combined cycle with varying cycle pressure ratio, turbine inlet temperature and ammonia mass fraction at ambient temperature of 50°C.

Fig.3 also depicts the variation with increase in the ambient temperature, as the ambient temperature is increased then decrease in work produced is observed because the cooling load on the refrigerant heat exchanger increases,which increases the inlet temperature of the fluid entering thecompressor thereby reducing the work produced by the combined cycle.

Variation of first law efficiency with respect to cycle pressure ratio, turbine inlet temperature, ammonia mass fraction and ambient temperature is shown in fig.4. It is observed that with increase in ammonia mass fraction, the decrease in efficiency is observed because of decrease in work output from the combined cycle. Whereas increasing the cycle pressure ratio and turbine inlet temperature increases the efficiency of the cycle. The effect of increase in ambient temperature decreases the first law efficiency of the cycle.

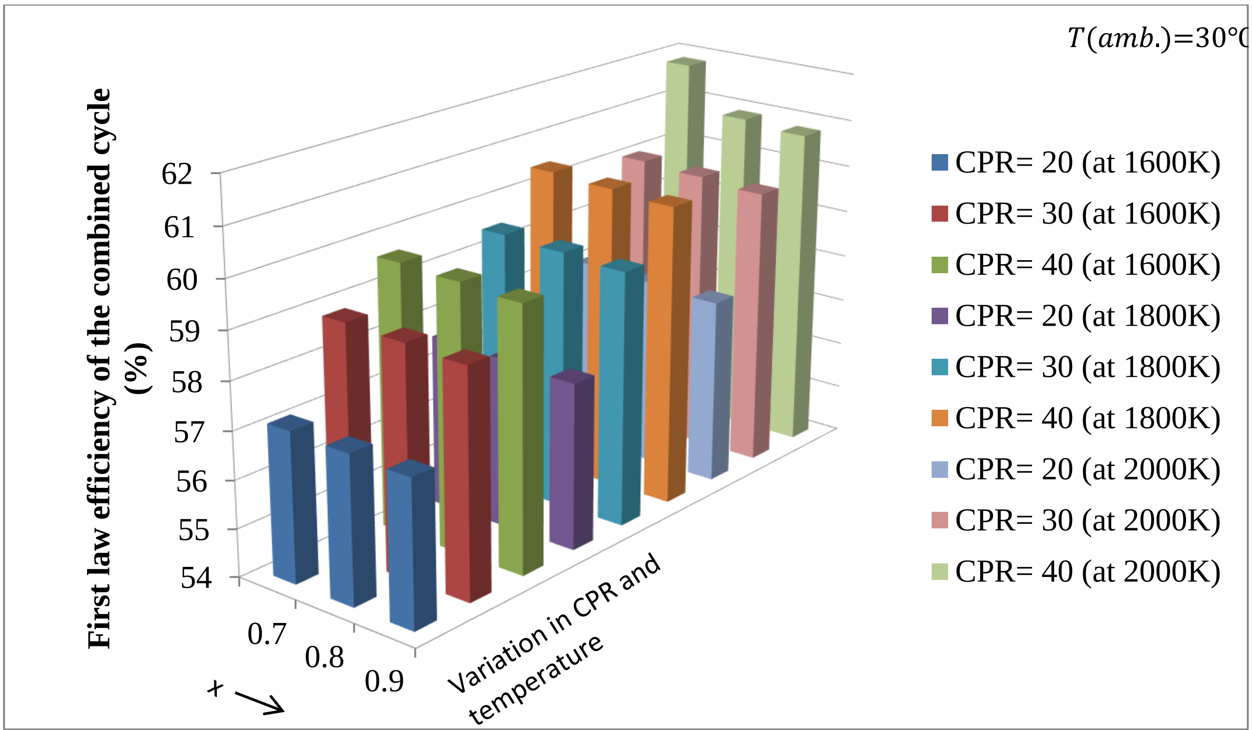

Fig.4(a) Variation in first law efficiency of the combined cycle with varying cycle pressure ratio, turbine inlet temperature and ammonia mass fraction at ambient temperature of 30°C.

