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Utilizing OF Thermal Energy OF Gas Turbines Exhaust Gases Working in Petroleum Sector

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Abstract

The gas turbine stations are used in petroleum sector for different application, such as compressing of gas, pumping of crude oil or water, and power generation. The main parts of these stations are compressor, combustion chamber, and gas turbine. The compressor provides a compressed air for combustion process, and for cooling process of different parts of the station. The fuel (liquid or gas fuel) is injected into the combustion chamber to perform the combustion process. The resulted hot flue gases are passed to the gas turbine to convert its' thermal energy into rotational mechanical energy to operate the air and gas compressors, crude oil or water pumps, and to generate electricity.

This work concerns with the applicability of utilizing the thermal energy content in flue gases exit from the gas turbine by adopting a field data received from one of the operating stations in north Oil Company in Iraq. The flue gases mass flow rate, and its' available energy have been calculated. Different operational conditions have been selected to produce superheated steam such as operating pressure range of (0.36-1.4 MPa), with a range of superheated temperature differs than the corresponding saturation temperature by (10-60 ^{o}C) for each selected pressure. This work is also concerned with the possibility of producing of hot water for a pressure range of (0.36-1.4 MPa) with a temperature lower than the corresponding saturation temperature by (5 ^{o}C). The results show that the power produced reaches (1.4-3.3 MW). Also the hot water produced reaches 90000 kg/hr at operating pressure of 0.36MPa.

The obtained results in this work are interesting since an elimination of thermal pollution by exhaust gases is accomplished. The available energy in these gases has been utilized to produce either superheated steam or hot water without consuming a conventional fuel.

1- Introduction

A considerable amount of heat is wasted with the exhaust of the gas turbine, adding high value of exergy with high concentration of pollutants (including carbon dioxide CO2, nitrogen oxides NOX, and sulfur oxides SOX) directly to the environment which are responsible for global warming [1,2]. Efficiency of energy conversion processes can be considerably improved if this waste heat is used. The complete use of the energy available to a system is called the total energy approach. In this approach all of the available heat energy in a power system at different temperature levels is used for work, steam, or hot water production, thereby rejecting a minimum amount of energy to the environment, and saving in fuel. The best approach is by passing the exhaust gases leave the gas turbine through a heat recovery steam generator (HRSG) [3] to supply steam to a steam turbine or to produce hot water.

The amount of the waste heat from industrial processes accounts for 20% to 50% of the input heat, and the recovery of this heat can improve energy efficiency by 10% to 50% [4]. About 50% of industrial waste heat has a low temperature ranging from 100°C to 120°C [5]. The key factors that determine the feasibility of waste heat recovery include flow rate, temperature, pressure, chemical composition, allowable temperature and pressure drop, operating schedules, availability, and other logistics of heat source. The selection of appropriate waste heat recovery system directly affects the efficiency and cost. Combined heat and power (CHP) systems are suitable for the simultaneous production of heat and power, whereas TGS are used for the simultaneous production of heating, and power [6]. The recovery of waste heat through CHP can increase the system exergy efficiency up to 70% [7, 8]. The economic aspect and feasibility of waste heat recovery-based CHP system have been analyzed in terms of their thermodynamic performance [7, 8]. The greenhouse gases GHG emission and exergo-environmental analysis of trigeneration system TGS was presented by Ahmadi et al.[9]. The comparison of the results with those of the conventional CHP system shows that the exergy efficiency of TGS is higher but its environmental effect is lower. The exergo-environmental optimization of a gas turbine-based CHP system was performed by [10,11] using a genetic algorithm.

The objectives of this work are to highlight the potential of waste-heat recovery technologies, in particular for Iraq north Oil Company, to enhance the energy efficiency of its natural gas compressing processing plants. To investigate the application of such technologies for Iraq gas power plants, which have generally made limited use of waste heat recovery up to date for power generation or hot water production finally to undertake a thermodynamic performance analysis of combined cycle for simultaneous generation of power generation, with waste heat sources?

2- System Analyses

The analysis of the Gas Cycle was adopted according to the available Data from (North Oil Company -Iraq) as given in Table (1). Figure (1) illustrates the Temperature Entropy and pressure volume diagrams of the gas turbine cycle.

Compressor Power =6400 kW

Net cycle power =9680 kW

Turbine power = Net cycle power +Compressor power (1)

=9680 +6400 = 16080 kW

Or = 16.08 MW

The work ratio = $\frac{Comp.\ work}{Turbine\ work} = \frac{6400}{16080} = 39.8\%$

Air flow rate =174500 kg/hr

2-1 Compressor Inlet

The enthalpy of air at compressor inlet is evaluated at T_1= 22°C or T_1=71.6F, from gas chart shown in Fig.2. So the enthalpy can be extracted from the chart as: h_1 =17.5 Btu/Ib.

