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ORIGINAL ARTICLE

Parametric analysis of biomass fired recompression supercritical CO_2 power cycle

Ramneek Singh[1*], Rupinder Pal Singh[2], RS Gill[2]
[1] College of Agricultural Engineering and Technology, Punjab Agricultural University, Ludhiana
[2] Department of Mechanical Engineering, College of Agricultural Engineering and Technology, Punjab Agricultural University, Ludhiana

ABSTRACT
Renewable energy resources are a potential substitute to reduce the dependency on the fossil fuels for energy production. Biomass can be converted into energy by utilizing thermodynamic power conversion cycles. In the present work, the supercritical carbon dioxide power cycle is used as power block to convert biomass into electricity. This cycle comprises more compact turbomachinery and stepping towards green energy. Thermodynamic investigation of the power conversion cycle assisted by biomass firing is presented through the development of a mathematical model. Further, the parametric analysis is conducted for s-CO_2 recompression with reheat power conversion cycle. The turbomachinery performance has also been analyzed for different operating conditions; Turbine and compressor inlet temperature/pressure, pressure ratio and intermediate pressure. The analysis concluded a thermal efficiency 49.09 % at an inlet temperature of 550°C, 20 MPa pressure.

***Keywords**: Turbomachinery, Renewable energy resources, CO_2 power cycle*

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INTRODUCTION

Fossil fuels are conventional energy resources and declining rapidly due to the economic growth. Moreover, these produces many pollutants during combustion namely carbon monoxide, the oxides of sulfur & nitrogen, and very fine soot/ash particles. The primary pollutants can further interact with the environment to generate additional deleterious effects as acid rain, smog and the greenhouse effect [1]. Hence, there is the urgent need of renewable energy resource due to faster depletion of conventional fossils fuels. Biomass is a form of renewable energy resource, which includes wastes from agricultural crops, animal waste, municipal solid waste, food processing industry etc. [2]. Biomass is carbon neutral as net transfer of carbon to the environment is zero by using biomass [3]. Biomass is abundant and easily available at almost everywhere on earth which is not possible in case of hydro and solar resources. Biomass to energy conversion technologies also contribute in to keep environment clean. It also enhances management of biomass based waste. Main Biomass combustion technologies for both heat and power applications are grate firing, pulverized fuel and fluidized bed combustion which is same as in case of coal combustion [4].

CO_2 is abundant, affordable, non-corrosive, non-toxic, non-flammable and non-explosive working fluid. CO_2 is environment-friendly with nil ozone depletion potential and a global warming potential of 1 over 100 years [5]. During the expansion process of supercritical CO_2 it exhibits gas properties with liquid density. Carbon dioxide reaches its critical point at a critical pressure, P_c = 7.4 MPa and critical temperature, T_c = 304.12 K. Moreover, critical temperature of carbon dioxide is near the ambient condition which makes it suitable to work in Brayton cycle. The supercritical fluid possesses specific properties and has an intermediate behavior between that of a liquid and a gas. In particular, supercritical phase of fluids display its density like liquid, viscosity like gas and diffusivity which is intermediate to that of a liquid and a gas [6]. Using s-CO_2 as working fluid will leads to reduction in CO_2 emissions and has a potential to contribute towards green energy. The supercritical cycle possess characteristics such as higher energy efficiency, large power/volume ratio, single phase fluid during condensation/cooling and no risk of turbine blade erosion which are desirable for practical applications [7]. In addition, Turbomachinery of s-CO_2 cycle is more compact in terms of size as compared with conventional Rankine cycle, ORC and Helium-Brayton cycle of same capacity. High density in supercritical phase allows achieving compactness of size of turbomachinery and heat exchanger. The compactness of turbomachinery gives privilege to enjoy great gains in saving space and initial capital investment [8] It is concluded that s-CO_2 power plants require 4 m^2 land per MW which is almost 6 times lesser than Steam rankine cycle[9].

Manente and Lazzaretto [10] demonstrated a supercritical closed CO_2 Brayton cycle with higher efficiency for electricity generation from biomass. Two cascaded s-CO_2 cycle were enabled for effective use of the available heat from flue gases. Optimization results show that the cascaded cycle configuration reaches highest biomass to electric power efficiency of 36%. A cascaded supercritical CO_2 cycle powered with both solar energy and biomass energy was proposed by Wang *et al.* [11]. The system contains recompression cycle and a simple cycle. Energy analysis was performed to evaluate the feasibility with considering the variations in solar irradiations and biomass complementary. The first law efficiency of the system reaches 40.1% in their work. Balafkandeh *et al.* [12] proposed a biomass-based cooling, heating and power (CCHP) system. The performance of s-CO_2 integrated with double effect LiBr-H_2O absorption chiller was analyzed and compared for two cases: fueling with syngas from the gasification process and, fueling with biogas from the digestion process. The results revealed that the digestion-based system performs better than the gasification in terms of efficiency. Dostal [13] worked on design of cycle components, turbo machinery and development of a suitable control scheme a recompression supercritical CO_2 cycle coupled with nuclear reactor and identification of the most appropriate supercritical configuration with best suitable operating conditions. The work also includes an economic analysis and quantification of the savings offer by supercritical CO_2 cycle over steam rankine cycle and helium cycles.

There are few studies available for the biomass fired supercritical recompression CO_2 cycle. In addition, the detailed parametric studies of different parameters on the system efficiency are scarce. In the present work, a detailed parametric study is conducted by varying compressor inlet temperature, turbine inlet temperature, pressure ratio and turbine inlet pressure on the energy efficiency of biomass fired s-CO_2 cycle. The last section describes the turbo-machinery optimization aspects of the cycle.

