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Thermodynamic Performance Evaluation of Gas Turbine Based on Tri-generation System

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***Abstract* - Thermodynamic analysis can be perfect tool for identifying the ways for improving the efficiency of fuel use, and determining the best configuration and equipment size for a Tri-generation plant. In this paper thermodynamic performance evaluation of gas turbine based on tri-generation system has been carried out. The operating parameter at inlet and outlet of each components involved in tri-generation system are determined. By using engineering equation solver (EES) parameters like enthalpy, entropy, exergy; etc are determined. The system performance parameters (first law, second law efficiency and exergy) are estimated with the help of these parameters. Parametric study has been done to investigate the effects of overall pressure ratio, turbine inlet temperature, and inlet air temperature on exergy destruction, first law efficiency, electrical to thermal energy ratio and second law efficiency of the components and overall system. The thermodynamic analysis shows the exergy destruction in combustion chamber and HRSG is significantly affected by the pressure ratio and TIT.**

***Keywords* - Gas Turbine, Tri-Generation, Vapour Absorption, Thermodynamic Analysis.**

I. Introduction

Tri-generation is the simultaneous production of power/electricity, hot water and/or steam, and chilled water from one fuel. Basically, a tri-generation power plant is a cogeneration power plant that has added vapour absorption system for producing chilled water from the heat that would have been wasted from a cogeneration power plant. The tri-generation system has efficiency higher than both simple and cogeneration system. Since during the operation of conventional power plant, large quantity of heat is rejected in atmosphere through steam condensers, cooling towers; etc. or with the exhaust gases hence most of this heat can be recovered and used to cover thermal needs thus increasing the efficiency from 30% - 40% of power plant to 80% - 90% of a cogeneration and tri-generation system.

One of the most interesting aspects of gas turbine is that waste heat of flue gases can be used in generating steam at two different pressures. Higher pressure steam can be used for injection in combustion chamber and low pressure steam can be used for running the vapour absorption refrigeration system. These attractive features can improve the power generation capacity and efficiency to a considerable extent.

A schematic diagram of Tri-generation system is shown in figure 1. It is based on Brayton cycle and vapour absorption refrigeration cycle comprising of an axial flow compressor, combustion chamber, turbine, counter current heat exchanger called heat recovery steam generator and vapour absorption refrigeration system.

Due to combustion of fuel the temperature of air further increases. Now the mixture of air and fuel named gases enters the turbine where the gases are expanded and produce the work output. The heat carried by the exhaust gases is recovered in the HRSG to generate the steam. Now the exhaust gases from HRSG enter the generator of vapour absorption refrigeration system where it gives the heat quantity to obtain effect.

The maximum energy that can be recovered by the HRSG is limited by the effectiveness of the HRSG and the outlet temperature of the flue gas. The effectiveness of the HRSG is set to be 0.8, and the exhaust from the stack is set to be $130^{O}C$ to prevent the possibility of vapour condensation. Because formation of vapour condensation leads to formation of sulphuric acid that causes the acidic corrosion. A typical single effect vapour absorption refrigeration system is used to cool the inlet air of the power generation system. The total heat required by the vapour absorption refrigeration system to cool the compressor inlet air is only a small fraction of the available energy.

Assumptions made for mathematical formulation

The mathematical formulation of the present analysis is based on the following assumptions:

a) The cogeneration system operates at steady state.
b) Ideal-gas mixture principles apply for the air and the combustion products.
c) The fuel (natural gas) is taken as methane modelled as an ideal gas. The fuel is provided to the combustion chamber at the required pressure by throttling from high-pressure source.
d) The combustion in the combustion chamber is complete and N_2 is assumed as inert gas.
e) Heat loss from the combustion chamber is 2% of the fuel lower heating value. All other components operate without heat loss.
f) Vapour absorption refrigeration system handles the binary mixture of water and lithium bromide where the water as refrigerant and lithium bromide as absorbent is used.
g) The pressure drop in vapour absorption refrigeration system is neglected.
h) The HRSG unit is a single pressure counter current heat exchanger with fixed effectiveness.

II. Thermodynamic Analysis for Tri-Generation System

The thermodynamic analysis of the proposed system has been carried out using equation of mass and energy balance. For this purpose, computer programme (EES) has been used. A set of governing equations for a particular component (k) is expressed as: Mass rate balance,

$$\sum_k \dot{m}_{i,k} = \sum_e \dot{m}_{e,k} \quad (1)$$

Energy rate balance,

$$\dot{Q}_{cv,k} - \dot{W}_{cv,k} = \sum_e \dot{m}_{e,k} h_{e,k} + \sum_i \dot{m}_{i,k} h_{i,k} \quad (2)$$

Exergy rate balance,

$$\dot{E}_{D,k} = \sum \dot{E}_{q,k} - \dot{W}_{cv,k} + \sum_i \dot{E}_{i,k} - \sum_e \dot{E}_{e,k} \quad (3)$$

Where $\dot{E}_D$ denotes the rate of exergy destruction and $\dot{E}_q$ denotes the associated exergy transfer rate due to heat transfer.

If the effect of kinetic and potential energy is ignored, the total exergy rate $\dot{E}_k$ consisting of physical and chemical exergy can be expressed as

$$\dot{E}_k = \dot{E}_k^{PH} + \dot{E}_k^{CH} \quad (4)$$

The chemical exergy of gases mixture is obtained by summing the overall compositions of air that includes N_2, O_2, CO_2, H_2O (g) and other gases.

III. Mathematical Modelling of Components

Compressor: Air enters the compressor at ambient conditions. This initial temperature and mass of air dictate the amount of work required for compression, the fuel that can be burnt, the fuel required to achieve a specified turbine inlet temperature. As a result, the net power output, the efficiency, the exhaust gas flow rate and temperature at the turbine exit (consequently the recoverable heat) are functions of the ambient conditions. The energy absorbed by the compressor in the form of work is given by

$$\dot{W}_C = \dot{n}_a(\bar{h}_1 - \bar{h}_2) = \dot{m}_a \frac{\bar{h}_1 - \bar{h}_2}{M_a} \quad (5)$$

Where $\dot{n}_a$ is molar flow rate of air, $\dot{m}_a$ is mass flow rate of air, M_a is molecular weight of air and h_2-h_1 is the difference of enthalpy between states $_1$ and $_2$. Across a compressor stage the temperature rise is:

$$\frac{T_2}{T_1} = \frac{\left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}}}{\gamma \times \eta} \quad (6)$$

Where γ is adiabatic index, η is isentropic efficiency of compressor.

The above equation holds true if the physical properties of working fluids is constant. But in real they vary with temperature and pressure.

To account for this we assume that the total pressure rise is occurring through a large number of stages, i.e. 17 for the axial flow compressor. The pressure and temperature rise across a stage is very small so the properties of working fluid assumed constant.

Each step properties of working fluid is calculated corresponding to pressure and temperature. In this way, the above equations evaluated across all stages and the summation of work across all stages gives the total compressor work. Final stage output pressure and temperature are used as the input to combustion chamber.

$$r_C = (T_2/T_1)^{(\gamma/\gamma-1)} \quad (7)$$

$$\eta_C = (\bar{h}_{2s} - \bar{h}_1)/(\bar{h}_2 - \bar{h}_1) \quad (8)$$

r_C is the pressure ratio of compressor, η_C is the isentropic efficiency of compressor and $\bar{h}_{2s}$is the isentropic enthalpy at outlet of compressor. Since S is a function of temperature, T_{2s} is determined in the following way instead of equation (7).

Change in entropy during isentropic compression in compressor

$$\begin{aligned}\bar{S}_{2S} - \bar{S}_1 = {} & X_{1,N_2}\left[\bar{S}_{2S}(T_{2S}) - \bar{S}(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{N_2} \\ & + X_{1,O_2}\left[\bar{S}(T_{2S}) - \bar{S}(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{O_2} \\ & + X_{1,CO_2}\left[\bar{S}(T_{2S}) - \bar{S}(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{CO_2} \\ & + X_{1,H_2O}\left[\bar{S}(T_{2S}) - \bar{S}(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{H_2O} = 0\end{aligned} \quad (9)$$

Where, $X_{1,N2}$ = mole fraction of N_2 in air, $X_{1,O2}$ =mole fraction of O_2 in air, $X_{1,CO2}$ = mole fraction of CO_2 in air, $X_{1,H2O}$ = mole fraction of H_2O in air.