2-2 Compressor outlet

The temperature and pressure at compressor outlet (state1) are 250°C (482F), and 5.1kg/cm^2 (500.31kN/m^2 or 73.29psia) respectively. Considering the compression process (1 to 2) as an isentropic process i.e $S_2 = S_1$ with p_2 =73.29psia and using gas chart Fig.2, the enthalpy of air at compressor outlet is evaluated as h_2=95 Btu/Ib, and compressed air temperature T_2=415F. From data sheet, it is given that $T_{\acute{2}}$ = 482 F (real value), so the real value of enthalpy at compressor outlet is obtained from p_2 and $T_{\acute{2}}$,

$$h_{\acute{2}} = 116 \text{ Btu/Ib}$$

The compressor efficiency $\eta_{comp.}$ is defined as

$$\eta_{comp.} = \frac{h_1 - h_2}{h_1 - h_{\acute{2}}}$$

So $\eta_{comp.} = \frac{17.5-95}{17.5-116} = 78.68\ \%$

2-3 Turbine inlet conditions

The pressure and temperature at gas turbine inlet as given in Table 1 are p_3=8.4 kg/cm^2 (824.04 kPa ; or 119.52 psia), and T_3=943 °C ; or (1729.4 F), so the enthalpy at turbine inlet can be evaluated from Fig.2, as h_3=430 Btu/Ib

2-4 Turbine outlet

The flue gases exits from gas turbine at temperature of 470 °C or (878F), and atmospheric pressure, so from Fig.2 (gas chart) the enthalpy is found as:

$$h_4 = 217 \text{ Btu/Ib}$$

so the mass flow rate of flue gases can be evaluated as

$$\dot{m_f} = \frac{\text{power}_{\text{turbine}}}{\Delta h_{\text{turb.}}} \quad (2)$$

where $\Delta h_{turb.}$ is the enthalpy drop across the turbine. It is evaluated as:

$$\Delta h_{turb.} = h_3 - h_4 \quad (3)$$

hence the flue gases mass flow rate is $\dot{m_f} = \frac{54880546}{430-217} = 257655.146\ Ib/hr$ or $= 71.571$ Ib/s

3- Analysis of the Steam Cycle

The temperature of flue gases is given in Table 1 as 470 °C (878F), so the available sensible energy in exhaust gases is taken from Fig.3 as 210 Btu/Ib. Hence the heat recovery from gas turbine is:

Heat recovery= $\dot{m_f} * avialable\ sensible\ energy$

or Heat recovery = 71.571*210 =15029.91 Btu/s

This energy is used as a source energy in heat recovery steam generator HRSG to produce steam or hot water. Three cases are presented for different steam turbine inlet conditions in this work, namely:

[Case I] Utilizing of the heat recovery with no heat loss, i.e the flue gases exhausted to the environment at 26.67 °C (80F). This approach is called total energy approach. The analyses starts with the heat balance principle that is:

$$Q=Q_1+Q_2+Q_3 \quad (4)$$

Where

Q is Heat source (heat rejected from flue gases), it is Q= 15029.91 Btu/s (15859.4 kW)

Q_1 is the energy required to heat water from inlet temperature to saturation temperature (sensible heat) (kW).

Q_2 is the latent heat required to phase change (to evaporate water) (kW).

Q_3 is the energy required to superheat the vapor (kW).

Assume the inlet temperature of water to the heat recovery steam generator (HRSG) is equal to (T_{in} =20 ^{o}C, and the inlet conditions of the steam turbine are:

Pressure is equal to 1.1 MPa (10.63 atm) and 60 ^{o}C superheated steam i.e the turbine inlet temperature is equal to ($T_{saturation}$ +60^{o}C), then:

$$Q_1 = \dot{m}_w * c_{pw} * (T_{saturation} - T_{in}) \quad (5)$$

$$Q_2 = \dot{m}_w * h_{fg} \quad (6)$$

$$Q_3 = \dot{m}_w * c_{pv} * (T_{superheated} - T_{saturation}) \quad (7)$$

or $$Q_3 = \dot{m}_w * c_{pv} * (\Delta T) \quad (8)$$

where

$\dot{m}_w$ is mass flow rate of water circulated in the steam cycle (kg/s).

c_{pw} is the specific heat of water (taken at inlet temperature) (kJ/kg.K).

c_{pv} is the specific heat of vapor (taken at saturation temperature) (kJ/kg.K).

$T_{saturation}$ is the saturation temperature corresponding to the turbine inlet pressure.

h_{fg} is the enthalpy of evaporation (latent heat) at the turbine inlet pressure (kJ/kg).

ΔT is the difference between the turbine inlet temperature and the saturation temperature.(^{o}C).

A set of steam turbine inlet condition have been selected as given in table (2).

Now let the quality (dryness fraction) at the turbine outlet be equal to(x= 0.9), and the condenser pressure be = 10 kPa, so the enthalpy at the exit of steam turbine is evaluated from steam tables as

h_{out}= h_f+xh_{fg}

where

h_f=191.83 kJ/kg (from steam table [12] at P=10kpa), and h_{fg}=2392.8 kJ/kg (from steam tables at P=10 kpa)

hence h_{out} =191.83+0.9*(2392.8) =2345.35 kJ/kg

The power produced by the steam turbine is evaluated as:

$$Power= h_{in}-h_{out} \qquad (9)$$

where h_{in} is inlet enthalpy to steam turbine, evaluated from steam tables at turbine inlet conditions.

