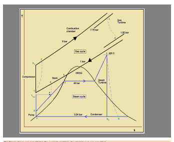
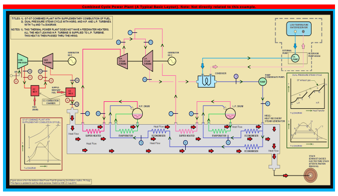


The figure above was provided in the example problem. No plant layout was provided.
We will work through each equipment's analysis, and then connect them to simple combined cycle process.
This will allow us to put together a simple plant layout for this example. A plant layout is highly suggested to solve the problem.

Combined Cycle Power Plant (A Typical Basic Layout). Note: Not directly related to this example.



Problem 4: *Main example on combined cycle power generation.*

The following data refer to a combined cycle power plant:

Compressor				
Inlet condition of air to compressor	:			
- air pressure	:	1	bar	
- air temperature	:	298	K	25 C
Pressure ratio of compressor	:	8		
Combustion chamber				
Pressure drop in the combustion chamber	:	3%	%	
Gas turbine				
Maximum gas temperature at inlet to the gas turbine	:	1173	K	900 C
HSRG				
Feedwater temperature to the HRSG	:	443.4	K	
Pressure drop of gas in the HRSG	:	0.05	bar	5 kPa
Steam turbine				
Condition of steam at inlet to the steam turbine				
- steam pressure	:	40	bar	
- steam temperature	:	698	K	425 C
Steam flow				
Steam flow rate	:	29.235	kg/s	
Specific heats				
Specific heat of air	:	1.006	kJ/kg K	
Specific heat ratio of air	:	1.4		
Specific heat of gas	:	1.148	kJ/kg K	
specific heat ratio of gas	:	1.333		
Efficiencies				
Efficiency of compressor	:	0.88		
Efficiency of gas turbine	:	0.88		
Efficiency of steam turbine	:	0.82		
Calorific value				
Calorific value of liquid octane (C_8H_{18}) used as fuel	:	44.43	MJ/kg	

Assume **$\Psi = (\Delta G_0 / \Delta H_0) = 1.0401 + 0.1728\ (h/c)$**
where **(h/c)** is the mass ratio of hydrogen to carbon in the fuel.

a). Determine the total power output and the overall efficiency of the combined cycle plant and estimate the "lost heat" coefficient x_L.

b). Give an energy balance of the plant and estimate the exergetic efficiency.

Solution:

Comments:
This problem does not provide a plant layout figure, we assume the given data is adequate to construct the layout.
There is only one combustion chamber, NOT two as in the previous example problem.
The textbook provides a Ts figure. We relate the Ts figure to the power plant process.
We will provide a figure for each major equipment.
We could assume the top cycle, (gas), is an open cycle. Ts shows 1 and 2 are connected. However, objective is to solve the problem the better we can. One major equipment at a time.
The Ts figure shows the output from the stack number 6 connected to number 1, this does not imply a closed cycle.
The stack gasses will not be sucked through the compressor because they are not combustible.
The connection to 1 and 6 imply they are connected to the outside air pressure and temperature.

Part (a).

Gas Turbine Plant.

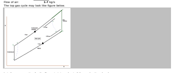
Compressor.

p1 1 bar
p2 8 bar — Compressor outlet pressure 8 bar, top cycle top of curve.
Air inlet temp **T1**: 298 K 25 C — Air inlet temperature at compressor.
γ air: 1.4 — Specific heat ratio of air
Equation: (T2/T1) = (p2/p1)^(γ-1)/(γη$_C$)
(γ-1)/γ air = 0.325
(p2/p1) = 8
(T2/T1) = 1.9642
T2 = 585.37 K — Temperature exiting compressor.
T2 = 312.37 C — In Celsius.

Combustor.

Pressure loss in CC 3%
Pressure loss: 0.24 bar
P3 ? 7.76 bar — Pressure entering GT? Is P3 after losses taken out at P2
Flow rate of combustion gas and fuel? Set to the following:
Flow rate gas **1** kg/s — 1 kg/s used in previous example OR Exergy problems for **per unit mass.**
Flow rate fuel **f** kg/s

In combustor fuel and air flowing through with gasses released. **Flow of fuel f + air = Flow rate of gas = 1 kg/s**
w(f + a) = 1, therefore wa = 1 - wf.
Flow of gas: 1 kg/s
Flow of air: **1-f** kg/s
The top gas cycle may look like figure below.



