Second Law Of Thermodynamics Analysis Of Triple Cycle Power Plant

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Abstract

Triple cycle power plant with methane as a fuel has been analyzed on the basis of second law of thermodynamics. In this model, ideal Brayton cycle is selected as a topping cycle as it gives higher efficiency at lower pressure ratio compared intercooler and reheat cycle. In trilple cycle the bottoming cycles are steam Rankine and organic Rankine cycle. Ammonia has suitable working properties like critical temperature, boiling temperature, etc. Steam cycle consists of a deaerator and reheater. The bottoming ammonia cycle is a ideal Rankine cycle. Single pressure heat recovery steam and ammonia generators are selected for simplification of the analysis. The effects of pressure ratio and maximum temperature which are taken as important parameters regarding the triple cycle are discussed on performance and exergetic losses. On the other hand, the efficiency of the triple cycle can be raised, especially in the application of recovering low enthalpy content waste heat. Therefore, by properly combining with a steam Rankine cycle, the ammonia Rankine cycle is expected to efficiently utilize residual yet available energy to an optimal extent. The arrangement of multiple cycles is compared with combined cycle having the same sink conditions. The parallel type of arrangement of bottoming cycle is selected due to increased performance.

***Key words** : Multiple cycles, triple cycle, exergetic loss, organic Rankine cycle*

Abstraksi

Pembangkit daya siklus gabung tiga dengan metana sebagai bahan bakar dianalisa berdasarkan hukum kedua termodinamika. Dalam model ini, siklus Brayton ideal dipilih sebagai siklus puncak yang memberikan efisiensi yang tertinggi pada perbandingan tekanan yang rendah dibandingkan siklus yang menggunakan pendingin antara dan pemanas ulang. Dalam siklus gabung tiga, siklus bagian bawah adalah siklus Rankine dan siklus Rankine organik. Amoniak mempunyai sifat-sifat kerja yang sesuai seperti temperatur kritis, temperatur didih, dll. Siklus uap terdiri dari deaerator dan pemanas ulang. Siklus amoniak bagian bawah adalah siklus Rankine ideal. Tekanan uap pemanas ulang dan pembangkit amoniak dipilih untuk penyederhanaan analisa. Pengaruh perbandingan tekanan dan temperatur maksimum yang mana diambil sebagai parameter penting berkenaan dengan unjuk kerja dan kerugian dayaguna dari siklus gabung tiga yang dibahas. Disisi lain, efisiensi siklus gabung tiga dapat dinaikkan terutama dalam penerapan pada recovering entalpi rendah kandungan pembuangan kalor. Oleh karena itu, semestinya penggabungan dengan sebuah siklus uap Rankine, siklus Rankine amoniak adalah diharapkan untuk efisiensi memanfaatkan gas sisa yang masih memiliki energi untuk sebuah tingkat optimal. Susunan siklus bertingkat dibandingkan dengan siklus gabungan mempunyai kesamaan kondisi. Susunan paralel pada bagian bawah siklus dipilih untuk meningkatkan unjuk kerja.

***Kata kunci** : Siklus bertingkat, siklus gabung tiga, kerugian dayaguna, siklus Rankine organik*

1. INTRODUCTION

The efficiency of the power plant may be improved to around 48% by employing a combined cycle. Still some heat is rejected to the surroundings through the exhaust and condenser. If this heat is utilized the efficiency of the plant can futher be increased. For this a heat recovery ammonia generator is placed after the steam turbine. The bottoming cycle is ammonia Rankine cycle. The heat from the out let of the steam turbine (in series type) or out let of exhaust (in parallel type) is utilized to raise the temperature of ammonia in evaporator. In the series type arrangement of triple cycle, the rejection in steam cycle goes to heat input to the ammonia cycle. In the parallel type arrangement of triple cycle both steam cycle and steam cycle recovers the heat from exhaust of gas turbine. These ammonia vapors are expanded in a low pressure turbine. Small amount of work is obtained from this turbine but the overall efficiency increases around 2% in addition to the combined cycle efficiency.

