Exergy Analysis of a Co-generation Cycle for Combined Production of Power and Refrigeration

Abstract: *In this paper, a noval industrial wasteheat recovery based cogeneration is proposed for the combined yielding of power and refrigeration. The cycle is an integration of Rankine power cycle and absorption refrigeration cycle. Combined first and second law approach is applied and computational analysis is performed to investigate the effects of exhaust gas inlet temperature, pinch point, and gas composition on first law efficiency, power to cold ratio, and second law efficiency of the cogeneration cycle and exergy destruction in each component. The variation in specific heat with exhaust gas composition and temperature are accounted in the analysis and results. The first-law efficiency decreases while power to cold ratio and second law-efficiency increases with increasing exhaust gas inlet temperature. Power to cold ratio and second law efficiency decreases while first-law efficiency increases with increasing pinch point. Second law efficiency is significantly varies with gas composition and oxygen content of the exhaust gas. Approximateing the exhaust gas as air, and the air standard analysis leads to either underestimation or overestimation of cogneneration cycle performance on second law point of view. Exergy analysis indicates that maximum exergy is destroyed during the steam generation process; which represents around 40% of the total exergy destruction in the overall system. The exergy destruction in each component of the system varies significantly with exhaust gas inlet temperature and pinch point. The present analysis contribute further information on the role of composition, exhaust gas temperature, and pinch point influence on the performance of a waste heat recovery based cogeneration system from second law pointof view.*

Rajesh Kumar
Amity School of Engineering and Technology, GGSIPU
New Delhi

Nomenclature

cp	=	specific heat, kJ/kg-K
E	=	exergy transfer rate, kW
h	=	specific enthalpy, kJ/kg
HRSG	=	heat recovery steam generator
m	=	mass flow rate (kg/s)
M	=	molar mass (kg/kmol)
P	=	pressure (bar)
PP	=	pinch point (K)
Q	=	rate of heat transfer (kW)
R	=	universal gas constant (kJ/kmol-K)
s	=	specific entropy (kJ/kg-K)
T	=	temperature (K)
W	=	rate of work output (kW)
ST	=	steam turbine

Symbols

η_I	=	first law efficiency
η_{ex}	=	exergetic efficiency
η_p	=	pump isentropic efficiency
η_t	=	turbine isentropic efficiency

Subscripts

f	=	solution circulation ratio
g	=	gas
O	=	environmental
p	=	pump
s	=	steam
t	=	turbine
w	=	water

1. INTRODUCTION

Congeneration cycles for the production of electric power and useful heat has received its fair share of attention, due to their energy efficiency enhancement and thus reduce net energy consumption in almost all these situations where both heat and power are required[1]. A method of second law analysis which provides an effective technique for measuring and optimizing performance of thermal system and widely gaining acceptance over traditional energy methods in both industry and academia has also been applied to those type of cogenerations by many investigators, and there is published literature based on second law analysis of those system for various operating conditions[2-4].

Cogeneration cycles for simultaneous production of power and refrigeration using low temperature heat sources (55°C-200°C) such as geothermal, solar collectors, was proposed by Goswami [5,6]. This cycle produced both power and cooling with one heat source. Other researchers have also investigated this new cycle. Xu et al.[7] presented a parametric analysis of this cycle. Hasan et al. [8] applied an analysis of the first and second laws of thermodynamics to optimize such cogeneration cycle. Detailed discussions on thermodynamic performance of this cycle are reported in the recent literaure [9-11].

A significant amount of heat is wasted as flue gases from industries [12,15]. The flue gases on the virtue of being at a higher temperature (400°C-500°C) relative to the surroundings and having a higher mass flow rate, posses considerable amount of available energy, which if no utilized properly will lead to huge undesirable energy loss and increase in environmental polluion. In recent years a great deal of attention is focused on to utilize the waste heat for various applications and to analyze the units which are used to absorb heat from waste flue gases [13]. There isa greater scope to recover waste heat from various industries and to generate power, useful heat, and cooling using a waste heat recovery steam generator HRSG. Therefore, in this study a new thermodynamic cycle for cogeneration of power and refrigeration is proposed which combined the Rankine power and absorption refrigeration cycles using high temperature waste heat from industry as a heat source.

