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A Comparative Investigation of Reheat In Gas Turbine Cycles

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ABSTRACT

The role of gas turbine power plants in electrical energy production has been considerably increased in the last 2-3 decades. Various methods have been proposed to improve the performance of gas turbine cycles. In this research two methods: reheat cycle (RC) and cycle with reheat and heat exchanger (RHC) were investigated.

Today to achieve a higher efficiency and capacity in gas turbines, higher turbine inlet temperatures like 1300 C and even more are used. Basically application of such temperatures without turbine blade cooling is impossible. Therefore analysis of gas turbine cycles without considering blade cooling in modeling will not certainly lead to a valid and correct results. The main object of this paper is to study the performance of RC and RHC under actual conditions. In this regard all processes treated as actual and particularly a relatively simple approach was used to predict the amount of cooling air.

The obtained results show that reheating in the context of a realistic study has no positive effect on gas turbine performance and of course this conclusion is not in agreement with some relevant published documents.

INTRODUCTION

Gas turbine cycle is very flexible so that its performance i.e. efficiency and specific net work can be improved by adding extra components to a simple cycle (SC).

In recent years many researches carried out regarding advanced gas turbine cycles like steam injected gas turbine cycle [1], humid air turbine [2], heat exchanger cycle [3], etc. Generally the main aim of theses researches are to achieve a higher efficiencies in gas turbine based cycles.

A reheat combustion chamber is a component that can be added to a gas turbine cycle to improve its performance [4]. In this method the expansion process in turbine is divided into two processes and an additional combustion chamber is placed between high pressure and low-pressure turbines. The exhaust gas from high-pressure turbine that contains sufficient oxygen enters a reheat combustion chamber and the temperature of gases can be increased as the result of supplementary combustion.

Previous researches on ideal RC show that reheating increases specific net work but decreases efficiency comparing to SC [4, 5]. Also, these investigations show that the maximum specific net work in RC is obtained if the pressure ratios for high and low - pressure turbines are assumed equal. Crane [5] has shown that although equal pressure ratios lead to a maximum specific net work but under this condition the efficiency is not maximum and the lower pressure ratio for high-pressure turbine increases the efficiency but this leads to the reduction of work output.

Many investigations concerning reheating in gas turbine were carried out assuming some simplifications including: constant specific heat, heat addition to working fluid instead of combustion process, and more importantly considering uncooled turbine model [5] instead of cooled turbine model. These assumptions in particular the last one leads to unrealistic results because almost all turbines today are usually cooled by air and neglecting the amount of cooling air fraction gives incorrect results. In this work it is attempted to model SC, RC, and RHC considering all actual factors that affect the performance of real cycles. In this regard, among the others, specific heat assumed to be function of both temperature and combustion product composition [6] and cooling air fraction were calculated using a relatively simple approach [7].

In this paper in addition to actual RC the RHC were also analysed and compared to one another as well as to SC. In the next sections it is shown that the inclusion of the cooled turbine model in modeling of gas turbine based cycles gives results that are far different from uncooled turbine model. Therefore by this approach the performance of reheat cycles can be predicted correctly.

THERMODYNAMICS ANALYSIS

General Description

Figure 1 illustrates the RC arrangement. The hot gas after leaving the first combustion chamber enters HP turbine where expands partially. Then gases which contains sufficient unused oxygen enters reheat

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combustion chamber where fuel is injected and supplementary firing occurs. The hot gases now having higher temperature (equal to first turbine inlet temperature) enters second or LP turbine where expands to atmospheric pressure.

Figure 2 shows RHC in which a heat exchanger is used to increase the air temperature before it enters to first combustion chamber by recovering heat from LP turbine exhaust gases.

