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Energy and exergy based thermodynamic analysis of reheat and regenerative Braysson cycle

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A B S T R A C T

The conventional Braysson cycle has not found practical use due to the difficulty in achieving isothermal compression. To make its implementation a reality, the original cycle has been modified by incorporating regenerator and a cooler before the final compression process. Reheating was included for augmenting the power output. Expressions for exergy efficiency and exergy destruction for all the components are derived along with the energy and exergy efficiencies of the complete cycle. The effects of maximum temperature, pressure ratio and number of compression stages on the cycle efficiencies have been evaluated. It has been found that the exergy destruction in the combustion chamber and reheater put together accounts for more than 55% of the total exergy destruction. The cycle efficiency is maximum at an optimum pressure ratio which itself is found to be a function of maximum temperature in the cycle. The energy and exergy efficiency of the cycle equals the efficiency of normal Braysson cycle at a much lower pressure ratio. The efficiency achieved through the modified cycle with 2 stages of compression is only 2.2% less than the efficiency through ideal isothermal compression for a pressure ratio of 3 and turbine inlet temperature of 1200 K.

1. Introduction

The Braysson cycle is a hybrid cycle composed of Brayton and Erricson cycles. It has high temperature heat addition process - a Brayton cycle and a low temperature heat rejection process - an Erricson cycle. A detailed parametric thermodynamic analysis is presented by Frost et al. [1] who actually proposed this cycle. Thereafter, the Braysson cycle was subjected to numerous studies and reviews by researchers based on both first and second laws.

Zheng et al. [2] carried out a second law analysis of Braysson cycle. He has shown that the exergy loss in the combustor is the largest in the Braysson cycle and both specific work and exergy efficiency of the cycle are larger than those of Brayton cycle. Performance optimization of endo-reversible Braysson cycle with heat resistant losses in the hot and cold side heat exchanger is performed by using finite time thermodynamics by Zheng et al. [3]. Zhou et al. [4] investigated the influence of multi-irreversibilities on the performance of the Braysson heat engine. Yasin et al. [5] performed the analysis of endo-reversible Braysson cycle based on ecological criteria that includes finite rate heat transfer irreversibility. The ecological objective function is defined as the power output minus the loss power, which is equal to the product of environmental temperature and entropy production rate. The design parameters for maximization of the objective function are determined.

Shiyan Zhang et al. [6] presented a novel model of the solar-driven thermodynamic cycle system which consists of a solar collector and a Braysson heat engine. They optimized the performance characteristics of the system on the basis of linear heat-loss model of solar collector and the irreversible cycle model of a Braysson heat engine. Srinivas et al. [7] performed second law analysis of an irreversible Braysson cycle. Lanmei et al. [8] investigated an irreversible solar driven Braysson heat engine by taking into account the temperature dependent heat capacity of the working fluid, radiation-convection heat losses of solar collector and irreversibilities resulting from heat transfer and non-isentropic compression and expansion processes. Demos et al. [9] proposed the incorporation of regulated water injection during the final compression to maintain constant temperature due to evaporation. They reported that the injection process adopted has a minimal

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Nomenclature

c_p	specific heat at constant pressure of both air and gases (kJ/kg K)
P	pressure (N/m^2)
T	temperature (K)
TIT	turbine inlet temperature (or) maximum temperature (K)
h	specific enthalpy (kJ/kg)
H	total enthalpy (kJ)
s	specific entropy (kJ/kg K)
m_f	mass flow rate of fuel in the combustion chamber (kg/s)
m_{f1}	mass flow rate of fuel in the reheater (kg/s)
m_g	mass flow rate of gases through the main gas turbine(kg/s)
m_{g6}	mass flow rate of gases through the reheat gas turbine(kg/s)
l.c.v	lower calorific value of fuel (kJ/kg)
e_f	specific exergy of fuel (kJ/kg)
r_p	pressure ratio across the main compressor
r_{p1}	pressure ratio across the main turbine
r_{p2}	pressure ratio across the reheat turbine
r_{pi}	pressure ratio across a stage of multi-stage intercooled compressor
r_{po}	overall pressure ratio across the multi-stage intercooled compressor
N	number of stages of multi-stage intercooled compressor
T_0	dead state temperature (K)
η_c	isentropic efficiency of the main compressor
η_{cs}	isentropic efficiency of a stage of multi-stage compressor
η_{en}	energy efficiency
η_{ex}	exergy efficiency
η_{reg}	first law efficiency of regenerator
γ	ratio of specific heats
Ψ	specific exergy (kJ/kg)
K	$(\gamma - 1)/\gamma$
ΔT	temperature rise in a stage of multistage compression process (K)

affect on the ideal cycle efficiency. Demos et al. [10] also proposed a multistep intercooled compression process on a solar-driven Braysson heat engine as a feasible solution for implementing isothermal compression in Braysson cycle. Zhang et al. [11] and Sadatsakkak et al. [12] have performed optimization of Braysson cycle using optimization tools.

The Braysson cycle has not found practical implementation due to the difficulty in achieving isothermal compression. Except by Demos et al. [9,10], no significant work has been reported on the means of achieving isothermal compression. Hence an attempt has been made in this work to modify the conventional Braysson cycle by introducing the regenerator and cooler before the final compression process to decrease the compression work. Reheating is also incorporated for augmenting the power output. Regeneration and reheating [13–21] has drawn a lot of attention from researchers for improving the performance of conventional cycles. As the first law based analysis does not give a true representation of the losses of work and scope for potential improvements of efficiency [22], the present work mainly focuses on the exergy analysis of the proposed cycle.

2. System description

A schematic diagram of Braysson cycle with regeneration and reheating is shown in Fig. 1. The cycle consists of a compressor, combustion chamber, main gas turbine, reheater, regenerator, reheat gas turbine, cooler and a multistage compressor. Both the main turbine and the reheat gas turbine are mechanically coupled to electric generators and compressors. The gases at the exit of the main turbine are reheated to the maximum temperature of the cycle before being passed into the reheat turbine where they further expand to sub-atmospheric pressure. The regenerator improves the cycle efficiency by utilizing the waste heat of the gases on the downstream of reheat turbine. The gases exiting the regenerator are cooled further before letting them into the multistage compressor. The gases are then pressurized to atmospheric pressure in the multistage compressor and exited to the ambient.

3. Description of thermodynamic cycle

The T-S plot of the Thermodynamic cycle is represented in Fig. 2 and described below.

3.1. Polytropic compression (1–2)

The compression process has been accomplished using a roto-dynamic compressor which is mechanically coupled to the main turbine.

