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ENERGY AND EXERGY ANALYSIS OF GAS TURBINE COGENERATION POWER PLANT.

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Abstract. In this paper, a comprehensive thermodynamic analysis of an indirect fired gas turbine cogeneration system with multistage reheat system for heating and electricity is reported. An exergy analysis was carried out in the present study for improving and optimizing the overall performance of the system. This cogeneration system consists of a gas turbine cycle, a heat recovery steam generator cycle and a steam turbine cycle. A comparative study between no heat and multistage reheat shows that there is a significant improvement in electric power output, process heat production, fuel utilization efficiency, and exergy efficiency due to reheat. It is also shown that the power to heat ratio decreases with the reheat because improvement in process heat generation is greater than the improvement in electric power output.

Keywords: Exergy, cogeneration, steam turbine, gas turbine, indirect fired

1. Introduction

Cogeneration involves the production of both thermal energy, generally in the form of steam or hot water, and electricity. The ratio of electric power to thermal energy varies depending on the plant type. A cogeneration plant may be conceived to supply thermal energy or electric power.

In the first case, electric power is considered to be a byproduct and is relatively small and revealed that fossil fuels account for about 80%, renewable energy resources contribute 14% and nuclear 6% of world annual energy use. These numbers will soon change as the world's population grows, energy demand rises, inexpensive oil and gas deplete, global warming effects continue to rise and urban pollution worsens the living conditions [1]. India consumed nearly 7% of coal of the world whereas China, the U.S, Japan and the rest of the world consumer 43%, 9%, 4% and 20%, respectively.68% of world's consumer of coal for the generation of electricity. Coal-fired generation increases by an annual average of 2.3 percent, making coal the second-fastest-growing source for electricity generation [2-3]

Thermal energy in the form of steam can be extracted from a point in the HRSG, in the live steam piping. The optimum extraction point depends on the required steam pressure, temperature and quantity over the load range. Gas turbine cogeneration is far more efficient than the typical steam utility central plants. About 75 % of heat utilization can be realized for power and heat, with about 25 % leaving in the exhaust gases. In fossil-fired steam power plants, only 35 percent of the fuel energy is obtained as power with condenser losses and boiler losses are accounting for 48 percent and 15 percent respectively. The combustion gas turbines are capable of producing significantly more electric power for a given amount of process heat. Since the cost-effectiveness of cogeneration system is directly related to the amount of power it can produce. But gas turbine cogeneration system has been recognized as a promising concept for energy conservation [4-5].

A lot of literature has been reported on the performance analysis of the cogeneration system based on exergy analysis of thermodynamics [6-10]. Rosen et al. [11] describe the efficiency analysis of the cogeneration system from first law of point of view the effect of some inefficiency parameter. Benelmir and Feidt [12] describe the concept of energy management strategy in cogeneration system which has been found quite beneficial for energy engineers to design the energy conversion system. A very few literature are available related to exergy analysis of cogeneration system. Huang et al. [13] performed an exergy analysis on a cogeneration system with steam- injected gas turbine. By specifying the balance equation of mass-energy and exergy of the components they determined the exergy loss. By taking the compressor pressure ratio, ratio of the vapour injected, Temperature of the vapour, and amount of the feed water as the parameters, they wrote down the outputs of the first and second law and calculated the heat power ratios. They also stated that while the highest exergy loss occurred in the combustion chamber, the highest exergy leakage occurred through the waste gases. Bandyapadhyay et al. [14] realized the thermo-economic optimization of the cogeneration facility through parameters such as the magnitude of the facility, investment costs, and power generation required for designing and operating the facilities. Al-attab and Zainal [15] were reviewed the externally fired heat engines widely used for solid fuels like coal. In their studied, they were utilized a wide range of thermal power sources such as concentrated solar power, fossil, nuclear and biomass as a working fuel. The methods used for improving cycle efficiency for externally fired combined cycle, humidified air turbine, fuel cells, and other cycles are reviewed thoroughly. Caglayan and Caliskan [16] were studied, an energy, exergy and sustainability analyses applied in the ceramic sector to simulate gas turbine-based cogeneration plant model. The cogeneration system mainly consists of a proposed gas turbine unit, a wall tile dryer, and a ground tile dryer. Results clearly indicated that the air compressor and combustion chamber are the two most efficient components in the system, while the minimum one may be the wall tile dryer as (7.98%). On the other hand cogeneration system had 17.51% energy efficiency; while its maximum exergy efficiency was found to be 29.94% at 10 ^{0}C at environmental condition. Caglayana and Caliskan [17] investigated the first and second laws of thermodynamics based environmental and enviroeconomic assessments of the natural gas-fired cogeneration system, including ground tile dryers and a gas turbine system. The results indicated that the thermoeconomic values was to be 2766.132 kW h/$ for the ground tile dryer, 2479.726 kW h/$ for the wall tile dryer, 1595.575 kW h/$ for the combustion chamber, 543.212 kW h/$ for the cogeneration (overall) system, 239.074 kW h/$ for the gas turbine, while the air

2. Literature review

The performance of a cogeneration cycle is defined not only by the efficiency but also by parameters such as fuel utilization and power to heat ratio. These parameters take into account the thermal as well as the electrical output.