Case II

Utilizing of the heat recovery with heat loss, i.e the flue gases exhausted to the environment at 120 ^{o}C (248F). So the heat loss is evaluated from figure.(7) as 40 Btu/Ib or

Heat loss= 40 Btu/Ib* mass flow rate of flue gases

= 40 *71.571 = 2862.84 Btu/s

or Heat loss =3020.3 kW

so the heat source for this case can be calculated as:

Heat source = Q-Heat loss

= 15859.434 – 3020.3 =12839.139 kW

The analyses procedure adopted for case I is used for this case to calculate the amount of steam or hot water produced.

4- Results and Discussion

4-1 Case I

The selected working conditions of steam turbine are given in table (3). For this case the full energy approach is adopted so the heat source used in heat recovery steam generator is:

Heat source is Q=15859.434 (kW).

According to the accomplished calculations for this case the mass flow rate and power generated from steam turbine are given in table (4), and presented in figures (8-12). Figure (4) shows that the power produced in steam turbine is increased with the increasing of turbine inlet temperature for turbine pressure of 0.36 MPa, also decreasing the effectiveness of the heat recovery steam generator (HRSG) causes a decrease of the power produced. The same trend is found for turbine inlet pressure of 0.7, 1.1 ad 1.4 MPa in figures 5, 6, 7,and 8 respectively. These figures also show that increasing the turbine inlet pressure decreases the power produced for the same temperature and for the same effectiveness of HRSG. From these figures the power produced for 100% effectiveness of HRSG was 2965 kW (≈3 MW) at 0.36 MPa and T_{in}=200 ^oC, while the power produced at 1.4 MPa and T_{in}=255 ^oC was 3350kW. The power produced for 70% effectiveness of HRSG at 0.36 MPa, and 1.4 MPa is 2075 kW and 2345 kW respectively. The produced steam flow rate at 0.36 MPa in HRSG reaches 20638 kg/hr, and 14447 kg/hr for HRSG effectiveness equal to 100% and 70% respectively as shown in Fig.s(13, 14, and 15).

4-2 CASE II

The powers produced from steam turbine for this case (when there is heat loss to ambient i.e hot gases are rejected to atmosphere at 120 ^oC) are shown in figures (16 – 17). It can be shown that there is a considerable and important amount of electrical energy can be produced for 100% HRSG effectiveness, reaches 2400 kW, and 2713 kW at 0.36 and 1.4 MPa respectively. While for 70% effectiveness the power produced is 1560kW,and 1900 kW at 0.36 and 1.4MPa respectively as shown in figures (16 – 17).

4-3 Hot Water Production

Figure (18) shows the amount of hot water produced from the energy content in exhaust gases from gas turbine noting that the exit temperature of hot water was taken as (5 ^oC) below the saturation temperature corresponding to the

Working pressure. This figure shows that the amount produced is decreased with increasing pressure, also with deceasing the effectiveness of HRSG.

5- Conclusion

Based on the previous discussion of the obtained results the following conclusions can be extracted:

1- According to the available data from the north oil company the power generated from the combined gas-steam cycle (for effectiveness of heat recovery steam generator equals to 60%) exceeds 1.4 MW when the inlet conditions to steam turbine are 0.36 MPa ,and 150 ^{o}C , while it reaches to 1.6 MW for conditions of 1.4 MPa and 190 ^{o}C.

2- Increasing the effectiveness of heat recovery steam generator HRSG increases the amount of steam generated and the power generated.

3- Approximately the same amount of power generated can be attained for different inlet pressures to steam turbine (such as 0.36 and 1.4 MPa) by increasing the turbine inlet temperature.

4- The amount of hot water produced in HRSG increases with its effectiveness.

5- The amount of hot water produced decreases with the and decreases with the pressure.

Table (1) Data and Specifications of Gas Turbine Cycle as Received From North Oil Com.

Specification	Required information	Specification	Required information
Compressor inlet conditions		**Turbine outlet conditions**	
Efficiency	Not available	Efficiency	32%
Temperature	Normal temp. 22 ^{o}C	Temperature exhaust temp.	470 ^{o}C
Power extracted of compressor	6400 kW	Turbine power outlet	14600 H.P
Flow rate of air	174.500 kg/hr	Flow rate exhaust	175.940 kg/h.p
Compressor outlet condition:		Pressure	Not available
Temperature	250 ^{o}C	Composition of exhaust	CO_2, CO , SO_2,(H_2O as vapor)
pressure	5.1 kg/cm^2	Net cycle power	Net available power kW 9680 Max
Fuel and combustion Chamber		Inlet water temp to heat exchanger	50 ^{o}C
Fuel mass flow rate (mf)	32.19*10 kcal/hr	Out let water temp from heat exchanger	60 ^{o}C
LHV of fuel	11.690 kcal/kg	Inlet water temp to chiller (average)	16 ^{o}C
HHV of fuel	Not available	Outlet water temp from chiller (average)	6 ^{o}C
Type of fuel	Sweet natural gas		
Burner efficiency	Not available		
Combustion chamber outlet Condition (temp. & press.)	(charging turbines) Temp.943 ^{o}C. Press. 84kg/cm		