3.2. Regeneration (2–3) & (7–8)

Regeneration has been accomplished by using a counter flow heat exchanger. The heat of exhaust gases from the reheat turbine is transferred to the high pressure air through regenerative heat transfer process. The process is assumed to be Isobaric.

3.3. Isobaric heat addition (3–4)

The heat addition process occurs in the combustion chamber by the introduction of fuel through spray injectors.

3.4. Polytropic expansion (4–5) & (6–7)

The expansion of working fluid takes place in the main and reheat gas turbines with intermediate reheat. The intermediate pressure at which reheating is carried out is optimized for maximum work output of the turbines. The downstream pressure of reheat turbine is considered as vacuum pressure.

3.5. Isobaric heat addition in reheater (5–6)

The power developed during expansion by the gases operating between two pressure levels can be augmented by utilizing multistage expansion with reheating. It has been assumed that this process doesn't involve any pressure losses due to friction (or) turbulence.

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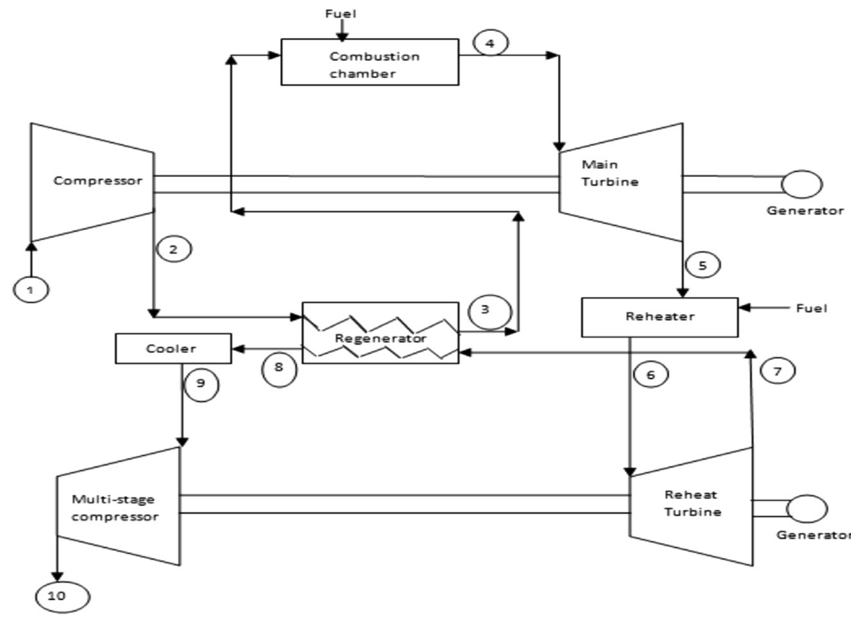

Fig. 1. Schematic diagram of Braysson cycle with reheat and regeneration.

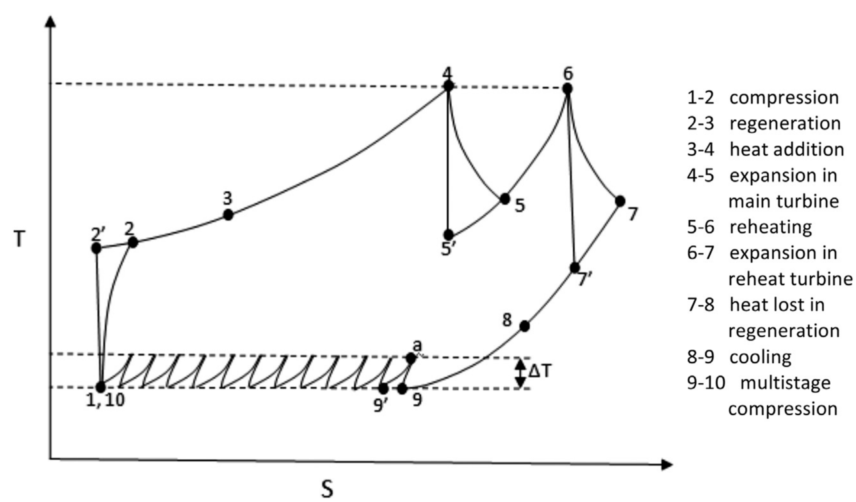

Fig. 2. T-S diagram of Braysson cycle with reheat and regeneration.

3.6. *Isobaric heat rejection (8–9)*

The gases from the regenerator are cooled in a heat exchanger before letting them into the multistage compressor to room temperature.

3.7. *Isothermal heat rejection (9–10)*

The isothermal heat rejection has been achieved with the help of a multistage intercooled compressor, which is powered by the reheat turbine. The compression and heat rejection take place simultaneously and the pressure rises to atmospheric pressure.

4. Thermodynamic analysis

The exergy of a system at a state is defined as the maximum work potential that can be obtained as the system undergoes a reversible process to dead state i.e., the state of the environment. Exergy is a property that depends not only on the state of the system but also on the state of the environment.

The exergy of a system of flowing fluid is given as Ψ

$$= \mathrm{h} - \mathrm{T_0 s} \tag{1}$$

For a single stream of fluid undergoing a change of state, the exergy change between the two states (1) and (2) is given by:

$$\Psi_2 - \Psi_1 = (\mathrm{h_2} - \mathrm{h_1}) - \mathrm{T_0}(\mathrm{s_2} - \mathrm{s_1}) \tag{2}$$

The exergy efficiency of a system in general is defined as the ratio of exergy recovered to the exergy supplied to the system. For example, in a turbine, the exergy recovered is the useful shaft work obtained while the exergy supplied is the difference of the exergies of the fluid stream from the inlet to the outlet. For a compressor, the exergy recovered is the difference of the exergies of the working fluid from the inlet to the outlet, while the shaft work input is the exergy supplied.

$$\eta_{ex} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} \tag{3}$$

Exergy gets destroyed due to the irreversibilities like friction, turbulence, heat transfer through finite temperature difference and so on, associated with the process. The exergy destroyed is simply the difference of exergy supplied and the exergy recovered.

$$\text{Exergy destroyed} = \text{Exergy supplied} - \text{Exergy recovered}$$

$$\eta_{ex} = 1 - \frac{\textit{Exergy destroyed}}{\textit{Exergy supplied}} \tag{4}$$

To pin point the zones where exergy destruction is large, it is necessary to derive the expressions for the exergy efficiency and destruction in individual components.