Fuel utilization is a measure of how much of the fuel energy supplied is usefully used in the plant. It is equal to the sum of electrical output and thermal output divided by the fuel input.

The power to heat ratio is defined as the ratio between the electrical output and the thermal energy produced. Combined cycles tend to have high power coefficients so they are more likely to be used for cogeneration applications with a relatively high power demand. This is because the electrical output of the gas turbine- about two-thirds of the total plant output- cannot be converted into thermal energy. In a conventional steam power plant, all of the energy produced could be exported to the process giving a possible power coefficient of zero.

Szargut [18] stated that exergy is the amount of work obtainable when some matter is brought to a state of thermodynamic equilibrium with the common components of the natural surroundings. The fundamental of the exergy method were laid down by Carnot in 1824 and Clausius in 1865.

Kapooria et al. [19] carried out a thermodynamic analysis of the Rankine cycle to enhance the efficiency and reliability of the steam power plant. Further, they identified factors such as reheating and affecting efficiency of the Rankine cycle and analyzed for improve working of the thermal power plant in subcritical range.

Srinivas et al. [20] conducted a thermodynamic analysis of the Rankine cycle with generalization of feedwater heater in subcritical range. They studied the number of feedwater heaters and bled temperature ratios on the overall performance of the Rankine cycle in subcritical range. They have develop developed computer code for the evaluation of the first law efficiency, irreversibilities and second law efficiency Rankine cycle with different number of feedwater heater. They concluded that greatest increment efficiency is bought by the first heater; the increment for each additional heater thereafter successively diminished. An increase in feed water temperature reduces the heat absorption from the outgoing flue gases in the economizer and causes a reduction in boiler efficiency.

Habib and Zubair [21] discussed first and second law procedures for the optimization of the reheat pressure level in the reheat regeneration thermal power plant in subcritical range. The procedure is general in form and is applies for a thermal power plant having to reheat pressure levels (low and high-pressure levels) and two open-type feedwater heaters. The second law efficiency of the steam generator, turbine cycle and plant were evaluated and optimized the reheat pressure ratio in both the pressure levels. The irreversibilities in the different components of the steam generator and turbine cycle sections were evaluated and discussed.

Ibrahim et al. [22] conducted a second law analysis of the reheat regenerative Rankine cycle in the subcritical range. They have performed the energy and exergy analysis for each component in the system at some influenced operating parameters such as turbine inlet pressure at 150 bar, turbine inlet temperature at 600 ^{0}C and condenser pressure at 0.1 bar. Their results show that exergy analysis is better in comparison with energy analysis, as it gives a clear understanding of actual losses in the system.

Srivastava [23] discussed the second law analysis of various types of coal from major mines of the world. He concluded that the first law analysis gives only the quantity of energy, while the second law defines the quality of energy. The projected increase in coal utilization in power plant makes it desirable to evaluate the energy content of coal both quantitatively and quantitatively.

Aljundi [24] studied the exergy analysis of Al-Hussein 396 MW power plant installed in Jordan. The performance of the plant was estimated by a component-wise modeling and a detailed break-up of energy and exergy losses for the consideration plant has been presented. It was found that the exergy destruction rate of the boiler is dominant over all other irreversibility in the cycle. It exergy analysis provides the tool for a clear distinction between energy losses to the environment and internal irreversibility in the process.

Sachdeva and Karun [25] determined the magnitude, location, and source of thermodynamic inefficiencies of a thermal power plant. The first law of thermodynamics introduces the concept of energy conversion, which states that energy entering a thermal system with fuel, electricity, flowing stream of matter and so on is conversed and cannot be destroyed. Exergy is a measure of the quality or grade of energy and it can be destroyed in the thermal system. The second law states that part of the exergy entering a thermal system with fuel; electricity flowing stream of matter and so on is conversed and cannot be destroyed.

Sanjay and Mehta [26] study of energy and exergy analysis on a 125 MW coal-based thermal power. Most of the power plant was designed by the energetic performance criteria based on first law thermodynamics only. The real useful energy loss cannot be justified by the first law of thermodynamics; it does not differentiate between the quality and quantity of the energy. Also present major losses available energy at combustor, super-heater, economizer and air-pre heater section. In this article exergy efficiency, exergy destruction and energy losses comparison charts are also defined. The definite value of thermal energy can only be obtained by qualitative or exergy analysis of its conversation, transport, and distribution.

3. System Description

A gas turbine system, in general, could have any number of reheat stages. The schematic diagram of gas turbine cogeneration system with one stage of reheat is shown in Fig. 1.

The energy input to the turbine cycle is heat added to the working fluid in the main air heater and the reheater. The fuel input to the system is the fuel supplied to the main air heater and the reheater. This quantity will primarily depend on the maximum cycle temperature at the main heater outlet and the reheat temperature at the reheater outlet. In this study, it should be considered, that the reheat temperature is the same as the maximum cycle temperature. The power output of the system will depend on the expansion ratio for each of the turbines. In the current study the same expansion ratio for each of the turbines. It should be further considered that the adiabatic turbine efficiency to be the same for each turbine. The hot turbine exhaust entering the heat recovery steam generator is the waste heat source for process heat generation. The quantity and quality of process steam produced will depend on the temperature of air entering and temperature of steam produced in the heat recovery steam generator.