Table (2): Specifications of Steam Turbine Inlet Conditions

Pressure (MPa) Atm.	$T_{saturation}$ (°C)	ΔT (°C)	Turbine inlet temperature (°C)	Vapor specific volume v (m^3/kg)	Specific enthalpy h (kJ/kg)
(0.36) 3.478 atm.	139.87	10	150	0.5252	2756.1
		20	160	0.5394	2777.9
		30	170	0.5535	2799.4
		40	180	0.5674	2820.6
		60	200	0.5949	2862.5
(0.7) 6.763 atm.	164.97	10	175	0.2895	2788.8
		20	185	0.2975	2812
		30	195	0.3053	2834.6
		40	205	0.3129	2856.9
		60	225	0.328	2900.7
(1.1) 10.63 atm.	184.09	10	194.1	0.182883	2807.232
		20	204.1	0.188094	2832.004
		30	214.1	0.1932256	2856.14
		40	224.1	0.1981554	2879.694
		60	244.1	0.207868	2925.566
(1.4) 13.526 atm.	195.07	10	205.1	0.1452	2816.6
		20	215.1	0.14944	2842.4
		30	225.1	0.15357	2867.3
		40	235.1	0.1576	2891.7
		60	255.1	0.1654635	2939.032

Table (3) Calculated Specifications for case I and case II

$_1$=501.0565∗ $\dot{m}_w$ (kW)	Q$_2$=2145.1∗ $\dot{m}_w$ (kW)		
Turbine pressure (MPa)	Turbine inlet Temp. (^{o}C)	Q_3*$\dot{m}_w$ (kW)	(Q_1+ Q_2 +Q_3)* $\dot{m}_w$ (kW)
(0.36) 3.478 atm	149.87	20.0525	2666.209
	159.87	40.105	2686.262
	169.87	60.1575	2706.314
	179.87	80.21	2726.367
	199.87	120.315	2766.472
Q_1=605.9746$\dot{m}_w$ (kW)	Q_2=2066.3$\dot{m}_w$ (kW)		
(0.7) 6.763 atm	174.97	19.8818	2692.156
	184.97	39.76361	2712.038
	194.97	59.64541	2731.92
	204.97	79.52721	2751.802
	224.97	119.2908	2791.565
Q_1=685.8962$\dot{m}_w$ kW	Q_2=2000.4$\dot{m}_w$ kW		
(1.1) 10.63 atm	194.09	19.80709	2706.103
	204.09	39.61418	2725.91
	214.09	59.42127	2745.718
	224.09	79.22836	2765.525
	244.09	118.8425	2805.139
Q_1=731.7926$\dot{m}_w$ kW	Q_2=1959.7$\dot{m}_w$ kW		

	205.07	19.81807	2711.311
	215.07	39.63614	2731.129
(1.4) 13.526 atm	225.07	59.45421	2750.947
	235.07	79.27228	2770.765
	255.07	118.9084	2810.401

Table (4) Steam Flow Rate and Power Produced by Steam turbine for Case I with Effectiveness=100%

Effectiveness= 100%									
Pressure (MPa)	Temp. (^oC)	Steam Mass flow rate (kg/hr) $*10^3$	Steam volume flow rate (m^3/hr) $*10^3$	Power (MW)	Pressure (MPa)	Temp. (^oC)	Steam Mass flow rate (kg/hr) $*10^3$	Steam volume flow rate (m^3/hr) $*10^3$	Power (MW)
0.36	149.87	21.41	11.25	2.44	1.1	194.09	21.1	3.86	2.71
	159.87	21.25	11.46	2.55		204.09	20.95	3.94	2.83
	169.87	21.1	11.68	2.66		214.09	20.79	4.02	2.95
	179.87	20.94	11.88	2.76		224.09	20.6	4.09	3.06
	199.87	20.64	12.28	2.96		244.09	20.35	4.23	3.28
0.7	174.97	21.21	6.14	2.61	1.4	205.07	21.1	3.06	2.76
	184.97	21.05	6.26	2.7		215.07	20.9	3.12	2.89
	194.97	20.9	6.38	2.84		225.07	20.75	3.19	3.01
	204.97	20.75	6.49	2.95		235.07	20.6	3.25	3.13
	224.97	20.45	6.71	3.16		255.07	20.3	3.36	3.35

Table (5) Steam Flow Rate and Power Produced by Steam turbine for Case II with Effectiveness = 100%

Effectiveness= 100%									
Pressure (MPa)	Temp. (^oC)	Steam Mass flow rate (kg/hr) *10^3	Steam volume flow rate (m^3/hr) *10^3	Power (MW)	Pressure (MPa)	Temp. (^oC)	Steam Mass flow rate (kg/hr) *10^3	Steam volume flow rate (m^3/hr) *10^3	Power (MW)
0.36	149.87	17.34	9.1	1.98	1.1	194.09	21.1	3.86	2.71
	159.87	17.21	9.28	2.07		204.09	20.95	3.94	2.83
	169.87	17.08	9.45	2.15		214.09	20.79	4.02	2.95
	179.87	16.95	9.62	2.24		224.09	20.64	4.09	3.06
	199.87	16.71	9.94	2.4		244.09	20.35	4.23	3.28
0.7	174.97	21.21	6.14	2.6	1.4	205.07	17.05	2.48	2.23
	184.97	21.05	6.26	2.73		215.07	16.92	2.53	2.34
	194.97	20.9	6.38	2.84		225.07	16.80	2.58	2.44
	204.97	20.75	6.49	2.95		235.07	16.68	2.63	2.53
	224.97	20.45	6.71	3.16		255.07	16.45	2.72	2.71

