



Comparative Analysis of Alternative and Conventional Regenerative Gas Turbine Cycle Using Entropy Generation Approach

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Abstract: A gas turbine cycle equipped with an Alternative Regeneration yields higher thermal efficiency than a conventional regenerative gas turbine cycle operating under the same conditions. The concept of alternative regeneration is explained first. Entropy generation approach which is a new method for evaluation of performance of any system which is suggested in the last few years is taken as the method of proving the new system of regeneration in gas turbine cycle better over the conventional one. For this entropy of each components is calculated for both type of regeneration and then total entropy is calculated for the systems. Finally, characteristics curves are plotted using MATLAB showing the results.

I. INTRODUCTION

An Alternative Regenerative gas turbine cycle always yields higher cycle efficiency over wide range of operating conditions over a conventional one. The idea behind this scheme is to preheat the compressed air with high temperature combustion gases before it is fully expanded across the turbine. So, the efficiency is higher because the compressed air enters the combustion chamber at a higher temperature i.e. heat addition in the combustor takes place at a higher average temperature. The heat exchanger for material selection is little difficult as it has to work under elevated temperature conditions over the conventional scheme. The highest value of efficiency of alternative cycle occurs at an optimum pressure ratio which is lower than that of a conventional cycle, which is a great advantage.

In the conventional regenerative cycle, gases leaving the final turbine are used to preheat the compressed air. In this scheme, the heat exchanger is located between the compressor and high pressure turbine (HPT). The gases leaving the power turbine (PT) are used as a heat source. But, in the alternative one, the exhaust gases after HPT is used as a heat source and passing through the heat exchanger and then it is expanded across the PT. So, the thermodynamic effect is to enhance the quantum of heat to the compressed air compared to the conventional one.

The analysis discussed above are done and explained with a new approach of Dimensionless Entropy Generation. In this, entropy of each of the components of the system is calculated and added together for getting the total entropy generated in the system. This method is adopted because entropy generation is a direct measure of inefficiency of the equipment. Earlier various expressions have been used to determine the performance of any equipment or system. In the method of entropy generation, instead of using expression for efficiency, performance can be evaluated directly with entropy production within the system. This method is adopted mainly because of two reasons. First, repetition which is present in current modeling procedure can be avoided. Second, this method is a unified approach for modeling equipment and it is independent of the type of equipment.

II. MATHEMATICAL FORMULATION FOR ENTROPY PRODUCTION

A. FOR CONVENTIONAL SCHEME:

For Compressor:

$$S = m (S_2 - S_1),$$

Where $S_2 = mC_{pa} \ln (T'_2/T_1)$ and

$$S_1 = mR \ln (P_2/P_1)$$

$$\text{Hence, } S = mC_{pa} \ln (T'_2/T_1) - mR \ln (P_2/P_1)$$

$$\text{Or } (S/mR) = (C_{pa}/R) \ln (T'_2/T_1) - \ln (P_2/P_1)$$

$$\text{Or } (T'_2/T_1) = [(P_2/P_1)^{R/C_{pa}} + 1] / \eta_c$$

$$(S/mR) = (C_{pa}/R) \ln \{ [(P_2/P_1)^{R/C_{pa}} + 1] / \eta_c \} - \ln (P_2/P_1)$$

$$\text{Or } \sigma_c = (C_{pa}/R) \ln \{ [r^{R/C_{pa}} + 1] / \eta_c \} - \ln (r) \quad (1)$$

The above expression indicates the dimensionless entropy generation for compressor for conventional scheme.

For High Pressure Turbine:

$$\begin{aligned}
 S &= m (S_3 - S_4) \\
 \text{Or } (S/mR) &= C_{pg} \ln (T_3/T'_4) - \ln (P_3/P_4) \\
 \sigma_c &= (T_3 - T_4) / (T_3 - T'_4) \\
 \text{and } (T'_4/T_3) &= 1 - \eta_{t1} (1 - T_4/T_3) \\
 \text{Hence, } (S/mR) &= (C_{pg}/R) \ln [1 - \eta_{t1} (1 - T_4/T_3)] - \ln (P_3/P_4) \\
 \text{Or } \sigma_{t1} &= (C_{pg}/R) \ln [\eta_{t1} \{1 - (P_4/P_3)^{C_{pg}/R}\} - 1] - \ln (P_3/P_4)
 \end{aligned} \tag{2}$$

The above expression shows the dimensionless entropy generation for High Pressure Turbine. Similarly, expression can be written for all the remaining components, which are:

For Power Turbine:

$$\sigma_{t2} = (C_{pg}/R) \ln [\eta_{t2} \{1 - (P_5/P_4)^{C_{pg}/R}\} - 1] - \ln (P_4/P_5) \tag{3}$$