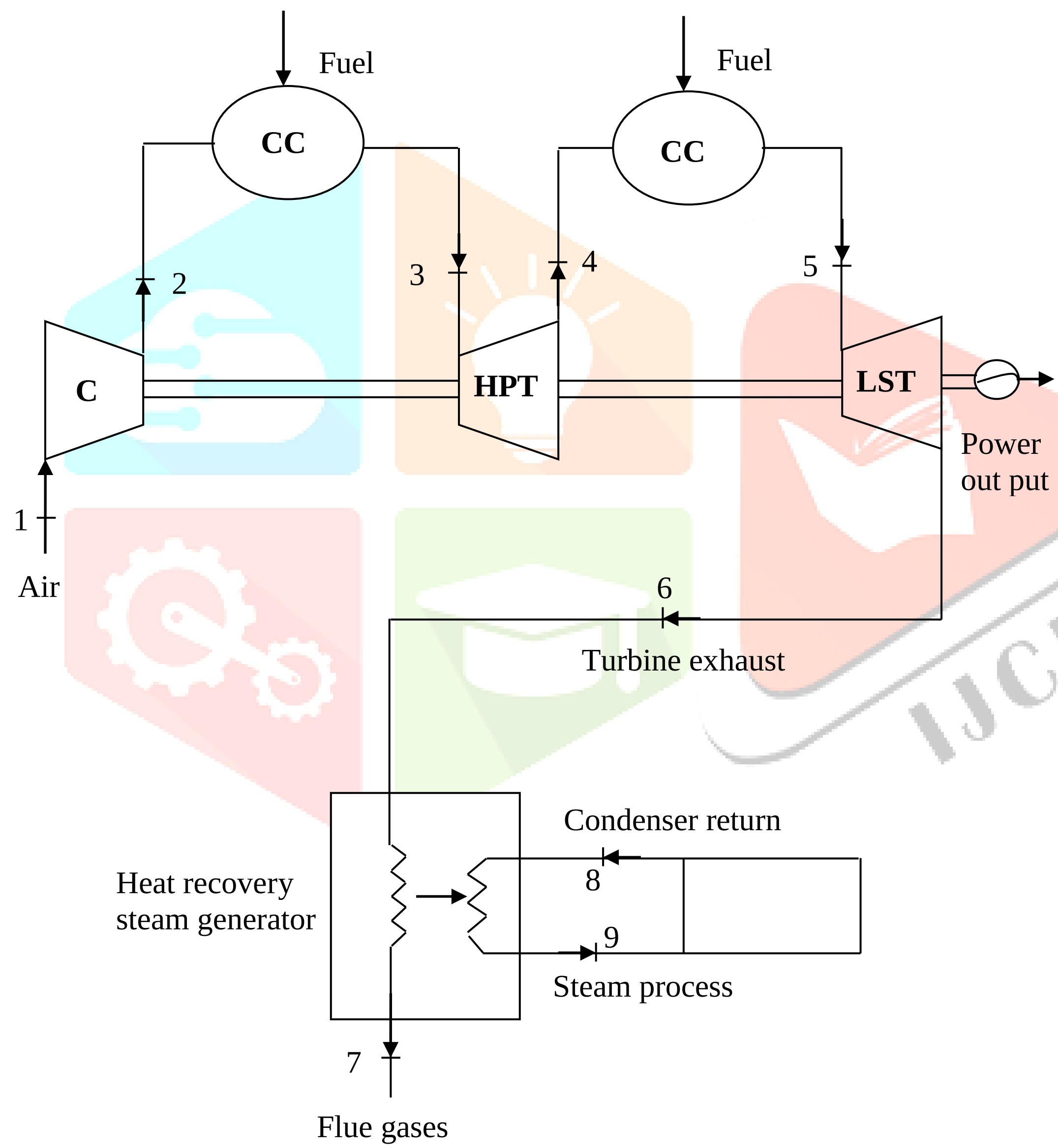

Fig. 1 Schematic diagram of gas turbine cogeneration system with reheat

3.1Thermodynamic analysis and the cycle performance

Power Output

The net power output of the cycle with one reheat is given by

$$\dot{W}_{net} = [(h_3 - h_4) + (h_5 - h_E) - (h_2 - h_1)] \tag{1}$$

Since air is an ideal gas with constant specific heats, we have

$$\dot{W}_{net} = Cp[(T_3 - T_4) + (T_5 - T_E) - (T_2 - T_1)] \tag{2}$$

The isentropic efficiency of turbine and compressor is defined as

$$\eta_{GT} = \frac{(T_3 - T_4)}{(T_3 - T_4)_{is}} \tag{3}$$

$$\eta_C = \frac{(T_2 - T_1)_{is}}{(T_2 - T_1)} \tag{4}$$

Pressure ratio across all the turbines will be same

$$\frac{P_3}{P_4} = \frac{P_5}{P_E} \tag{5}$$

Using equations (3), (4) and (5), equation (2) may be expressed as

$$\dot{W}_{net} = C_p\left[2T_{max}\left(1 - \frac{1}{\pi_{GT}^{\alpha}}\right)\eta_{GT} - \frac{T_1}{\eta_C}\left(\pi_C^{\alpha} - 1\right)\right] \tag{6}$$