4.1. Polytropic compression (1–2)

Assuming the mass flow rate of air (m_a) = 1 kg/sec,

The actual temperature of air at the exit of a compressor is

$$T_2 = T_1 + \frac{T_1 \cdot \left[r_p^{(\gamma-1)/\gamma} - 1\right]}{\eta_c} \tag{5}$$

Rate of exergy supplied to the compressor $= m_a c_p (T_2 - T_1) = c_p (T_1 \cdot [r_p^{(\gamma-1)/\gamma} - 1])/\eta_c$

On substituting of eq. (5);

Rate of exergy supplied to the compressor

$$= c_p \frac{T_1 \cdot \left[r_p^{(\gamma-1)/\gamma} - 1\right]}{\eta_c} \tag{6}$$

$$\begin{aligned}\text{Rate of exergy recovered} &= m_a\{c_p(T_2 - T_1) - T_0(S_2 - S_1)\} \\ &= c_p\left(T_1 \cdot \left[r_p^{(\gamma-1)/\gamma} - 1\right]\right)\Big/\eta_c \\ &\quad - T_0 c_p \left[\ln\left(\frac{T_2}{T_1}\right) - \frac{\gamma-1}{\gamma}\ln(r_p)\right]\end{aligned} \tag{7}$$

Using equations (3), (6), (7) the exergy efficiency is evaluated.

Rate of exergy destroyed in the compressor

$$= T_0 C_p \left[\ln\left(\frac{T_2}{T_1}\right) - \frac{\gamma-1}{\gamma}\ln(r_p)\right] \tag{8}$$

4.2. Determination of mass flow rates of fuel added in the combustion chamber and reheater

The first law efficiency of a regenerator is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

$$\eta_{reg} = \frac{H_3 - H_2}{H_7 - H_2} \tag{9}$$

Assuming specific heat of air and gases as 1.005 kJ/kg K, the temperature of air at the outlet of regenerator after simplification of eq. (9) is given by

$$T_3 = \eta_{reg} m_{g6}(T_7 - T_2) + T_2 \tag{10}$$

where, m_{g6} is the mass flow rate of gases flowing through the reheat turbine.

$$m_{g6} = m_a + m_f + m_{f1} = 1 + m_f + m_{f1}$$

The temperature at the exit of reheater is equal to the maximum temperature in the cycle, i.e., $T_4 = T_6 = T_{max}$

The intermediate pressure P_3 between the two stages of expansion has been evaluated based on the condition that the gross work output of the turbines is maximum, and is determined as

$$P_3 = \left\{\left(\frac{m_{g6}}{m_g}\right)^{\frac{\gamma}{2(\gamma-1)}}\right\}\sqrt{P_2 P_4} \tag{11}$$

where m_g is the mass flow rate of gases flowing through the main turbine,

$$m_g = m_a + m_f = 1 + m_f$$

Expression (11) can be rewritten in terms of pressure ratios of the main and the reheat turbines as

$$r_{p2} = r_{p1}\left[m_{g6}/m_g\right]^{\gamma/2(\gamma-1)} \tag{12}$$

The temperature of gases at the exit of main turbine is given as

$$T_5 = T_4\left[1 - \eta_T\left(1 - r_{p1}^{(1-\gamma)/\gamma}\right)\right] \tag{13}$$

Similarly, the temperature of gases at the exit of reheat turbine is given as

$$T_7 = T_6\left[1 - \eta_T\left(1 - r_{p2}^{(1-\gamma)/\gamma}\right)\right] \tag{14}$$

Making an energy balance for the combustion chamber we have,

$$m_a h_3 + m_f(\text{l.c.v}) = m_g h_4 \tag{15}$$

Substituting for T_3 from eq. (10) and $m_a = 1$ kg/s and rearranging the terms, we have

$$\begin{aligned}&m_f\{\eta_{reg} \cdot c_p(T_7 - T_2) - c_p T_4 + \text{l.c.v}\} + m_{f1} \cdot \eta_{reg} c_p (T_7 - T_2) \\ &\quad = c_p(T_4 - T_2) - \eta_{reg} \cdot c_p(T_7 - T_2)\end{aligned} \tag{16}$$

Making an energy balance for the reheater,

$$m_g h_5 + m_{f1}(\text{l.c.v}) = m_{g6} h_6$$

On rearranging the terms,

$$m_f c_p (T_6 - T_5) - m_{f1}(\text{l.c.v} - c_p T_6) = c_p (T_5 - T_6) \tag{17}$$

An iterative procedure is followed in determining T_5, T_7, m_f & m_{f1} from equations (12)–(14), (16), (17)

Making an energy balance for the regenerator, we have

$$m_{g6}(h_7 - h_8) = m_a(h_3 - h_2) \tag{18}$$

On simplification, the temperature of gases at the exit of regenerator is given as

$$T_8 = T_7 - \left(\frac{1}{(1 + m_f + m_{f1})}\right)(T_3 - T_2) \tag{19}$$

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4.3. Isobaric heat addition in the regenerator (2–3) & (7–8)

The exergy efficiency of regenerator is defined as the ratio of the rate of increase in the exergy of air to the rate of decrease in the exergy of hot gases.

$$\eta_{ex} = \frac{m_a(e_3 - e_2)}{m_{g6}(e_7 - e_8)} = \frac{[(h_3 - h_2) - T_0(s_3 - s_2)]}{m_{g6}[(h_7 - h_8) - T_0(s_7 - s_8)]} = \frac{\left[c_p(T_3 - T_2) - T_0 c_p \ln\left(\frac{T_3}{T_2}\right)\right]}{m_{g6}\left[c_p(T_7 - T_8) - T_0 c_p \ln\left(\frac{T_7}{T_8}\right)\right]} \tag{20}$$

The exergy destroyed is simply the difference of the decrease in exergy of gases and the increase in exergy of air in unit time.

$$\text{Rate of exergy destruction} = m_{g6}c_p\left[(T_7 - T_8) - T_0\ln\left(\frac{T_7}{T_8}\right)\right] - m_a c_p\left[(T_3 - T_2) - T_0\ln\left(\frac{T_3}{T_2}\right)\right] \tag{21}$$

4.4. Isobaric heat addition in combustion chamber (3–4)

Heat addition in the combustion chamber is assumed to take place at constant pressure by continuous combustion of fuel. Liquid octane is considered as the fuel. Exergy destruction in the combustion chamber is due to irreversible combustion process.