Using the inbuilt specific entropy expressions for N_2, O_2, CO_2 and H_2O from software, the above expression is solved for T_{2s} and corresponding enthalpy of h_{2s} is determined.

Physical Exergy at state 1

At this state $T_1 = T_0$ and $p_1 = p_0$. Accordingly, $h_0 = h_1$ and $s_1 = s_0$ and the physical exergy component vanishes:

$$\dot{E}_1^{PH} = 0 \quad (10)$$

Chemical Exergy at state 1,

$$\dot{E}_1^{CH} = 0 \quad (11)$$

Total Exergy at state 1,

$$\dot{E}_1 = \dot{E}_1^{PH} + \dot{E}_1^{CH} = 0 \quad (12)$$

Physical Exergy at state 2,

$$\dot{E}_2^{PH} = \dot{m}_{a2} \frac{\bar{h}_2 - \bar{h}_0 - T_0(\bar{S}_2 - \bar{S}_0)}{M_a} \quad (13)$$

Chemical Exergy at state 2,

$$\dot{E}_2^{CH} = 0 \quad (14)$$

Total Exergy at state 2,

$$\dot{E}_2 = \dot{E}_2^{PH} + \dot{E}_2^{CH} \quad (15)$$

Exergy Destruction in Compressor-

$$\dot{E}_{D,C} = \sum_i \left(1 - \cancel{\frac{T_0}{T_1}}\right)\dot{Q}_1 - \dot{W}_C + \dot{E}_1 - \dot{E}_2$$

$$\dot{E}_{D,C} = \dot{E}_1 - \dot{E}_2 - \dot{W}_C \quad (16)$$

Combustion Chamber: Denoting the fuel-air ratio on a molar basis as a, the molar flow rates of the fuel, air and combustion products are related by

$$\frac{\dot{n}_f}{\dot{n}_a} = \lambda \quad (17)$$

and
$$\frac{\dot{n}_p}{\dot{n}_a} = 1+\lambda \quad (18)$$

where the subscripts f, p and a denote fuel, combustion product and air respectively; and λ is the fuel-air ratio.

For complete combustion of natural gas (methane), chemical equation takes the following form

$$\lambda CH_4 + [X_{1,N2}N_2 + X_{1,O2}O_2 + X_{1,CO2}CO_2 + X_{1,H2O}H_2O]$$
$$\rightarrow [1+\lambda]\quad [X_{2,N2}N_2 + X_{2,O2}O_2 + X_{2,CO2}CO_2 + X_{2,H2O}H_2O] \quad (19)$$

Balancing carbon, hydrogen, oxygen, and nitrogen the mole fractions of the components of the combustion products are: Mole fraction of N_2

$$X_{2,N2} = \frac{X_{1,N2}}{1+\lambda} \quad (20)$$

Mole fraction of O_2

$$X_{2,O2} = \frac{X_{1,O2} - 2\lambda}{1+\lambda} \quad (21)$$

Mole fraction of CO_2

$$X_{2,CO2} = \frac{X_{1,CO2} + \lambda}{1+\lambda} \quad (22)$$

Mole fraction of H_2O

$$X_{2,H2O} = \frac{X_{1,H2O} + 2\lambda}{1+\lambda} \quad (23)$$

The molar analysis of the combustion products is fixed once the fuel-air ratio λ has been determined. The fuel-air ratio can be obtained from an energy rate balance as follows.

$$0 = \dot{Q}_{CV} - \dot{W}_C + \dot{n}_f \bar{h}_f + \dot{n}_a \bar{h}_a - \dot{n}_p \bar{h}_p$$
$$0 = \dot{Q}_{CV} + \dot{n}_f \bar{h}_f + \dot{n}_a \bar{h}_a - \dot{n}_p \bar{h}_p$$

(24)By assumption 1.1(e) of the model, the heat transfer rate is

$$\dot{Q}_{CV} = -l_f n_f \eta_{cc} \overline{LHV} \quad (25)$$
$$= \dot{n}_a\left(-l_f \lambda \eta_{cc} \overline{LHV}\right) \quad (26)$$

Where l_f =0.02, LHV=Lower Heating Value, η_{cc}= Efficiency of combustion chamber

Combining equation (23) to (26)

$$0 = -l_f \lambda \eta_{cc} \overline{LHV} + \bar{h}_a + \lambda \bar{h}_f - (1+\lambda)\bar{h}_p \quad (27)$$

Using ideal gas mixture principles, the enthalpies of air and combustion products are

$$\bar{h}_a = [X_{1,N2}N_2 + X_{1,O2}O_2 + X_{1,CO2}CO_2 + X_{1,H2O}H_2O]_{T2} \quad (28)$$

$$(1+\lambda)\bar{h}_p = \begin{bmatrix} X_{1,N2}h_{N_2} + (X_{1,O2} - 2\lambda)\bar{h}_{O_2} + (X_{1,CO2} + \lambda)\bar{h}_{CO_2} \\ + (X_{1,H2O} + 2\lambda)\bar{h}_{H_2O} \end{bmatrix}_{T3} \quad (29)$$

Combining the last three equations and solving for λ yields

$$\lambda = \frac{[X_{1,N2}\Delta\bar{h}_{N_2} + X_{1,O2}\Delta\bar{h}_{O_2} + X_{1,CO2}\Delta\bar{h}_{CO_2} + X_{1,H2O}\Delta\bar{h}_{H_2O}]}{[\bar{h}_p - 0.02\,\eta_{CC}\overline{LHV} + (2\bar{h}_{O_2} + \bar{h}_{CO_2} + 2\bar{h}_{H_2O})]_{T3}} \quad (30)$$

Mass flow rate of air is given by

$$\dot{m}_a = \frac{M_a \dot{W}_C}{(1+\lambda)(\bar{h}_3 - \bar{h}_4) + (\bar{h}_1 - \bar{h}_2)} \quad (31)$$

and Mass flow rate of fuel:

$$\dot{m}_f = \lambda \frac{M_f}{M_a} \dot{m}_a \quad (32)$$

Where, M_f = Molecular weight of fuel (Methane), M_a = Molecular weight of air

Fuel (methane) is injected in combustion chamber at pressure P_f and temperature T_1. Physical Exergy of fuel (methane) at state f,

$$\dot{E}_f^{PH} = \dot{m}_f[\bar{h}_f - \bar{h}_0 - T_0(\bar{S}_f - \bar{S}_0)] \quad (33)$$

Since T_o= T_1, using ideal gas equation, the above equation reduces to

$$\dot{E}_f^{PH} = \dot{m}_f RT_1 \ln\frac{p_f}{p_1} \quad (34)$$

Chemical Exergy of fuel (methane) at state f-

$$\dot{E}_f^{CH} = \dot{m}_f \times \frac{e_{methane}^{CH_4}}{M_f} \times 10^{-3} \quad (35)$$

where $e_{methane}^{CH_4}$=424348 kJ/kmol

Total Exergy at state f,

$$\dot{E}_f = \dot{E}_f^{PH} + \dot{E}_f^{CH} \quad (36)$$

Physical Exergy at state 3,

$$\dot{E}_3^{PH} = \dot{m}_3 \frac{\bar{h}_3 - \bar{h}_0 - T_0(\bar{S}_3 - \bar{S}_0)}{M_p} \quad (37)$$

At state 3, there are combustion products (flue gases) after combustion. The restricted dead state corresponding to the mixture at state 3 consists of liquid water phase and a gas phase with mole fractions x'_{N2}, x'_{O2}, x'_{CO2}, $x'_{H2O(g)}$. The contribution of the liquid water to the chemical exergy is determined for state 3. The contribution of the gas phase to the chemical exergy is evaluated as

$$\sum(x'_k \times e_k^{CH}) + \bar{R}\,To \sum(x'_k \times \ln x'_k) \quad (38)$$