Where $\pi_{GT} = \frac{P_3}{P_4} = \frac{P_5}{P_E}$, $\alpha = \frac{\gamma - 1}{\gamma}$, $\pi_C = \frac{P_2}{P_1}$, $T_{max} = T_3 = T_5$

For a system with 'n' stages of reheat the power output becomes.

$$\dot{w}_{net} = Cp\left[(n+1)\eta_{GT}T_{max}\left(1 - \frac{1}{\pi_{GT}^{\alpha}}\right) - T_1\left(\frac{\left(\pi_C^{\alpha} - 1\right)}{\eta_C}\right)\right] \tag{7}$$

Divide equation (7) across by (CpT1) to get the specific cycle power output, we have

$$\dot{W}_{net} = \left[n\theta\eta_{GT}\left(1 - \frac{1}{\pi_{GT}^{\alpha}}\right) - \left(\frac{\left(\pi_{C}^{\alpha} - 1\right)}{\eta_{C}}\right)\right] \tag{8}$$

It may also be written as

$$\dot{W}_{net} = \left[n\theta\eta_{GT}\,\psi_{GT} - \frac{\psi_{C}}{\eta_{C}} \right] \tag{9}$$

Where $1 - \frac{1}{\pi_{GT}^{\alpha}} = \psi_{GT}$, $\psi_{C} = \pi_{C}^{\alpha} - 1$, $\theta = \frac{T_{max}}{T_{min}} = \frac{T_3}{T_1} = \frac{T_5}{T_1}$

The electrical power output of the system may be obtained from

$$\dot{W}_{el} = \eta_g \dot{W}_{net} \tag{10}$$

$$\dot{w}_{el} = \eta_g \dot{w}_{net} \tag{11}$$

Where η_g is electrical conversion efficiency

Energy Input

Heat input to the cycle with one stage of reheat is given by

$$\dot{Q}_{in} = [(h_3-h_2) + (h_5-h_4)] \tag{12}$$

Assuming ideal gas with constant specific heats we have on the basis of full reheat

$$Q_{in} = Cp[(T_3-T_2) + (T_5-T_4)] \tag{13}$$

It may also be written as

$$\dot{Q}_{in} = Cp\left[T_{max} - T_1 - \frac{T_1\psi_C}{\eta_C} + T_{max}\psi_{GT}\eta_{GT} \right] \tag{14}$$

For a system with 'n' stages of reheat, we have

$$\dot{Q}_{in} = Cp\left[T_{max} - T_1 - \frac{T_1\psi_C}{\eta_C} + T_{max}\psi_{GT}\,n\eta_{GT} \right] \tag{15}$$

Divide equation (15) across by CpT_1, we have

$$\dot{q}_{in} = (\theta - 1) - \left(\frac{\psi_C}{\eta_C}\right) n\eta_{GT}\theta\Psi_{GT} \tag{16}$$

Energy of fuel input ΔH_f may be obtained as

$$\Delta H_f = \frac{\dot{Q}_{in}}{\eta_H} \tag{17}$$

Where η_H is the air heater efficiency

Process Heat Production

The amount of process heat produced is given by

$$\dot{Q}_P = (h_E - h_F) \tag{18}$$

Assuming ideal gas with constant specific heats, we have

$$\dot{Q}_P = C_P(T_E - T_F) \tag{19}$$

Turbine efficiency may be defined as

$$\eta_{GT} = \frac{(T_E - T_5)_{is}}{(T_{Eis} - T_5)} \tag{20}$$

and also $T_{Eis} = T_{max}(1 - \Psi_{GT})$ (21)

Using equation (20) and (21), equation (19) may be written as

$$\dot{Q}_P = Cp\left[T_{max} - \eta_{GT}\psi_{GT}T_{max} - T_F\right] \tag{22}$$

Divide equation (22) across by CpT_1 we have

$$\dot{q}_p = [\theta - \theta\Psi_{GT}\eta_{GT} - \tau] \tag{23}$$

where $\tau = \frac{T_F}{T_0}$

$$T_F = (T_P + T_{PP}) - [T_E - (T_P + T_{PP})]\left(\frac{h_F - h_c}{h_g - h_F}\right) \tag{24}$$

First Law Efficiency

The ratio of all the useful energy extracted from the system (electricity and process heat) to the energy of fuel input is known as the fuel-utilization efficiency (η_1), which is also known as the first-law efficiency since only energy accounting is involved. According to this definition η_1 is given by

$$\eta_1 = \frac{\left(\dot{W}_{el} + \dot{Q}_P\right)}{\Delta \dot{H}_F} \tag{25}$$