$$\text{Rate of exergy recovered} = \text{Exergy of gas at outlet of combustion chamber} - \text{Exergy of air at inlet of combustion chamber}$$
$$\text{Rate of exergy recovered} = m_g\left\{c_p\left(T_4 - T_0\ln\left(\frac{T_4}{T_0}\right)\right)\right\} - m_a\left\{\left(T_3 - T_0\ln\frac{T_3}{T_0}\right)\right\} \tag{22}$$

$$\text{Rate of exergy supplied} = m_f \cdot e_f \tag{23}$$

$$\text{Rate of exergy destruction} = m_f e_f - \left\{m_g c_p\left[T_4 - T_0\ln\left(\frac{T_4}{T_0}\right)\right] - m_a c_p\left[T_3 - T_0\ln\left(\frac{T_3}{T_0}\right)\right]\right\} \tag{24}$$

4.5. Polytropic expansion in turbine (4–5)

$$\text{Rate of exergy supplied} = m_g\left\{c_p(T_4 - T_5) - c_p T_0\left[\ln\left(\frac{T_4}{T_5}\right) - ((\gamma - 1)/\gamma)\ln(r_{p1})\right]\right\} \tag{25}$$

The power output of the turbine is the exergy recovered.

$$\text{Rate of exergy recovered} = m_g c_p(T_4 - T_5) \tag{26}$$

$$\text{Exergy destruction rate} = m_g\left\{c_p(T_4 - T_5) - c_p T_0\left[\ln\left(\frac{T4}{T5}\right) - ((\gamma - 1)/\gamma)\ln(r_{p1})\right] - c_p(T_4 - T_5)\right\} \tag{27}$$

4.6. Isobaric heat addition in reheater (5–6)

$$\text{Rate of exergy recovered} = m_{g6}[h_6 - T_0 s_6] - m_g[h_5 - T_0 s_5] = m_{g6}C_p\left(T_6 - T_0\ln\frac{T_6}{T_0}\right) - m_g C_p\left[T_5 - T_0\ln\left(\frac{T_5}{T_0}\right)\right] \tag{28}$$

$$\text{Rate of exergy supplied} = m_{f1} * e_f \tag{29}$$

$$\text{Rate of exergy destruction} = e_f m_{f1} - \left\{m_{g6}c_p\left(T_6 - T_0\ln\left(\frac{T_6}{T_0}\right)\right) - m_g c_p\left(T_5 - T_0\ln\left(\frac{T_5}{T_0}\right)\right)\right\} \tag{30}$$

4.7. Polytropic expansion in reheat turbine: (6–7)

$$\text{Rate of exergy supplied} = m_{g6}c_p(T_6 - T_7) - c_p T_0\left[\ln\left(\frac{T_6}{T_7}\right) - \frac{\gamma - 1}{\gamma}\ln r_{p2}\right] \tag{31}$$

$$\text{Rate of exergy recovered} = m_{g6}c_p(T_6 - T_7) \tag{32}$$

$$\text{Rate of exergy destruction} = m_{g6}\left\{c_p(T_6 - T_7) - c_p T_0\left[\ln\left(\frac{T_6}{T_7}\right) - \frac{\gamma - 1}{\gamma}\ln r_{p2}\right] - c_p(T_6 - T_7)\right\} \tag{33}$$

4.8. Isobaric heat rejection in cooler: (8–9)

It is assumed that the gases are cooled up to the ambient temperature at the end of isobaric heat rejection in cooler. The exergy efficiency of the cooler is the ratio of the increment in the exergy of cooling water to the decrease in the exergy of gases. However, the exergy of cooling water is not recovered and is lost as waste heat to the environment. Hence it can be concluded that the exergy recovered from the cooler is zero and therefore the second law efficiency is zero.

4.9. Perfectly intercooled multi-stage compressor: (9–10)

Consider an intermediate stage of multi-stage compression in which the temperature at the end of actual compression is T_a and at the end of isentropic compression is T'_a.

$$\frac{T_{a'}}{T_9} = r_{pi}^{(\gamma-1)/\gamma} \tag{34}$$

The isentropic efficiency of any stage is given as:

$$\eta_c = \frac{T_{a'} - T_9}{T_a - T_9} \quad (35)$$

For 'N'stages, the pressure ratio per stage (r_{pi}) is related to overall pressure ratio (r_{p0}) by the expression:

$$r_{pi} = r_{p0}^{1/N} \quad (36)$$

$$\text{Specific exergy recovered} = N[e_a - e_9] - (N-1)[e_a - e_{9'}] = N[(h_a - h_9) - T_0(s_a - s_9)] - (N-1)[(h_a - h_{9'}) - T_0(s_a - s_{9'})]$$

Substituting the expressions (34)–(36), the specific exergy recovered can be rewritten as

$$N c_p T_9 \left[\frac{r_{p0}^{\frac{K}{N}} - 1}{\eta_{cs}}\right] - N c_p T_0 \left[\ln\left(1 + \frac{r_{p0}^{\frac{K}{N}} - 1}{\eta_{cs}}\right) - \frac{K}{N} \ln r_{p0}\right] - (N-1) c_p \left\{T_9 \left[\frac{r_{p0}^{\frac{K}{N}} - 1}{\eta_{cs}}\right] - T_0 \ln\left(1 + \frac{r_{p0}^{\frac{K}{N}} - 1}{\eta_{cs}}\right)\right\} \quad (37)$$

$$\text{Specific exergy supplied} = N[(h_a - h_9)] = N c_p T_9 \left[\frac{r_{p0}^{\frac{K}{N}} - 1}{\eta_{cs}}\right] \quad (38)$$

The rate of exergy destruction = m_{g6}*{Specific exergy supplied − Specific exergy recovered}

4.10. Overall energy efficiency of the cycle

It can be defined as the ratio of the net power output of the cycle to the rate of heat input to the system.

Net rate of work output of the cycle:

$$W_{net} = m_g c_p (T_4 - T_5) - m_a c_p T_1 \left[\left(r_p^K - 1\right)/\eta_c\right] + m_{g6} c_p (T_6 - T_7) - N m_{g6} c_p T_9 \left[(r_{po})^{K/N} - 1\right]/\eta_{cs} \quad (39)$$

The rate of heat supplied to the cycle consists of heat input in the combustion chamber and reheater. It is given by the expression:

$$Q = \text{l.c.v}\left(m_f + m_{f1}\right) \quad (40)$$

$$\eta_{en} = \frac{\textit{Net rate of work output of the cycle}}{\textit{rate of heat supplied to the cycle}}$$

$$\eta_{en} = \frac{m_g c_p (T_4 - T_5) - m_a c_p T_1 \left[\left(r_p^K - 1\right)/\eta_c\right] + m_{g6} c_p (T_6 - T_7) - N m_{g6} c_p T_9 \left[(r_{po})^{K/N} - 1\right]/\eta_{cs}}{\text{l.c.v}\left(m_f + m_{f1}\right)} \quad (41)$$

4.11. Overall exergy efficiency of the cycle

The limitation associated with first law is taken care of by the exergy efficiency. The overall exergy efficiency is computed considering the chemical exergy inputs through fuel to the system.

$$\text{Rate of chemical exergy supplied to the system} = e_f\left(m_f + m_{f1}\right) \quad (42)$$

The net rate of exergy recovered is the power output of the cycle, which is given by the expression (39).