On the basis of 1 kmol of mixture at state 3, we have (x_{1N2} + x_{1O2} +x_{1CO2}+x_{1H2O}) kmol as a gas phase and 1-(x_{1N2}+x_{1O2}+x_{1CO2}+x_{1H2O}) kmol as liquid water where x_{1N2}, x_{1O2}, x_{1CO2} and x_{1H2O} is initial mole fraction of air, thus

Chemical exergy at state 3,

$$\dot{E}_4^{CH} = (\dot{m}_3 / M_p) \times [(x_{1N2} + x_{1H2O} \times \sum(x'_k \times e_k^{CH}) + \bar{R}\,To \sum(x'_k \ln x'_k) \times (1 - (x_{1N2} + x_{1O2} + x_{1CO2} + x_{1H2O})) \times e_{ow}]\,10^{-3}) \quad (39)$$

Exergy Destruction in Combustion Chamber-

$$\dot{E}_{D,CC} = \sum_i \left(1 - \frac{T_0}{T_1}\right)\dot{Q}_1 - \dot{W}_C + \dot{E}_2 + \dot{E}_f - \dot{E}_3$$
$$\dot{E}_{D,CC} = \dot{E}_2 + \dot{E}_f - \dot{E}_3 \quad (40)$$

Turbine: The purpose of using a turbine is to generate power from expansion of a gas. After combustion the molar composition of gas changes to

$$[X_{2,N2}N_2 + X_{2,O2}O_2 + X_{2,CO2}CO_2 + X_{2,H2O}H_2O]_{T4} \quad (41)$$

The value of specific enthalpy h_4 can be evaluated using the turbine isentropic efficiency. Solving the expression for expansion for turbine isentropic efficiency is as,

$$\eta_{st} = \frac{\bar{h}_3 - \bar{h}_4}{\bar{h}_3 - \bar{h}_{4s}} \quad (42)$$

We get

$$\bar{h}_4 = \bar{h}_3 - \eta_{st}\left(\bar{h}_3 - \bar{h}_{4S}\right) \quad (43)$$

where η_{st} is isentropic efficiency of turbine, h3 and h_4 are enthalpies of flue gas at states 3, 4 and h_{4s} is the isentropic enthalpy at turbine exit.

The change in entropy of combustion gas after isentropic expansion in turbine

$$\bar{S}_{4S} - \bar{S}_3 = X_{2,N_2}\left[\bar{S}(T_{4S}) - \bar{S}(T_3) - \bar{R}\ln\frac{p_4}{p_3}\right]_{N_2} + X_{2,O_2}\left[\bar{S}(T_{4S}) - \bar{S}(T_4) - \bar{R}\ln\frac{p_4}{p_3}\right]_{O_2} + X_{2,CO_2}\left[\bar{S}(T_{4S}) - \bar{S}(T_4) - \bar{R}\ln\frac{p_4}{p_3}\right]_{CO_2} + X_{2,H_2O}\left[\bar{S}(T_{4S}) - \bar{S}(T_4) - \bar{R}\ln\frac{p_4}{p_3}\right]_{H_2O} = 0 \quad (44)$$

Physical Exergy at state 4-

$$\dot{E}_4^{PH} = \dot{m}_3 \frac{\bar{h}_3 - \bar{h}_0 - T_0(\bar{S}_4 - \bar{S}_0)}{M_p} \quad (45)$$

Exergy Destruction in Turbine-

$$\dot{E}_{D,turbine} = \sum_i \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_i + \dot{W}_T + \dot{E}_3 - \dot{E}_4 \quad (46)$$

Heat recovery steam generator: The waste heat is recovered in the heat recovery steam generator. In this process water enters the economiser in the form of compressed liquid at feed water temperature T_{fw}. As the water receives heat from the hot exhaust gases, it becomes saturated, starts boiling, and is superheated. On the hot side, the exhaust gases leaving the turbine enter the steam generator and get cooled finally to the stack temperature. For maximum heat recovery, the stack temperature should approach the acid dew point of exhaust gases, while keeping the pressure drops within desirable limits.

The factors which affect the cost and effectiveness of any HRSG are pinch point, approach point, allowable back pressure, stack temperature, steam temperature and pressure. The minimum temperature difference for heat transfer, which is known as pinch point plays an important role in identifying the optimum heat recovery and size of heat exchangers. Approach point is the difference between the saturation temperature and the temperature of water leaving the economizer; lowering approach point will increase probability of steaming in the economizer which may cause hammering and blanketing.

The gas side pinch point temperature (T_P) and economizer exit temperature are calculated, by assuming the drum saturation pressure (P_{Drum})

$$T_P = T_{Drum} + PP$$

$$T_{ECO} = T_{Drum} - AP$$

The steam generated (m_{st}) for each kg/s of exhaust gases can be determined by applying the mass and energy conservation principles across superheater and evaporator as-

$$\dot{m}_{st} \times (\bar{h}_{sup} - \bar{h}_{fw}) = \dot{m}_p \times (\bar{h}_4 - \bar{h}_5) \quad (47)$$

where $\bar{h}_{sup}$ is the enthalpy of superheated steam generated in HRSG, $\bar{h}_{fw}$ is enthalpy of feed water at inlet to HRSG, m_p is mass flow rate of gases, $\bar{h}_4$ and $\bar{h}_P$ is enthalpy of flue gases at inlet and pinch point of HRSG respectively

Physical Exergy at state 5,

$$\dot{E}_5^{PH} = \dot{m}_5[\bar{h}_5 - \bar{h}_0 - T_0(\bar{S}_5 - \bar{S}_0)] \quad (48)$$

Chemical Exergy at state 5,

$$\dot{E}_5^{CH} = \dot{m}_w \times \frac{e_{water}}{M_w} \times 10^{-3} \quad (49)$$

where e_{water} = 45 kJ/kmol (sp. Entropy of water)

Physical Exergy at state 6-

$$\dot{E}_6^{PH} = \dot{m}_6[\bar{h}_6 - \bar{h}_0 - T_0(\bar{S}_6 - \bar{S}_0)] \quad (50)$$

Exergy Destruction in HRSG

$$\dot{E}_{D,HRSG} = \dot{E}_6 - \dot{E}_7 + \dot{E}_5 - \dot{E}_6 \quad (51)$$

Generator: The exhaust gases from HRSG enters the generator of Vapour Absorption Refrigeration System where exhaust gases gives heat to the solution in generator the heat transfer equation are as follows

$$\dot{Q}_g = \dot{m}_p(\bar{h}_5 - \bar{h}_8) \quad (52)$$

$$f_c = [\{1 - \left(1 - \frac{X_9}{100}\right)\} / \{\left(1 - \frac{X_{12}}{100}\right) - \left(1 - \frac{X_9}{100}\right)\}] \quad (53)$$

Here, Qg is the heat given by the exhaust gases to the generator, f_c is the circulation rate.

The exergy destruction in the generator is given by following equation

$$\dot{E}_{D,g} = \dot{E}_5 + \dot{E}_{14} - \dot{E}_9 - \dot{E}_8 - \dot{E}_{15} \quad (54)$$

Absorber: From absorber the strong solution enters the generator through pump and solution heat exchanger and weak solution from generator enters to absorber also the refrigerant in vapour state enters to absorber so mixing process takes place in absorber. The heat transfer equation for absorber given below

$$\dot{Q}_{abs} = \dot{m}_r(\bar{h}_{18} - \bar{h}_{11}) + f_C(\bar{h}_{11} - \bar{h}_{12}) \quad (55)$$

where f is the circulation rate and m_r is the mass of refrigerant. The exergy destruction in absorber is given by the following equation

$$\dot{E}_{D,Abs} = \dot{E}_{11} + \dot{E}_{18} + \dot{E}_{22} - \dot{E}_{12} - \dot{E}_{23} \quad (56)$$

Solution Heat Exchanger: In solution heat exchanger the weak solution in hot state from generator enters it and heat up the strong solution which enters in cold state the heat exchange between two solutions is given by the following equation

$$\dot{Q}_{SHX} = \dot{m}_1(\bar{h}_{14} - \bar{h}_{13}) \quad (57)$$

where $\dot{m}_1$ is mass flow rate of strong solution. The exergy destruction in solution heat exchanger is given by the following equation

$$\dot{E}_{D,SHX} = = \dot{E}_9 + \dot{E}_{13} - \dot{E}_{14} - \dot{E}_{10} \quad (58)$$

Solution Pump: The function of the solution pump is that it transfers the solution from absorber (low pressure region) to generator (high pressure region).The work consumed by the pump is given by following equation

$$\dot{W}_{Pump} = V \times (P_g - P_a) \times \dot{m}_1/\eta_{pump} \quad (59)$$

$$= \dot{m}_1 \times (\bar{h}_{12} - \bar{h}_{11}) \quad (60)$$

Where, $\dot{m}_1 = f_C \times \dot{m}_r$

η_{pump} is the efficiency of pump, P_g &P_a be the pressure of generator and absorber, m_1 is the mass of strong solution enters the pump and V be the volume of strong solution entering the pump.