Nag [2001] presented the detailed analysis of steam cycle and combined cycle power generation. Turns [2002] presented the combustion mechanism, reactions, gas table and equations as function of temperature. These equations are curve fittings for various gases are used to determine the properties of the gases. Hung [2002] proposed conceptual arrangement to found a maximum output. El-Wakil [1984] presented the detailed discussion regarding the combined cycle its advantages and applications. A detailed thermodynamic analysis of the combined power cycle without supplementary firing was given in Horlock [1992]. The chemical physical exergetic components are determined by the method given by Kotas [1984].

The present work is aimed to study the following objectives : to select the best optimized multiple cycle arrangement, to estimate the overall efficiency of the triple cycle based on the second law of thermodynamics, and to study the effect of pressure ratio, maximum temperature of gas cycle in the performance and exergetic losses.

2. THERMODYNAMIC ANALYSIS OF THE TRIPLE CYCLE

In this study it is found that the series-type triple cycle exhibits no significant difference as compared with the combined cycle. On the other hand, the efficiency of the parallel type triple cycle can be raised, especially in the applicationof recovering low-enthalpy content waste heat. Therefore, the parallel type cycle is selected for the analysis.

2.1 Assumptions Used in The Analysis

The following assumption have been made during the analysis of the combined cycle :

1. Atmospheric condition is taken as 25 ^{o}C and 1 bar.
2. Gas cycle pressure ratio and maximum temperature is fixed at 15 and 1200 ^{o}C.
3. The inlet condition for high pressure steam turbine is taken as 120 bar and 550 ^{o}C.
4. The condenser pressure is assumed to be 0,085 bar.
5. The temperature difference between condensed steam, ammonia, and outlet cooling water is taken as 8.
6. Steam reheat pressure is taken as 25% of HRSG pressure.
7. Isentropic efficiency of gas turbine is taken as 90%.
8. Isentropic efficiency of compressor steam and ammonia turbine are taken as 85%.
9. Pressure drop in HRSG, HRAG, and condenser is neglected.
10. Heat loss in HRSG, HRAG, turbines, condenser, and feed water heater (deaerator) are neglected.
11. Pinch points in both steam and ammonia cycle is taken as 10.

2.2 Overview of The Termodynamic Model

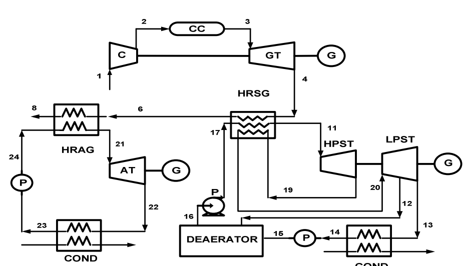

Figure 1. Schematic diagram of components for parallel type cycle

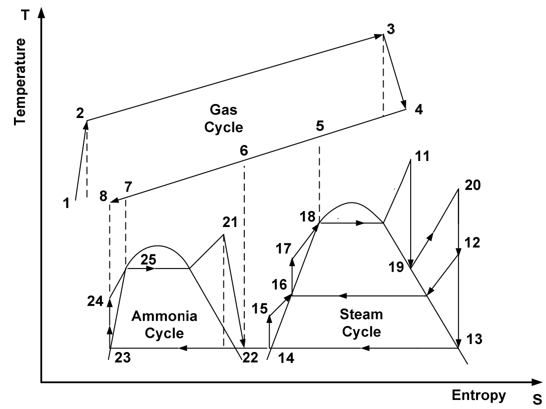

Figure 2. T – S diagram of thermodynamic model of parallel type triple cycle

The schematic diagram of triple cycle with parallel arrangement of multiple cycles is shown in figure 1 and the corresponding temperature – entropy in the figure 2. This work mainly deals with the detailed study and analysis of triple cycle based on the second law of thermodynamics. The expanded gases are made to passthrough a single pressure heat recovery steam generator and heat recovery ammonia generator in case of parallel type. The remaining is utilized to raise the temperature of the ammonia in the ammonia Rankine cycle. Ammonia is having a very low boiling point and easily vaporized by the low temperature hot fluid. The ammonia vapors after expanding from the turbine are at a low temperature.