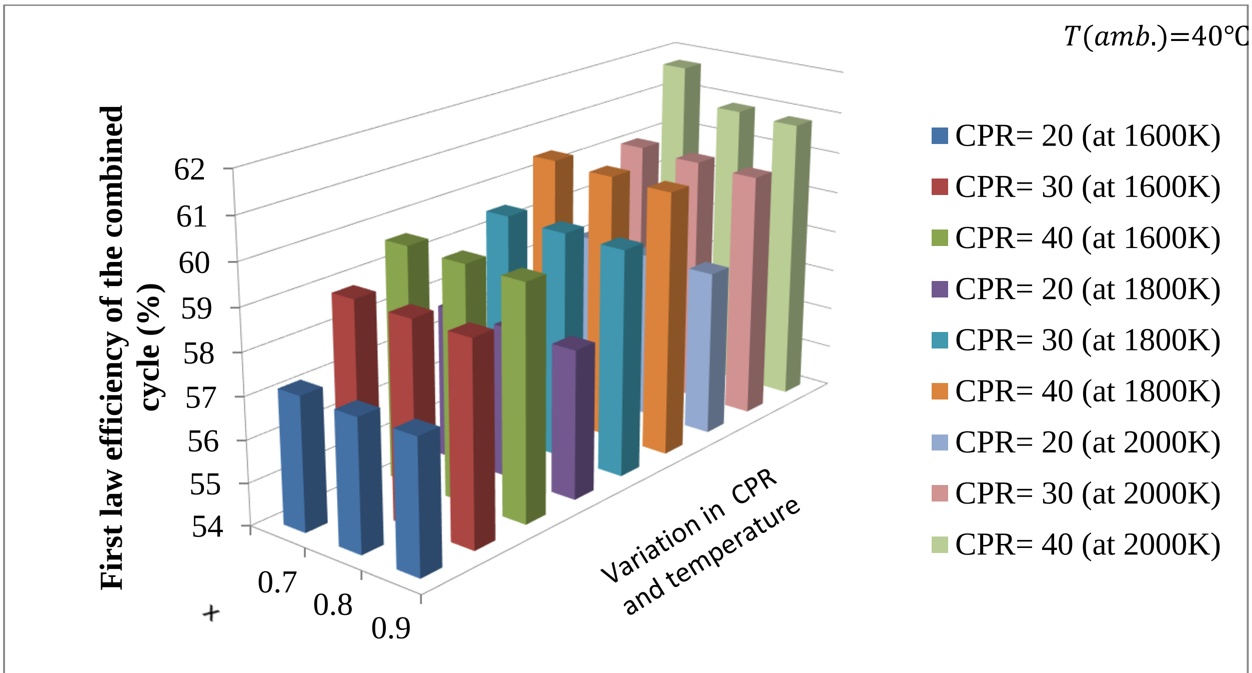

Fig.4 (b) Variation in first law efficiency of the combined cycle with varying cycle pressure ratio, turbine inlet temperature and ammonia mass fraction at ambient temperature of 40°C.