Condenser: The refrigerant in steam state enters to the condenser where it condenses and exchanging heat with water which passes through the condenser the heat transfer equation for condenser given below

$$\dot{Q}_{cond} = \dot{m}_r \times (\bar{h}_{15} - \bar{h}_{16}) \quad (61)$$

The exergy destruction in condenser is given by the following equation

$$\dot{E}_{D,Cond} = \dot{E}_{15} - \dot{E}_{16} + \dot{E}_{24} - \dot{E}_{25} \quad (62)$$

Evaporator: The condensed refrigerant from condenser after throttling enters to the evaporator where it evaporates and also the air from ambient is passed through the evaporator where it cools and then it is send to compressor of gas turbine unit, the heat transfer equation for evaporator is given below

$$\dot{Q}_{evap} = \dot{m}_r \times (\bar{h}_{17} - \bar{h}_{16}) \quad (63)$$

The exergy destruction in evaporator is given by the following equation

$$\dot{E}_{D,Evap} = \dot{E}_{19} + \dot{E}_{17} - \dot{E}_{18} - \dot{E}_{20} \quad (64)$$

IV. Performance Parameters

The relevant parameters required for the combined first and second law analysis of gas turbine cogeneration system are summarised below:

First – Law Efficiency (η_I): The ratio of all the useful energy extracted from the system (electricity and process heat) to the energy of fuel input is known as first-law efficiency. This reflects the first law of thermodynamics, which is concern with quantity not energy quality. It is also known as energetic or fuel utilization efficiency.

$$\eta_I = (\dot{W}_{el} + \dot{Q}_{process} + \dot{Q}_{evap})/\dot{Q}_{fuel} \quad (65)$$

Where $\dot{W}_{el}$= Electrical work rate, $\dot{Q}_{process}$=Process heat rate, $\dot{Q}_{fuel}$= evaporator Cooling rate

Second –Law Efficiency (η_{II}): Since electrical power is more valuable than process heat according to the second law of thermodynamics unlike energy, the exergy is always destroyed in any real process. It is useful to consider both output and input in terms of exergy. The amount of exergy supplied in the product to the amount of exergy associated with the fuel is a more accurate measure of thermodynamic performance of a system, which is defined as:

$$\eta_{II} = (\dot{W}_{el} + \dot{E}_{process} + \dot{E}_{evap})/\dot{E}_{fuel} \quad (66)$$

Where $\dot{E}_{process}$= process exergy rate, $\dot{E}_{evap}$= exergy rate in evaporator, $\dot{E}_{fuel}$= fuel exergy rate

Electrical to Thermal Energy Ratio (R_{ET}): The cost effectiveness of any tri-generation system is directly related to the amount of power it can produce for a given amount of process heat and cold needed. Thus the electrical to thermal energy ratio (R_{ET}) is an important parameter used to assess the performance of such a system which is defined as

$$R_{ET} = \dot{W}_{el}/(\dot{Q}_{process} + \dot{Q}_{evap}) \quad (67)$$

Exergy Destruction Ratio: It can be defined as the ratio of rate of exergy destruction in a system component to the exergy rate of the fuel provided to the overall system.

$$y_D = \dot{E}_D / \dot{E}_{f,tot} \quad (68)$$

Alternatively, the component exergy destruction rate can be compared to the total exergy destruction rate within the system.

$$y_D^* = \dot{E}_D / \dot{E}_{f,tot} \quad (69)$$

The two exergy destruction ratios are useful for comparisons among various components of the same system. The exergy destruction ratio y_D^* can also be invoked for comparisons among similar components of different systems using the same, or closely similar, fuels.

Exergy Loss Ratio: It is defined as the ratio of rate of exergy loss in a system component to the exergy rate of the fuel provided to the overall system

$$y_L = \dot{E}_L / \dot{E}_{f,tot} \quad (70)$$

Exergetic Efficiency: It is a parameter for evaluating thermodynamic performance. The exergetic efficiency (second-law efficiency, effectiveness, or rational efficiency) provides a true measure of the performance of an energy system from the thermodynamic viewpoint. Considering a system at steady state where, in terms of exergy, the rates at which the fuel is supplied and the product is generated are E_F and E_P, respectively. An exergy rate balance for the system reads

$$\dot{E}_F = \dot{E}_P + \dot{E}_D + \dot{E}_L \quad (71)$$

Where $\dot{E}_D$ and $\dot{E}_L$ denotes the rate of exergy destruction and exergy loss respectively

The exergetic efficiency η_{exgt} is the ratio between product and fuel,

$$\eta_{exgt} = \frac{\dot{E}_P}{\dot{E}_f} = 1 - \frac{\dot{E}_D + \dot{E}_L}{\dot{E}_f} \quad (72)$$

The exergetic efficiency shows the percentage of the fuel exergy provided to a system that is found in the product exergy.

V. Results and Discussion

The parametric studies of tri-generation system and model validation are discussed here with different input values as given in table 1. The formulation of gas turbine tri-generation system has been discussed earlier. In order to analyze the tri-generation power plant system using engineering equation solver (EES) the operating

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parameters and calculated parameters at points (fig. 1) of the tri-generation system are given at table 2 and 3. Figure 2 shows the variation of exergy destruction amongst the tri-generation system components. Here the maximum exergy destruction is found in combustion chamber i.e. 36.97 MW. Similarly figure 3 & 4 show the variation of percentage exergy destruction with respect to both overall plant and fuel exergy for tri-generation system. Also, figure 5 shows the variation of exergy destruction for VAR system Components. Likewise figure 6 & 7 show the variation of percentage exergy destruction with respect to both overall plant and fuel exergy for VAR system. Figure 2 to 7 reveals that maximum exergy destruction rate found is in combustion chamber and generator respectively.

It is vetted from reference data [1] that first and second law efficiency in tri-generation system is more than cogeneration in both part and full load mode , but electrical to thermal energy ratio is less as compared to cogeneration system. The value of first and second law efficiency in tri-generation system and electrical to thermal energy ratio is shown in figure 8.

Parametric studies: The effect of pressure ratio (r_p) across the compressor, turbine inlet temperature (TIT), ambient temperature and inlet air temperature on the first law efficiency (η_I) and power-to-heat ratio is obtained by the energy balance approach or the first law analysis of the cycle. However, the exergy destruction or thermodynamic losses of each component and second law efficiency (η_{II}) of the cycle has also been investigated under the exergy balance approach.

Effect of pressure ratio: As pressure ratio increases the air temperature at the inlet of combustion chamber increases which results in decreasing the heat added to the cycle. The ratio of net work output to heat added represent the first law efficiency, hence as pressure ratio increases the first law efficiency of tri-generation system increases as shown in Figure 9. Figure also shows the variation of second law efficiency which is more accurate measure of thermodynamic performance, since the quality of fuel i.e. exergy associated with heat addition is higher than heating value or energy of fuel, the exergy of the fuel would increase while bringing it from ambient pressure to combustion pressure at ambient temperature. Hence exergy associated with the heat addition will be equal to exergy associated with the heat addition with the heating value of fuel plus exergy increase, therefore the second law efficiency is slightly lower than first law efficiency of tri-generation system. It is shown in figure 10.

