THERMODYNAMIC MODEL FOR COMBINED OPEN-CYCLE-TWIN-SHAFT GAS TURBINE AND EXHAUST GAS OPERATED ABSORPTION REFRIGERATION UNIT

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ABSTRACT

thermodynamic analysis of combined open-cycle-twin-shaft gas turbine (brayton cycle) and exhaust gas operated absorption refrigeration unit carry a significant amount of thermal energy that is usually expelled to the atmosphere without taking any further part in the power generation processes. The low grade thermal energy can however be put to beneficial use. This paper explores the utilisation of the exhaust gases of an open-cycle-twin-shaft gas turbine. An air standard cycle is assumed for the gas turbine, first with the aid of thermodynamics laws the specific net work and the efficiency of the cycle as a function of temperature ratio and pressure ratio of the cycle are calculated, and the realistic bounds placed on the cycle by the thermodynamic analysis is shown. Then the temperature of the exhaust gases and the heat that can be put into beneficial for precooling in terms of temperature ratio and pressure ratio of the cycle are determined. The specific net work and efficiency of a precooled cycle have been calculated and compared to the conventional systems. It has been concluded that the precooling has a marked effect on the specific net work and efficiency at low temperature ratio. Also without increasing the maximum cycle temperature the precooled cycle can work at a higher compressor pressure ratio and at higher temperature ratio.

Keywords - Brayton cycle, Absorption refrigeration, Gas turbine CHP Charge air cooling Energy recovery, Thermodynamic analysis

Notation

$$a = \frac{n_c - 1}{n_c} = \frac{1}{\eta_c} \times \frac{\gamma - 1}{\gamma}$$

C_p Specific heat at constant pressure

h Specific enthalpy
m mass flow rate
n Polytropic exponent
Q Heat
S Entropy
W Work
η Efficiency
η_C
η_{HE} Heat exchange efficiency
π Cycle pressure ratio
T Temperature
θ Cycle temperature ratio

Subscripts:

1 Beginning of compression
2 End of compression and beginning of combustion
3 End of combustion and beginning of expansion in high pressure turbine
4 End of the expansion in high pressure turbine and beginning of expansion and beginning of expansion
5 End of the expansion in low pressure turbine
amb Ambient
C Compressor, compression
c Charge air cooling capacity required
ex Exhaust
cooling Exhaust gases cooling capacity
HPT high pressure turbine
LPT Low pressure turbine
pre Precooled

INTRODUCTION

Man has a natural tendency to try to improve the efficiency of devices which convert thermal energy into mechanical work. Moreover the rise in fossil fuel costs is a simulus for this kind of research. The energy consumption of the world has been increased very rapidly. It appears that an energy gap, i.e., the difference between consumption and the naturally growing productions of non-renewable sources of energy might develop. In order to bridge this energy gap, and also consider the additional problems of atmospheric pollution, some policy should be considered.

Among several options, one of them is, the use of combined heat and power plant, in which the energy supplied to a thermal power station is more effectively used. The thermal efficiency of most power plants lies between 30% to 40% [1], [2], [3] and [4], consequently a large amount of energy released during combustion of fuel is rejected to the ambient as low grade heat. The exhaust gases of a power plant carry a significant amount of thermal energy that is usually expelled to the atmosphere without taking any further part in the power generation process. The low grade thermal energy can however be put to beneficial use.

Cogeneration is an engineering concept involving the production of both electricity and useful thermal energy in one operation, there by utilising fuel more effectively than if the desired products were produced separately. Since the heart of a cogeneration system is a prime mover with waste heat at a useful temperature, it is not surprising that the requirements of cogeneration may be met in many ways.

Serious development of the gas turbine began not long before the Second World War with power shaft in mind, but attention was soon transferred to the turbojet engine for aircraft propulsion. The gas turbine began to compete successfully in other fields only in the mid nineteen fifties, but since then it has made a progressively greater impact in an increasing variety of applications. Gas turbine exhaust gases is relatively clean, especially the exhaust gases from the natural gas turbines. The use of such gases has the advantage of reducing the maintenance costs of cleaning the equipment and reducing the maintenance costs of the heat recovery equipment used in the application [5].