2.3 Thermodynamic Analysis

For chemical compounds, molar chemical exergy can be calculated from the values of Gibbs functon of formation and the values of the chemical exergy of the constituent chemical elements taken from Kotas [1984]. Considering the reversible reaction of formation of the compound, in which exergy is conserved, the standard molar chemical exergy of the compound can be written as :

$$\overline{\varepsilon^{0}} = \overline{\Delta g_{f}^{0}} + \sum E_{el}^{\ 0} \qquad (1)$$

where $\overline{\Delta g_{f}^{0}}$ is the standard molar Gibbs function of formation and $E_{el}^{\ 0}$ is the standard chemical exergy of constituent elements per mole of the chemical compounds.

$$\left(\overline{\Delta g_{f}^{0}}\right)_{CH_4(g)} = -\ 50810 \text{ kJ/kgmol}$$

The reaction of formation is,

$$C + 2H_2 \rightarrow CH_4 \qquad (2)$$

The equation $\overline{\varepsilon^0} = \overline{\Delta g_f^0} + \sum E_{el}^0$ becomes

$$\overline{\varepsilon^0}_{CH_4(g)} = \left(\overline{\Delta g_f^0}\right)_{CH_4(g)} + \overline{\varepsilon^0}_{C(s)} + 2\,\overline{\varepsilon^0}_{H_2(g)} \quad (3)$$

$$\overline{\varepsilon^0}_{CH_4(g)} = -50810 + 410530 + 2(238350)$$
$$= 836420 \text{ kJ/kg mol}$$

Therefore, $E_{CH_4} = 836420$ kJ/kg mol.

Without using the value of gamma, with balancing the entropies in isentropic processes the outlet temperatures can be calculated. For an isentropic compression process in the compressor, the expression is given as,

$$\Delta S = 0 = S_{O_2(T_2')} - S_{O_2(T_0)} + 3.76\left(S_{N_2(T_2')} - S_{N_2(T_0)}\right) - R_u\left[\log(P_2/P_1) + 3.76\log(P_2/P_1)\right] \quad (4)$$

The temperature of the air at state 2' can be estimated from the above equation. By considering compressor isentropic efficiency, the actual temperature of the compressed air T_2 can be calculated. The combustion equation in gas turbine combustion chamber is,

$$CH_4 + x\left(O_2 + 3.76\,N_2\right) \rightarrow CO_2 + 2H_2O + (x-2)O_2 + 3.76\,x\,N_2 \quad (5)$$

Where "x" is the amount air to be supplied per kg mole of methane for gas turbine combustion chamber which can be calculated by energy balance for the temperature of T_2 and T_3. For an isentropic expansion process in the gas turbine, the expression is given as,

$$\Delta S = 0 = S_3 - S_{4'} \quad (6)$$

Total molecules of products combustion chamber, $m_{CC} = 1 + 2 + (x-2) + 3.76x$

2.4 Exergy Analysis

To determine the exergetic losses, the irreversibilities associated in all the components to be estimated for second law analysis. The chemical exergy and physical exergy can be determined for each state by using the following equations, Chemical exergy,

$$E_{ch} = \sum_k n_k \overline{\varepsilon^0}_k + \overline{R}\,T_0 \sum_k n_k \ln\left(P\,x_k\right) \quad (7)$$

Physical exergy,

$$E_{ph} = H - \sum_k T_0\, S_k \quad (8)$$

Exergy,

$$E = E_{ch} + E_{ph} \quad (9)$$

Exergetic loss (irreversibility) in compressor,

$$I_c = E_1 + W_c - E_2 \quad (10)$$

Exergetic loss in gas turbine combustion chamber,

$$I_{gtcc} = E_{CH_4} + E_2 - E_3 \quad (11)$$

Exergetic loss in gas turbine,

$$I_{gt} = E_3 - E_4 - W_{gt} \quad (12)$$

Exergetic loss in heat recovery steam generator,

$$I_{hrsg} = m_{CH_4}\left(E_4 - E_6\right) - m_s\left[\begin{matrix} h_{11} - h_{17} + h_{20} - h_{19} - \\ T_0\left(S_{11} - S_{17} + S_{20} - S_{19}\right)\end{matrix}\right] \quad (13)$$