Evaluation of the thermodynamic performance of this cycle has proven to be a difficult task. The output of this cycle consists of power and refrigeration, which are too different quantities. Evaluating cycle performance involves adding these outputs, which is akin to adding "sweet and sour". Exergy analysis which is a field of research on energy systems devoted to the study of identification and quantification of malfunctions of the plant or its components that results in an increase of the amount of resource needed to obtain the same output, or in more general terms, in a decrease of the overall efficiency of the plant. This method uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the analysis, design and improvement of energy systems, and answer why actual operation performance of an exergy system differ from the design one [3,14].

As is seen, in most of the studies in the above cited literature, there is no study which is conducted to observe the effect of operating parameters on the energetic and exergetic performance of an industrial waste heat recovery based cogeneration for combined producion of power and refrigeration. The main objective of this paper is to identify and quantify the malfunctions due to irreversibilities (losses) associated with every component and have negative effect on the performance of the system. The exergy balance for the cycle and its components are presented and compared to energy balances. The effects of gas composition, specific heat, pinch point, and gas inlet temperature to HRSG have been observed on the thermodynamic performance parameters of the cycle viz; energy efficiency, exergy efficiency, power to cold ratio, and also their effect on irreversibilities associated with various components of the system in order to determine where improvements could be made to enhance performance. This will contribute some original information on the role of operating variables and will be useful in the design of an industrial waste heat recovery based cogeneration system.

2. PROBLEM FORMULATION

The waste heat from the industry enters the HRSG at 1 where it transfers the heat to the feed water which enters at 3 and converted into steam at 4. The steam generated is then expanded in a steam turbine at 4 to generate mechanical power and an alternator is connected to the turbine to drive the load and steam then enters the condenser at 5 where most of its heat is rejected to the cooling water. The water is then pumped at 6 to the HRSG. The stack gas coming out of HRSG at 2 is sent to the generator of vapor absorption system. The refrigereant (H_2O) separated from LiBr-H_2O in the generator by means of the heat driven by the stack gas. After refrigerant has reached the desired temperature it goes through the condenser at 8 and evaporator at 10 through the expansion value at 9. The water vapour mixture that enters the evaporator at 10 is boiled and exits the evaporator in a saturated state at 11. The saturated steam at 11 enters the absorber where it mixes with a weak solution at 17, generating heat that has to be dissipated to increase the efficiency of mixing process. The mixing process results in a strong solution that exists the absorber at 12 and pumped to the upper pressure of the cycle at 13. The high pressure strong solution at 13 is heated to a high temperature.

The following assumptions are made in the analysis[15,16].

1. System is at steady state.
2. No pressure drops on the steam side.
3. Pressure drop on gas side does not affect its temperature.
4. Lithium bromide solution in the generator and the desorber are assumed to be in equilibrium at their respective temperatures and pressures.
5. Refrigerant (water) at condenser and evaporator exit is in saturated states.
6. Strong solution of the refrigerant leaving the absorber and the weak solution of refrigerant leaving the generator are saturated.
7. To avoid crystallization of the solution, the temperature of the solution entering the throttle value should be at least 7-8° above crystallization temperature.
8. The system uses waste heat of industry to generate steam and then to drive the load, and stack gases to drive the generator which produces chilled water in the evaporator.
9. Heat losses towards the surroundings air are not been taken into account, as they are generally a minute fraction of the other energy transfers.

3. PROPERTY EVALUATION

The exhaust gas is considered as an air in one case and the actual gas composition in other cases. The specific heat of air is calculated using standard thermodynamic tables. The specific heat of the actual exhaust gas is determined using the relation from Moran and Shapiro [17].

$$C_P = \frac{R}{M}(\alpha + \beta T + \gamma T^2 + \delta T^3 + \varepsilon T^4) \quad ...(1)$$

T is in K, equation valid from 300-1000K.

Where R is the Universal gas constant, M is the molar mass of the gas, and $\alpha, \beta, \gamma, \delta, \varepsilon$ are gas constants for various ideal gases.