The heater efficiency and air turbine cycle thermal efficiencies are

$$\eta_H = \frac{\dot{Q}_{in}}{\Delta \dot{H}_F} \tag{26}$$

$$\eta_{TH} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{27}$$

After using equation (10), (26) and (27) in equation (25), it may be written as

$$\eta_1 = \eta_H \left(\eta_g \eta_{TH} + \frac{\dot{Q}_P}{\dot{Q}_{in}} \right) \tag{28}$$

Again by using equation (9), (16) and (23) for 'n' stages of reheat η_f may further be expressed as

$$\eta_1 = \eta_H \left[\eta_g \left\{ \frac{(n+1)\theta\psi_{GT}\eta_{GT} - (\psi_C/\eta_C)}{(\theta-1) - (\psi_C/\eta_C) + n\eta_{GT}\theta\psi_{GT}} \right\} + \left\{ \frac{\theta - \eta_{GT}\theta\psi_{GT} - \tau}{(\theta-1) - (\psi_C/\eta_C) + n\eta_{GT}\theta\psi_{GT}} \right\} \right] \tag{29}$$

Electrical to thermal energy ratio (Power to Heat Ratio)

$$R_{PH} = \frac{\dot{W}_{el}}{\dot{Q}_{in}} \tag{30}$$

After using equation (9) and (23) in equation (30), we have

$$R_{PH} = \left[\eta_g \left\{ \frac{(n+1)\theta\psi_{GT}\eta_{GT} - (\psi_C/\eta_C)}{\theta - \theta\eta_{GT}\psi_{GT} - \tau} \right\} \right] \tag{31}$$

Second-Law Efficiency (Exergetic Efficiency)

Efficiency is defined as the ratio of output to input. If we consider both output and input in terms of energy, we have the so-called first-law efficiency. Since exergy is more valuable than energy according to the second-law of thermodynamic, it is useful to consider both output and input in terms of exergy.

The second-law efficiency is given by the following expression as

$$\eta_2 = \frac{\left(\dot{W}_{el} + \dot{E}_P\right)}{\dot{E}_P} \qquad (32)$$

Where $\dot{W}_{el}$ is all exergy, $\dot{E}_P$ is the exergy content of process heat, and $\dot{E}_F$ is the exergy content of fuel input.

After using equation (28) in (32) we may have

$$\eta_2 = \frac{\eta_1}{\in_f}\left[\frac{\frac{\dot{W}_{el}}{\dot{Q}_P} + \in_P}{\frac{\dot{W}_{el}}{\dot{Q}_P} + 1}\right] \qquad (33)$$

Where $\in_P = \frac{\dot{E}_P}{\dot{Q}_P}$ and $\in_F = \frac{\dot{E}_F}{\Delta \dot{H}_f}$

For most fuels, the exergy factor $\in_F$ is close to unity. For process steam, the exergy factor $\in_P$ is always less than unity, but it increases with pressure of process steam produced.

$\in_P$ for our system can be obtained from

$$\dot{E}_P = Cp\left[\left(h_E - h_F\right) - T_0\left(S_E - S_F\right)\right] \qquad (34)$$

$$\dot{Q}_P = Cp\left(h_E - h_F\right) \qquad (35)$$

$h_E = h_g$ and $h_F = h_c$

$$\in_P = 1 - \frac{T_0\left(S_g - S_c\right)}{\left(h_g - h_c\right)} \qquad (36)$$

Compressor-Compression Ratio for Maximum Cycle Power Output

The compressor-compression ratio for maximum cycle power output could be useful and obtained as

$$\frac{\partial \dot{W}_{net}}{\partial \pi_C} = 0 \tag{37}$$

After using equations (9) and (32), and (37) for single stage of reheat we have

$$(\pi_C)_{opt} = \left[\frac{\eta_{GT}\eta_C\theta}{(\beta_{23}\beta_{45}\beta_{EF})^{\alpha/2}}\right]^{2/3\alpha} \tag{38}$$

For two stage of reheat by using equations (9) and (33) it may be given as

$$(\pi_C)_{opt} = \left[\frac{\eta_{GT}\eta_C\theta}{(\beta_{23}\beta_{45}\beta_{EF})^{\alpha/3}}\right]^{3/4\alpha} \tag{39}$$

The quantity $(\beta_{23}\ \beta_{45}\ \beta_{EF})^{\alpha/2}$ in equation (5.38) and $(\beta_{23}\ \beta_{45}\ \beta_{67}\ \beta_{EF})^{\alpha/3}$ in equation (39) may be shown to be close to unity. This implies that $(\pi_C)_{opt}$ in general is independent of pressure drops in cycle but increases with number of reheats.

4. Results and Discussion

My study is limited to the generation of power and saturated steam approximately at (10 bar). To determine the effect of compression ratio on the performance of our system operating under different conditions, the following common specifications were chosen in Table 1.

Effect of number of reheat stages

The performance of the system due to no reheat stage, single reheat stage, and double reheat stages is shown in Tables 2 and Table 3 From these tables it is observed that there is significant improvement in electrical power output, process heat generation, fuel utilization efficiency (first law efficiency) and second law efficiency due to reheat. But the power to heat ratio decreases with reheat because improvement in process heat generation is greater than the improvement in electrical power output. Optimum compressor compression ratio is examined by second law efficiency results in all cases and it was found that optimum compressor compression ratio increases with reheat. There is a wide range for compressor compression ratio that may used to give good second law efficiency in the cases of single reheat and double reheat stages. It was also found that the variation of electrical power output with the compressor compression ratio essentially parallels that of the second law efficiency in all cases.