Effect of variation of Turbine inlet air temperature: Fig 10 shows the variation of first law efficiency (η_I), second law efficiency (η_{II}) and electrical to thermal ratio (R_{ET}) with respect to change in TIT for ($r_{p=}$ 10, $T_{e\,=}$ 5˚C). It is found that the first law efficiency increases with increase in TIT. This is because increasing TIT leads to significant increase in net work output and insignificant increase in heat addition of cycle. Therefore first law efficiency increases with increase in TIT. It also shows that the variation of second law efficiency is slightly lower than first law efficiency. This difference between second law efficiency and first law efficiency for tri-generation occurs due to exergy associated with cold is much less than the energy of process heat. It is also seen from figure 10 that TIT does not have significant effect on electrical to thermal energy ratio for tri-generation cycle because increase in TIT causes significant increase in power output, process heat and cold and the improvement in process heat and cold.

Effect of variation of turbine evaporator temperature: As shown in figure 11 the first law efficiency of tri-generation system slightly increases with the increase in evaporator temperature. This is because at higher evaporator temperature, the cooling load will be higher. The second law efficiency also increases with evaporator temperature but the magnitude of increase is very small as the exergy associated with the cold is very small. The electrical to thermal ratio decreases with increase in evaporator temperature due to increase in cold.

Effect of ambient temperature: Figure 12 shows the variation of power output with ambient temperature. As air temperature rises, its density falls. Thus, the volumetric flow rate remains constant; the mass flow rate is reduced as air temperature rises. Power output is also reduced as air temperature rises because power output is proportional to mass flow rate. As per ISO condition optimum power output occurs at 15˚C.

Effect of pressure ratio on exergy destruction: It is found the exergy destruction in combustion process dominates it represents over 70% of the total exergy destruction in overall system. As the pressure ratio increases the exergy destruction in the combustion chamber decreases significantly. This is because of the increase in pressure ratio implies lower difference of exergy between combustion products and compressed air but its difference with exergy carried by fuel drops. It is shown in Table 5 that as the pressure ratio increases the exergy destruction in HRSG decreases. This is because of the higher pressure ratio results in higher exergy of combustion products and lower turbine exhaust exergy which leads to the higher turbine output.

Effect of turbine inlet temperature on exergy destruction: Table 6 shows the variation of magnitude of exergy destruction in each component of system with change in turbine inlet temperature. As TIT increases the exergy destruction in the combustion chamber increases, because the mean temperature of heat addition increases. The exergy destruction in HRSG increases because the temperature difference between two heat exchanging fluid increases for the given pressure ratio of the cycle, more process heat is produced due to more steam generated by HRSG at higher TIT.

VI. Conclusions

Based on input parameters the analysis of tri-generation system is done and the following points are concluded.

a) Maximum energy is destroyed during the combustion and steam generation process; it represents over 82% of the total energy destruction in the overall system.

b) The first law efficiency, second law efficiency and electrical to thermal energy of Tri-generation increases with the increase in pressure ratio up to 16, after that it almost remains constant.

c) The first law efficiency, electrical to thermal energy ratio, and the second law efficiency Tri-generation system increases with the increase in turbine inlet temperature.

d) The energy destruction in combustion chamber and heat recovery steam generator decreases significantly with the in pressure ratio but increases significantly with the increase in turbine inlet temperature.

e) The energy destruction in the generator, the absorber, and the condenser increases slightly with the evaporator temperature, while it decreases in the throttling valve, the evaporator and the heat exchanger solution.

f) The first law efficiency and second law efficiency for Tri-generation is found to be almost constant with the variation of evaporator temperature but the electrical to thermal energy ratio for Tri-generation decreases slightly with the increase in evaporator temperature.

g) At a given TIT, process heat pressure, and evaporator temperature, the energy destruction in compressor and turbine increases with the increase in pressure ratio.

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AUTHOR'S PROFILE

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Nomenclature

E	Exergy (MW)	R	Universal gas constant (kJ/kmolK)
E^{CH}	Specific chemical exergy (kJ/kmol)	r_p	Pressure ratio of the compressor
E^{PH}	Specific physical exergy (kJ/kmol)	s	Specific entropy of the fluid (kJ/kgK)
f_c	Circulation rate	T	Temperature (K)
$h^{\cdot}$	Specific enthalpy of the fluid (kJ/mole)	TIT	Turbine Inlet Temperature (K)
M	Molecular weight (kg/kmol)	U	Internal energy (kJ)
m	Mass flow rate (kg/s)	v	Specific volume (m^3/kg)
n	Molar flow rate (kmol/s)	W	Work (kW)
P	Pressure (bar)	x	Molar fraction
PP	Pinch point (˚C)	X	Concentration of LiBr
Q	Heat transfer (kJ)	y	Exergy destruction ratio
Subscripts			
a	Air	p	Product or gases
A	Absorber	P	Pinch point

COMP	Compressor	r	Refrigerant
CC	Combustion Chamber	S	Isentropic
C	Condenser	SHX	Solution Heat Exchanger
fw	Feed water	REG	Regenerator
cv	Control volume	sat	Saturated
D	Destruction	sup	Superheated
ECO	Economizer	st	Steam
E	Evaporator	T	Turbine
f	Fuel	w	Water
G	Generator	CH	Chemical
HRSG	Heat Recovery Steam Generator	KN	Kinetic
k	k^{th} component	PH	Physical
L	loss	PT	Potential
o	reference	T	Total
Greek symbol			
η	Efficiency	λ	Fuel-Air ratio
Acronyms and abbreviations			
VAR	Vapor Absorption Refrigeration System	HHV	Higher Heating Value
COP	Coefficient of Performance	IAC	Inlet Air Cooling
EES	Engineering Equation Solver	LHV	Lower Heating Value
GT	Gas Turbine	AHE	Air Heat Exchanger

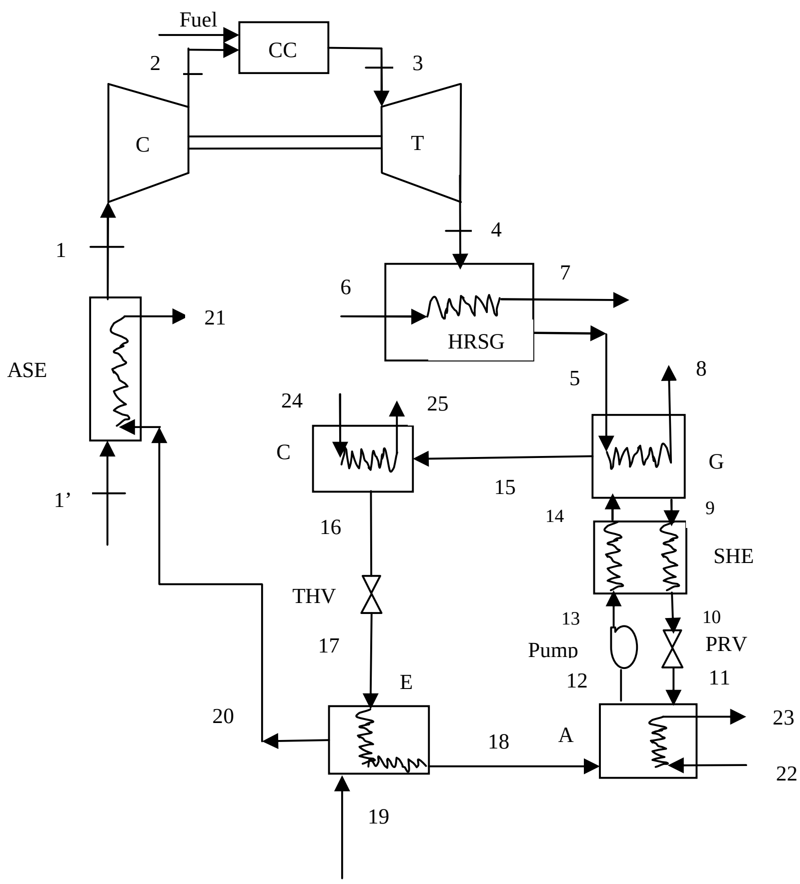

Fig. 1 Schematic of Tri-generation System

State points in Figure 1

1’ Cooling water inlet in AHE
1 Inlet air to Compressor
2 Compressor outlet or combustion chamber inlet
3 Combustion chamber outlet or turbine inlet
4 Exit from Gas Turbine or Entry to HRSG
5 Exit from HRSG or Inlet to Generator of VARS
6 Feed water inlet to HRSG
7 Steam outlet from HRSG
8 Exit from generator
9 Inlet to solution heat exchanger
10 Inlet to Pressure Reduction Valve
11 Exit from Pressure Reduction Valve or Inlet to absorber
12 Exit from Absorber or inlet to solution pump
13 Exit from solution Pump
14 Exit from solution heat exchanger
15 Inlet to condenser
16 Exit from condenser
17 Exit from throttle valve
18 Exit of evaporator
19 Cooling water inlet to evaporator
20 Chilled water outlet from evaporator
21 Exit from AHE
22 Cooling water inlet to absorber
23 Cooling water outlet from absorber
24 Cooling water inlet to condenser
25 Cooling water outlet from condenser
F Fuel input to combustion chamber