The specific heat of mixture of gases is expressed as the sum of the specific heats of each component and their mass fraction y_i.

$$C_P(T) = \sum_{i=1}^{n} y_i C_{Pi}(T) \quad ... (2)$$

The entropy change for an ideal gas mixture is expressed as

$$C_P(T) = \sum_{i=1}^{n} y_i C_{Pi}(T) \quad ... (3)$$

Thermodynamic analysis

Using the temperature profile and pinch point, the stack gas temperature of HRSG can be calculated by[4].

$$T_{g2}\ (T_P+PP) - [T_{g1}-(T_P+PP)] \left(\frac{h_f - h_c}{h_g - h_f}\right) \quad ... (4)$$

Where T_P is the process heat temperature of steam in HRSG and PP is the pinch point. The steam generation rate in HRSG and by performing the energy balances as:

$$m_{s1} = m_g \left(\frac{C_{P_{Tg1}} T_{g1} - C_{P_{Tg2}} T_{g2}}{h^{S4} - h^{W3}}\right) \quad ... (5)$$

Where m_g is the mass flow rate of exhaust gas from industry, C_P is the specific heat of exhaust gas, h_{s4} and h_{w3} are the enthalpies of steam at HRSG outlet and feed water at HRSG inlet respectively.

The rate of heat input Q_{in} to the system is given by

$$Q_{in} = m_g\, C_{P_{Tg1}}\, T_{g1} \quad ... (6)$$

The net power output for the system is

$$W_{net} = W_{ST} - W_{p1} - W_{p2} \quad ... (7)$$

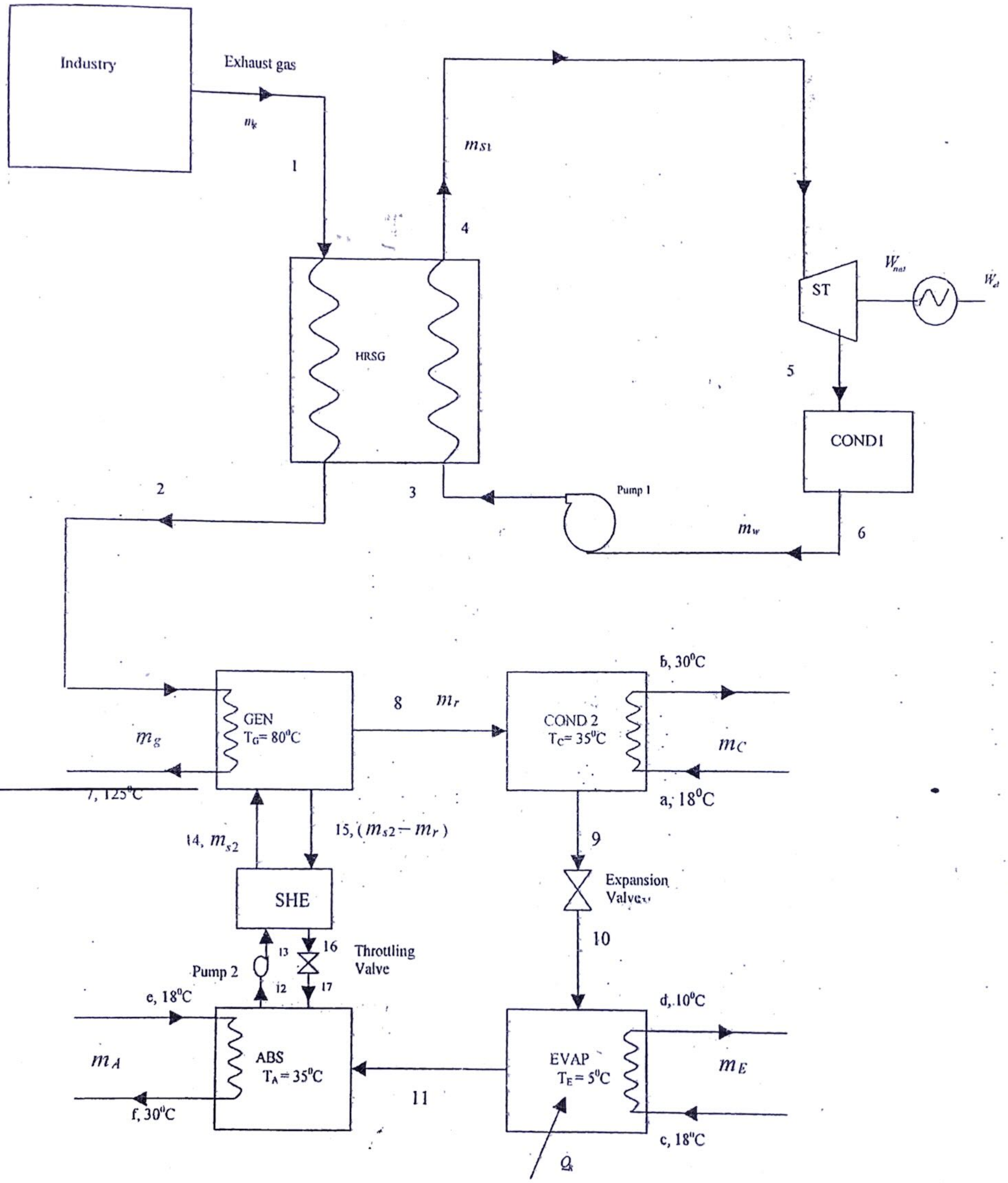

Fig. 1. Schematic of Combined Power and Refrigeration Cycle, as used in the analysis