The overall exergy efficiency of the cycle:

$$\eta_{ex} = \frac{\text{Net rate of exergyrecovered}}{\text{Rate of chemical exergysupplied}}$$

$$\eta_{ex} = \frac{m_g c_p (T_4 - T_5) - m_a c_p T_1 \left[\left(r_p^K - 1\right)/\eta_c\right] + m_{g6} c_p (T_6 - T_7) - N m_{g6} c_p T_9 \left[(r_{po})^{K/N} - 1\right]/\eta_{cs}}{e_f\left(m_f + m_{f1}\right)} \quad (43)$$

5. Results and discussion

The modified Braysson cycle with reheat and regeneration added is analysed in this report. From the expressions developed, the exergy efficiency and exergy destruction of each device is obtained. The overall energy and exergy efficiencies of the cycle are also evaluated. The following assumptions have been considered in the analysis. Isentropic efficiency of the compressor, main turbine and reheat turbine are 0.9, 0.9 & 0.9 respectively. Stage efficiency of the Multi-stage compressor is 0.9 and effectiveness of regenerator is 0.85. It has been assumed that there are no pressure drops of the working fluid either in the devices or in the ducting. Consideration has also been given to the fuel added in the combustion chamber and reheater in order to study the exergy destruction rate during combustion phenomena. The analysis has been performed considering octane as fuel which has a lower calorific value of 44,427 kJ/kg and specific chemical exergy 47,346 kJ/kg, Nag [23].

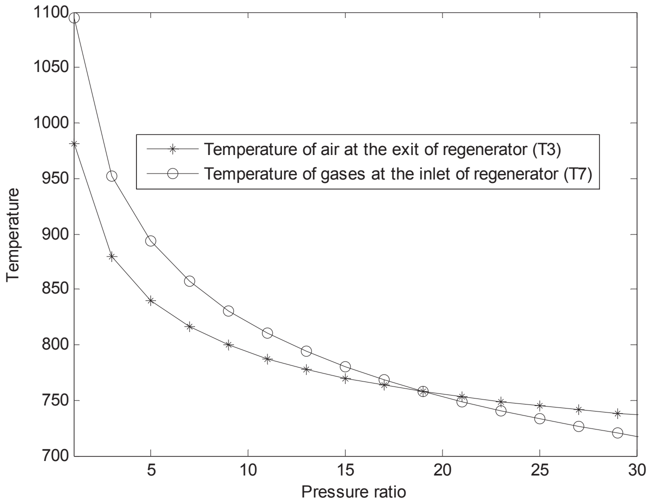

Fig. 3. Pressure ratio Vs temperature of air at the exit of regenerator (T_3) and temperature of gases at the inlet of regenerator (T_7) (TIT = 1200 K, N = 10).

5.1. Effect of pressure ratio on cycle efficiencies

It has been observed that there is a limiting pressure ratio, when regeneration is adopted. Though regeneration increases the thermal efficiency of the cycle, it imposes a limit on the maximum pressure ratio in the cycle. Pressure ratios higher than the limiting value would mean that the temperature of compressed air exceeds the temperature of gases at the exit of reheat turbine. In such eventuality, regeneration becomes impracticable. Fig. 3 exhibits the variation of temperature of gases and compressed air with pressure ratio for TIT (turbine inlet temperature) of 1200 K. The limiting pressure ratio in this particular case is 19. The limiting pressure ratio is a function of TIT as seen from Fig. 4. At higher TITs, the gases can expand through higher pressure without adversely affecting the regeneration downstream.

Table 1, shows the variation of energy and exergy efficiencies of the cycle with pressure ratio for TIT = 1200 K. There is an optimum pressure ratio at which the efficiencies reach a peak value. The optimum pressure ratio of the cycle is also a function of TIT as depicted in Fig. 5. The efficiencies increase abruptly with pressure ratio up to the optimum value, and thereafter decline gradually with further rise in pressure ratio as shown in Figs. 6 and 7. For lesser pressure ratio, the work output of the turbines is minimal, though regeneration is effective and reduces the heat input from the external source. The combined effect is to produce lower cycle efficiencies. At higher pressure ratios above optimum value, the influence of regeneration diminishes, thereby lessening the cycle efficiency.

Table 1
Effect of pressure ratio on the energy and exergy efficiencies of the cycle (N = 10, TIT = 1200 K).

Sl. No.	Pressure ratio	η_{en} (%)	η_{ex} (%)
1	2.8	52.36	49.13
2	3.0	52.43	49.19
3	**3.06**	**52.44**	**49.21**
4	3.4	52.43	49.19
5	3.6	52.38	49.15
6	3.8	52.32	49.09
7	4.0	52.23	49.01

The bold lettering refers to maximum values of energy and exergy efficiency that are obtained at the optimum pressure ratio.

5.2. Effect of maximum temperature on cycle efficiencies

From Table 2, it can be seen that as the inlet temperature to the gas turbines is enhanced, there is an appreciable increase in both the efficiencies.

As the maximum temperature rises from 800 K to 1600 K the energy and exergy efficiencies increase by about 20.36% and 19.11% respectively. These efficiencies represent the maximum values at the corresponding maximum temperatures as they have been evaluated at optimum pressure ratio.