The utilisation of the exhaust waste heat from power plants is now well known and form the basis of many combined heat and power installations. The combination of a thermal power cycle and a refrigeration cycle, has however, been the subject of little interest in the past. The objective of this study is to determine the thermodynamic basis to enable a full understanding of the potential and limitation of a cogeneration system which consists of an open-cycle-twin-shaft gas turbine combined with an absorption refrigeration unit. An air standard cycle is assumed for the gas turbine, first by the aid of thermodynamics laws the specific net work and the efficiency of the cycle as a function of temperature ratio (3 to 6) and pressure ratio (6 to 18) [2] of the cycle are calculated, and the realistic bounds placed on the cycle by the thermodynamic analysis has been shown. Then the temperature of the exhaust gases and consequently the amount of energy available in the exhaust gases in the form of

heat that can be put into beneficial use in terms of temperature ratio and pressure ratio of the cycle are determined.

This energy can be introduced to an absorption unit and by considering the efficiency of heat exchangers and the coefficient performance of the absorption refrigeration machine the amount of cooling capacity available in the exhaust gases can be calculated, this can be used for charge air cooling or air conditioning purposes or both.

Thermodynamic model for combined open-cycle-twin-shaft gas turbine and absorption refrigeration unit

The cycle examined in this study is an open-cycle-twin-shaft gas turbine combined with an absorption refrigeration machine. The object of the study is the examination of the thermodynamic performance of each of the components of this to determine the most suitable configuration for combing it with an absorption unit. The specific net work and efficiency of the cycle as a function of pressure ratio and temperature ratio of the cycle are calculated, then by writing a heat balance equation for the cycle the amount of heat available in the exhaust gases is determined.

The schematic diagram of the cycle is shown in Fig (1) and the processes are given in P-V and T-S diagrams, Fig (2). Based on the thermodynamics laws, the specific net work and the efficiency will be:

$$\frac{W_{net}}{mC_p T_{amb}} = \theta\eta_T\left(1-\frac{1}{\pi^a}\right)-\frac{1}{\eta_c}\left(\pi^a-1\right)$$

$$\eta_{th} = \left(1-\frac{1}{\pi^a}\right)\left[\frac{\eta_c\eta_T\theta-\pi^a}{\eta_c\theta-\left(\eta_c+\pi^a-1\right)}\right]$$

Heat energy available in the exhaust gases

In order to calculate the amount of heat available in the exhaust gases we have to

calculate the value of T_5 in terms of cycle parameters, which will be as follows

$$T_5 = \theta \times T_{amb}\left[(1-\eta_T) + \frac{\eta_T}{\pi^a}\right]$$

Considering the efficiency of heat exchangers, the amount of energy available in the exhaust gases is equal to

$$\frac{Q_{ex}}{m_{ex}Cp_{ex}T_{amb}} = \eta_{HE}\left\{\frac{1}{2}\left[\theta\left(1-\eta_T+\frac{\eta_T}{\pi^a}\right)+1\right]-\frac{T_{out,ex}}{T_{amb}}\right\}$$

Cooling capacity available in the exhaust gases

It is possible to express the exhaust gas temperature in terms of the cycle pressure ratio and cycle temperature ratio, so the heat available in the exhaust gases and

consequently the cooling capacity can be determined Considering the efficiency of heat exchangers the amount of energy available in the exhaust gases in the form of heat will be:

$$\frac{Q_{ex}}{m_{ex}Cp_{ex}T_{amb}} = \eta_{HE}\left\{\frac{1}{2}\left[\theta\left(1-\eta_T+\frac{\eta_T}{\pi^a}\right)+1\right]-\frac{T_{out,ex}}{T_{amb}}\right\}$$

And assuming a 3% transmission loss (stray energy to the environment) [3] and following refs. [4], [8], [9] and [10], the coefficient of the performance of the absorption unit has been set to 0.50, the cooling capacity is then equal to

$$\frac{Q_{ex}}{m_{ex}Cp_{ex}T_{amb}} = \eta_{HE}(0.97)(0.50)\left\{\frac{1}{2}\left[\theta\left(1-\eta_T+\frac{\eta_T}{\pi^a}\right)+1\right]-\frac{T_{out,ex}}{T_{amb}}\right\}$$

Cooling capacity required for charge air Cooling

The amount of cooling capacity required for charge air cooling is directly related to the temperature of the charge air due to the cooling process and the efficiency of the heat exchangers that have been used. In this study the maximum temperature difference for precooling is assumed to be 50 ° C and the calculations are based on the assumption that,

the specific heat through the cycle is constant and the mass of air is equal to the mass of the exhaust gases, i.e.,

$$\frac{Q_{ex}}{m_{ex} Cp_{ex} T_{amb}} = \frac{\Delta T_{pre}}{T_{amb} \times \eta_{HE}}$$