If η_g is the mechanical to electrical conversion efficiency then electrical power output is given by

$$W_{el} = \eta_g W_{net} \quad \text{... (8)}$$

The rate of cold production (refrigeration) may be obtained after applying the energy balance on evaporator as

$$Q_R = m_r (h_{11} - h_{10}) = m_E (h_c - h_d) \quad \text{... (9)}$$

The enthaply and entropy values of LiBr-H_2O mixture at inlet and outlet of the evaporator of vapor absorption refrigeration can be obtained from Chua etal.[18].

PERFORMANCE PARAMETERS

The relevant parameters required for the energetic and exergetic analysis of combined production of power and refrigeration cycle may be considered as follows:

1. **First law efficiency** (η_I) : The first law measure of efficiency is simply a ratio of useful output energy to input energy. This quantity is normally referred to simply as efficiency, in the case of power cycles, and as a coefficient of performance for refrigeration cycles. For the case of cycle considered here, it may be defined as the ratio of all the useful energy extracted from the system (electric power and refrigeration) to the energy of exhaust gas from the industry. By definition.

$$\eta_I = \frac{W_{el} + Q_R}{Q_{in}} \quad \text{... (10)}$$

2. **Power to cold ratio** ($R_{P/C}$): The effectiveness of this type of cogeneration system is directly related to the amount of power it can generate for a given amount of refrigeration produced. Hence another parameter used to assess the thermodynamic performance of a given cogeneration system is ($R_{P/C}$) which is defined as:

$$R_{P/C} = \frac{W_{el}}{Q_R} \quad \text{... (11)}$$

Second-law efficiency (η_{ex}): The first law fails to accoun for the quality of energy. Therefore a first-law efficiency does not reflect all the losses due to irreversibilities in a system. Second-law efficiency measures the fraction of the exergy going into the cycle that comes out as useful output [17]. The remaining exergy is lost due to irreversibilities in the components. In the present system, the exergy output is the exergy of the electric power output (W_{el}) and the exergy of refrigeration (E_R). The second-law efficiency is given by

$$\eta_{II} = \frac{W_{el} + E_R}{E_{in}} \quad \text{... (12)}$$

Where Ein is the exergy of the exhaust gas from the industry that enters the HRSG, and is given by [15].

$$E_{in} = m_g\{[h(T_{g1})-h(T_0)]-T_0[S(T_{g1})-S(T_0)]\} \quad \text{... (13)}$$

When T_0 is the environmental or ambient temperature and s(T) is specific entropy.

The exergy of refrigeration, E_R, is the refrigeration capacity divided by the coefficient of performance of a Carnot refrigeration cycle operating between the ambient and cycle low temperature (T_E) and is given by[9].

$$E_R = Q_R \left(\frac{T_0 - T_E}{E_{in}} \right) \quad \text{... (14)}$$

Irreversibility analysis

In realistic cycles, there are irreversibilities associated with its every component. The irreversibilities will have negaive effective on the system performance, and they can be evaluated via exergy analysis which is a useful tool in determining they way for achieving better performances. According to Bejan et al. [19], the exergetic balance applied to a fixed control volume is given by the following equations:

$$\Sigma Q_j \left(1- \frac{T_0}{T_j} \right) W + \sum_{in} m_{in} e_{in} - \sum_{out} m_{out}\, e_{out} - I_D = 0 \quad \text{... (15)}$$

Where Qj is the heat transfer rate from or to the system, W the mechanical power supplied by or to the system, I_D the exergy destruction rate because of the irreversibilities, and e is the exergy transfer associated with the stream of matter. The kinetic and potential energies are usually neglected in air case, because the velocity of the fluid and the height changes are small, and since there is departure of chemical substances from the cycle to the environment, the chemical exergy is zero [14]. Therefore, the specific exergy e can be evaluated as:

$$e = (h - h_0) - T_0 (S - S_0) \quad \text{... (16)}$$

In the following, the equations that make the physical model of the industrial waste heat recovery based cogeneration system for combined production of electric power and cold are presented. These are the mass, energy and exergy balances of each component of the system.