Exergetic loss in exhaust from hrsg,

$$I_{ex} = m_{CH_4} E_8 \quad (14)$$

Exergetic loss in steam turbines,

$$I_{st} = T_0\, m_s\left[\begin{matrix} S_{19} - S_{11} + S_{12} - S_{20} + \\ (1-m_1)\left(S_{13} - S_{12}\right)\end{matrix}\right] \quad (15)$$

Mass of cooling water required for condenser for steam cycle,

$$m_w = \frac{\left[m_s(1-m_1)(h_{13} - h_{14})\right]}{\left[c_{pw}\left(T_{out} - T_{in}\right)\right]} \quad (16)$$

Exergetic loss in condenser,

$$I_{co} = T_0\left[\begin{matrix} m_w\, c_{pw}\left(\dfrac{T_{out}}{T_{in}}\right) + \\ m_s\,(1-m_1)\left(S_{14} - S_{13}\right)\end{matrix}\right] \quad (17)$$

Exergetic loss of hot water from condenser,

$$I_w = m_w\, c_{pw}\left(T_{out} - T_0\right) - T_0 \log\left(\frac{T_{out}}{T_0}\right) \quad (18)$$

Total (internal + external) exergetic loss in condenser,

$$I_{cot} = I_{co} + I_w \quad (19)$$

Similarly exergetic loss in condenser for ammonia cycle can be determined. Exergetic loss in deaerator,

$$I_{deae} = T_0\, m_s\left[m_1\left(S_{16} - S_{12}\right) + \left(S_{16} - S_{15}\right)\right] \quad (20)$$

Exergetic loss in heat recovery ammonia generator,

$$I_{hrag} = m_{CH4}\left(E_6 - E_8\right) - m_a\begin{bmatrix} h_{21} - h_{24} - \\ T_0\left(S_{21} - S_{24}\right)\end{bmatrix} \quad (21)$$

Exergetic loss in ammonia turbines,

$$I_{at} = T_0\, m_a\left(S_{22} - S_{21}\right) \quad (22)$$

For 1 kg mole of fuel the gas cycle work inputs and outputs are : Work input to compressor,

$$W_c = h_2 - h_1 \quad (23)$$

Work output by high pressure gas turbine,

$$W_{gt} = h_3 - h_4 \quad (24)$$

Net work output by gas cycle,

$$W_{netgc} = W_{gt} - W_c \quad (25)$$

Steam turbine work,

$$W_{st} = m_s\begin{bmatrix}\left(h_{11} - h_{19} + h_{20} - h_{12}\right) + \\ \left(1 - m_1\right)\left(h_{12} - h_{13}\right)\end{bmatrix} \quad (26)$$

Pump work,

$$W_{ps} = m_s\begin{bmatrix}\left(1 - m_1\right)\left(h_{13} - h_{14}\right) + \\ \left(h_{17} - h_{16}\right)\end{bmatrix} \quad (27)$$

Net work from steam cycle,

$$W_{netsc} = W_{st} - W_{ps} \quad (28)$$

Similarly in ammonia cycle
Ammonia turbine work,

$$W_{at} = m_a\left(h_{21} - h_{22}\right) \quad (29)$$

Pump work,

$$W_{pa} = m_a\left(h_{24} - h_{23}\right) \quad (30)$$

Net work from ammonia cycle,

$$W_{netac} = W_{at} - W_{pa} \quad (31)$$

Net work from triple cycle,

$$W_{net} = W_{netgc} + W_{netsc} + W_{netac} \quad (32)$$

Second law efficiency of gas cycle,

$$\eta_{2,gc} = \frac{W_{netgc}}{m_{CH4}\ \varepsilon^0_{CH4}} \times 100\% \quad (33)$$