Lets form an equation for the flow rate into and out of the combustion chamber.
Apostrophe used to represent rate, example mass flow rate of gas m'-g.

f x CV (Calorific value of fuel) = m'-g (cp-g) (T3 - T4) - m'-f (cp-a) (T2 - T1)
f x CV (Calorific value of fuel) = **m'-g (cp-g) (T3 - T4) - (1 - f) (cp-a) (T2 - T1)**
Plugin and solve for f. Use deg C for temperature values since CV is in MJ/kg, no K unit here.
CV = 44.43 **MJ/kg** 44430 **kJ/kg**
T3: 900.00 C **T4-->T1:** 25 C — ***To is ambient condition, T1 = To. Inlet air temperature.***
T2: 312.37 C **T1:** 25 C — ***T4 above eventually is exhausted to To.***

(f)	44430	=	1	1.148	875.00	-	(1 - f)	1.006	287.37
	44430(f)	=		1004.500		-	(1 - f)		289.090
	44430(f)	=		1004.500		-	289.090	-	289.090(f)
	44430(f)	=		715.410		-		289.090(f)	
	44140.91(f)	=		715.410		-			
	f	=		0.0162	kg/s				

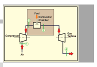
Air to Fuel Ratio? (1-f)/f. Correct!

Air Fuel Ratio = 60.70 ***Based on mass flow rate.***

Chemical reaction in the cumbustor is approximately: Octane + Oxygen = Carbon Dioxide + Water
$C_8H_{18} + O_2 = CO_2 + H_2O$.
This combustion reaction: $C_8H_{18} + 12.5O_2 = 8CO_2 + 9H_2O$.

Element	Atm Weight	Name	Mol
Carbon	12		
Hydrogen	1		
Oxygen	16		
C8H18	**114**	Octane	1
12.5 O2	**400**		12.5
8 CO2	352		8
9 H2O	42		9

See figure ---------------------------------->



Air contains 23.2% of oxygen **by mass**.
Proportion of oxygen in **1 kg of air**: 0.232 23.2%
Stoichiometric Air Fuel ratio = (Atomic weight of Air/ Atomic weight of Fuel) x (1/proportion of oxygen in 1 kg)
Stoichiometric Air Fuel Ratio = 15.124

Discussion-Explanation:

The chemical reaction during combustion has an output resulting with Stoichiometric Air to Fuel Ratio. The combustor performance based on the quantity of material quantity, fuel calorific value, and flow rates has an output for Air to Fuel Ratio.
The flow rate (process in combustor) compared to the chemical equation's Air to Fuel ratio shows a difference, that difference is excess air.

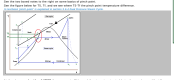
Air Fuel Ratio (mech) = 60.70 *Based on mechanical mass flow rate.*
Air Fuel Ratio (chem) = 15.12 *Based on chemical equation.*
Excess air 3.013 *((Air fuel mech) - (Air fuel chem)) / (air fuel chem)*
% Excess air 301% ***Percentage.***

Gas Turbine.

Pressure drop coming out of combustor 3%.
p3 = 7.76 bar — Pressure into gas turbine
Pressure drop in the HRSG 5kPa = 0.05bar. Pressure into compressor is 1 bar, so the pressure out of turbine has to rise above 1 bar. So it enters HRSG at 1.05, then drops within HRSG 0.05 bar, and compressor inlet P1 = 1 bar.
p4 = 1.05 bar — Pressure out of gas turbine line set to p4 at HRSG inlet pressure.
T3 = 1173 bar — Temperature into the gas turbine.
We need to calculate T3.
γ gas: 1.333 — Specific heat ratio of air
Equation: (T3/T4) = (p3/p4)^(γ-1)(η$_T$)/(γ)
(γ-1)(ηT)/(γ) = 0.220
(p4/p3) = 0.135
(T4/T3) = 0.6442
T4 = 755.67 K — Turbine exhaust gas temperature.
T4 = 482.67 C — Turbine exhaust gas temperature.