The irreversibility or thermodynamic losses in each component of the system may be obtained with the combined application of first and second laws of thermodynamics and may be reported as:

Heat recovery steam generator (HRSG):

The stack gas temperature at the exist of HRSG (T_{g2}) and mass flow rate of steam (m_s) are calculated in the equations (4) and (5) respectively.

The irreversibility rate or exergy destruction rate in HRSG can be evaluated after making exergy balance as:

$$I_{D,HRSG} = m_g\{[h(T_{g1})-h(T_{g2})]-T_0[S(T_{g2})-S(T_{g1})]\} - m_{s1}[(h_{s4}-h_{w3})-T_0(S_{s4}-S_{w3})] \quad ...(17)$$

Steam Turbine (ST):

The exergy destruction or irreversibility rate in steam turbine is given by

$$I_{D,ST} = m_{s1}[(h_{s4}-h_{s5})-T_0(S_{s4}-S_{s5})]-W_{st} \quad ...(18)$$

Condenser 1(C1):

$$I_{D,C1}=m_{s1}[h_{s5}-h_{s6})-T_0(S_{s5}-S_{s6})] \quad ...(19)$$

Pump 1 (P1):

$$I_{D,P1}=W_{P1}-m_w[h_{w3}-h_{w6})-T_0(S_{w3}-S_{w6})] \quad ...(20)$$

Generator (Gen):

The irreversibility in the components of vapor absorption refrigeration part of the system used for cooling may be evaluated as;

Energy and mass balances allows us to determine the heat transferred by the external fluid to the solution within the generator

$$m_{s2}h_{14}+m_Gh_{g2}-(m_{s2}-m_r)\ h_{15}-m_rh_g-m_Gh_7=0 \quad ...(20)$$

The exergy of the fluids that enter the generator is:

$$\Sigma m_{in,G}e_{in,G}= m_G[(h_2-h_0)-T_0(S_2-S_0)]+m_{s2}[h_{14}-h_0)-T_0(S_{14}-S_0)] \quad ...(21)$$

The exergy of the mass flow that comes out being:

$$\Sigma_{mout,G}\ e_{out,G}= (m_{s2}-m_r)[(h_{15}-h_0)-T_0(S_{15}-S_0)]+m_G[(h_7-h_0)-T_0(S_7-S_0)]+m_r[(h_8-h_0)-T_0(S_8-S_0)] \quad ...(22)$$

We obtain the exergy destruction rate because of the irreversibilities in the heat transfer between the external and the solution by applying equation (15), and taking into account equation (22).

$$I_{D,G} = T_0[m_r(S_8-S_{15})+m_{s2}(S_{15}-S_{14})]+m_GT_0(S_7-S_2) \quad ...(23)$$

The first two terms arepositive while the third is negative. The global balance is positive.

In the similar fashion, the irreversibility rate or exergy destruction rate in the solution heat exchanger, solution pump absorber, condenser 2, evaporator, and expansion valve can be obtained, and may be reported as;

The irreversibility rate in solution heat exchanger is giveny by,

$$I_{D,SHE} = T_0\ [m_{s2}\ (S_{14}-S_{13})+(m_{s2}-m_r)(S_{16}-S_{15})] \quad ...(24)$$

The pump transport the dilued soluion in liquid state from the absorber to the generator, raising its pressure. The irreversibiliy rate in the pump is given by

$$I_{D,P2} = m_{S2}T_0(S_{13}-S_{12}) \quad ...(25)$$

The throttling value reduces the pressure of the concentrated solution from the high pressure in the impulsion of the compressor(generator) towards the low pressure in the suction side of the compressor (absorber). The irreversibility rate in the expansion valve may be given as,

$$I_{D,TV}= (m_{s2}-m_r)\ T_0(S_{17}-S_{16}) \quad ...(26)$$

$I_{D,TV}$ is the higher than zero, since the difference is positive.