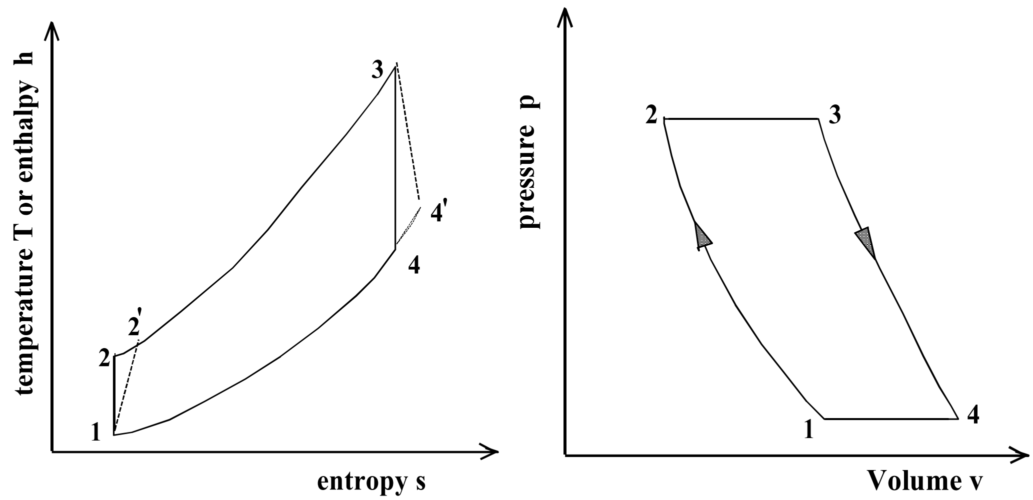

Fig.(1) T-s and p-v diagram of simple Gas Cycle

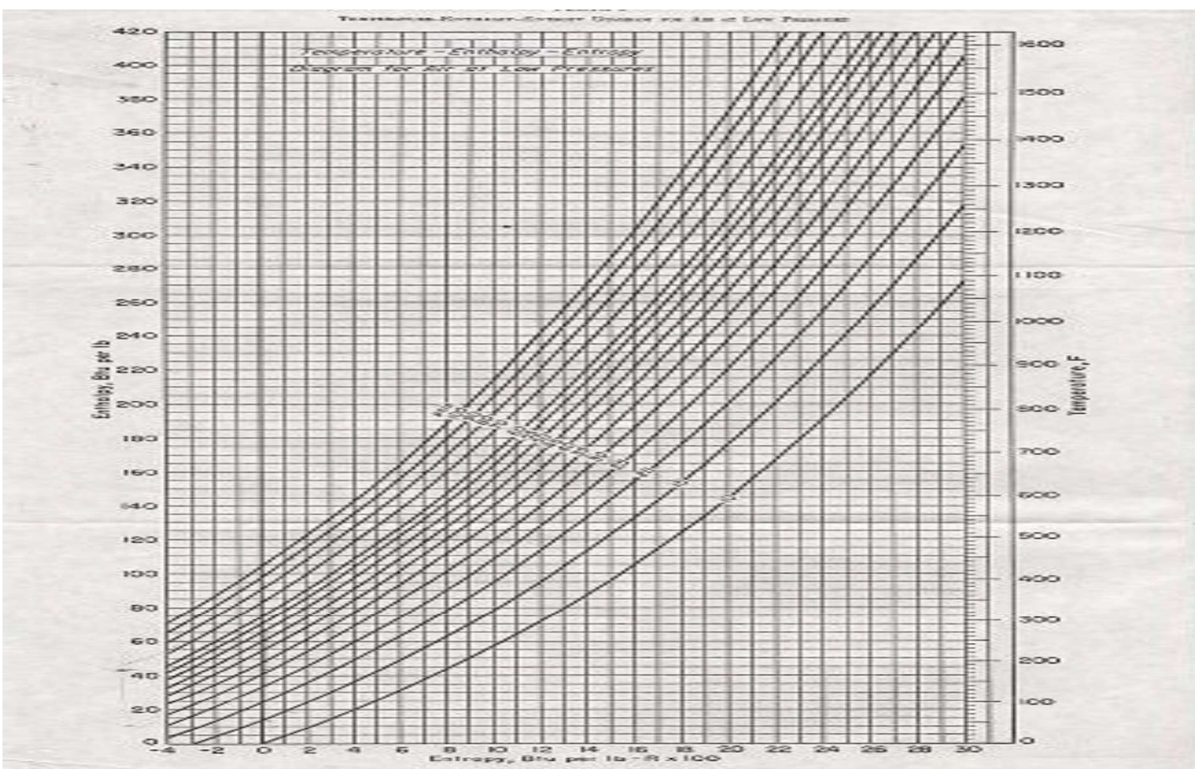

Fig. (2) T-S diagram for Flue-Gases [13]