5.3. Effect of the number of stages of multistage compressor on cycle efficiencies

Figs. 6 and 7 represent the effect of variation of number of stages of multistage compressor on both the efficiencies of the cycle (TIT = 1200 K). The efficiencies increase marginally with the increase in number of stages from 1 to 10 and thereafter the effect of number of stages on the efficiencies is minimal. For N = 2, η_{ex} = 48.8% and for N = 10, η_{ex} = 49.21%. The maximum exergy

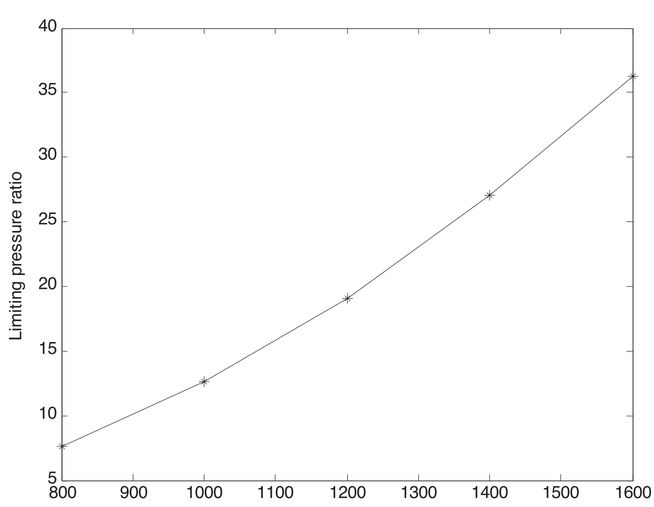

Fig. 4. Maximum temperature in the cycle Vs Limiting pressure ratio (N = 10).

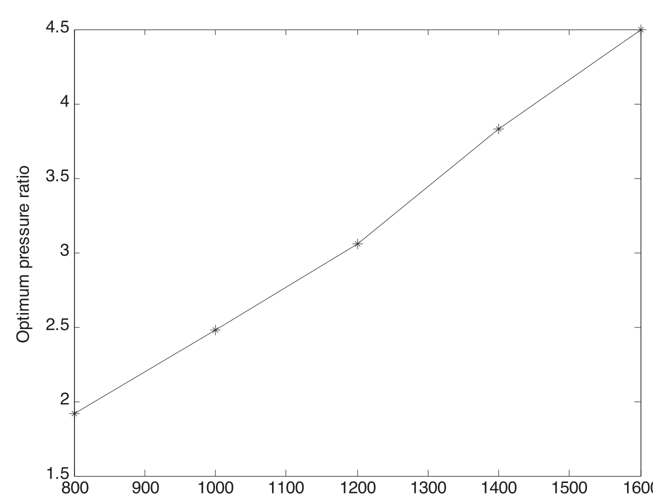

Fig. 5. Maximum temperature in the cycle Vs Optimum pressure ratio (N = 10).

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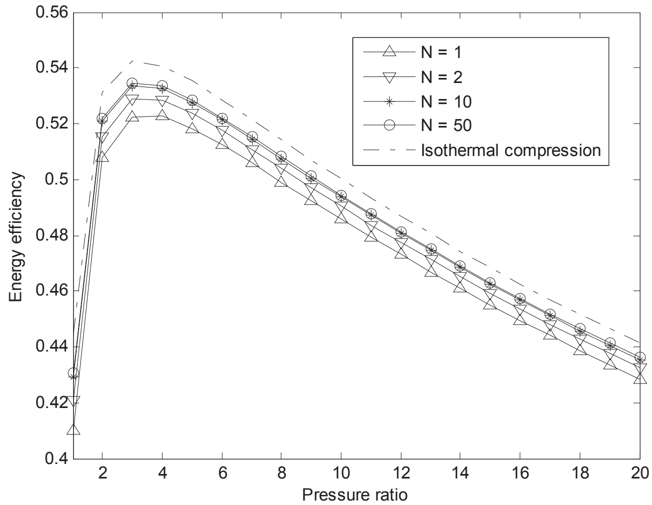

Fig. 6. Pressure ratio Vs overall energy efficiency of the cycle for varying number of stages of multi-stage compressor (TIT = 1200 K).

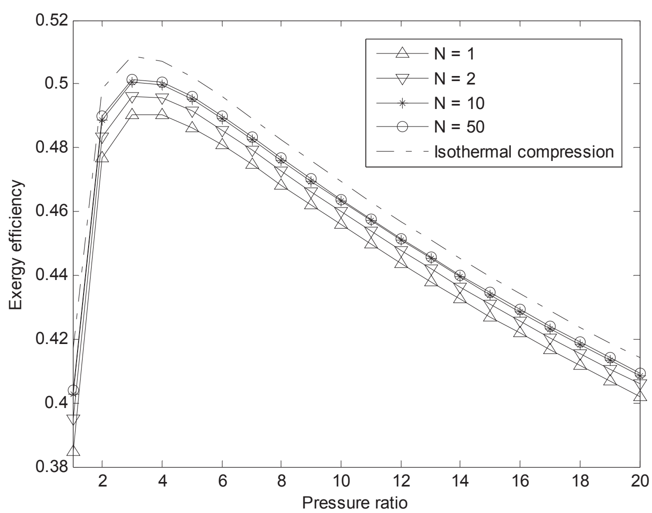

Fig. 7. Pressure ratio Vs overall exergy efficiency of the cycle for varying number of stages of multi-stage compressor (TIT = 1200 K).

efficiency that is obtained under ideal isothermal condition is 51%. Thus the increase in efficiency is only 2.2% for increase in number of stages from 2 to a very high value, which represents isothermal compression. Hence the effect of number of stages of final compression is highly nullified in the proposed cycle and therefore this cycle can be considered as a viable alternative to combined cycle power plant.

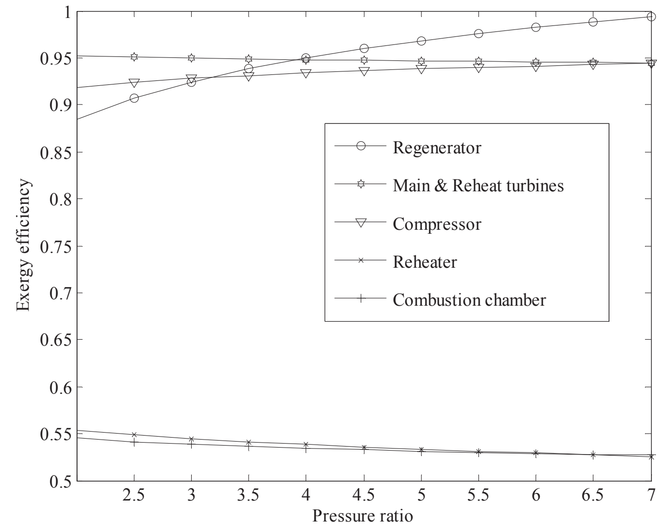

Fig. 8. Exergy efficiency of various components for TIT = 800 K & N = 10.

5.4. Component wise exergy efficiency

It is necessary to evaluate the exergy efficiency and rate of exergy destruction in the various devices that the system comprises. Figs 8–10 show the variation of efficiencies with pressure ratio drawn at different TITs. From these plots, the following conclusions can be drawn for the various components.