The absorber absorbs the refrigerant vapor at low pressure and low temperature, which condenses in the solution. By means of mass, energy, and exergy balance, the irreversibility rate in the absorber may be defined as:

$$I_{D,A} = T_0[m_r(S_{17}-S_{11})+m_s(S_{12}-S_{17})+m_A(S_f-S_e)] \quad ...(27)$$

The refrigerant vapor goes to the condensor2 from the generator. The irreversibility rate or exergy destructionrate in the condenser is given by:

$$I_{D,Cond2} = T_0[m_r(S_9-S_8) + m_c(S_b-S_a)] \quad \text{... (28)}$$

The condensed refrigerant from the condenser goes to the throttling valve, and after throtting it goes to the evaporator. The irreversibility rate inthe evaporator is given by,

$$I_{D,Evap} = T_0\ [m_r(S_{11}-S_{10}) + m_E(S_d-S_c)] \quad \text{... (29)}$$

The condensed refrigerant from the condenser goes to the expansion valve where its pressure reduces to the evaporator pressure. The irreversibility rate is given by,

$$I_{D,EV} = m_r T_0(S_{10}-S_9) \quad \text{... (30)}$$

Results and discussion:

The effects exhaust gas inlet temperature to the HRSG, gas composition, and pinch point on the first-law efficiency and power to cold ratio is obtained by energy balance approach or the first-law analysis of the cycle. However, the exergy destruction in each component, and the second law efficiency of the cogeneration cycle have also been investigatedunder the exergy balance approach or the second law analysis of the cycle.

Fig. 2 shows the effect of change in exhaustgas inlet temperature and gas composition on first law efficiency of cogeneration cycle for a particular pinch point pp=20°C. It is observed that as gas inlet temperature (T_{g1}) increases the first lawefficiency of cogeneration cycle decreases. This is because higher gas inlet temperature causes improvement in power output, cold production and heat input rate to the system and the increase in heat input rate is much greater than

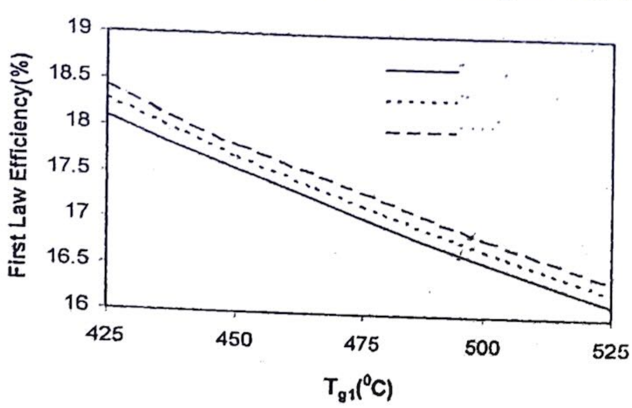

Fig. 2.Variation of first law efficiency with gas composition and exhaust gas inlet temperature; PP=20°C, mg=1100kg/s

the increase in power output and cold production. It is also observed that the first law efficiency of cogeneration cycle varies with gas composition and oxygen content in the gas and it decreases with the increase in oxygen content in the exhaust gas as demonstrated in Fig. 2. The specific heat varies with gas composition which influences heat transfer. The analysis based on air standard approach under estimates first-law efficiency and actual gas composition shall be used to accurately predict first-law performance.

Fig. 3 shows the variation of power to coldratio of cogeneration cycle with the change in exhaust gas inlet temperature and gas composition. For the same pinch-point power to cold ratio increases with exhaust gas inlet temperature. This is because the power output and cold production increases with the increase in exhaust inlet gas temperature as expected due to more steam production in HRSG, and improvement in power output is much greater than the improvementt in cold production. However, the power output and cold production for any gas inlet temperature depends on the exhaust gas composition and power to cold ~~ratio~~ decreases with an increase in oxygen content of exhaust gas. The power to cold ratio increases more rapidly at higher gas inlet temperature for a particular pinch-point and gas composition as demonstrated in Fig. 3.