For Regenerator:

$$\begin{aligned}
 \sigma_r &= (C_{pg}/R) \ln [1 + (1/\varepsilon_c)(T_6/T'_2 - 1)] \\
 &\quad - (C_{pg}/R) \ln (T_7/T_6)
 \end{aligned} \tag{4}$$

where, ε_c is the effectiveness of regenerator and is given by

$$\varepsilon_c = (T_6 - T'_2) / (T'_5 - T'_2)$$

For Combustor:

$$\sigma_{cc} = (C_{pg}/R) \ln (T_3/T_6) - \ln (P_3/P_6) \tag{5}$$

Hence, we can write that the total dimensionless entropy generated in the conventional scheme is

$$\sigma_{conv} = \sigma_c + \sigma_{t1} + \sigma_{t2} + \sigma_r + \sigma_{cc} \tag{6}$$

B. FOR ALTERNATIVE SCHEME:

Similar to the previous pattern, expressions can be derived for dimensionless entropy generated in this case also for individual components, which are:

$$\sigma_c = (C_{pa}/R) \ln \{[r^{R/C_{pa}} + 1] / \eta_c\} - \ln (r) \tag{7}$$

$$\sigma_{t1} = (C_{pg}/R) \ln [\eta_{t1} \{1 - (P_4/P_3)^{C_{pg}/R}\} - 1] - \ln (P_3/P_4) \tag{8}$$

$$\sigma_{t2} = (C_{pg}/R) \ln [\eta_{t2} (1 - T_6/T_5)] - \ln (P_5/P_6) \tag{9}$$

$$\sigma_r = (C_{pg}/R) \ln (T_5/T'_2) - (C_{pg}/R) \ln (T'_4/T_7) \tag{10}$$

$$\sigma_{cc} = (C_{pg}/R) \ln (T_3/T_7) - \ln (P_3/P_7) \tag{11}$$

Hence, we can write that the total dimensionless entropy generated in the alternative scheme is

$$\sigma_{alt} = \sigma_c + \sigma_{t1} + \sigma_{t2} + \sigma_r + \sigma_{cc} \tag{12}$$

III. BLOCK DIAGRAM AND T-S REPRESENTATION

A. CONVENTIONAL SCHEME:

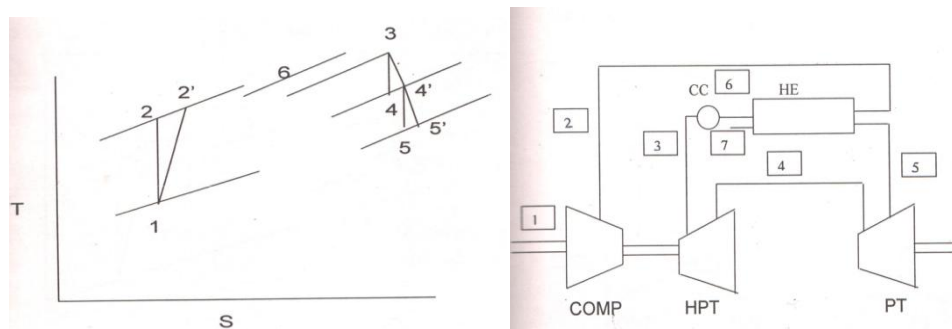


Fig. 1 Conventional Regenerative Gas Turbine Cycle

B. ALTERNATIVE SCHEME:

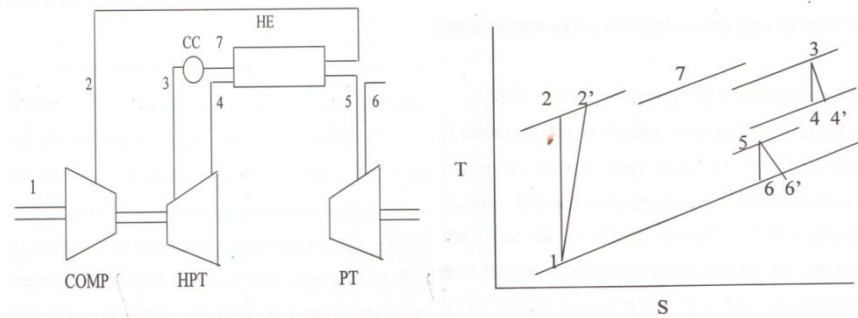


Fig. 2 Alternative Regenerative Gas Turbine Cycle

For various operating conditions, heat exchangers can improve the performance of the cycle. It is clear from Fig.1 that the heat exchanger uses the exhaust gases coming out from the final turbine so that high amount of work is extracted from the high enthalpy gases. But, if the heat exchanger is placed between the two turbines as shown in Fig.2, then the cycle thermal efficiency can further be improved, which is referred as Alternative Regenerative gas turbine cycle. The essence of the scheme is that the quantum of heat that delivered to the compressed air is much more than that of what conventional regenerative cycle is delivering, which in turn results in higher average temperature in the process of heat addition in the combustor.

IV. RESULTS AND DISCUSSIONS

Different expressions for individual components are expressed and total dimensionless entropy is formulated for both the arrangements i.e. conventional and alternative type. These complex expressions are then solved for getting the final solutions using computer software MATLAB. Also, various characteristics curves are obtained using MATLAB, which are drawn ahead.