MATERIAL AND METHODS

Cycle Description

The configuration of s-CO_2 recompression with reheating Brayton power conversion cycle is given in Fig.1. The CO_2 initially expands in high pressure (HP) turbine with maximum cycle pressure (turbine inlet pressure, P1), enters the reheater and further expands in low pressure (LP) turbine. The reheater raises the temperature of exit stream of HP turbine before entering in low pressure turbine at intermediate pressure. The LP stream from LTR is divided into two different streams. One of the stream enters the compressor & other enters the low pressure compressor which further combines with the main stream exiting the LTR. The main CIT (T7) and CIP (P6, P7) are determined as closer to critical point. The thermodynamic comparison of different cycle arrangements show that recompression with reheat achieves the highest efficiency. [14-15]. The effect of recompression overcomes the occurrence of pinch point problem. Feher [15] studied several different working fluids for the supercritical cycle and concluded carbon dioxide to be the most desirable working fluid. The precooler is also used in the cycle to cool the working fluid before it enters the compressor. In addition, with the most desirable operating conditions of TIP and TIT (20 MPa and 550°C), cycle achieves the highest efficiency [16].

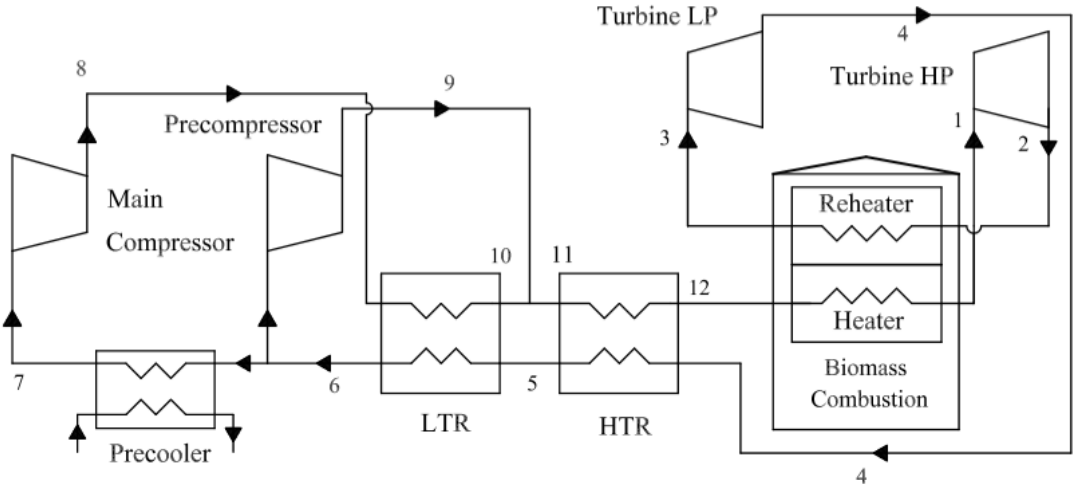

Fig.1. Configuration of recompression supercritical CO_2 (with reheating) Brayton power cycle

Assumptions

1. All processes follow steady state conditions.
2. Isentropic expansion and compression.
3. Split turbine into two stages and incorporate fluid reheating.
4. Effectiveness values are considered for the LTR and HTR.
5. The components of the cycle are well insulated pressure losses are neglected.

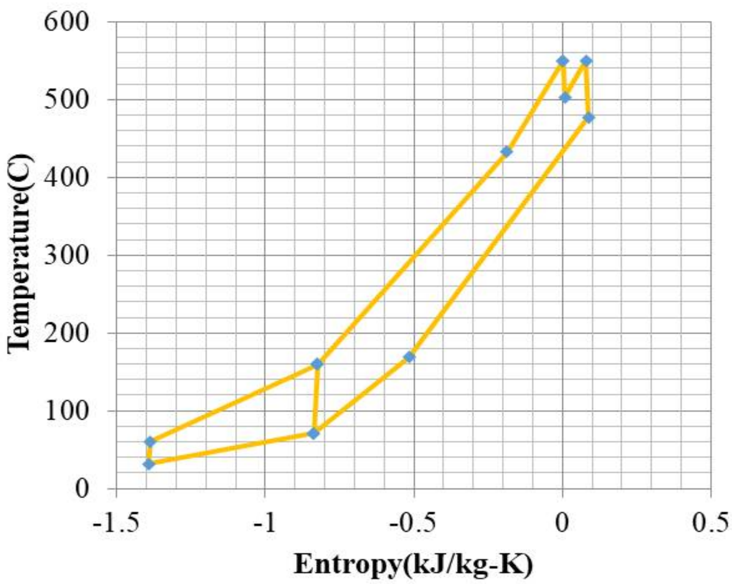

Fig.2. Temperature – Entropy plot of recompression with reheating Brayton power conversion cycle

Numerical simulation

The mathematical model is developed to evaluate the thermal performance of s-CO_2 recompression with reheating cycle. The state point values for all the components is obtained, based on the energy balance equations of the individual components. Span and Wagner correlations [19] are used to evaluate the thermodynamic properties of CO_2. The relation between pressure ratio (PR), maximum pressure (P_{max}) and minimum pressure (P_{min});

$$PR = \frac{P_{\max}}{P_{\min}} \quad (1)$$

Table.1. Standard cycle design considerations and input parameters [17][18]

Input Parameter	Value
Turbine Inlet Temperature,TIT,T1	550 ^oC
Main Compressor Inlet Temperature ,CIT,T6	32 ^oC
Compressor Efficiency *(Main and Pre)*	0.90
Turbine Efficiency *(HP and LP)*	0.93
LTR Effectiveness	0.97
HTR Effectiveness	0.97
Turbine Inlet Pressure,TIP,P_max,P1	20000 KPa
Pressure Ratio ,PR	2.6

Energy model

The energetic analysis of the s-CO_2 recompression power conversion cycle is based on the first law of thermodynamics. Compression duty is split between main compressor and recompressor and a fraction of mass flow exist according to the split ratio. The design points at 9, 10 and 11 exhibits similar thermo physical properties [20].