Second law efficiency of steam cycle,

$$\eta_{2,sc} = \frac{W_{netsc}}{m_{CH4}\ \varepsilon^0_{CH4}} \times 100\% \quad (34)$$

Second law efficiency of ammonia cycle,

$$\eta_{2,ac} = \frac{W_{netac}}{m_{CH4}\ \varepsilon^0_{CH4}} \times 100\% \quad (35)$$

Second law efficiency of triple cycle,

$$\eta_{2,tc} = \frac{W_{net}}{m_{CH4}\ \varepsilon^0_{CH4}} \times 100\% \quad (36)$$

3. RESULTS AND DISCUSSION

In this triple cycle organic Rankine cycle is added to the combined cycle (Brayton – steam Rankine cycle). The ammonia is selected as working fluid for the organic Rankine cycle. There are two possibilities of arrangement of ammonia cycle to the combined cycle which are series or parallel combinations. Both the arrangement are compared with the combined cycle (without organic Rankine cycle) performance. Gas cycle parameters, gas turbine inlet temperature and maximum temperature have more domination then the other steam and ammonia cycle parameter. Therefore, the effect of pressure ratio of gas cycle and maximum cycle temperature are studied on second law efficiency and exergetic destruction in gas, steam, and ammonia cycles. In all the three models of the arrangements the sink temperature is kept constant. The exergetic loss in gas cycle includes compressor, combustion chamber, gas turbine, and exhaust. Steam cycle exergetic loss includes heat recovery steam generator, steam turbines, condensor, pump, and deaerator. Ammonia cycle exergetic loss includes heat recovery ammonia generator, ammonia turbine, pump, and condenser.

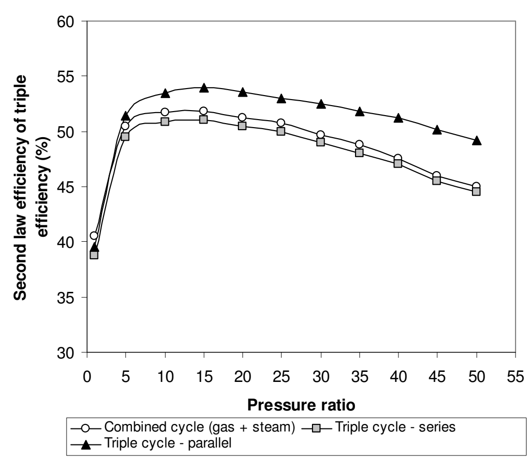

Figure 3. Comparison of triple cycle with series and parallel arrangement of bottoming cycle with combined cycle and with respect to pressure ratio of gas cycle.

Figure 3 compares the second law efficiency of combined cycle with triple cycle with series arrangement and parallel arrangement with respect to pressure ratio of gas cycle. The series arrangement is not improving the efficiency with the pressure ratio for the same heat rejection temperature. This is because the temperature limits of source and sink are not altered and addition of components in series increases the losses. The addition of ammonia cycle to the combined cycle in parallel improves the performance. At the pressure ratio of 15, the second law efficiency of combined cycle, series type triple cycle and parallel type triple cycle are 52%, 51.3% and 54% respectively. The figure shows that the parallel type the arrangement improves the efficiency to 2%. In the arrangement the waste heat recovery is improved. Exhaust losses are decreased. Therefore the efficiency increases with the arrangement.