Effect of cycle pressure drops

Table 4 and Table 5 show the performance of our system with relative pressure drops 4 % and 6 % in each heat transfer device. The fuel utilization efficiency is independent of pressure drops. But, the second-law efficiency and power to heat ratio clearly reflect the fact that greater pressure drop will degrade the thermodynamic performance more than the lower pressure drop. It was also observed that optimum compressor compression ratio is almost independent of pressure losses in the cycle.

Effect of pinch-point on cycle performance

Table 6 shows the effect of pinch-point on the thermodynamic performance of the system when a number of reheat and cycle pressure drops are fixed. From this table, it was observed that the power to heat ratio increases with an increase in the pinch-point that is expected because a larger pinch-point will result in a higher temperature of outlet air. Consequently, less process heat will be produced when larger pinch-point is used. The fuel-utilization efficiency decreases with an increase in the pinch-point. This is consistent with the fact that the fuel utilization efficiency may be given by

$$\eta_1 = \eta_{TH}\left(1+\frac{1}{R}\right)$$

The second-law efficiency decreases with a larger increase in pinch-point. This is consistent with the fact that a larger pinch-point would mean larger exergy destruction (entropy generation) for the system. The rates of decrease for both η_1 and η_2 are essentially a constant. But the rate of decrease for η_2 is smaller than the rate of decrease for η_1 by the factor $\in_p$. Since a larger value for pinch-point would mean a smaller (less extensive) heat recovery steam generator but a less efficient system tradeoffs used on η_1 could lead to a wrong decision as the second-law efficiency does not decrease as much as the first-law efficiency.

5.3 Validation of Results

For the validation of results computed for the cogeneration system, a comparison of these results is done with the data reported by Huang and Egalfopoulus [27] for existing cogeneration plant, which is shown in Table 6 Forgiven operating parameters like number of reheat stages, pressure drop and pinch-point temperature it is shown that the computed fuel utilization efficiency (first law efficiency) decreases with the pressure ratio, but increases significantly up to two reheat stages then slowly thereafter, similar to the observed performance. Hence it gives a suitable comparison of our computed results with the experimental reported data.

5. Conclusions

In this part of the paper is useful expressions have been derived for the study of an indirect-fired air turbine cogeneration system with reheat. Performance data generated from these expressions should be useful to decision-makers in the selection of optimal parameters at the system design stage. Some important conclusions that can be made from this study are as follows:

i) Specific electrical power, specific process heat generation, fuel utilization efficiency, and second-law efficiency are improved due to one and two stages of reheat, but the power to heat ratio decreases with reheat.

ii) Optimum compressor compression ratio for maximum second-law efficiency increases with reheat. For a system with no reheat, this optimum compressor compression ratio is 10 for a system with one stage of reheat it is about 14, and for a system with two-stage of reheats it is about 20.

iii) Maximum second-law efficiency does not vary too much over a fairly wide range of compressor compression ratio for one stage of reheat as well as for two stages of reheat.

iv) As a first approximation, the optimum compression ratio for maximum second-law efficiency may be taken as the optimum compressor compression ratio for maximum cycle power output.

v) Optimum compression ratio for maximum second-law efficiency is essentially independent of pressure losses in the cycle.

vi) The electrical power output, process heat generation and second-law efficiency decreases, with larger pinch-point but fuel-utilization efficiency and power to heat ratio increases with pinch-point.

vii) Optimum pinch-point temperature for maximum second-law efficiency is about 24ºC.

viii) Second-law efficiency and power to heat ratio are better indicators of thermodynamic performance than fuel utilization efficiency.

Nomenclature

T	Temperature	(K)
p	Pressure	(bar)
V	Volume	(m^3)]
t	time	(sec)
$\dot{W}$	work transfer rate (kW)	
E	Exergy (kJ)	
$\dot{Q}$	Heat transfer rate (kW)	
$\dot{m}$	Mass flow rate (kg/sec)	
$\dot{h}$	Methalpy (kJ/K)	
S	Entropy (kJ/K)	
s	specific entropy (kJ/kg-K)	
$\dot{S}$	Entropy rate (kJ/K-sec)	
$\dot{E}$	Exergy rate (kW)	
U	Velocity (m/s)	
Z	Elevation (m)	
g	Acceleration due to gravity (m/s^2)	
h	Enthalpy (kJ/kg)	
m	mass (kg)	
e	specific exergy (kJ/kg)	
P	Power (kW)	
W	Work (kJ/kg)	
HPT	High pressure turbine	
c_p	Specific heat at Constant pressure (kJ/kg-K)	
CC	Combustion Chamber	
ΔH_f	Heat supplied by fuel (kJ)	
Δe_r	Specific thermal exergy of fuel (kJ/kg)	