Work done by High pressure Turbine is determined by

$$\eta_{HP} = \frac{(h_1 - h_2)}{(h_1 - h_{2,is})} \tag{2}$$

$$W_{HP} = \dot{m}(h_1 - h_2) \tag{3}$$

Work done by low pressure turbine

$$\eta_{LP} = \frac{(h_3 - h_4)}{(h_3 - h_{4,ise})} \tag{4}$$

$$W_{LP} = \dot{m}(h_3 - h_4) \tag{5}$$

Work done by Main compressor

$$\eta_{MC} = \frac{(h_{8,ise} - h_7)}{(h_8 - h_7)} \tag{6}$$

$$W_{MC} = \dot{m}(h_8 - h_7) * SR \tag{7}$$

Work done by Recompressor

$$\eta_{RC} = \frac{(h_{9,ise} - h_6)}{(h_9 - h_6)} \tag{8}$$

$$W_{RC} = \dot{m}(h_9 - h_6) * (1 - SR) \tag{9}$$

$$SR = \frac{\dot{m}_7}{\dot{m}_6} \tag{10}$$

Heat Supplied

$$Q_{heat} = \dot{m}(h_1 - h_{12}) \tag{11}$$

$$Q_{reheat} = \dot{m}(h_3 - h_2) \tag{12}$$

$$Q_s = Q_{heat} + Q_{reheat} \tag{13}$$

Net work done

$$W_{net} = W_{HP} + W_{LP} - W_{MC} - W_{RC} \tag{14}$$

First Law Efficiency

$$\eta_{thermal} = \frac{W_{net}}{Q_S} \tag{15}$$

Energy-mass Balance for HTR,

$$(h_{12} - h_{11}) = (h_4 - h_5) \tag{16}$$

Effectiveness for HTR

$$\varepsilon_{HTR} = \frac{(h_4 - h_5)}{(h_4 - h(T_{11}, P_6))} \tag{17}$$

Energy-mass Balance for LTR,

$$SR * (h_9 - h_8) = (h_5 - h_6) \tag{18}$$

Effectiveness for LTR,

$$\varepsilon_{LTR} = \frac{(h_5 - h_6)}{(h_5 - h(T_8, P_6))} \tag{19}$$

Thermal model validation

The developed thermal model is validated with Turchi et al. [21] and results are presented in the Table 2. The model is sufficiently validated and can be further used for parametric analysis.

Table.2. Validation results at TIT=700°C

	Turchi *et al.*[21]		Present work	
CIT (°C)	45	60	45	60
PR	2.7	2.5	2.7	2.5
ε-HTR	0.97			
ε-LTR	0.88			
η_{cycle}	0.5230	0.4970	0.5226	0.4974

RESULTS AND DISCUSSION

The present work discusses the effect of various parameters on the thermal efficiency and work done by turbo machinery of recompression (with reheat) supercritical CO_2 Brayton cycle. The efficiency is evaluated at varying pressure ratio (PR), turbine inlet pressure (TIP) and compressor inlet temperature (CIT). The parametric analysis is conducted within the following range of variables.

$$550^\circ C \le TIT \le 950^\circ C \tag{20}$$

$$32^\circ C \le TIT \le 50^\circ C \tag{21}$$

$$2.1 \le PR \le 2.7 \tag{22}$$

$$20MPa \le TIP \le 28MPa \tag{23}$$

Effect of varying compressor inlet temperature on cycle performance

The effect of compressor inlet temperature (main compressor) on the cycle efficiency is shown in Fig. 3. The thermal efficiency of the supercritical cycle is inversely proportional to the compressor inlet temperature at all the TIT. The higher inlet temperature of compressor is responsible for the increased compressor work done which further reduces the overall thermal efficiency. The rise in turbine inlet temperature from 550 °C to 950 °C improves the thermal efficiency by 28 % at a compressor inlet temperature of 50 °C. The maximum efficiency of the cycle is achieved as 60% at the minimum compressor and maximum turbine inlet temperature. The net work produced in the system is plotted at different compressor inlet temperature as shown in Fig. 4. The trend is quite similar to the efficiency curve as the both the parameters are directly proportional. The maximum net work available is obtained as 3329 kW at 32°C CIT and 950 °C TIT. The efficiency of the cycle at different inlet pressures is shown in Fig.5. The efficiency decreases with increase in compressor inlet temperature. As turbine inlet pressure increases the thermal efficiency remains almost constant up to certain CIT and start decreasing because of the variation in work done by turbo machinery. This happens due to variations in enthalpy of design points of LTR and turbo machinery performance. The highest thermal efficiency of 49.1% is achieved at TIP of 20 MPa and CIT of 32 °C. The reason for higher efficiency is that the exit pressure of turbine reaches near the critical point at a PR of 20 MPa. Also, the lower temperature of CIT means the less work done by the compressor.