HRSG.

Set the pinch point temperature difference (DTp) to be 30 C, i.e. (T5 - Tf) = 30C.
DTp = 30 C
See the two boxed notes to the right on some basics of pinch point.
See the figure below for T5, Tf, and we see where T5-Tf the pinch point temperature difference.
In textbook 'pinch-point' is explained in section 2.6.4 Dual Pressure Steam Cycle.

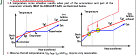


Basis of Pinch Analysis (Source: The Chemical Engineers Resource Page - Internet)

Pinch technology presents a simple methodology for systematically analysing chemical processes and the surrounding utility systems with the help of the First and Second Laws of Thermodynamics. The **First Law** of Thermodynamics provides the energy equation for calculating the enthalpy changes (H) in the streams passing through a heat exchanger. The **Second Law** determines the direction of heat flow. That is, heat energy may only flow in the direction of hot to cold. This prohibits *'temperature crossovers'* of the hot and cold stream profiles through the exchanger unit. In a heat exchanger unit neither a hot stream can be cooled below cold stream supply temperature nor a cold stream can be heated to a temperature more than the supply temperature of hot stream. In practice the hot stream can only be cooled to a temperature defined by the *'temperature approach'* of the heat exchanger. The temperature approach is the minimum allowable temperature difference (DTmin) in the stream temperature profiles, for the heat exchanger unit. The temperature level at which DTmin is observed in the process is referred to as ***"pinch point"*** or ***"pinch condition"***. The pinch defines the minimum driving force allowed in the exchanger unit.

Pinch point temperature difference, Δt_{pp}

- The pinch point temperature difference (Δt_{pp}) is the difference between the temperature of the exhaust exiting the evaporator (t_{g3}) and the temperature of water evaporation (t_s): $\Delta t_{pp} = t_{g3} - t_s$
- The water evaporation takes place at constant temperature (isothermal process) at the pressure prevailing in the drum.
- Without considering pressure losses in the water/steam tubes, the pressure in the boiler drum equals the pressure after the feed water pump.
- The pinch point temperature difference, Δt_{pp}, is usually between 5°C and 15°C. The lower the pinch point temperature difference, the higher the cost for the heat exchanger.
- On the other hand, the lower the pinch point temperature difference, the more heat is recovered from the flue gas to the water-to-steam process.
- The aim in introducing the pinch point temperature difference into HRSG designs is to avoid a temperature cross situation.
- A temperature cross situation results when part of the economizer and part of the evaporator virtually HEAT the EXHAUST GAS, as illustrated below.



- Observe that all temperatures, t_{g4}, t_{FW}, t_{g1} and t_{sh}, may be very reasonable.
- The temperature cross situation can occur if the engineer assumes a stack temperature for his/her calculations without checking the pinch point temperature difference.
- It is thus important for all HRSG designers to always base their calculations on the pinch point temperature difference and not on the stack temperature.

Source:

At the **steam cycle side of HRSG** the pressure is 40 bar and the steam is saturated (mix of vapour/gas and liquid).

SATURATED STEAM - PRESSURE TABLE

P bar	T °C	Sat. liq. v_f X1000	Sat. vap. v_g	Sat. liq. u_f	Sat. vap. u_g	Sat. liq. h_f	Sat. vap. h_g	Sat. liq. s_f	Sat. vap. s_g
40	250.4	1.252	0.050	1082	2602	1087	2801	2.796	6.070

See location of Tf in the TS figure above and that is the temperature found in the saturated table above.
Tf = **250.4** C
Pinch point temperature difference DTp = 30.
T5? Tf + DTp.
T5 = **280.4** C
We have found the temperature on the steam cycle side of the HRSG.