Comparison of conventional cycle and precooled cycle

A-Cycle pressure ratio is the same for both cycles

Let it be assumed that the cycle pressure ratio for both cycles is the same, so we can write

$$T_{2,non-cooled} = T_{amb}\left[1 + \frac{\pi^a - 1}{\eta_c}\right]$$

And

$$T_{2,cooled} = T_{amb}\left(1 - \frac{\Delta T_{pre}}{T_{amb}}\right)\left[1 + \frac{\pi^a - 1}{\eta_c}\right]$$

or

$$\frac{T_{2,non-cooled} - T_{2,cooled}}{T_{2,cooled}} = \frac{\Delta T_{pre}}{T_{amb}}$$

B-Temperature introduced to the combustion process is the same for both cycles

Based on the assumption we have

$$T_{2,non-cooled} = T_{amb}\left[1 + \frac{\pi_1{}^a - 1}{\eta_c}\right]$$

and

$$T_{2,cooled} = T_{amb}\left(1 - \frac{\Delta T_{pre}}{T_{amb}}\right)\left[1 + \frac{\pi^a - 1}{\eta_c}\right]$$

Or

$$\pi_2 = \left(\frac{\eta_c + \pi_1^a - 1}{1 - \frac{\Delta T_{pre}}{T_{amb}}} + 1 - \eta_c \right)^{1/a}$$

Cooling capacity available in the exhaust gases

It is possible to express the exhaust gas temperature in terms of the cycle pressure ratio and cycle temperature ratio, so the heat available in the exhaust gases and consequently the cooling capacity can be determined Considering the efficiency of heat exchangers the amount of energy available in the exhaust gases in the form of heat will be:

$$\frac{Q_{ex}}{m_{ex} Cp_{ex} T_{amb}} = \eta_{HE} \left\{ \frac{1}{2} \left[\theta \left(1 - \eta_T + \frac{\eta_T}{\pi^a} \right) + 1 \right] - \frac{T_{out,ex}}{T_{amb}} \right\}$$

And assuming a 3% transmission loss (stray energy to the environment) [4] and following refs. [5], [6], [7] and [8], the coefficient of the performance of the absorption unit has been set to 0.50, the cooling capacity is then equal to

$$\frac{Q_{ex}}{m_{ex} Cp_{ex} T_{amb}} = \eta_{HE} (0.97)(0.50) \left\{ \frac{1}{2} \left[\theta \left(1 - \eta_T + \frac{\eta_T}{\pi^a} \right) + 1 \right] - \frac{T_{out,ex}}{T_{amb}} \right\}$$

Discussion of the results

The results of the above analysis are shown in figures (3)to (5). The efficiency and net specific work have been plotted for a range of cycle pressure ratios from 6 to 18, whilst the cycle temperature ratio has been varied from 3 to 6 in increments of 0.50. For cycle temperature ratio of 3 and cycle pressure ratios higher than 11.2 the

values of efficiency are below zero and the net specific work is negative which clearly is not acceptable. In this study the efficiency of the compressor is assumed to be 0.87 and the efficiency of turbine is assumed is equal to 0.85, while in the calculations, the minimum stack temperature of 130 °C set by the dew point of the exhaust gases [2] has been used for the exhaust gases leaving the heat exchanger.

It has been shown that at a fixed pressure ratio, the gas turbine cycle efficiency increases with an increase in cycle temperature ratio, and at a fixed temperature ratio, the cycle efficiency increases with an increase in cycle pressure ratio, until a maximum value is reached, then decreases with an increase in cycle pressure ratio. The point of maximum value depends on the cycle temperature ratio. The non-dimensional net work shows the same trend as the efficiency.

In this study the specific net work and efficiency of a precooled cycle have been calculated and compared to the conventional systems. For a reduction in the inlet air temperature the specific net work increases for all values of the cycle temperature ratio .The efficiency of the cycle also increases due to the redaction in compressor work requirement.