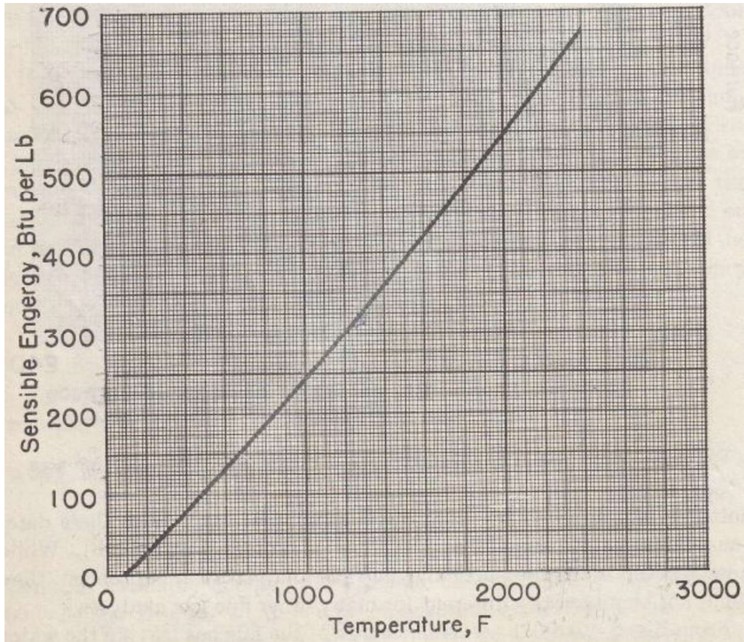

Fig.(3) Sensible energy of flue gases [13]

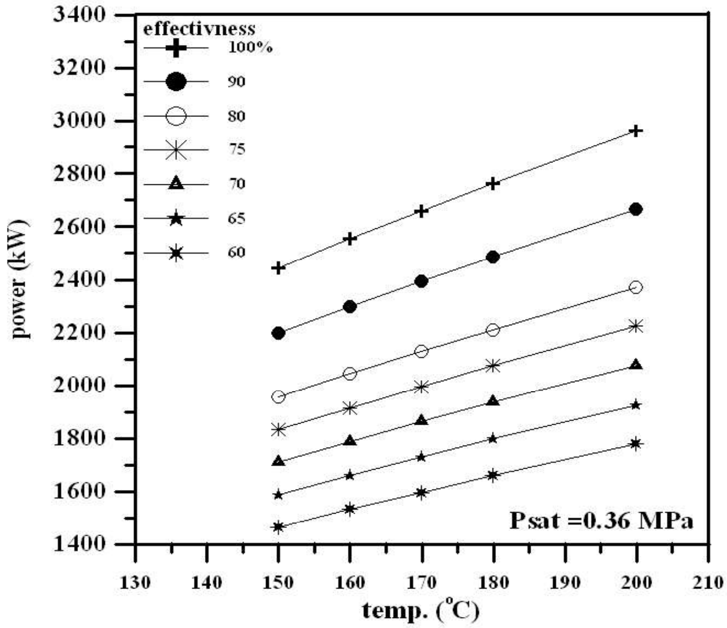

Fig.(4) Power Produced for Steam Turbine Inlet Pressure=0.36 MPa

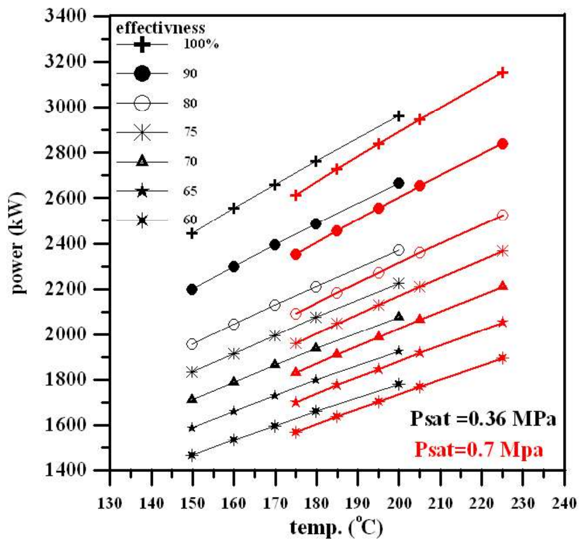

Fig.(5) Power Produced for Steam Turbine Inlet Pressures (0.36& 0.7 MPa)