R	Characteristic gas constant (kJ/kg-K)
ΔH_r	Heat of reaction (kJ)
T_{pp}	pinch point temperature (°C)
T_C	Temperature of Condensat return (°C)
P_P	Process heat pressure (MPa)
Q	temperature ratio (T_{max} / T_{min})
HRSG	Heat recovery Steam generator
GT	Gas turbine
ST	Steam turbine
W_g	Gross Work Output (kW)
C_p	Heat Capacity (kJ/K)
dT_s	Change in temperature in isentropic process
r	pressure ratio
ad	a diabetic
π_t	Turbine Expansion ratio
UA	Exporter Heat Conductance (kW/K)
Tce	Refrigerant Lower Isotherm (K)
X	Defined by equation (6.15)
T_{nc}	Refrigerant higher Isotherm (K)
COP	Coefficient of performance
η_r	refrigeration cycle 1 symbolic efficiency
W_r	Refrigeration power (kW)
b, C	Parameter defined by eq. (6.19) & (6.20)
$\dot{S}ce$	Entropy transfer rate from the evaporator
$\dot{S}hc$	Entropy transfer rate from in the condenser
th	throttling, thermal
x	quality of mixture

Subscript

0	Atmosphere
in	Inlet
out	outlet
gen	generation
w	work
rev	reversible
el	Electrical
Q	Heat
D	Destruction
B	Boiler
I	Turbine
I, II …..	Stages
c	Condenser, condensate
S	Steam
P	Pump, process
H	Heater
TV	throttling value
g	generator, saturated vapor
reg	regenerator
reh	reheat
C	Compressure
a	air
f	fuel, saturated limited
p	product
r	reactant
HPT	High pressure turbine

PT	Power turbine
max	Maximum
CV	Control Volume
S.T.	Steam Turbine
GC	Gas turbine cycle
COMB	Combined Cycle power plant
is	Isentropic
Opt	Optimum
ex	Exergetic
ave	Average

Superscript

λ	$\frac{r-1}{r}$
Su	Superheat
δ_{th}	Throttling

Table 1. Effect of number of reheat stages on fuel utilization efficiency (First law efficiency) η_f, power to heat ratioR_{PH}, and second law efficiency or exergetic efficiency η_2

$\eta_{\boldsymbol{H}}$ = 85%, $\eta_{\boldsymbol{g}}$ = 95%, Pinch point temperature = 22 ^{0}C, Process steam pressure = 10 bar, Maximum temperature = 820 ^{0}C, Condensing temperature = 100 0C, Pressure drops 4% in each heat transfer device

S. No	Without Reheat Stage					With Single Reheat Stage					With Double Reheat Stage				
	π_C	τ	η_f	R_{PH}	η_2	π_C	τ	η_f	R_{PH}	η_2	π_C	τ	η_f	R_{PH}	η_2
1	4	1.436	0.674	0.317	0.340	4	1.360	0.730	0.260	0.352	4	1.330	0.746	0.235	0.3520
2	6	1.480	0.638	0.440	0.342	6	1.401	0.717	0.340	0.3680	6	1.370	0.740	0.3050	0.3703
3	8	1.506	0.607	0.526	0.347	8	1.428	0.707	0.390	0.3050	8	1.400	0.737	0.3560	0.3820
4	10	1.525	0.580	0.598	0.348	10	1.447	0.700	0.427	0.3800	10	1.420	0.733	0.3910	0.3890
5	12	1.540	0.552	0.646	0.333	12	1.462	0.695	0.450	0.3816	12	1.434	0.731	0.4150	0.3940
6	14	1.550	0.527	0.684	0.323	14	1.475	0.688	0.460	0.3819	14	1.447	0.729	0.4308	0.3960
7	16	1.561	0.501	0.713	0.310	16	1.485	0.683	0.478	0.3818	16	1.457	0.726	0.4410	0.3970
8	18	1.570	0.475	0.729	0.295	18	1.492	0.679	0.490	0.3817	18	1.465	0.725	0.4470	0.3980
9	20	1.577	0.448	0.732	0.280	20	1.500	0.675	0.495	0.3806	20	1.472	0.723	0.4570	0.3998
10	22	1.583	0.420	0.733	0.262	22	1.505	0.670	0.496	0.3780	22	1.480	0.722	0.4590	0.3992
11	24	1.590	0.390	0.736	0.243	24	1.512	0.660	0.498	0.3720	24	1.485	0.721	0.4600	0.3990

Table 2. Effect of number of reheat stages **'n'** on specific electrical power output, specific process heat production with relative pressure drop of 4% in each heat transfer device and T_{PP} = 22 ^{0}C