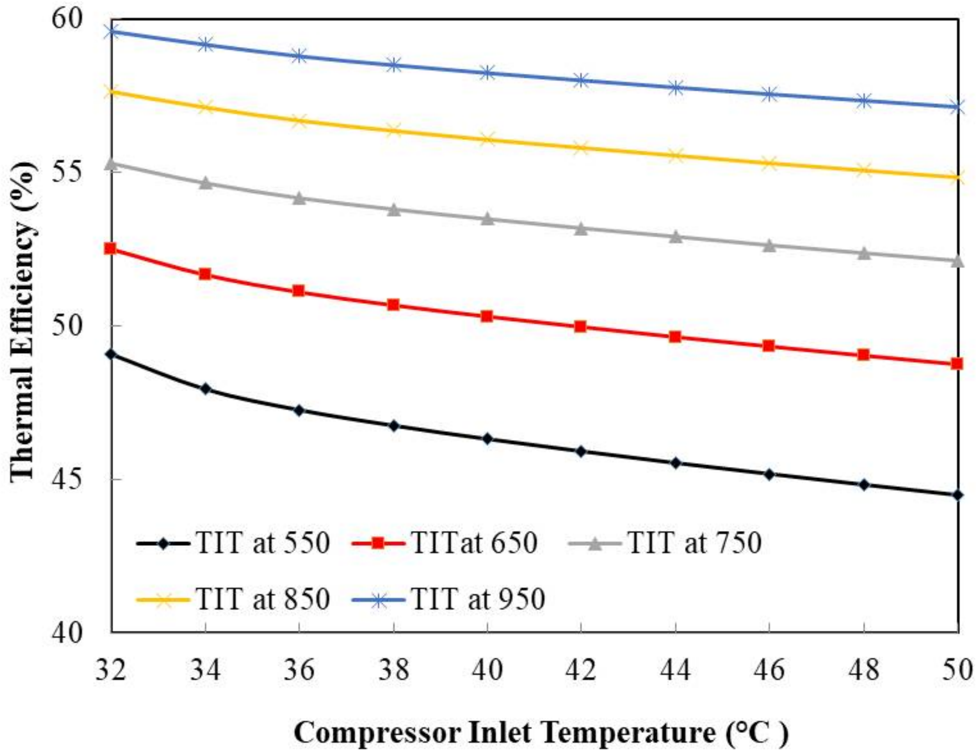

Fig.3. Effect of compressor temperature on the thermal efficiency of recompression (with reheat) supercritical CO_2 cycle

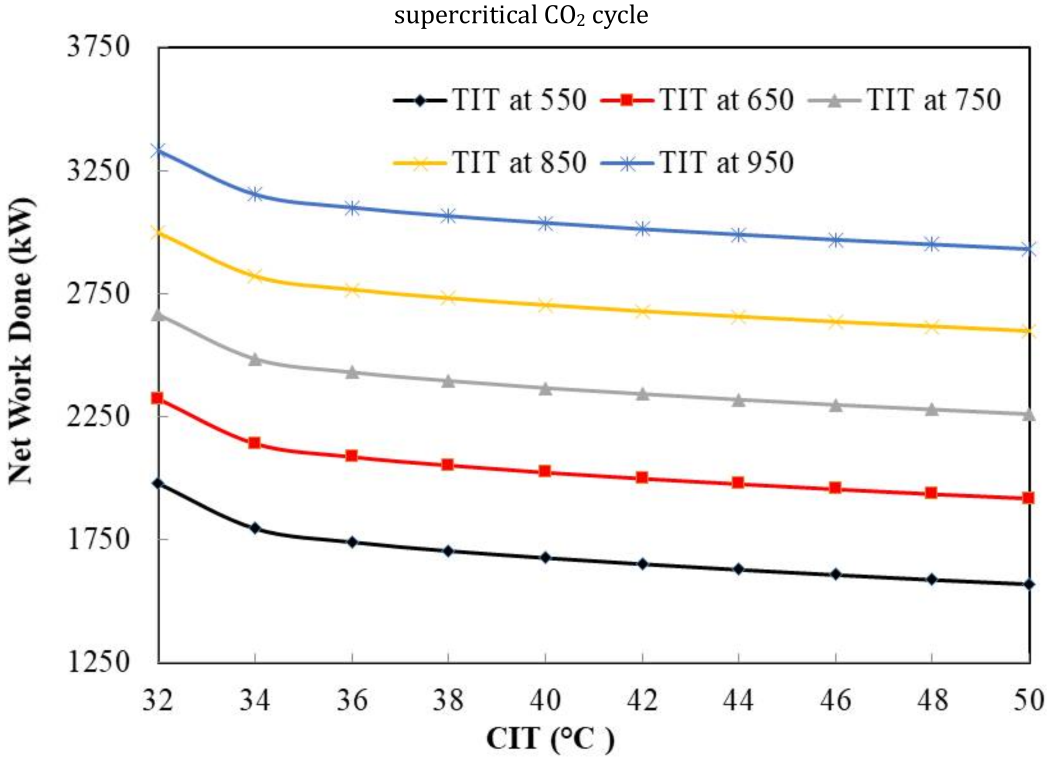

Fig.4. Effect of CIT on the net work done in a recompression (with reheat) supercritical CO_2 Brayton cycle at different TIT