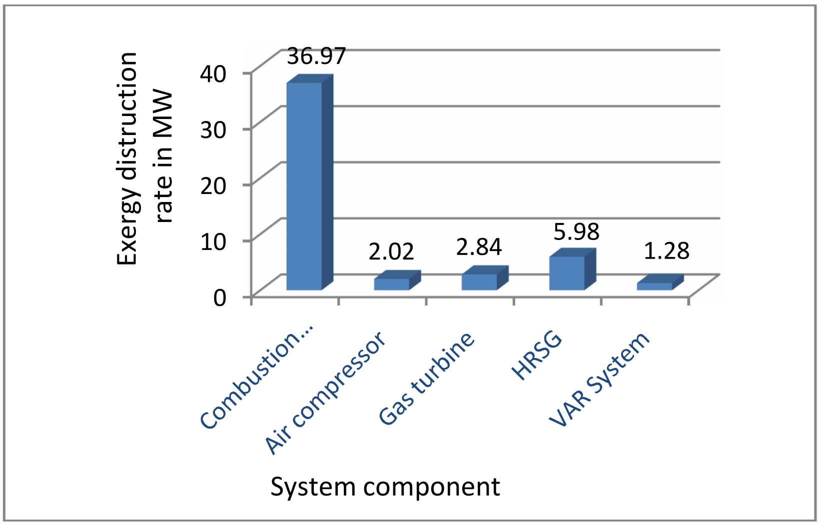

Fig.2. Variation of exergy destruction for tri-generation system Component

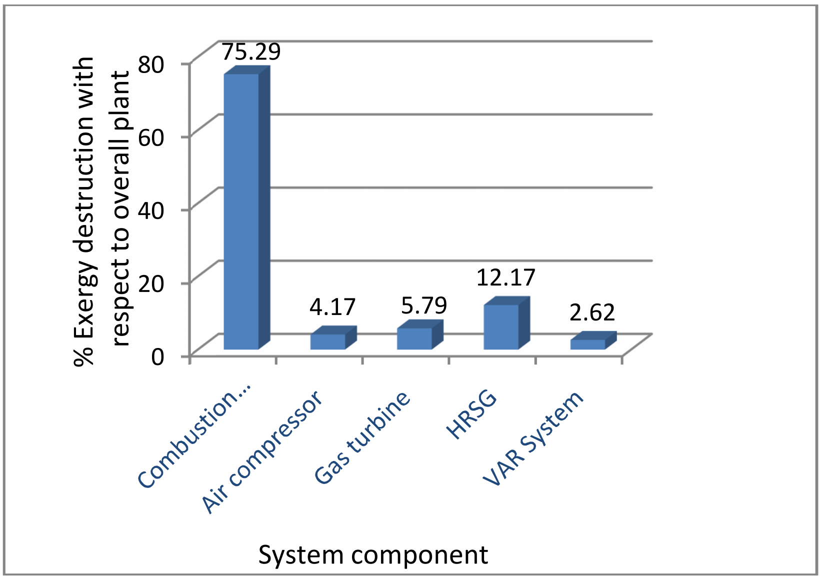

Fig.3. Variation of percentage exergy destruction with respect to overall plant for tri-generation system

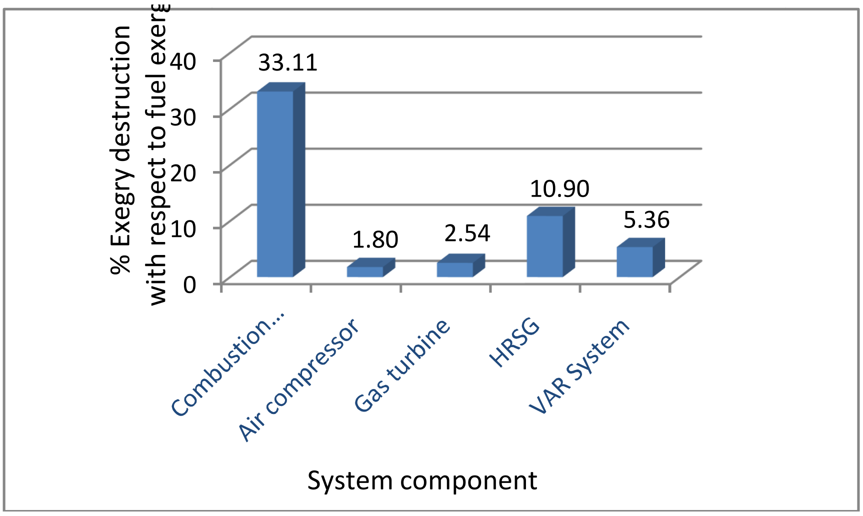

Fig.4. Variation of percentage exergy destruction with respect to Fuel exergy for tri-generation system

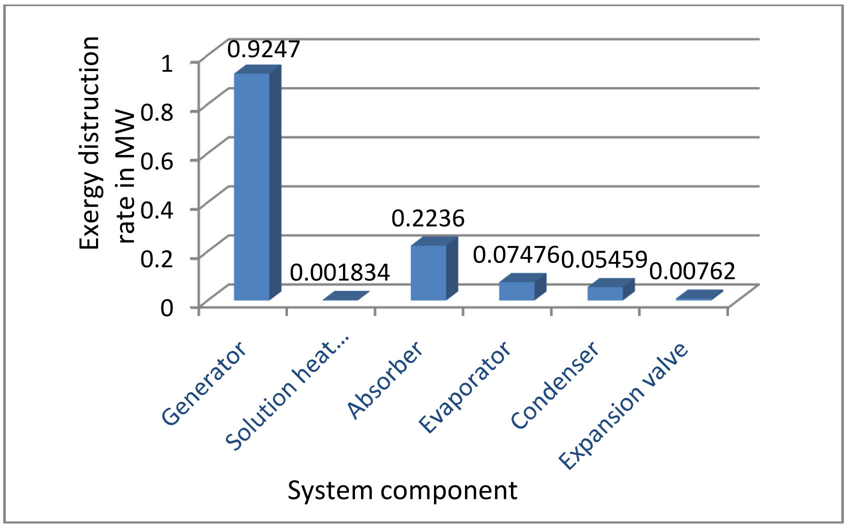

Fig.5. Variation of exergy destruction for VAR system Component

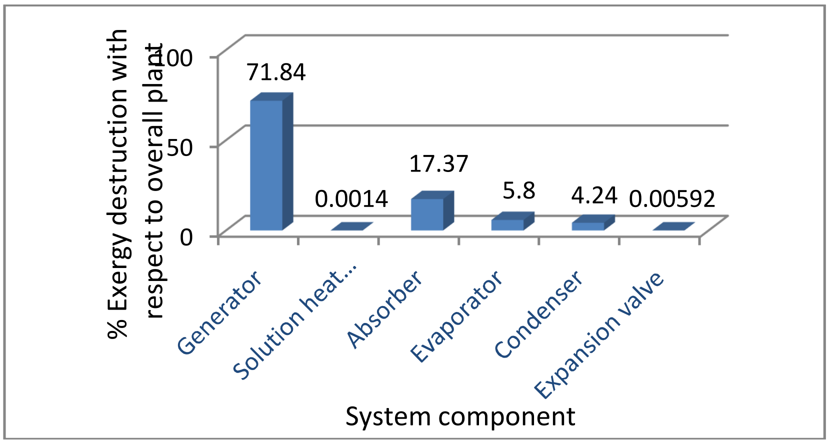

Fig.6. Variation of percentage exergy destruction with respect to overall plant for VAR system