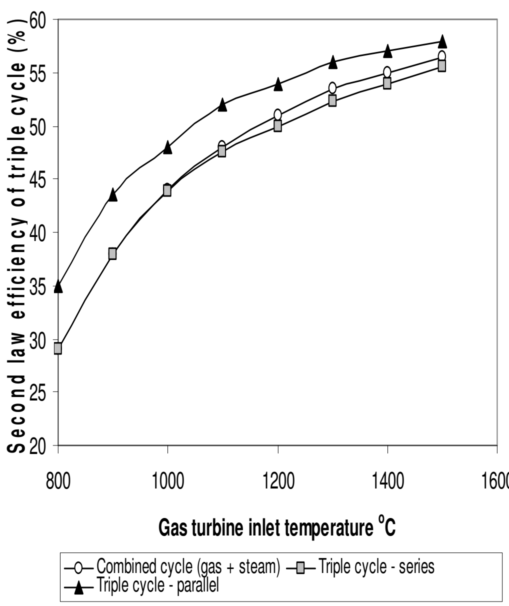

Figure 4. Comparison of triple cycle with series and parallel arrangement of bottoming cycle with combined cycle and with respect to gas turbine inlet temperature of gas cycle.

Figure 4 compares the second law efficiency of combined cycle with triple cycle with incorporation of ammonia cycle in series arrangement and in parallel arrangement with respect to gas turbine inlet temperature i.e. maximum temperature of the cycle. Here also the series arrangement not improves the efficiency with the temperature for the same heat sink temperature. The efficiency decreases with series addition of organic cycle to the combined cycle due to increased losses. The additon of ammonia cycle to the combined cycle in parallel improves the performance. At 1200 ^{o}C the second law efficiency of parallel arrangement is 54% which is high compared of others. As mentioned before in this arrangement the waste heat recovery is improved. Parallel type arrangement increases the efficiency compared to others ones on the basis of pressure ratio and maximum temperature. Therefore in the subsequent simulation work parallel arrangement is fixed.

The influence of gas pressure ratio on second law efficiency of various cycle in shown figure 5. Up to pressure ratio of 35, the gas cycle efficiency increases and then decreases with increases in pressure ratio. The steam cycle efficiency decreases where as the ammonia cycle efficiency increases with increase in the pressure ratio.

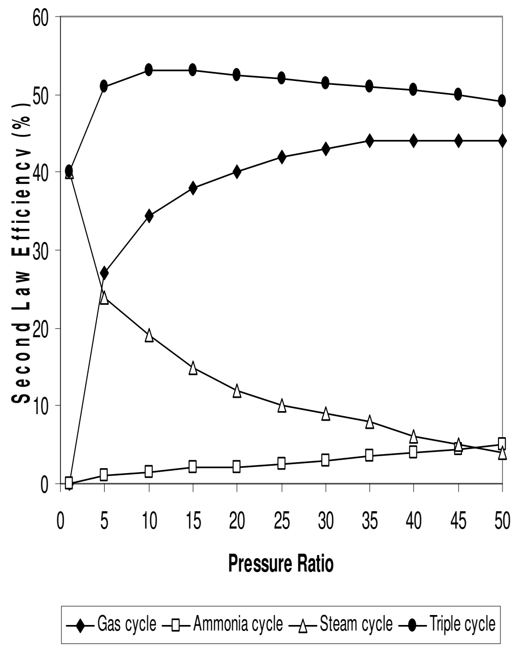

Figure 5. Effect of pressure ratio on second law efficiency of gas, steam, ammonia and triple cycle with parallel type.

For fixed maximum temperature, the increase in pressure ratio leads to decrease the gas turbine exhaust temperature. This reduces the steam cycle efficiency. To compensate the reduction in steam efficiency, the ammonia cycle efficiency increases as the flow rate of ammonia increases where as steam flow rate decreases with increase in pressure ratio. The triple cycle efficiency increases with the pressure ratio up to 15 and then decreases with increase in pressure ratio. Therefore the optimum pressure ratio for triple cycle is 15. At 15 pressure ratio, the second law efficiency of gas, steam, ammonia and triple cycles are 38%, 14%, 2% and 54% respectively.