Compressor Analysis

Following equations can be written assuming air as an ideal gas and using polytropic efficiency [4] to model compressor actual behaviour.

$$\eta_{\infty c} = \frac{dh_s}{dh} = \frac{\overline{R}T\frac{dp}{p}}{\overline{C}_{pair}..dT} \Rightarrow \qquad (1)$$

$$\int_{T_1}^{T_2} \overline{C}_{pair} \times \frac{dT}{T} = \int_{p_1}^{p_2} \frac{\overline{R}}{\eta_{\infty c}} \times \frac{dp}{p} \qquad (2)$$

In above relations constant-pressure specific heat is function of temperature and relevant functions for various gases are given in [6]. The only unknown parameter in Equation 2 is compressor outlet temperature which can be found using numerical methods. Compressor specific work in terms of KJ/Kg is calculated from following equation:

$$w_{comp} = \frac{\int_{T_1}^{T_2} \overline{C}_{pair}.dT}{M_a} \qquad (3)$$

Combustion Chamber Analysis

Following chemical reaction can be written by assuming a general formula of C_nH_m for a hydrocarbon fuel and a theoretical air coefficient of A:

$$C_nH_m + A(n+m/4)[O_2 + 3.76N_2] \rightarrow$$
$$(n)CO_2 + (m/2)H_2O + [(A-1)(n+m/4)]O_2 +$$
$$[3.76A(n+m/4)]N_2 \qquad (4)$$

The first law of thermodynamics for an adiabatic combustion process is:

$$\sum_{i=1}^{n}(n_i h_i)_{reac\tan ts} = \sum_{i=1}^{n}(n_i h_i)_{products} \qquad (5)$$

$$h = h_f^{\circ} + \int_{298}^{T} \overline{C}_p.dT \qquad (6)$$

Theoretical air and then fuel air ratio can be calculated from above equation for any turbine inlet temperature.

High-Pressure Turbine Analysis

Actual behaviour of turbine can be modeled by following equations. In this model, combustion products is assumed as ideal gas and polytropic efficiency is used for turbine [4].

$$\eta_{\infty ht} = \frac{dh}{dh_s} = \frac{\overline{C}_{pg1}.dT}{\overline{R}T(\frac{dp}{p})} \Rightarrow \qquad (7)$$

$$\int_{p_3}^{p_4} \eta_{\infty ht}\overline{R}(\frac{dp}{p}) = \int_{T_3}^{T_4} \overline{C}_{pg1}(\frac{dT}{T}) \qquad (8)$$

where:

$$\overline{C}_{pg1} = \frac{\sum_{i=1}^{n}(n_i\overline{C}_{pi})_{products}}{\sum_{i=1}^{n}(n_i)_{products}} \qquad (9)$$

Equation 8 can be used to evaluate HP turbine outlet temperature. Following equation gives turbine specific work (KJ/Kg).

$$w_{hp} = \frac{\int_{T_3}^{T_4} \overline{C}_{pg1}.dT}{M_{g1}} \qquad (10)$$

Similar relations can also be written for LP turbine.

Reheat Combustion Chamber Analysis

The equation of combustion reaction for reheat combustion chamber can be written as:

$$
\begin{aligned}
&xC_nH_m + nCo_2 + (m/2)H_2o + \\
&(A-1)(n+m/4)O_2 + 3.67A(n+m/4)N_2 \\
&\Rightarrow (x+n)Co_2 + (2x+m/2)H_2o + \\
&3.76A(n+m/4)N_2 + \\
&\left[(n+m/4)(A-1) - 2x\right]O_2
\end{aligned}
\qquad (11)
$$

In the above equation, x denotes consumed fuel in reheat combustion chamber for one mole of consumed fuel in primary combustion chamber. x can be found by applying the energy balance equation to the reheat combustion chamber.

Heat Exchanger Analysis

In heat exchanger the exhaust gas from LP turbine is used to heat air before it enters to first combustion chamber and therefore fuel consumption is reduced in the combustion chamber.