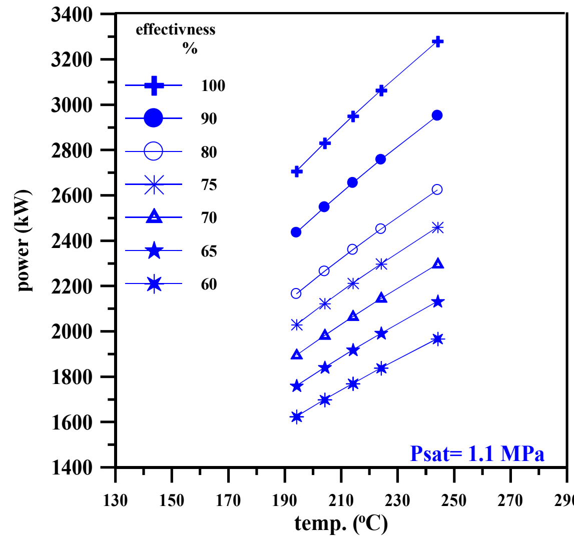

Fig.(6) Power Produced for Steam Turbine Inlet Pressure 1.1 MPa

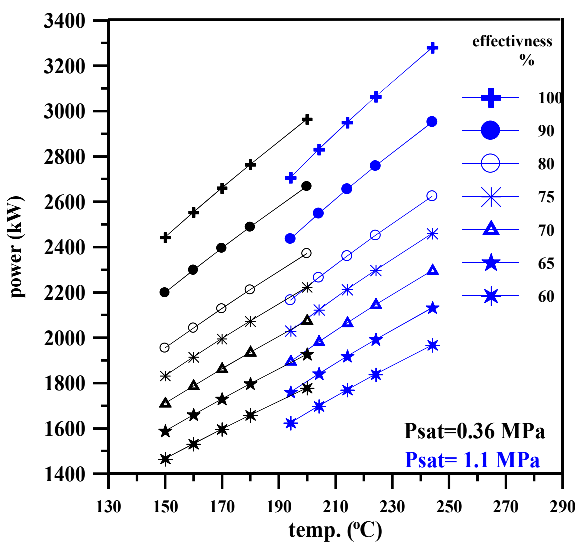

Fig.(7) Power Produced for Steam Turbine Inlet Pressures (0.36 & 1.1 MPa)

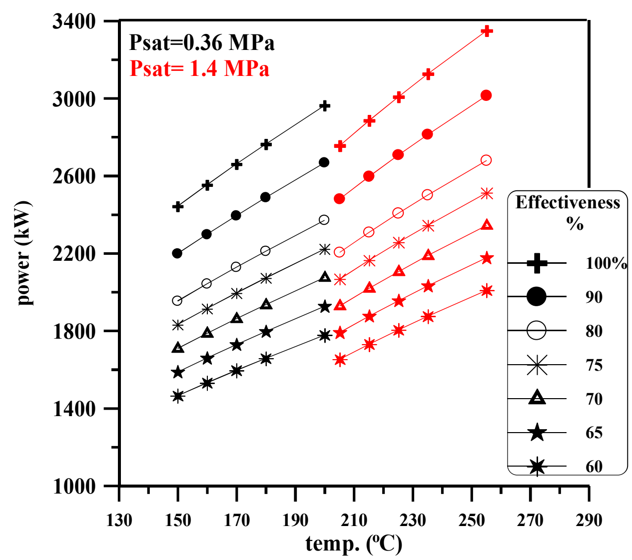

Fig.(8) Power Produced for Steam Turbine Inlet Pressures (0.36 & 1.4 MPa)Case

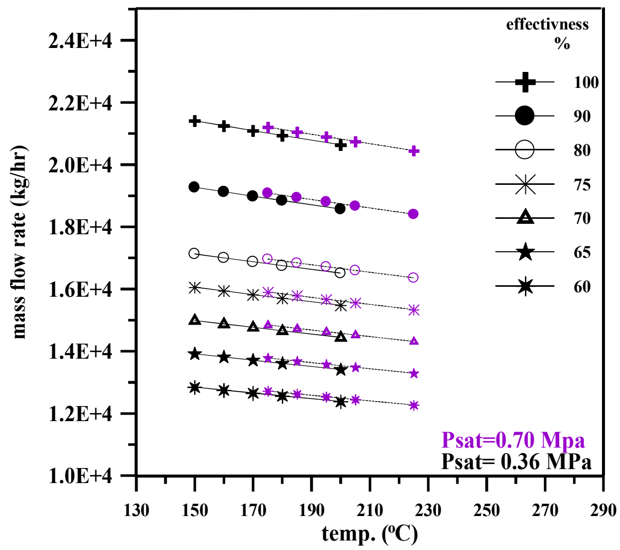

Fig.(9) Variation of Steam Mass Flow Rate Produced in HRSG for Steam Turbine Inlet Pressures of (0.36& 0.7MPa) Case I

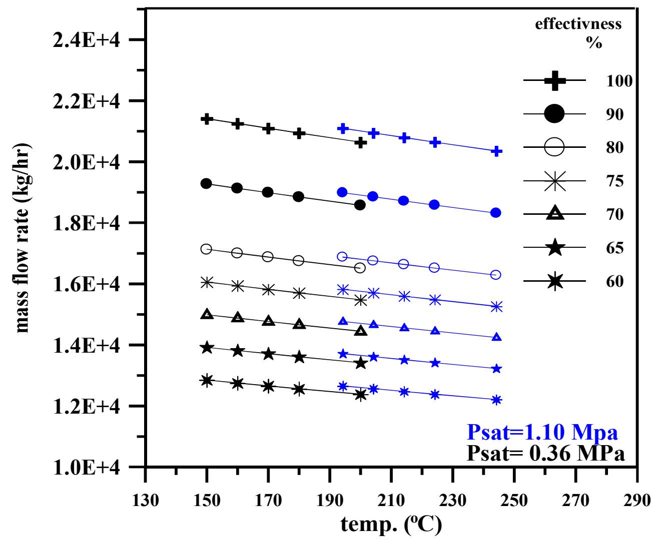

Fig.(10) Variation of Steam Mass Flow Rate Produced in HRSG for Steam Turbine Inlet Pressures of (0.36& 1.1MPa) CASE I