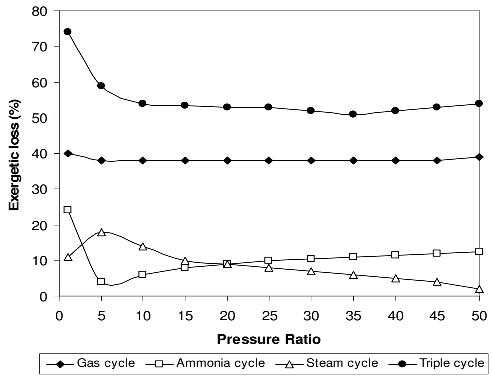

Figure 6. Effect of pressure ratio of gas cycle on exergetic destruction of various cycles.

The exergetic loss in individual fluid cycle and overall triple cycle is shown in figure 6. The major exergetic destruction occurs in gas cycle due to combustion chamber. At low pressures steam cycle exergetic loss is more than the ammonia cycle where as from pressure ratio of 20, ammonia cycle loss dominating the steam cycle. On overall basis, for triple cycle the exergetic loss decreases and then increases with increase in the pressure ratio.

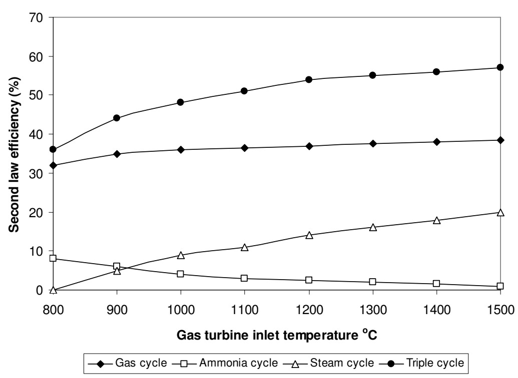

Figure 7. Effect triple cycle maximum temperature on second law efficiency of gas, steam, ammonia and triple cycle with parallel type

The effect of maximum temperature of triple cycle on second law efficiency of individual cycles and triple cycle is shown in figure 7. For fixed pressure ratio, the increased maximum temperature also increases the gas turbine exhaust temperature. Therefore gas cycle and also steam cycle efficiency increases with increase in temperature. Due increased steam flow rate and decreased ammonia flow rate, the ammonia cycle decreases with the increase in the maximum temperature.

Figure 8 shows the exergetic destruction in gas, steam, ammonia and overall triple cycle with parallel type arrangement. Only the steam cycle exergetic loss increases with increase in the maximum temperature due to increase in the steam mass flow rate. The gas and ammonia cycle exergetic loss decreases with increase in the maximum temperature. On overall basis the triple cycle exergetic loss decreases with increase in the temperature. Therefore the efficiency of triple cycle increases with rise in temperature.

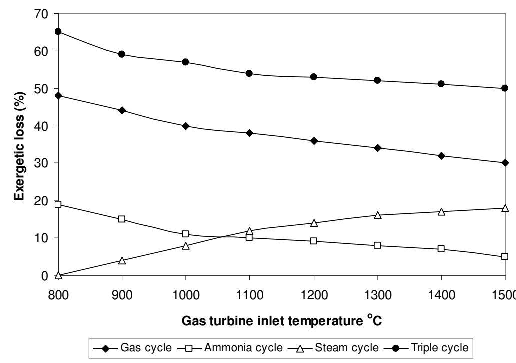

Figure 8. Effect of maximum temperature of triple cycle on exergetic loss of various cycles.

4. Conclusions

In this work, first to arrangements of series and parallel arrangements are compared with the combined cycle to select the best triple cycle arrangement. The comparison shows that the series type does not improve the efficiency. Therefore, parallel type arrangement of ammonia cycle with the steam cycle in triple cycle is selected. The variations in performance of gas, steam, ammonia and triple cycle are discussed with respect to gas cycle parameters, which are taken as pressure ratio and maximum temperature. The second law efficiency and exergetic losses are plotted against to gas pressure ratio and temperature. The optimum pressure ratio of gas cycle is found as 15 to get maximum triple cycle efficiency. The effect of gas temperature on performance has more effect than the pressure. Due to incorporation of ammonia cycle the efficiency is increased by 2%.

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