Heat exchanger effectiveness is defined as Equation 12 Considering that the air is a minimum fluid [8].

$$
\varepsilon = \frac{q}{q_{\max}} = \frac{\int_{T_2}^{T_a} \overline{C}_{pair}.dT}{\int_{T_2}^{T_6} \overline{C}_{pair}.dT} \qquad (12)
$$

Also, the energy balance for this component is:

$$
\int_{T_2}^{T} \overline{C}_{pair}.dT = (1+f_1+f_2)\int_{T_b}^{T_6} \overline{C}_{pg2}.dT \qquad (13)
$$

As T_2 and T_6 can be obtained respectively from compressor and LPT models, therefore, combustion chamber inlet temperature (T_a) and heat exchanger exit temperature (T_b) can be found from above equations.

Cooling Air Evaluation.

Analysing the performance of modern gas-turbine engines without prediction of cooling air flows leads to inaccurate results as cooling air is a large fraction of inlet air flow in these machines. On the other hand, to accurately model cooling airflow is a complex task. Cooling flows are a complex function of turbine operating and design parameters such as compressor efficiencies, bleed pressures, turbine gas temperatures, and allowable metal temperatures [9]. A large number of detailed gas turbine design information is therefore required to accurately model the performance of modern gas turbines.

A simple model based on figure 3 was also developed for estimation of cooling air [7]. In this model compressor and turbine are modelled as single stage compression and expansion devices respectively. Cooling air is assumed to be totally extracted from the compressor outlet, whilst the remaining air flows to the combustion chamber. It is further assumed that half of the cooling air is mixed with combustor outlet gas in the first nozzle section. The temperature after this mix is called rotor inlet temperature. The rest of the cooling air is mixed after the gas expanded in the nozzle.

Figure 4 represents existing relationship between turbine and rotor inlet temperatures for various gas turbines. Based on this figure, a simple correlation can be established between turbine inlet temperature (combustor outlet temperature) and turbine rotor inlet temperature. It was found that this relationship could be expressed with good approximation by equation:

$$
T_r = 0.8451\ T_3 + 136.2 \qquad (14)
$$

In this equation temperature is in terms of $^{\circ}$C.

Energy balance for nozzle section (for 1 Kg of air at compressor inlet) is given by:

$$
(1-caf+f)h_3 + \frac{caf}{2}.h_2 = (1-\frac{caf}{2}+f)h_r \qquad (15)
$$

This equation can also be simplified as :

$$
\frac{h_3 - h_r}{h_3 - h_2} = \frac{caf}{2 - caf + 2f} \qquad (16)
$$

Eventually cooling air for 1Kg of inlet air can be estimated using above equations.

Evaluation of Cycle Performance Parameters

After calculating compressor specific work, turbine specific work, fuel air ratio in both combustion chambers, and air cooling ratio, the specific net work of the cycle can be estimated using Equation 17. Gas turbine specific net work is defined as the work output of the cycle for one Kg of compressor inlet airflow.

$$
w_{net} = w_{lpt} + w_{hpt} - w_{comp} \qquad (17)
$$

Where we have:

$$
\begin{aligned}
w_{hpt} &= (1+f_1-\frac{caf}{2})(h_3-h_4) \\
&+ \frac{caf}{2}(h_2-h_3)
\end{aligned}
\qquad (18)
$$

$$
w_{lpt} = (1+f_1+f_2)(h_5-h_6) + \frac{caf}{2}(h_2-h_5) \qquad (19)
$$

Knowing specific net work and heating value of the fuel ,thermal efficiency can also be calculated using following equation:

$$\eta_{th} = \frac{w_{net}}{LHV(f_1 + f_2)} \qquad (20)$$

RESULTS AND DISCUSSION

Based on models presented above, a computer code constructed for performance prediction and parametric analysis of SC, RC, and RHC. In this investigation following assumptions were made:

1) Atmospheric air conditions are: T_0=15 C, P_0=1.0132 bar, and relative humidity = 60%.