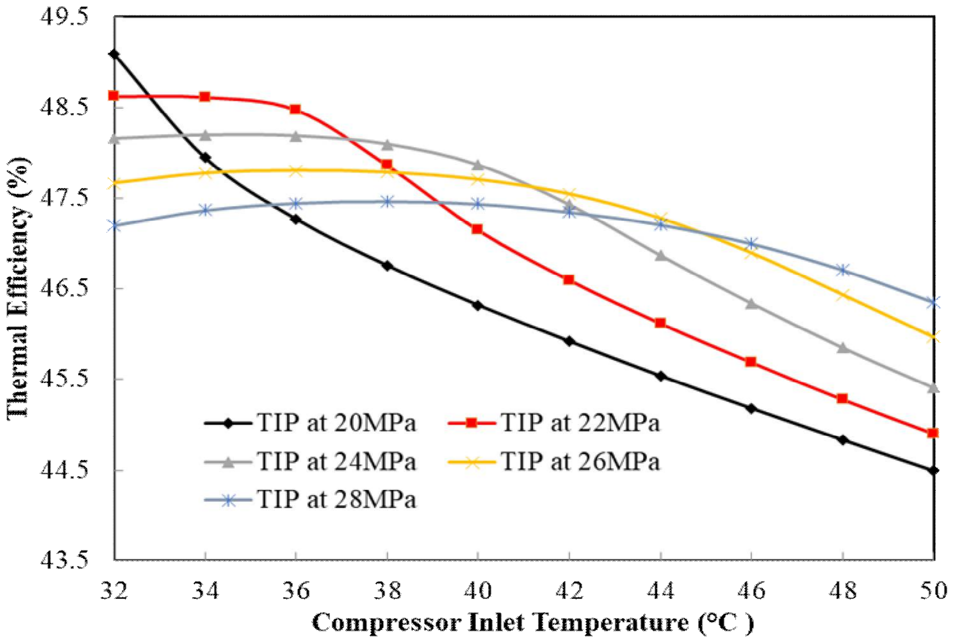

Fig.5. Effect of compressor inlet temperature on the thermal efficiency in a recompression supercritical CO_2 cycle at different inlet pressure of turbine

Effect of turbine inlet temperature on cycle performance

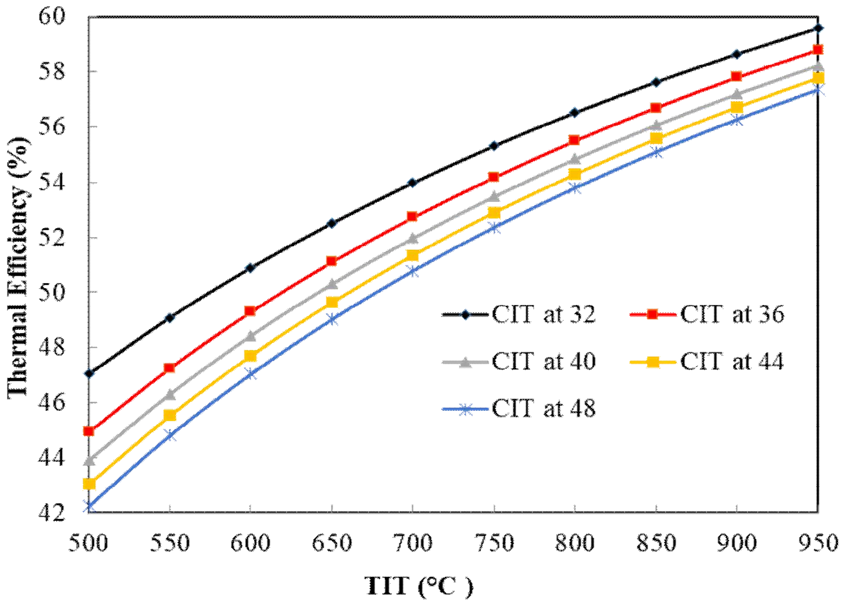

Fig. 6. Effect of TIT on the efficiency at varying CIT

Fig .6. represents the variation in thermal efficiency due to influence of turbine inlet temperature at different CIT. The higher inlet temperature of turbine is responsible for the increased turbine work done which further increases the overall thermal efficiency of cycle. The cycle exhibit highest thermal efficiency of 59.6 % at lowest CIT =32 °C and highest TIT of 950 °C. The decrease in TIT and increased CIT gives the lowest efficiency of 42.27% at TIT=550 °C & CIT=48 °C. CIT is generally taken 45 °C for dry cooling conditions. The work done by Turbomachinery is directly proportional to the turbine inlet temperature as the difference in enthalpies of exit and entry point increases with rise in TIT.

Influence of turbine inlet pressure on thermodynamic performance

Fig.7. shows the effect of inlet pressure on thermal efficiency at different CIT. Curve for CIT=32°C decreases linearly with increase in TIP. Thermal efficiency for CIT = 48°C is directly proportional to TIP. For CIT within range from 36°C to 44°C, the curve first increases, reaches maximum and then decreases.

Fig.8. shows the effect of turbine inlet pressure on thermal efficiency at different TIT. Thermal efficiency curve first increases, reaches maximum and then decreases. It is observed from the plot that an optimum TIP exists at same point for all TIP.