5.4.1. Compressor

The exergy efficiency of the compressor increases with pressure ratio and is independent of the maximum temperature in the cycle for a given pressure ratio. For pressure ratio of 7.0, the exergy efficiency of the compressor is 94.51%. It can be inferred that for higher TITs higher pressure ratios can be adopted and thus the exergy efficiency of the compressor will increase correspondingly.

5.4.2. Regenerator

The exergy efficiency of a regenerator increases with pressure ratio, as can be observed from the plots. The irreversibility associated with the heat transfer process during regeneration reduces with decrease in temperature difference between the hot gases and the compressed air. This condition is achieved at higher pressure ratio values though the heat recovered through regeneration will lessen with the decrease in temperature difference between the two fluids. Hence the highest possible exergy efficiency of 100% can be achieved ideally at the limiting value of pressure ratio. From the plots, it can also be observed that for a given pressure ratio value of, say 7.0, the exergy efficiency diminishes from 99% to 95% for rise in TIT from 800 K to 1200 K.

Table 2
Energy and exergy efficiencies of the cycle at optimum pressure ratio for different maximum temperatures (N = 10).

Sl. No.	Maximum temperature, TIT (K)	Optimum pressure ratio	η_{en} (%)	η_{ex} (%)
1	800	1.92	39.31	36.88
2	1000	2.48	47.03	44.13
3	1100	2.78	49.95	46.27
4	1200	3.06	52.44	49.21
5	1400	3.83	56.5	53.01
6	1600	4.5	59.67	55.99

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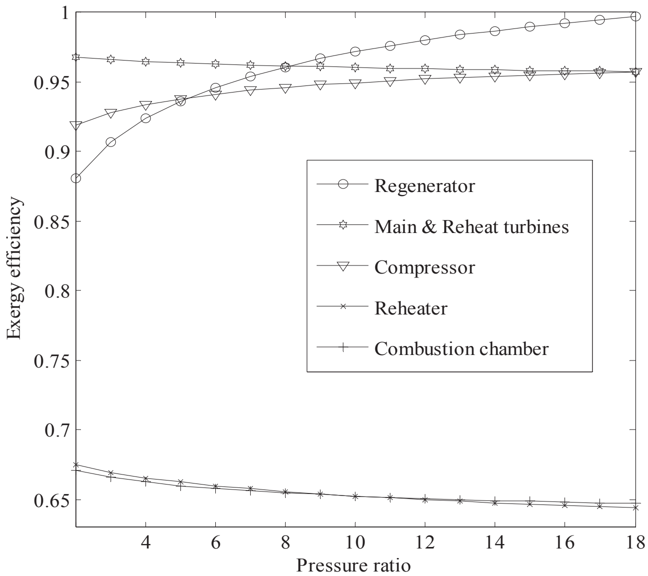

Fig. 9. Exergy efficiency of various components for TIT = 1200 K& N = 10.

5.4.3. Combustion chamber

The combustion phenomenon is an irreversible process where the transformation of chemical exergy into thermo-mechanical exergy takes place. The exergy efficiency of the combustion chamber varies with both the pressure ratio and the TIT. It can be seen that with the increase in maximum temperature in the cycle, say from 800 K to 1200 K, the second law efficiency increases substantially from 54.5% to 67.08% for a pressure ratio of 2. An increase of about 12.58% occurs, which can be attributed to the decrease in exergy destruction.

For TIT of 800 K, the exergy efficiency of combustion chamber drops marginally from 54.5% to 52.7% with the pressure ratio increasing from 2.0 to 7.0. The exergy efficiency of the regenerator is affected by the range of expansion of gases.

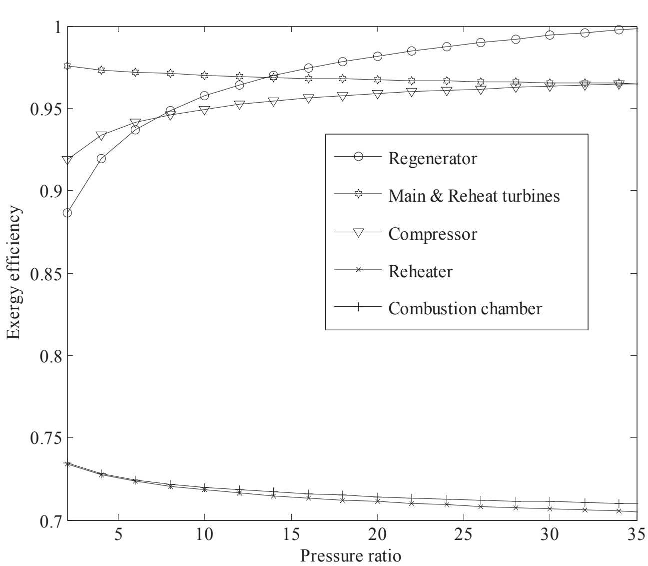

Fig. 10. Exergy efficiency of various components for TIT = 1600 K& N = 10.

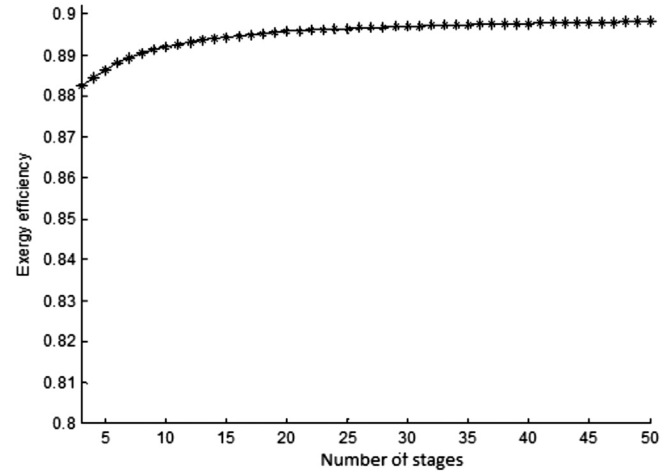

Fig. 11. Exergy efficiency Vs Number of stages for multistage compressor.

5.4.4. Reheater

The excess air available in the hot gases is utilized for combustion of the fuel added in the reheater. The exergy efficiency of the reheater shows the same tendency of variation with pressure ratio and TIT as that of combustion chamber.

5.4.5. Main and reheat turbines

The exergy efficiency of both the turbines increases with increase in the inlet temperature of the gas turbines, but decreases marginally with the increase in pressure ratio at a given TIT. For a TIT of 800 K, the efficiency falls by 0.81% for an increment in pressure ratio from 2.0 to 7.0. The efficiency increases by 2.33% for a pressure ratio of 2.0, when the TIT is raised from 800 K to 1600 K.