2) As calculations have been performed over a range of pressure ratio, polytropic efficiency, for compressor and turbine analysis, have been used instead isentropic efficiency [4]. A typical value of 0.90 has been assumed for both compressor and turbine polytropic efficiencies.

3) Although a general formula of C_nH_m has been used for the fuel in the model, for calculation purposes, the fuel has been assumed to be methane with lower heating value of 50010 KJ/Kg [6] throughout this work. It is also assumed that the fuel is available at atmospheric temperature and the combustor inlet pressure P_2, so there is no need for gas compression work.

4) No pressure losses were assumed for either compressor inlet or turbine outlet (i.e. p_1=p_4=p_0).

5) All internal pressure losses are assumed to be concentrated in the combustion chamber and equates to 3% (p_3=0.97p_2).

6) Combustion efficiency [4] has been assumed to be 99%.

7) An overall 2% power loss has been assumed. This includes turbine-driven auxiliaries.

8) The effectiveness of heat exchanger is assumed as 0.90.

9) The equal pressure ratios are assumed for high and low pressure turbines in reheat cycles.

10) HP and LP turbine inlet temperatures are assumed the same. In this research all results are obtained assuming inlet temperature of 1200 C.

The results of this analysis are shown as diagrams which are discussed below.

Figures 5 and 6 shows respectively specific net work and thermal efficiency for three cycles versus pressure ratio. These curves have been plotted based on uncooled turbine model. It is observed that reheating increases specific net work of RC comparing to SC. The maximum value of specific net work for the former is about 36% higher than that for the the latter. In contrast, reheating by itself reduces thermal efficiency.

Combined usage of reheating and heat exchanger leads to a considerable increase in thermal efficiency because the combustion chamber inlet air is heated by recovering heat from exhaust gases in heat exchanger. Of course the work output for RHC is slightly lower than that for RC and SC because of pressure drop in heat exchanger. This results are qualitatively in agreement with given results in [5].

Figures 7 and 8 demonstrate the differences between air cooled and uncooled turbine models to predict the performance parameters of SC. Examination of these curves show that uncooled turbine model predicts specific net work and efficiency respectively by 22% and 24% more than cooled turbine model. Another interesting point is that pressure ratios corresponding to maximum specific net work and efficiency are higher for uncooled turbine model rather than cooled turbine model.

These results clearly show that the performance criteria of gas turbine based cycles including reheat cycles can not be predicted accurately by ignoring the amount of cooling air which is required for turbine blade cooling.

Figures 9 compares the variation of specific net work with compressor pressure ratio for SC, RC, and RHC using cooled turbine model. It is seen that RC has higher w_{net} than SC for pressure ratios of higher than 10, but for pressure ratios of lower than 10 the w_{net} of SC is higher. In addition, the important point is that the maximum value of w_{net} are almost equal for both cycles. Therefore it can be said that the main advantage of reheating which is the increament of w_{net} vanishes considerably if we consider turbine air cooling in our analysis. Of course this result is in contrast to the result that obtained from uncooled turbine model [5, 10].

Figure 10 depicts the efficiency variation of SC, RC, and RHC with pressure ratio using cooled turbine model. As expected, the efficiency of RC is lower than that for SC. In this case the difference between this two is higher than that for uncooled turbine model (Figure 6). An important result is that the efficiency of RHC although is higher than RC, but it is slightly lower than that for SC. This result is in contrast to the result obtained from uncooled turbine model which gives the efficiency of RHC much higher than SC (Figure 6).

The above results can be justified knowing that the amount of cooling air for reheat cycles are much higher than that for SC and this destroys the positive effect of reheating on gas turbine performance.

Finally it should be noted that in this paper cooling air is assumed to be totally extracted from the compressor outlet. Actually cooling air for the LPT does not need to have the pressure of the compressor outlet. It can be extracted from the intermediate stages of the compressor where the pressure is slightly higher than LPT inlet pressure. By this way the performance of RC and RHC may be improved due to reduction of both compressor input work and cooling air for LPT. Therefore it is recommended that a further investigation should be carried out considering this point.