Conclusion

The following conclusions can be drawn from this work

At a fixed cycle temperature ratio, higher cycle pressure ratio cycles has less heat energy in the exhaust gases. At a fixed cycle temperature ratio, higher cycle temperature ratio has more heat energy in the exhaust gases. At a fixed cycle temperature ratio, higher cycle temperature ratio has more heat energy in the exhaust gases. For high cycle temperature ratios (4.5 to 6), at all cycle pressure ratios considered in this study (6 to 18), there is enough cooling capacity for charge air cooling of up to 50 °C at an ambient temperature of 60 °C. For low cycle temperature ratios (3 and 4), the maximum pressure ratio at which these conditions can be met is 12. The temperature of the exhaust gases and consequently the amount of energy available in the exhaust gases in the form of heat that can be put into beneficial use in terms of temperature ratio and pressure ratio of the cycle are determined, considering the high temperature of the exhaust gases which is around 450-600 °C[2], There is sufficient amount of energy in the exhaust gases for precooling purposes. This energy can be introduced to an absorption unit and by considering the efficiency of heat exchangers and the coefficient performance of the absorption refrigeration machine, the amount of cooling capacity available in the exhaust gases can be used for charge air cooling or air conditioning or both.

From the result of the analysis it can be concluded that the precooling has a marked effect on the specific net work and efficiency at low temperature ratios. Also without applying any more thermal stress to the power plant the precooled cycle can work at higher compressor pressure ratio up to 37.82%, which depends on the cycle pressure ratio and the degree of precooling, and for the same compressor pressure ratio the temperature introduced to the combustion process will be lower by up to 15%, which depends on the degree of precooling.