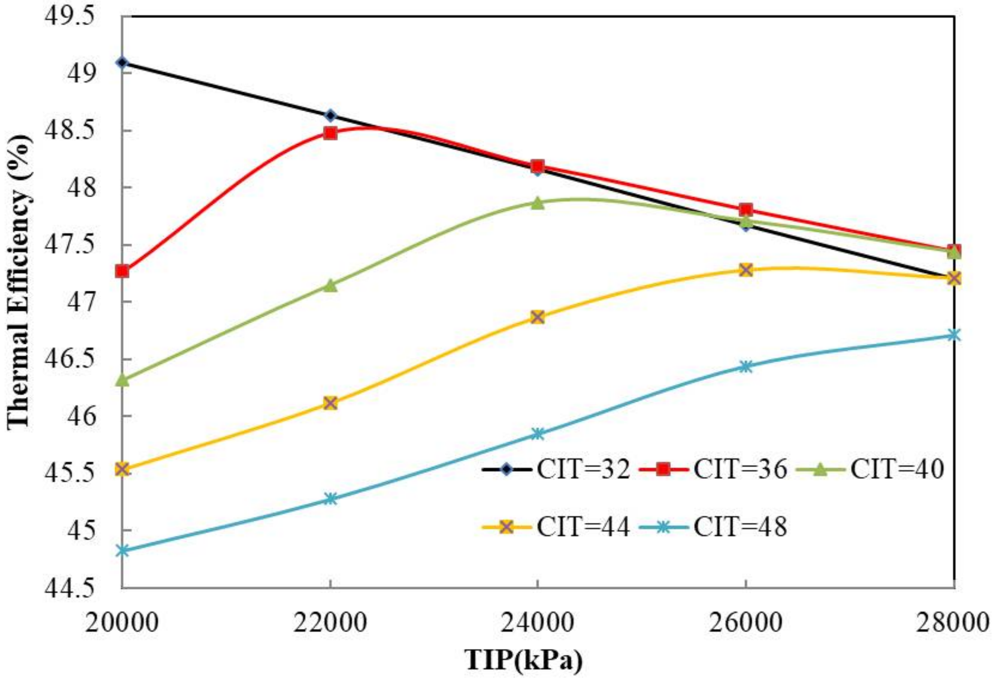

Fig.7. Effect of TIP on the thermal efficiency in a recompression (with reheat) supercritical CO_2 Brayton cycle at different Compressor Inlet Temperature

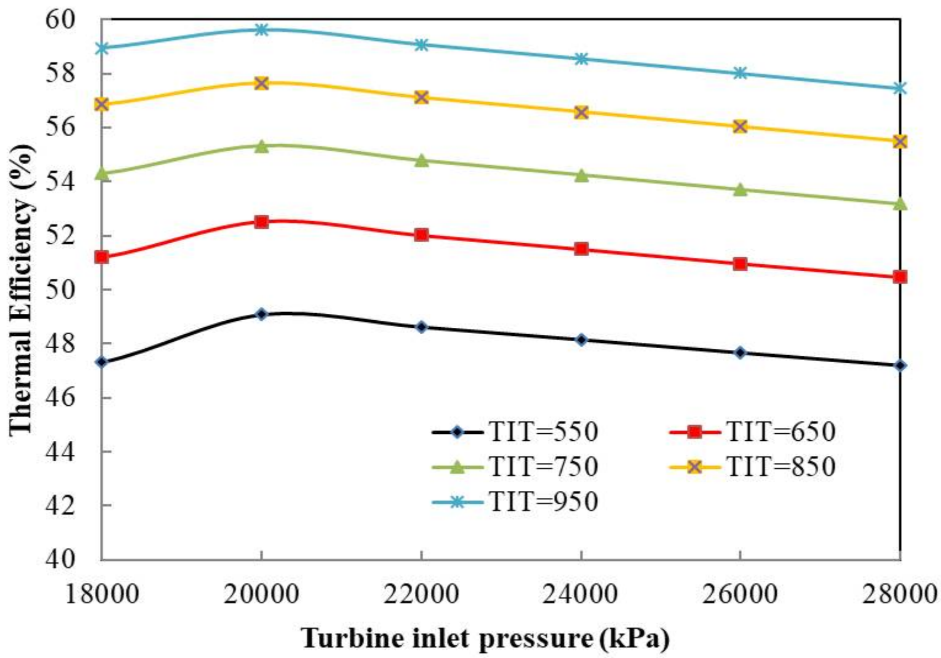

Fig.8. Effect of TIP on the thermal efficiency at different turbine inlet temperature

Influence of cycle pressure ratio on the thermal efficiency

The effect of pressure ratio on thermal efficiency is shown in fig.9. The curve first rises, reaches maximum and then decreases. For PR above 2.7 the cycle becomes transcritical. As CIT increases the slope of curve becomes gentler. An optimum value of pressure ratio exists for each particular CIT. The value of optimum PR is inversely proportional to CIT. Optimum PR of 2.6 exists for cycle having CIT at 32 °C and optimum

PR of 2.2 exists for cycle having CIT at 48 °C. However, optimum conditions never depend on a single input variable parameter. Although it depends on operating conditions and performance characteristics.

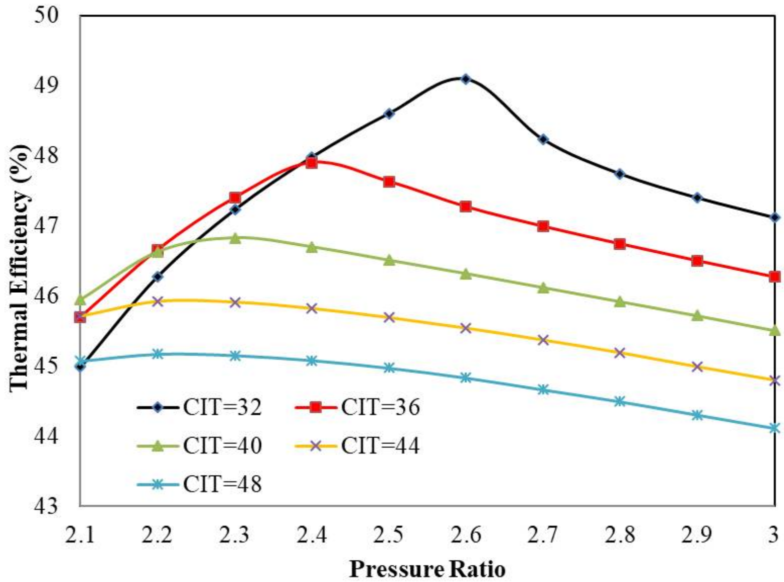

Fig.9. Influence of pressure ratio on the thermal efficiency in a recompression supercritical CO_2 cycle at different CIT

Influence of intermediary pressure on cycle performance

The effect of intermediate pressure on thermal efficiency is shown in fig.10. Thermal efficiency curve first increases, reaches maximum and then decreases. It is observed from the plot that an optimum intermediate pressure exists at same point for all turbine inlet temperature. Moreover, thermal efficiency decreases with rise in TIT. The highest efficiency of 49.08% is attained at an intermediary pressure of 14000 kPa and 550°C TIT. Fig.11. shows the effect of intermediate pressure on net work done at TIP. The curve for work done first increases, reaches maximum and then decreases. For TIP=28MPa the curve initially increases and then tends to be constant.Net work done is increase with increase in TIP. It is observed that an optimum value of Intermediate pressure based on net work done exists for TIP below 28MPa.