CONCLUSION

Following results are obtained from this investigation:

- In the context of uncooled turbine model, reheating increases specific net work but reduces thermal efficiency comparing to simple cycle. Also combined use of reheat and heat exchanger increases thermal efficiency too.
- In the context of cooled turbine model the main advantages of RC which is higher specific net work disappears and its specific net work does not differ considerably from simple cycle work output. Also in this case combination of reheat and heat exchanger does not increase efficiency comparing to simple cycle.
- Pressure ratios correspond to maximum specific net work and maximum efficiency for a given turbine inlet temperature in cooled turbine model are smaller than those for uncooled turbine model for RC, RHC, and SC.
- In both models, pressure ratio corresponds to maximum specific net work for simple cycle is smaller than those for RC and RHC, whereas, pressure ratio corresponds to maximum efficiency for RHC is much smaller than those for simple cycle and RC.
- Finally it should be noted that the cooled turbine model should be improved, and the detailed examination on the influence of cooling air on performance parameters of each cycles should be examined in the future.

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NOMENCLATURE

A = excess air coefficient
Cp = constant-pressure specific heat [KJ/Kg-K]
caf = cooling air fraction
f = fuel air ratio
h = enthalpy [KJ/Kg]
LHV = lower heating value [KJ/Kg]
M = molecular weight
N = denotes spices
P = pressure [bar]
$\overline{R}$ = universal constant of gases [KJ/Kmole-K]
T = temperature [C]
w = specific work [KJ/Kg]
x = fuel ratio for reheat combustion chamber
η_∞ = polytropic efficiency
ε = heat exchanger effectiveness

Subscripts

a = air
c = compressor
f = formatrion
g = gas

Acronyms

HPT = high pressure turbine
LPT = low pressure turbine
RC = reheat cycle
RHC = reheat heat exchanger cycle
SC = simple cycle

Figures and Diagrams:

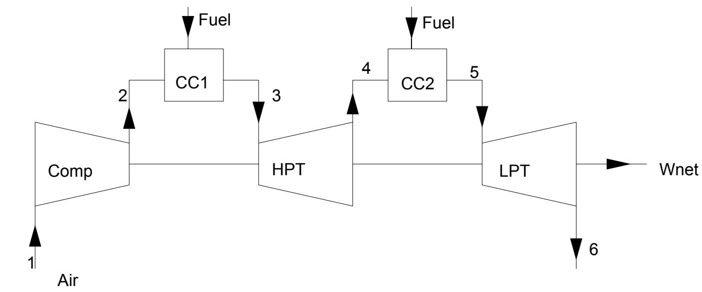

Figure 1: Flow diagram of reheat cycle (RC)

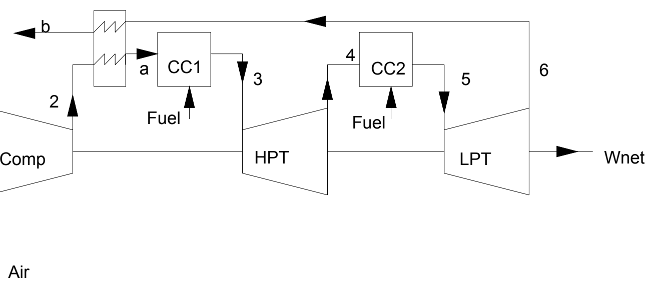

Figure 2: Flow diagram of reheat heat exchanger cycle (RHC)