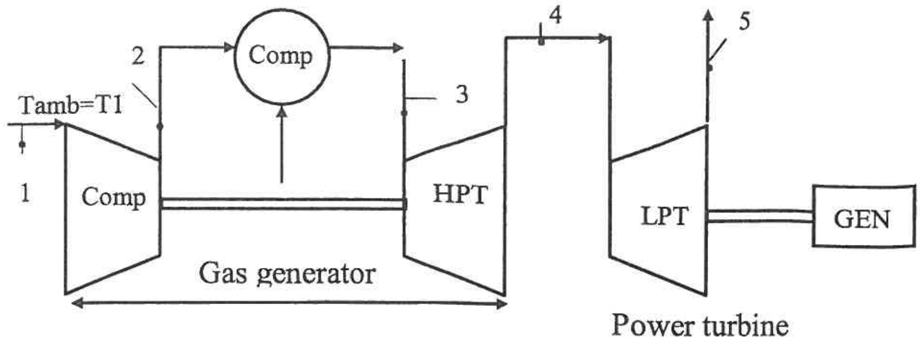

Fig (1) schematic diagram of open-cycle -twin-shaft gas turbine

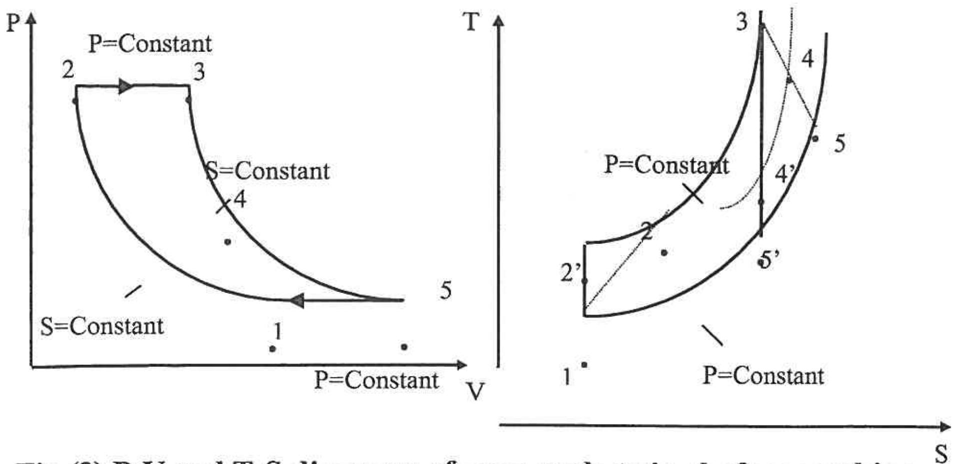

Fig (2) P-V and T-S diagrams of open-cycle-twin-shaft gas turbine

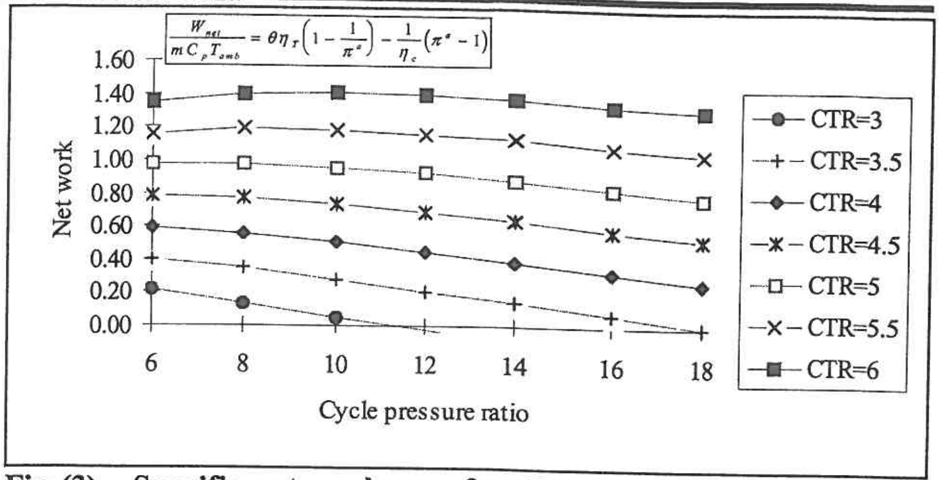

Fig (3) - Specific net work as a function cycle pressure ratio and cycle temperature ratio

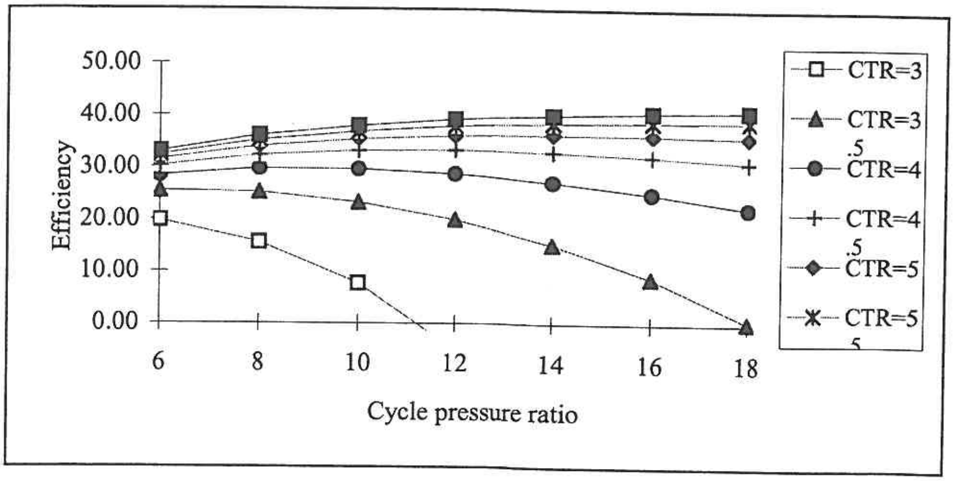

Fig (4) - Efficiency as a function cycle pressure ratio and cycle temperature ratio

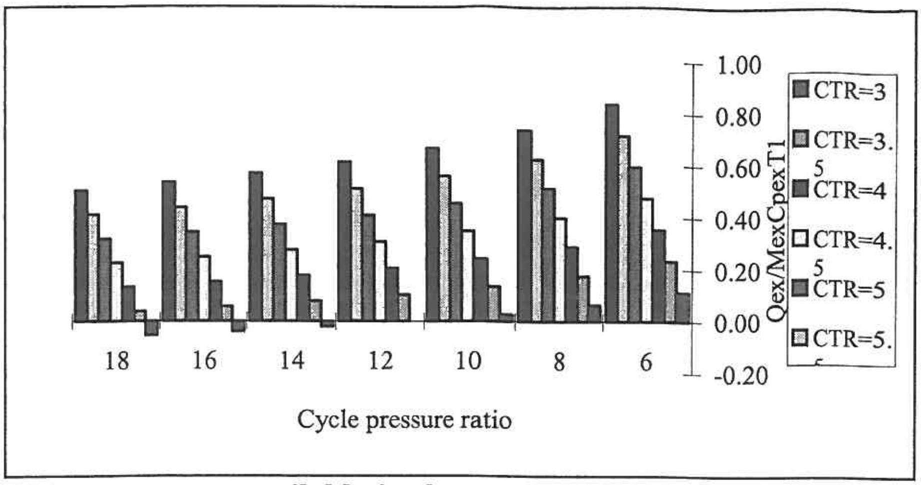

Fig (5) - Heat energy available in the exhaust gases as a function cycle pressur

ratio and cycle temperature ratio

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