A comparative performance curves drawn between total dimensionless entropy generation and other important parameters like pressure ratio, turbine inlet temperature and ambient temperature for both conventional and alternative regenerative cycles are illustrated. Fig. 3(a) & 3(b) clearly indicates that the entropy production for both the schemes first reduces and then after reaching to minimum value, they again climbs up, just opposite to the efficiency curve. Also, the least value of entropy i.e. maximum efficiency is attained by alternative cycle with minimum value of pressure ratio.

A. Variation of total entropy against Pressure Ratio

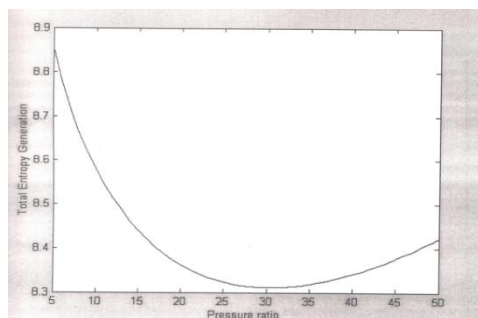


Fig. 3(a) For Conventional Cycle

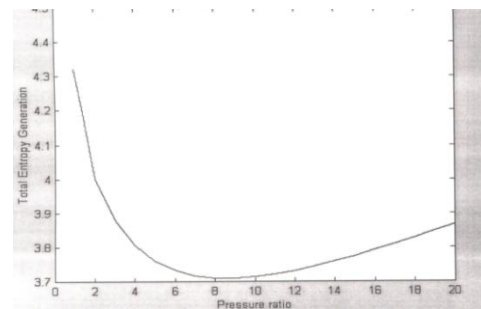


Fig. 3(b) For Alternative Cycle

4.1 Variation of total entropy against Turbine Inlet Temperature

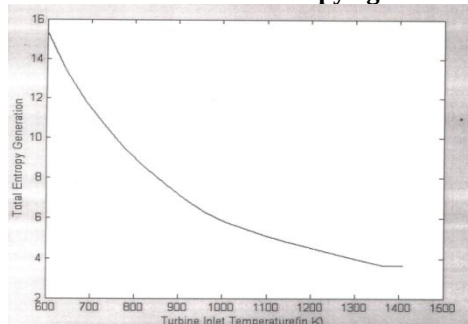


Fig. 4(a) For Conventional Cycle

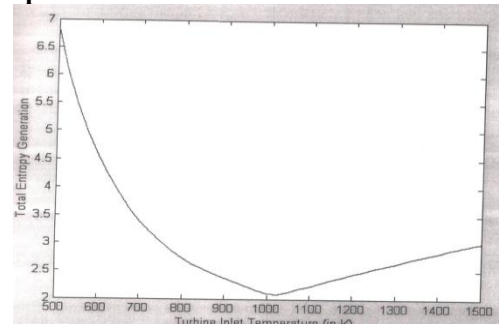


Fig.4(b) For Alternative Cycle

Similarly, Fig .4 & 5 reveals the fact that least value of entropy generation is attained for the case of alternative cycle for minimum value of turbine inlet temperature and ambient temperature. In other words, higher efficiency is obtained in the case of alternative regenerative cycle for lower value of turbine inlet temperature and ambient temperature.

B. Variation of total entropy against Ambient Temperature

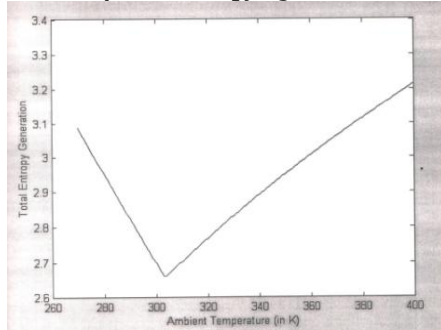


Fig. 5(a) For Conventional Cycle

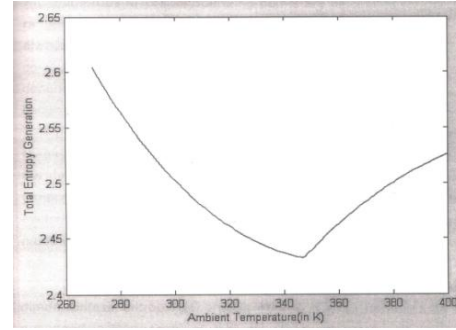


Fig. 5(b) For Alternative Cycle

V. CONCLUSION

An alternative regenerative cycle with various operating parameters are explained for practical range of values and reveals the fact that an alternative cycle always possesses higher cycle efficiency or in other words dimensionless entropy generation for an alternative cycle is always smaller compared to the conventional one. But, at the same time, requirements of heat exchangers are more challenging in the case of alternative cycle. The alternative regenerative cycle has the capacity to enhance the efficiency of the cycle by some percentage, but on the cost of decrease in power output. To compensate this drawback, bigger engine components may be used making higher initial cost of the plant. But, for engines which has to run continuously, efficiency of the cycle could be taken as more important parameter while designing the plant.

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