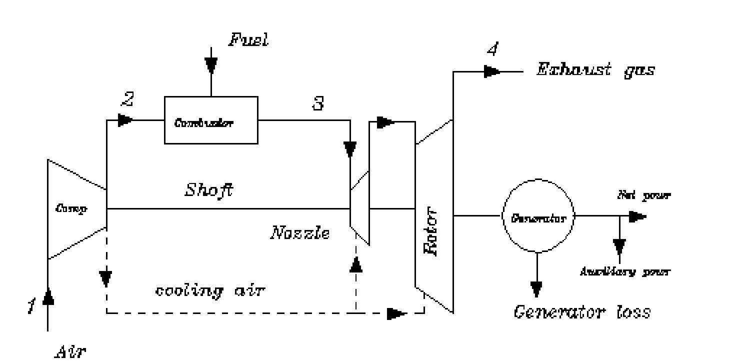

Figure 3: Flow diagram for cooled turbine model

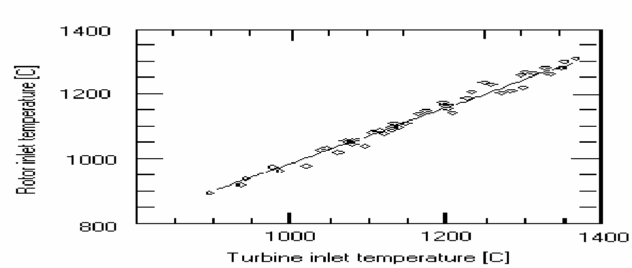

Figure 4: Rotor inlet temperature against turbine inlet temperature for various gas turbines