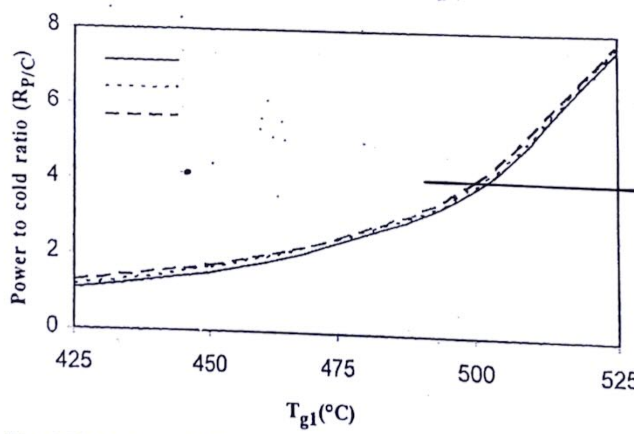

Fig. 3.Variation of Power to cold ratio ($R_{P/C}$) with gas composition and exhaust gas inlet temperature; PP=20°C, mg=100kg/s

Fig. 4 shows the variation of second-law efficiency of cogeneration cycle with a change in exhaust gas inlet temperature (T_{g1}) and gas composition for a particular pinch point = 20°C. It is observed that second law efficiency increases with exhaust gas inlet temperature. This is because higher gas inlet temperatures increases streamgeneration rate in HRSG, power output and availabiliy of the gas. The gas composition influences second law efficiency of the

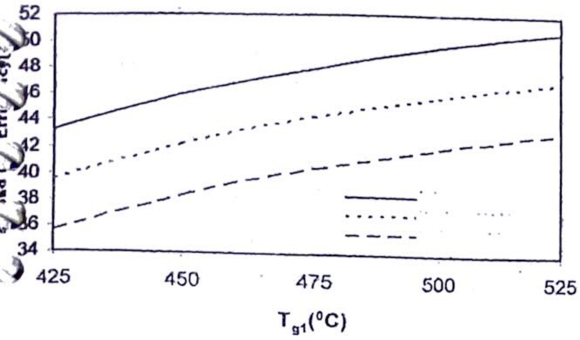

Fig. 4. Variation of Second Law Efficiency (%) with gas composition and exhaust gas inlet temperature; PP=20°C, mg = 100 kg/s

cogeneration system.The same pinch point, and gas inlet temperature the second law efficiency is different for different gas compositions. This clearly demonstrates that treating gas as air and doing the second law analysis based on air results in prediction of the cogeneration system performance on low or high side. Though the second law variation trends are same with gas inlet temperature, the gas composition has influence on heat recovery, irreversibilities in components, first and second law efficiencies of cogeneration unit. Modeling the exhaust gas as an air can significantly overestiamte the second law efficiency of cogeneration cycle.

Table 1. Exhaust Gas Composition [15, 17]

Combustion Products	Mass Fraction (%)		
	Gas I	Gas II	Air
CO_2	4.42	10.28	–
H_2O	3.43	8.41	–
O_2	16.55	7.48	20.95
N_2	75.6	73.82	78.08
Ar	–	–	0.93
Other	–	–	0.01

Table 2. System Test Parameters

Test Parameter	
Gas mass flow rate (kg/s)	100.00
Gas inlet temperature (K)	773.00
Saturation temperature (K)	549.73
Steam pressure (bar)	60.00
Steam outlet temperature (K)	623.00
Pinch point (K)	20.0
Pressure loss	5.00
Turbine isentropic efficiency (%)	85.00
Pump isenropic efficiency (%)	85.00
Condenser pressure (bar)	0.10
Environmental temperature (K)	293.00

Conclusion

Exergy analysis which is a method uses conservation of mass, and conservation of energy principles together with the second law of thermodynamics has been applied to industrial waste heat recovery based cycle for combined production of power and refrigeration. From thermodynamic point of view, the combination of Rankine power cycle with absorption chilling machine in these cogeneration proves to be highly efficient because the flue gas from heat recovery steam generator is used as a heat sourcefor vapor absorption refrigeration as described in this study. Combined first and second law analysis of the given system leads to the following conclusions:

1. The first law efficiency decreases while power to cold ratio and second law efficiency increases with increasing exhaust gas inlet temperature.
2. Power to cold ratio and second law efficiency are sensitive to pinch point and the pinch point should be low for better performance based on first and second law point of view.
3. Second law efficiency is significantly varies with gas composition and oxygen content in the gas while power to cold ratio and first law efficiency shows small variations with the change in oxygen content.
4. The second law efficiency is higher than first-law efficiency as it provides a measure of how efficiently the system is using thermodynamic resources based on quality point of view.
5. Exergy analysis provides ranking among the components for exergy destruction and present study results shows 40% irreversibility in the HRSG, 20% in steam turbine, 10% in condenser of power cycle, and 10% in the generator of absorption system and than in others.
6. Modeling the exhaust gas as an air significantly overestimates the second-law efficiency of industrial waste heat cogeneration system.