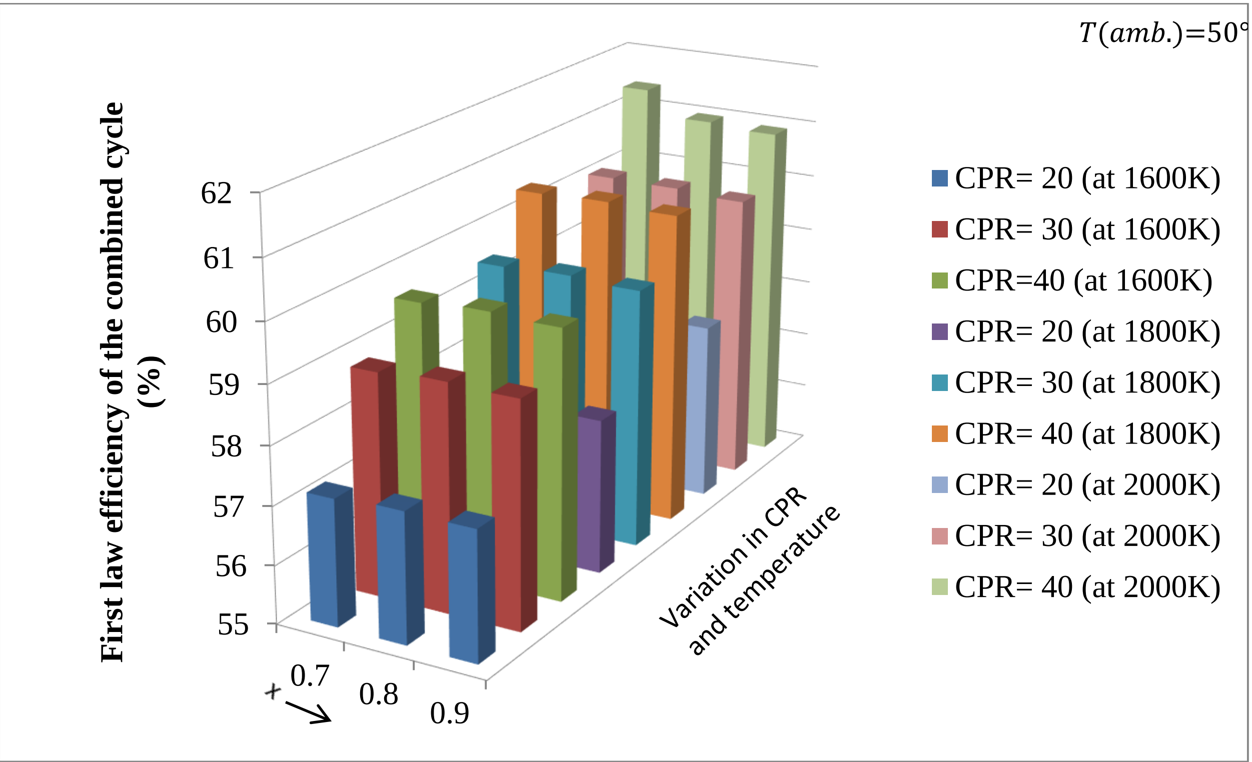

Fig.4(c) Variation in first law efficiency of the combined cycle with varying cycle pressure ratio, turbine inlet temperature and ammonia mass fraction at ambient temperature of 50°C.

5. Conclusion

First law analysis of combined power cycle having simple gas turbine, ammonia water cycle and trans critical carbon dioxide is performed for different ammonia mass fraction, cycle pressure ratio, ambient temperature and turbine inlet temperature. The result depicts that-

1. Maximum work output of is obtained for ammonia mass fraction of 0.6 and ambient temperature of 30°C 1982.31kJ/kg of inlet air to compressor is obtained.
2. A first law efficiency and second law efficiency of 62.07%. is obtained for turbine inlet temperature of 2000K and ambient temperature of 30°C

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Details of Parameter, Term, Symbol and Unit considered:

Parameter	**Term, Symbol**	**Unit**
Gas Properties	Specific heat, $c_p = f(T)$ Enthalpy, $h = c_p(T)dT$	kJ/kg.K kJ/kg
Compressor	Polytropic efficiency, (η_{pc}) = 93.0 Mechanical Efficiency, (η_M) = 99.5	% %
Combustor	Efficiency of combustion chamber, (η_{cc}) = 99.0 Loss of pressure in combustion chamber , (p_{loss}) = 1.5% of inlet pressure Lower heating value, (LHV) = 50000	% bar kJ/kg
Gas turbine	Polytropic efficiency, (η_{pgt}) = 93.0 Exhaust pressure = 1.08	% bar
HRVG (Triple Pressure)	Effectiveness = 98.0 Loss in pressure = 5% of entry pressure (for bottoming cycle fluid) Loss of pressure on the gas side = 5% of pressure at inlet Minimum exhaust temperature from HRVG = 353.0 Pressure of steam in the HD = 180 Maximum temperature attained due to superheating in H.P. = 873 (Max.) Exhaust pressure from high pressure steam turbine= 100 Pressure loss in reheater = 3% of pressure at inlet Intermediate pressure= 100.0 bar Superheating of steam in I.P. = 673.0 (Max.) Inlet pressure of ammonia water mixture cycle = 70.0 L.P Turbine Inlet Temperature= 348 to 373 Pressure at which deaerator is operating = 1.5 Pressure at which condenser is operating = 0.07 Approach temperature difference between gas/steam or gas/ammonia water mixture = 20.0(min.) Pinch point = 20.0 (min.)	% bar bar K bar K bar bar bar K bar K bar bar K K
Bottoming cycle turbine	Isentropic efficiency of bottoming cycle turbines = 88.0	%
Ammonia Mass Fraction	0.6-0.9	
Pump	Isentropic efficiency of pump’s, (η_{pump}) = 88.0	%
Generator	Efficiency of generator, ($\eta_{gen.}$) = 88.0	%