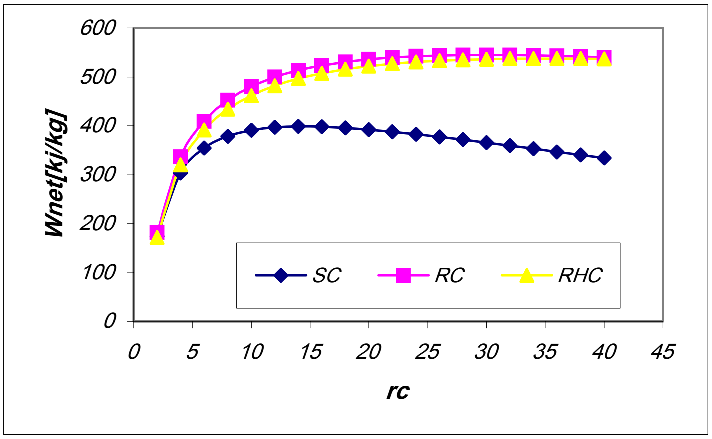

Figure 5: Specific net work versus pressure ratio using uncooled turbine model

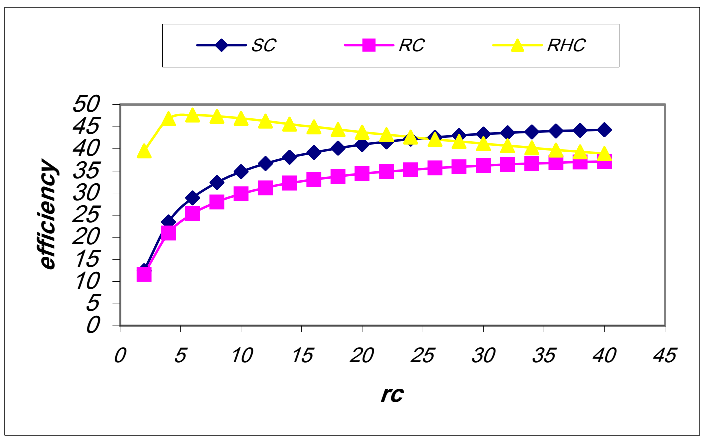

Figure 6: Efficiency versus pressure ratio using uncooled turbine model

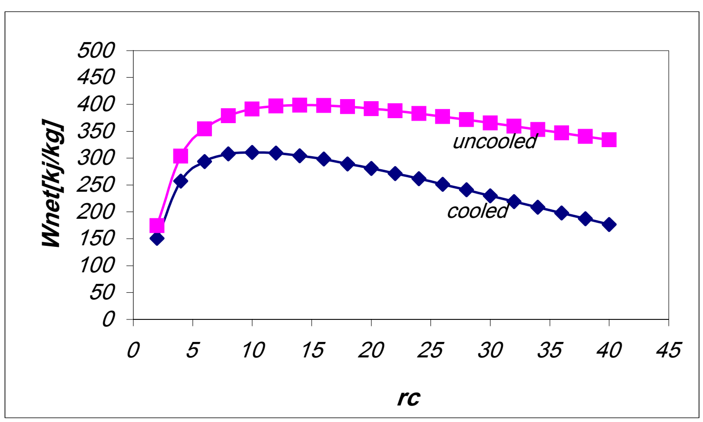

Figure 7: Comparison of variation of SC specific net work against pressure ratio using both cooled and uncooled turbine models

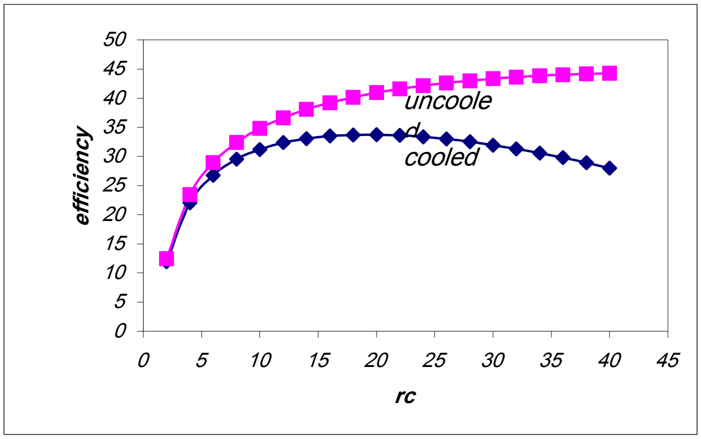

Figure 8: Comparison of variation of SC efficiency against pressure ratio using both cooled and uncooled turbine models

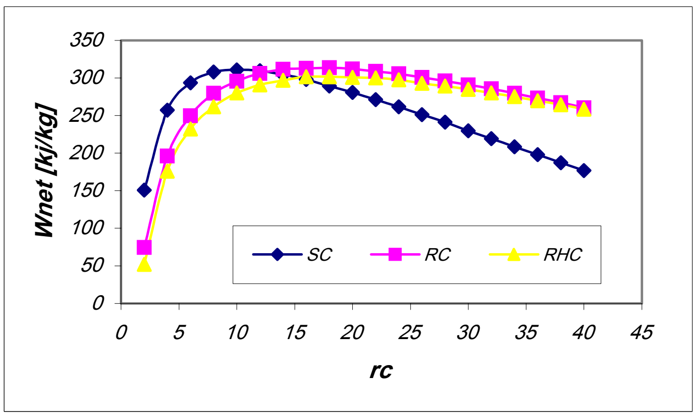

Figure 9: Specific net work versus pressure ratio using cooled turbine model

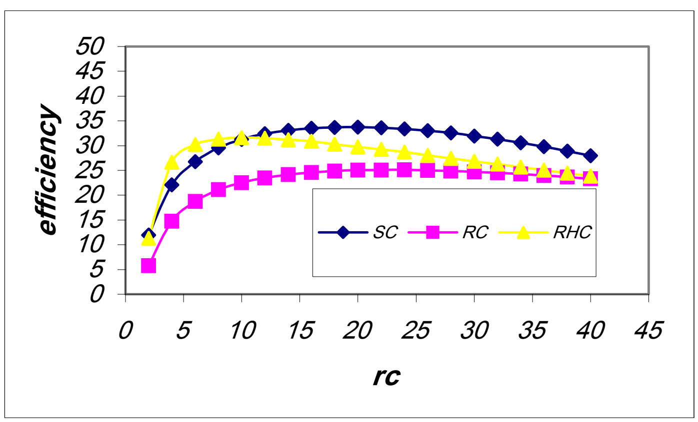

Figure 10: Efficiency versus pressure ratio using cooled turbine model