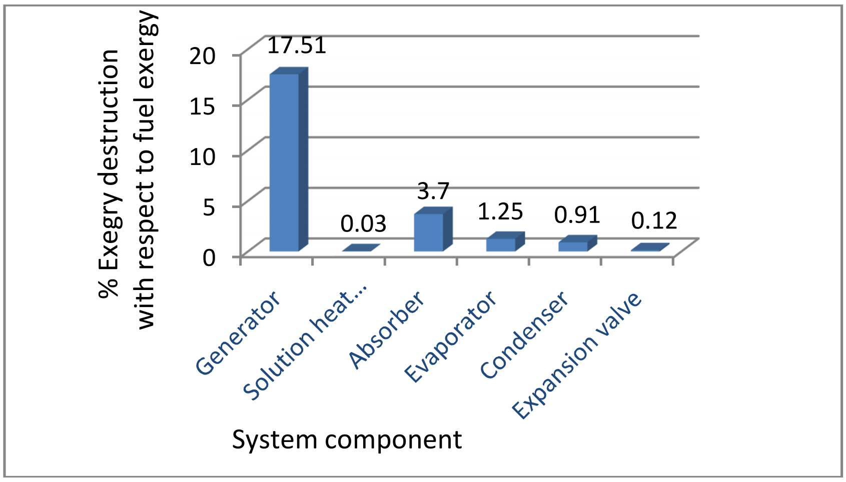

Fig.7. Variation of percentage exergy destruction with respect to Fuel exergy for VAR system

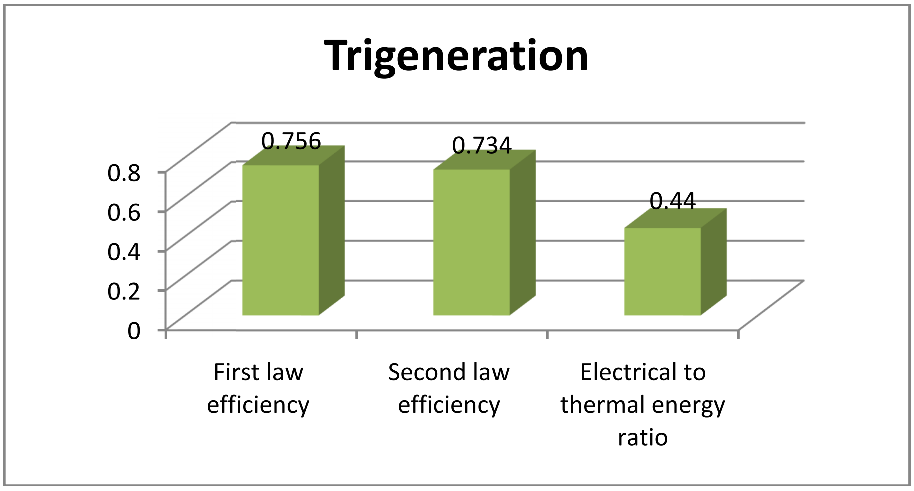

Fig.8. Performance parameter for tri-generation system

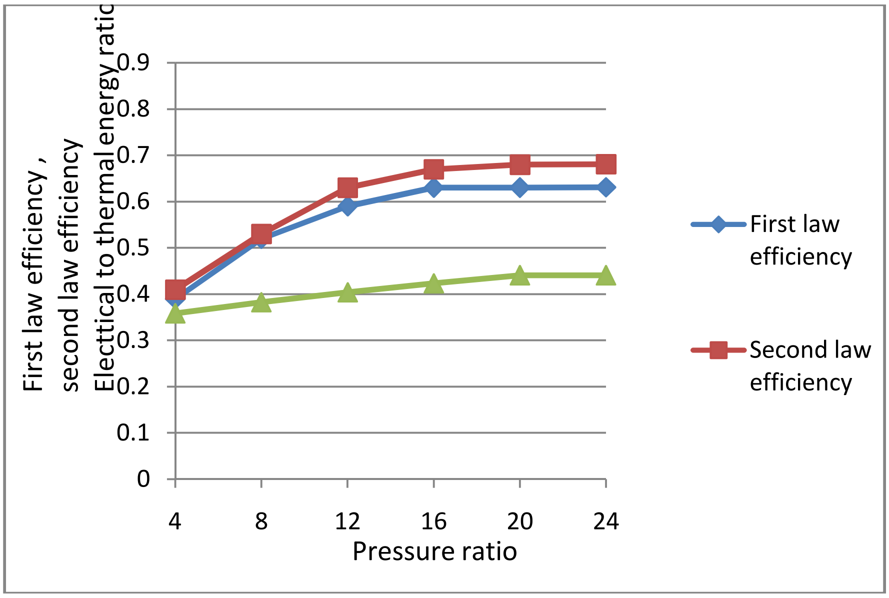

Fig. 9- Effect of variation of pressure ratio on first law efficiency, second law efficiency

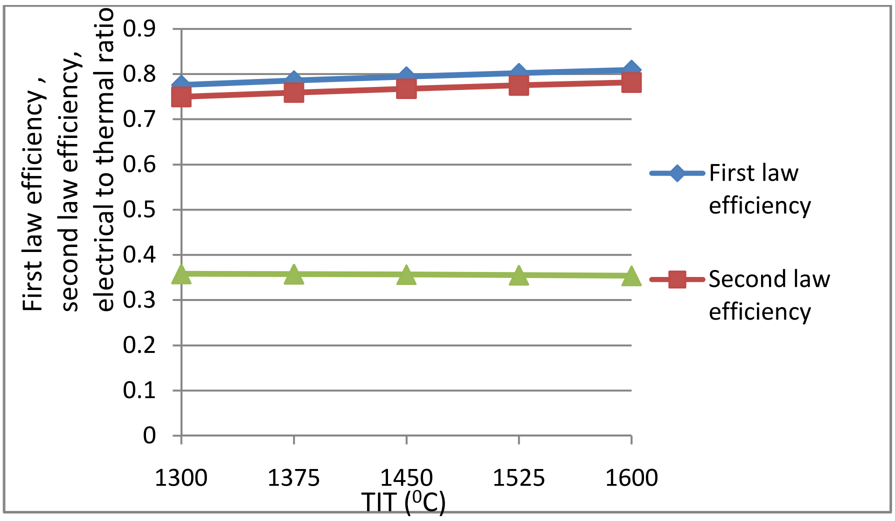

Fig. 10- Effect of variation of TIT on first law efficiency, second law efficiency and electrical to thermal energy ratio

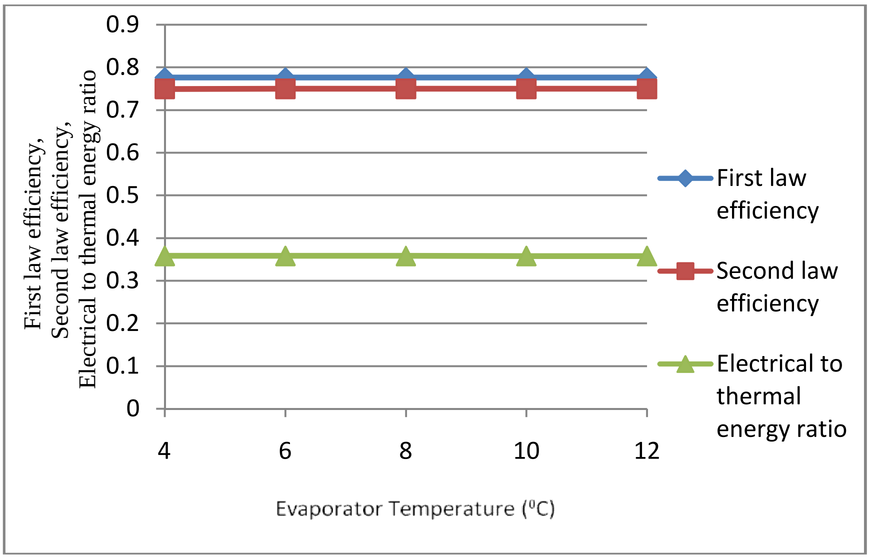

Fig.11. Effect of variation of evaporator temperature on first law efficiency, second law efficiency and electrical to thermal ratio