5.4.6. Multistage compressor

The variation of exergy efficiency of multistage compressor with the number of stages is shown in Fig. 11. It can be seen that the exergy efficiency increases with the number of stages reaching an asymptotic value of about 90%.

5.5. Exergy destruction rate

It is also of prime importance to evaluate the overall rate of exergy destruction in the system and identify the devices in which the exergy destruction rate is large. This analysis is limited to TITs of 1200 K and 1600 K. It can be seen from Figs. 12 and 13 that 55%–65% of the total exergy is destroyed in the combustion chamber and reheater. This can be due to the reason that the combustion process is highly irreversible and leads to maximum exergy loss. This can be reduced to a certain extent by increasing the number of reheat stages.

It is observed that the rate of exergy destruction in the cooler at the limiting pressure ratio is almost twice that of the value at the lowest pressure ratio. This can be attributed to the fact that the regenerative heat recovery process becomes less effective at higher pressure ratio. As the cycle considered for analysis is an open cycle, certain amount of exergy is invariably lost with the outgoing gases from the system. It is, therefore, imperative that the gases are cooled before letting them into the multistage compressor, lest the power required in multistage compression should be high.

The contribution of the regenerator to the total exergy destruction stands at around 14%–18% at the minimum pressure ratio. This value decreases with increase in pressure ratio and approaches zero at the limiting value.

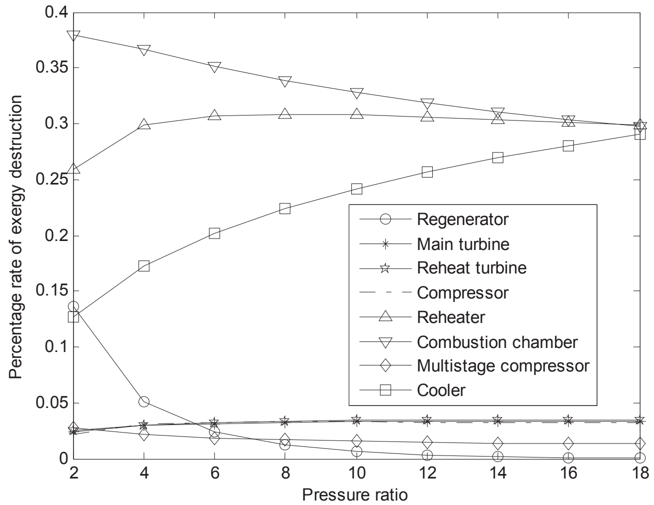

Fig. 12. Percentage rate of exergy destruction Vs Pressure ratio for various devices (TIT = 1200 K, N = 10).

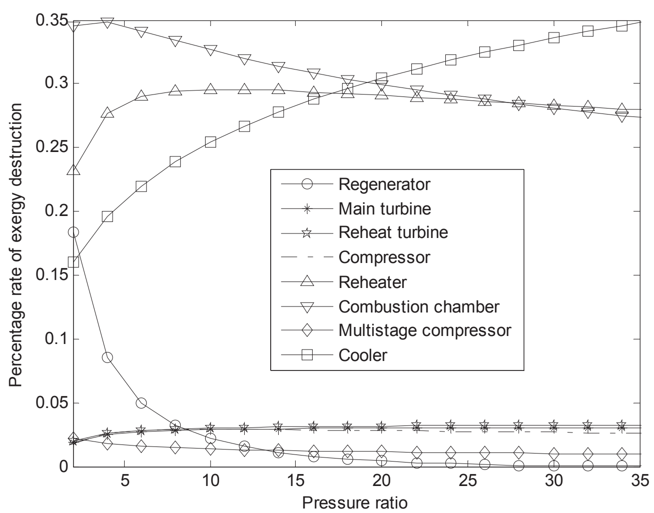

Fig. 13. Percentage rate of exergy destruction Vs. Pressure ratio for various devices (TIT = 1600 K, N = 10).

The remaining devices do not contribute appreciably towards the overall exergy destruction. Figs. 12 and 13 clearly indicate that the contribution of the compressor, turbines and multistage compressor combined is less than 0.05%.

5.6. Comparison

From Table 3, it is evident that the energy efficiency of this reheat and regenerative Braysson cycle reaches 60% at a maximum temperature of 1630 K and pressure ratio of 4.0.

Table 3
Energy efficiency of the cycle at different maximum temperatures (N = 10).

Sl. No	Maximum temperature (K)	η_{en} (%)
1	1580	59.33
2	1600	59.60
3	1620	59.87
4	**1630**	**60.00**
5	1640	60.12

The bold lettering refers to the condition that energy efficiency of 60% is obtained at a maximum temperature of 1630 K in this cycle as compared to the same efficiency being obtained at higher temperature of 1850–1950 K in the conventional Braysson cycle.

In a normal Braysson cycle as proposed by Frost et al. [1], this efficiency can be attained at a maximum temperature of 1850–1950 K and pressure ratio of 19.7. Thus there is a reduction in pressure ratio of about 79% and reduction in maximum temperature of about 12%. This shows a clear improvement in the proposed reheat and regenerative Braysson cycle.

The exergy efficiency of the cycle is also compared with the results reported by Zeng et al. [2]. At a TIT of 1100 K and pressure ratio of around 10, the efficiency reported by Zeng et al. is about 46%. From Table 2 it can be inferred that the same efficiency can be obtained in the present cycle at a much lower pressure ratio of 2.78 and TIT of 1100 K. Thus there is a reduction in pressure ratio of about 72%. Hence the cycle proposed in this paper stands better with reference to both energy and exergy efficiency.

6. Conclusions

The energy and exergy efficiency of Reheat and Regenerative Braysson cycle achieves the efficiency of normal Braysson cycle at a much lower pressure ratio. The pressure ratio is reduced by 79% for 60% energy efficiency and by 72% for 46% exergy efficiency. The analysis has included the exergy destruction in the components and an evaluation of the effects of pressure ratio, maximum temperature and number of compression stages on the performance of the cycle. The exergy destruction in the combustion chamber and reheater put together accounts for more than 55% of the total exergy destruction. The pressure ratio and maximum temperature have a pronounced effect on the exergy efficiency of the cycle. The increase in efficiency is only 2.2% for increase in number of stages from 2 to a very high value, which represents isothermal compression. Hence the effect of number of stages of final compression is highly nullified in the proposed cycle and it may not be a necessity to attain isothermal compression. Therefore this cycle can be considered as a viable alternative to combined cycle power plant.

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