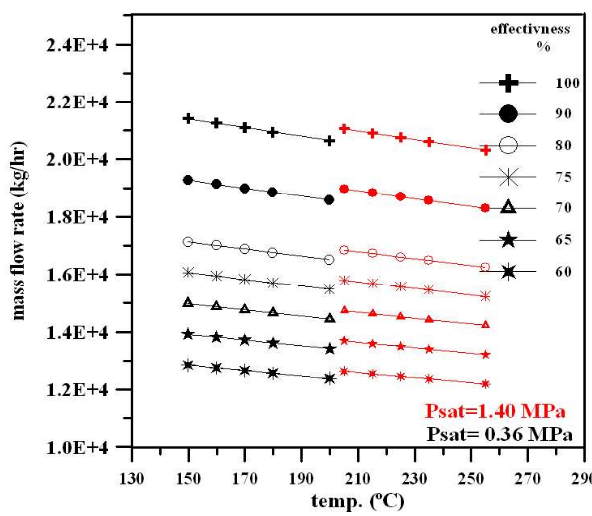

Fig.(11) Variation of Steam Mass Flow Rate Produced in HRSG for Steam Turbine Inlet Pressures of (0.36& 1.4MPa) CASE I

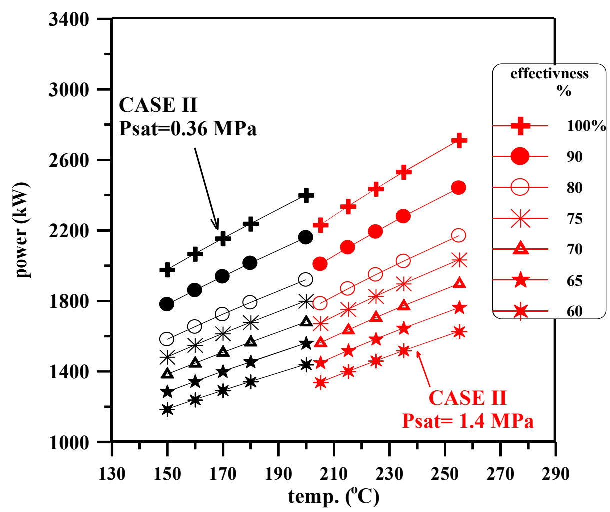

Fig.(12) Power Produced for Steam Turbine Inlet Pressures (0.36& 1.4MPa) Case II

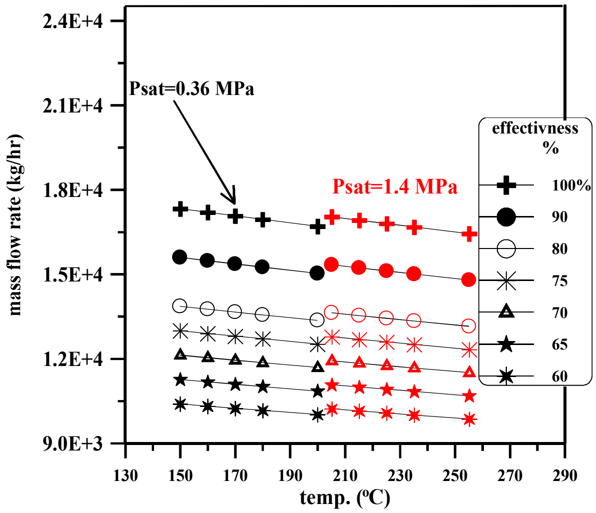

Fig.(13) Variation of Steam Mass Flow Rate Produced in HRSG for Steam Turbine Inlet Pressures of (0.36& 1.4MPa) CASE II

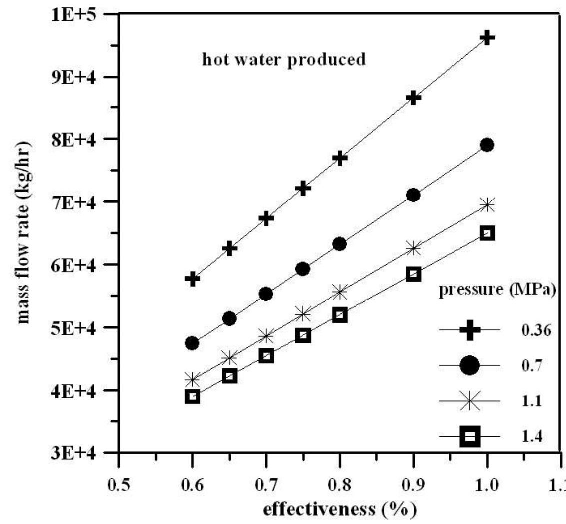

Fig.(14) Variation of mass flow rate of the hot water produced with the effectiveness of the heat exchanger.

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