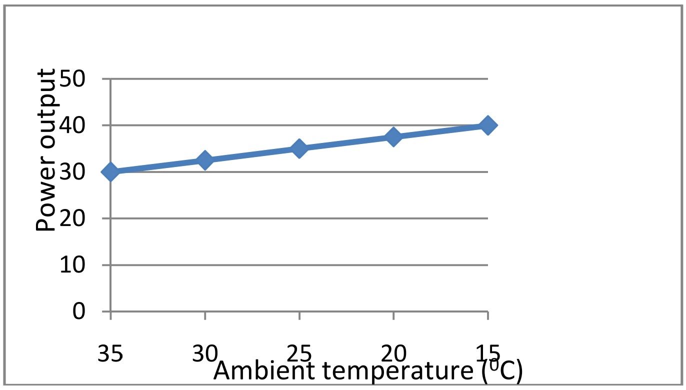

Fig. 12- Effect of variation of ambient temperature on power output

Table 1 : Input data for analysis of tri-generation plant

Air inlet pressure to compressor (P_1), bar	1.013
Exhaust pressure of combustion product after HRSG (P_5), bar	1.14
Pressure of feed water at inlet of HRSG (P_6), bar	90
Pressure of steam generation (P_7), bar	90
Injection pressure of fuel (methane) (P_f), bar	22
Mass flow rate of steam generated in HRSG (m_{st}), kg/s	27.77
Pressure ratio of compressor (r_p)	10 (4-24)
Net power output of the plant (W_{net}), MW	40
Natural gas lower heating value, kJ/kmole	802361
Isentropic efficiency of compressor (η_{SC}), %	85
Isentropic efficiency of Turbine (η_{GT}), %	86
Efficiency of combustion chamber (η_{cc}), %	95
Generator efficiency (η_g), %	98
Inlet air temperature to compressor (T_1), K	283.15
Turbine inlet temperature (TIT), K	1527 (1300-1700)
Temperature of feed water at inlet to HRSG (T_6), K	383
Injection temperature of fuel (methane) (T_f), K	298.15
Pressure drop in HRSG on the gas side, %	5
Pressure drop in combustion chamber, %	5
Pressure loss in generator of VARS, %	5
Generator temperature ˚C	90

Evaporator temperature ˚C	5
Condenser temperature ˚C	35
Absorber temperature ˚C	35
Inlet temperature of cooling media at absorber ˚C	15
Outlet temperature of cooling media at absorber ˚C	35
Inlet temperature of cooling media at condenser ˚C	25
Outlet temperature of cooling media at condenser ˚C	35
Inlet temperature of cooling media at evaporator ˚C	15
Outlet temperature of cooling media a evaporator ˚C	10
Effectiveness of solution heat exchanger, %	85
Effectiveness of condenser, %	85
Effectiveness of evaporator, %	85
Effectiveness of generator, %	80

Table 2 : Operating parameters at state points of the tri-generation system

State	Substance	Mass flow rate (kg/s)	Temperature (K)	Pressure (bar)
1	Air	85.65	298.15	1.013
2	Air	85.65	611.34	10.5
3	Combustion Products	87.71	1527	9.61
4	Combustion products	87.71	1005.5	1.099
5	Combustion products	87.71	507.9	1.213
6	Water	19.44	383	90
7	Steam	19.44	760.1	90
8	Combustion products	87.71	210	1.213
9	Weak solution	65.07	90	5.625
10	Weak solution	65.07	43.25	5.625
11	Weak solution	65.07	43.25	0.87
12	Strong solution	55.7	35	.87
13	Strong solution	55.7	35	5.624
14	Refrigerant	0.121	69.92	5.624
15	Refrigerant	0.121	90	5.624
16	Refrigerant	0.121	35	5.625
17	Refrigerant	0.121	5	0.87
18	Refrigerant	0.121	5	0.87
19	Cooling water	13.66	20	1.013
21	Cooling water	13.66	10	1.013
22	Cooling water	13.66	25	1.013
23	Cooling water	18.92	35	1.013
24	Cooling water	7.924	25	1.013
25	Cooling water	7.924	35	1.013
F	Methane	2.062	311	16

Table 3 : Calculated parameters at state points of tri-generation system

State	Substance	Enthalpy (kJ/kmol)	Entropy (kJ/kmolK)	Total exergy (MW)
1	Air	-4713	199.9	0
2	Air	4632	201.3	27.619
3	Combustion product	694.5	235.2	103.978
4	Combustion product	-17309	238.4	41.299
5	Combustion product	-32258	217	9.427
6	Water	8428	210.5	0.647
7	Steam	60598	102.3	8.791
8	Combustion product	-37975	207.7	110.45
9	Weak solution	235.6	0.4665	22.587
10	Weak solution	152	0.4665	25.553
11	Weak solution	152	0.287	25.553
12	Strong solution	85.88	0.2207	29.791

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13	Strong solution	85.88	0.2207	29.791
14	Refrigerant	157.5	0.4365	29.823
15	Refrigerant	48074	156	7.118
16	Refrigerant	48074	9.097	4.428
17	Refrigerant	2641	9.506	4.428
18	Refrigerant	45213	162.6	6.721
19	Cooling water	1135	4.039	49.0123
20	Cooling water	758.2	2.7	48.7749
21	Cooling water	2266	6.61	5.732
22	Cooling water	1889	6.61	26.4797
23	Cooling water	2642	9.096	26.7507
24	Cooling water	1889	6.61	26.4287
25	Cooling water	2642	9.096	26.6992
F	Methane	-74875	-	111.657

Table 4 : Exergy destruction data of tri-generation system

Component	**Exergy Destruction Rate (MW)**	**Exergy Destruction (Percentage a)**	**Exergy Destruction (Percentage b)**
Combustion chamber	36.970	75.2890	33.11
Air compressor	2.020	4.1137	1.80
Gas turbine	2.847	5.7978	2.54
HRSG	5.98	12.1782	10.90
VAR System	1.2871	2.6211	5.36
Overall plant	49.1041	100	53.71

Percentage a is the exergy destruction rate within a component as a percentage of the total exergy destruction rate within the cogeneration system.
Percentage b is the exergy destruction rate within a component as a percentage of the exergy rate entering the cogeneration system with the fuel.

Table 5 : Effect of variation of pressure ratio on exergy destruction in different components of tri-generation system for TIT=1400˚C

Prc	**$E_{D,CC}$ (kW)**	**$E_{D,C}$ (kW)**	**$E_{D,HRSG}$ (kW)**	**$E_{D,T}$ (kW)**	**$E_{D,VARS}$ (kW)**
4	85698	2577	5564	2721	1798
6.222	62483	2568	4327	3151	1675
8.444	53154	2634	3965	3540	1686
10.67	48003	2720	3859	3903	1732
12.89	44707	2814	2853	4247	1792
15.11	42412	2911	2133	4581	1858
17.33	40727	3011	1987	4907	1927
19.56	39444	3113	1901	5228	1999
21.78	38444	3216	1881	5548	2072
24	37651	3321	1671	5867	2147

Table 6 : Effect of variation of TIT on exergy destruction in different components of tri-generation system for prc=10

TIT (°C)	**$E_{D,CC}$ (kW)**	**$E_{D,C}$ (kW)**	**$E_{D,HRSG}$ (kW)**	**$E_{D,T}$ (kW)**	**$E_{D,VARS}$ (kW)**
1300	53075	2441	3234	812	1360
1356	51669	2278	3022	987	1049
1411	50390	2134	2835	1024	757.9
1467	49220	2007	2669	1165	482.7
1522	48143	1892	2521	1346	220.1
1578	47146	1789	2388	1496	202
1633	46220	1696	2267	1765	182
1689	45354	1612	2158	1897	145
1744	44542	1534	2058	2087	132
1800	43777	1463	1967	2341	109