Reference

(1) Boycle, M.P., 2002, "Handbook for Cogeneration and Combined Cycle Power Plants," ASME Press, New York.

(2) Khaliq, A., and Kaushik, S.C., 2004, "Thermodynamic Performance Evaluation of Combstion gas Turbine Cogeneration System with Reheat," Appl. Thermal Eng., **24**, pp. 1785-1795.

(3) Oh, S.D., Pang, H.S., Him, S.M., and Kwak, H.Y., 1996, "Exergy Analysis for a Gas Turbine Cogeneration System," ASME Tras., J. Eng. For Gas Turbine and Power, **118**, pp. 782-791.

(4) Khaliq, A., and Choudhary, K., 2007, "Combined First and Second Law Analysis of Gas Turbine Cogeneration System with Inlet Air Cooling and Evaporative After Cooling of the Compressor Discharge" ASME Trans. J of Eng. for Gas Turbine and Power, 129, pp. 1005-1012.

(5) Goswami, D.Y., 1995, "Solar Thermal Power-Status of Technologies and Opportunities for Research," Proc. 2nd ISHMT-ASME Heat and Mass Transfer Conference, Tata McGraw Hill, New **27(1)**, Delhi, pp. 57-60.

(6) Goswami, D.Y., 1998, "Solar Thermal Power Technology: Present Status and Ideas for the Future," Energy Source, 20, pp. 137-145.

(7) Xu, F., Goswami, D.Y., and Bhagwat, S.S., 2000, "A Combined Power/Cooling Cycle," Energy, **25**, pp. 233-245.

(8) Hasan, A.A., Goswami, D.Y., Vijayaraghavan, S., 2002, "First and Second Law Analysis of a New Power and Refrigeration Thermodynamic Cycle Using a Solar Heat Sources," Solar Energy, **73(5)**, pp. 385-393.

(9) Tamm, G., Goswami, D.Y., Lu, S., and Hasan, A.A., 2003, "A Novel Combined Power and Cooling Cycle for Low Temperature Heat Sources-Part I: Theoretical Investigations," J. Solar Energy Eng., **125(2)**, pp. 218-222.

(10) Liu, M., Zhang, N., and Cai, R.X., 2006, "A Series Connected Ammonia Absorption Power/Cooling Combined Cycle," J. Eng. Thermophysi, **27(1)**, pp. 9-12.

(11) Liu, M., and Zhang, N., 2007, "Proposal and Analysis of Noval Ammonia-Water Cycle for Power and Refrigeration and Cogeneration," Energy, **32**, pp. 961-970.

(12) Nag, P.K. and De, S., 1997, "Design and Operation of a Heat Recovery Steam Generation with Minimum Irreversibility," Appl. Thermal Eng., **17**, pp. 385-391.

(13) Reddy, B.V. Ramkiran, G., Kumar, K.A., and Nag, P.K. 2002, "Second Law Analysis of a Waste Heat Recovery Steam Generator," Int. J. Heat and Mass Transfer, **45**, pp. 1807-1814.

(14) Wall, G., 2003, "Exergy Tools" Proc. Instn. Mech. Engrs, Part-A, J. of Power and Energy, **217**, pp. 125-136.

(15) Butcher, C.J., and Reddy, B.V., 2007, "Second law Analysis of a Waste Heat Recovery Based Power Generation System, Int. J. Heat and Mass Transfer, **50**, pp. 2355-2363.

(16) Khaliq, A., and Kumar, R., 2008, "Second Law Based Thermodynamic Analysis of Double Effect Vapor Absorption Refrigeration System, Int. J. Energy Research, **32**, pp. 161-174.

(17) Moran, M.J., and Shapiro, H.N., 2008, "Fundamentals of Engineering Thermodynamics," John Wiley & Sons, Inc., 6th Edition USA, 3rd Chapter.

(18) Chua, H.T., Toh, H.K., Malek, A., Ng, K.C., and Srinivasan, K, 2008, "Improved Thermodynamic Property Fields of LiBr-H_2O Solutions," Int. J. Refri., **23**, pp. 412-429.

(19) Bejan, A., 2002, "Fundamentals of Exergy Analysis," Entropy Generation Minimization, and the Generation of Flow Architecture," Int. J. Energy Research, **26**, pp. 545-565.

□