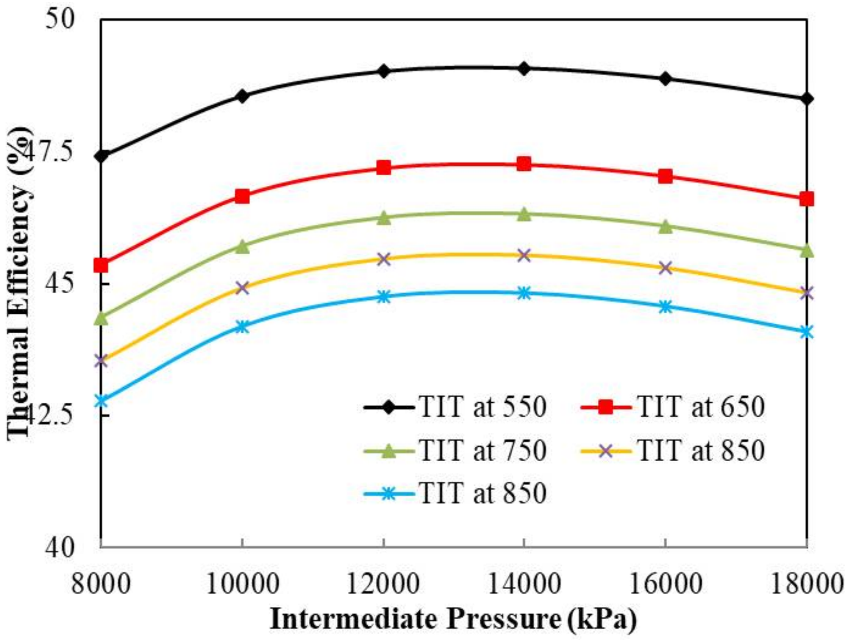

Fig.10. Effect of Intermediate Pressure on the Efficiency in a recompression (with reheat) supercritical CO_2 cycle at different TIP

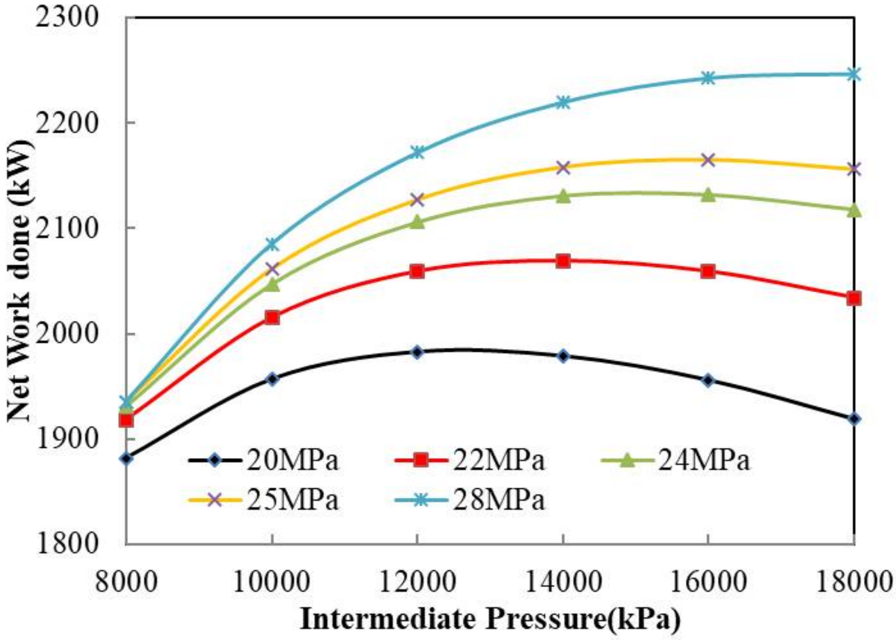

Fig.11. Effect of Intermediate Pressure on the Net Work done in a recompression (with reheat) supercritical CO_2 Brayton cycle at different TIP

Effects of cycle input parameters on performance of Turbomachinery

Parametric analysis of performance of Turbomachinery is shown in following section. Fig.12. illustrate the effect of TIT on work done by turbomachinery. Work done by HP and LP turbine is directly proportional to the Turbine inlet temperature. However, work done by main compressor and pre compressor remains constant at any TIT.

Fig.13. represents the effect of compressor inlet temperature on work done by turbomachinery. The work done by main compressor is directly proportional to the compressor inlet temperature because inlet of main compressor is exit of pre-cooler and work done by Pre compressor is inversely proportional to CIT. Although, work done by both turbine remains constant for all given values of CIT.