S. No	Without Reheat Stage				With Single Reheat Stage				With Double Reheat Stage			
	π_C	τ	$\dot{W}_{el}$	$\dot{q}_p$	π_C	τ	$\dot{W}_{el}$	$\dot{q}_p$	π_C	τ	$\dot{W}_{el}$	$\dot{q}_p$
1	4	1.436	0.424	1.3330	4	1.360	0.488	1.867	4	1.330	0.494	2.070
2	6	1.480	0.461	1.0510	6	1.401	0.581	1.676	6	1.370	0.608	1.923
3	8	1.506	0.458	0.8670	8	1.428	0.622	1.547	8	1.400	0.667	1.820
4	10	1.525	0.437	0.7350	10	1.447	0.641	1.453	10	1.420	0.704	1.743
5	12	1.540	0.411	0.6334	12	1.462	0.646	1.377	12	1.434	0.714	1.684
6	14	1.550	0.380	0.5540	14	1.475	0.643	1.315	14	1.447	0.730	1.628
7	16	1.561	0.347	0.4860	16	1.485	0.636	1.262	16	1.457	0.734	1.592
8	18	1.570	0.313	0.4300	18	1.492	0.626	1.220	18	1.465	0.729	1.556
9	20	1.577	0.279	0.3810	20	1.500	0.613	1.179	20	1.472	0.728	1.552
10	22	1.583	0.247	0.3370	22	1.505	0.604	1.145	22	1.480	0.727	1.495
11	24	1.590	0.233	0.2960	24	1.512	0.593	1.112	24	1.485	0.717	1.470

Table 3. Effect of pressure drop in heat transfer devices on fuel utilization efficiency, second law efficiency and power to heat ratio for system with single reheat stage and T_{PP} = 22 ^{0}C

S. No	With 6 % pressure drop					With 4 % pressure drop				
	π_C	τ	η_f	R_{PH}	η_2	π_C	τ	η_f	R_{PH}	η_2
1	4	1.356	0.731	0.2330	0.3440	4	1.360	0.730	0.260	0.3520
2	6	1.380	0.719	0.3129	0.3620	6	1.401	0.717	0.340	0.3680
3	8	1.400	0.709	0.3638	0.3770	8	1.428	0.707	0.390	0.3750
4	10	1.412	0.703	0.3970	0.3749	10	1.447	0.700	0.427	0.3800
5	12	1.422	0.695	0.4160	0.3748	12	1.462	0.695	0.450	0.3820
6	14	1.431	0.688	0.4350	0.3750	14	1.475	0.688	0.460	0.3807
7	16	1.438	0.683	0.4450	0.3748	16	1.485	0.683	0.478	0.3817
8	18	1.444	0.677	0.4560	0.3738	18	1.492	0.679	0.490	0.3818
9	20	1.450	0.672	0.4580	0.3714	20	1.500	0.675	0.495	0.3806
10	22	1.455	0.667	0.4650	0.3700	22	1.505	0.670	0.496	0.3780
11	24	1.460	0.662	0.4670	0.3650	24	1.512	0.660	0.498	0.3720

Table 4. Effect of pressure drop in heat transfer devices on specific electrical power output and specific process heat production for single stage reheat system and T_{PP} = 22 ^{0}C

S. No	With 6 % pressure drop				With 4 % pressure drop			
	π_C	τ	$\dot{W}_{el}$	$\dot{q}_p$	π_C	τ	$\dot{W}_{el}$	$\dot{q}_p$
1	4	1.356	0.4430	1.896	4	1.360	0.4880	1.867
2	6	1.380	0.5380	1.721	6	1.401	0.5810	1.676
3	8	1.400	0.5820	1.600	8	1.428	0.6220	1.547
4	10	1.412	0.6000	1.511	10	1.447	0.6413	1.453
5	12	1.422	0.6010	1.443	12	1.462	0.6460	1.377
6	14	1.431	0.6020	1.383	14	1.475	0.6430	1.315
7	16	1.438	0.5950	1.334	16	1.485	0.6360	1.262
8	18	1.444	0.5880	1.289	18	1.492	0.6260	1.220
9	20	1.450	0.5739	1.252	20	1.500	0.6130	1.179
10	22	1.455	0.5660	1.215	22	1.505	0.6040	1.145
11	24	1.460	0.5430	1.188	24	1.512	0.5830	1.112

Table 5. Effect of pinch point temperature on various cycle performance parameters ($\dot{W}_{el}$, $\dot{q}_p$, R_{PH} and η_2) with 4% pressure drop and single stage of reheat system

S. No	π_C	τ	T_{PP}	$\dot{W}_{el}$	$\dot{q}_p$	η_f	R_{PH}	η_2
1	4	1.350	20	0.4870	1.8790	0.7335	0.2590	0.3530
2	6	1.380	21	0.5680	1.7050	0.7210	0.3330	0.3680
3	8	1.403	22	0.6170	1.5770	0.7090	0.3910	0.3760
4	10	1.420	23	0.6350	1.4830	0.6990	0.4280	0.3790
5	12	1.434	24	0.6428	1.4089	0.6920	0.4560	0.3820
6	14	1.447	25	0.6310	1.3530	0.6850	0.4670	0.3505
7	16	1.458	26	0.6280	1.2950	0.6760	0.4850	0.3790
8	18	1.468	27	0.6210	1.2470	0.6700	0.4980	0.3780
9	20	1.447	28	0.6080	1.2060	0.6630	0.5040	0.3750
10	22	1.486	29	0.5940	1.1680	0.6570	0.5080	0.3730
11	24	1.495	30	0.5770	1.1340	0.6490	0.5088	0.3680

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