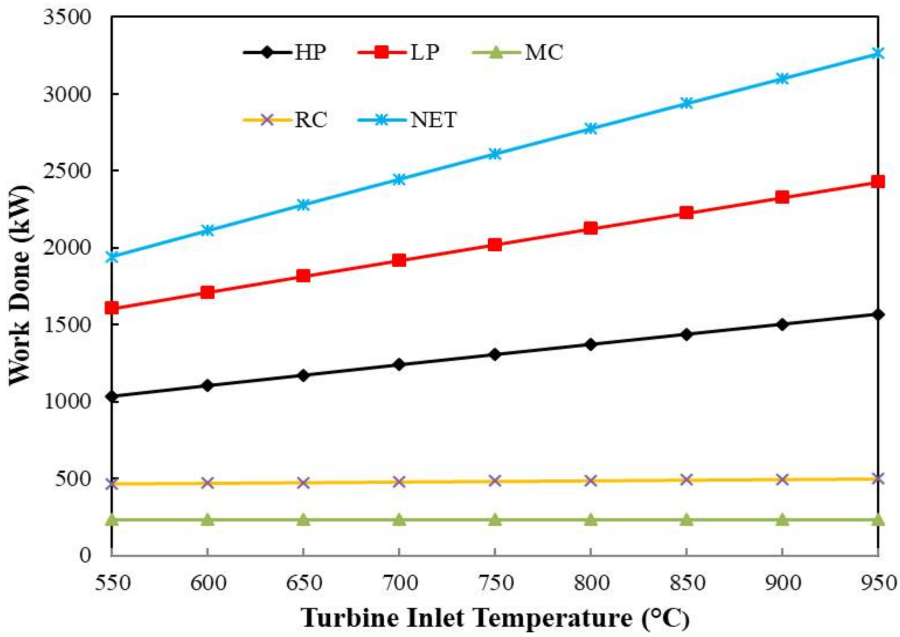

Fig.12. Effect of TIT on the Work done by Turbomachinery in a recompression (with reheat) supercritical CO_2 cycle

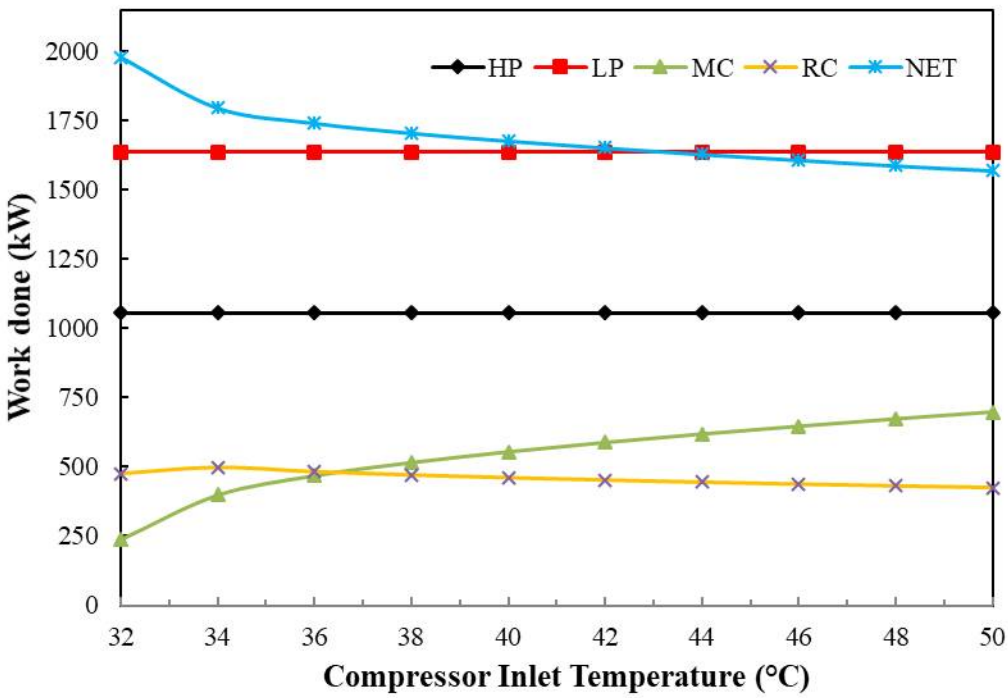

Fig.13. Effect of Compressor Inlet Temperature on the Work done by Turbomachinery in a recompression supercritical CO_2 cycle

CONCLUSIONS

The present study provides a detailed parametric study of a biomass fired s-CO_2 cycle. It proves a huge potential of the supercritical cycle for power production efficiently. The main conclusions drawn from the study are summarizes as:

- For supercritical CO_2 Brayton cycle, CIT should be low as possible and near to critical point. If, CIT declines below Critical point then two-phase region may occur inside compressor which will surely cause damage to blade.
- The energy efficiency of thermodynamic cycle rises with increase in TIT.
- An optimum PR and TIP exists for cycle at which it reaches maximum thermal efficiency for particular operating parameters.
- Turbine work done increases with increase in TIT while compressor work done remain constant. Hence, net work done is directly proportional to TIT.
- Net work done decrease with increase in CIT. However, the work produced by turbine is unaffected by change in CIT.
- Net work done increases with TIP but thermal efficiency decreases. The turbine work remains constant with change in TIP.
- An optimum intermediate pressure exists at which cycle exhibits maximum thermal efficiency.

Ultimately, it can be stated that this study has proven the high potential of the supercritical CO_2 recompression with reheat cycle for power production through biomass.

Nomenclature

Abbreviations

CIT-Compressor inlet temperature; CIP-Compressor inlet temperature; HTR –High temperature recuperaturer; HP-High pressure; IP-Intermediate pressure; LP-Low pressure; LTR- Low temperature recuperaturerMC-Main compressor; NET-Net work done; PR-Pressure ratio RC –Re-compressor; SR-Split ratio; TIT-Turbine inlet temperature; TIP -Turbine inlet pressure

Symbols

W-Work done (kW); Q-Heat(kW); h- Enthalpy (kJ/kg); T-Temperature (C); P-Pressure (kPa or Mpa); $\dot{m}$ - Mass flow rate (kg/s)

Greek Leters

η -efficiency ε -effectiveness

Subscript

max-maximum*;* min-minimum*;* in-inlet*;* out-outlet*;* ise-isentropic

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