SUPERHEATED STEAM
v in m³/kg, u in kJ/kg, h in kJ/kg, s in kJ/(kg°K)
P= 40 bar

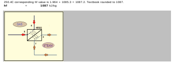
T	v	u	h	s
400	0.0734	2920	3213	6.769
450	0.08	3010	3330	6.936

Through interpolation the enthalpy at 425c at 40 bar is 3271.5. Textbook rounded to 3272.
ha = **3272.0** kJ/kg

SATURATED STEAM - TEMPERATURE TABLE

T °C	P bar	Spec. vol. m³/kg Sat. liq. v_f X1000	Sat. vap. v_g	Int. Ener. kJ/kg Sat. liq. u_f	Sat. vap. u_g	Enthalpy kJ/kg Sat. liq. h_f	Sat. vap. h_g	Entropy kJ/(kg°K) Sat. liq. s_f	Sat. vap. s_g
250	39.73	1.251	0.05	1080	2603	1085.3	2802	2.793	6.073
260	46.88	1.275	0.042	1128	2600	1134.4	2797	2.884	6.002

Notice on the saturated steam tables at 250C the pressure is 39.73 bar. Our pressure from d-a is 40 bar.
From 250 to 260 the pressure P bar increases and so does the enthalpy hf.
We choose the hf column because the working fluid coming from the pump into the HRSG is in liquid state.
250.4C corresponding hf value is 1.964 + 1085.3 = 1087.2. Textbook rounded to 1087.
hf = **1087** kJ/kg



Applying energy balance to the HRSG plant, shown above with 4-5, and f-a marked on the figure.
(w-g)(cp-g)(T4 - T5) = (w-s)(cp-a)(ha - hf)

Specific heat of air = 1.006 kJ/kg K
Specific heat of gas = 1.148 kJ/kg K
Flow rate of gas = 1.000 kg/s

What are we doing here? Trying to solve for the flow rate of steam, **(w-s)**, on the steam side in the HRSG plant.
(w-g)(cp-g)(T4 - T5) = (w-s)(ha - hf) .....Energy balance. Note: RHS enthalpy no cp.
1.000 1.148 202.272 = (w-s) 2185.0
Mass flow rate of steam **w-s**: = 0.1063 kg/s In HRSG Textbook rounded off to **0.106** kg/s

Next we need to solve for T6.

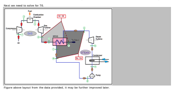


Figure above layout from the data provided, it may be further improved later.

From previous notes and examples, we know **m1' w-1 (T1 - T2) = m2' w-2 (Ta - Tb)** in general.
An energy balance need to be constructed to **capture T6.**
We have calculated T5, and from the Ts graph we can see how to construct this with the layout above.
If we can find the enthalpy **he** in the steam cycle then that would correspond to **h6** on the gas cycle.
Using the steam table calculations we may arrive to enthalpy **he**.
This means working through the steam cycle plant, similar to past examples from steam turbine through pump.
We are given Te in the problem data this is **170.4 deg C!** This would be saturated steam.
How do we find the enthalpy **he**?
he is between pump and HRSG
Te = **170.4** C **pe** = 40 bar
We know at 170.4 is past boiling point too low for superheated its saturated steam. Leaving pump mostly is liquid.

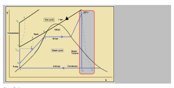
SATURATED STEAM - TEMPERATURE TABLE

T °C	P bar	Spec. vol. m³/kg Sat. liq. v_f X1000	Sat. vap. v_g	Int. Ener. kJ/kg Sat. liq. u_f	Sat. vap. u_g	Enthalpy kJ/kg Sat. liq. h_f	Sat. vap. h_g	Entropy kJ/(kg°K) Sat. liq. s_f	Sat. vap. s_g
170	7.916	1.114	0.243	718.3	2576	719.2	2768	2.042	6.663
180	10.02	1.127	0.194	762.1	2584	763.2	2778	2.140	6.586

he = 720.96 kJ/kg — Through interpolation of the table above.
he-txbook = 721.1 kJ/kg — Slight different value, method is the same.
Difference = *0.14 kJ/kg*
he = **721.1** kJ/kg — We will use the textbook value for he.

Allow for total heat transfer in the HRSG
This heat transfer results with the stack temperature T6.
T6 thus is the temperature of the gasses leaving the HRSG and entering the stack.
Applying energy balance between (T4 - T6) and (ha - he)
(w-g)(cp-g)(T4 - T6) = (w-s)(ha - he)
Notice the match between (T6 - T4) and (ha - he). So we have an **energy balance** here.
(w-g)(cp-g)(T4 - T6) = (w-s)(ha - he) ..... (RHS remember no cp-a on the enthalpy side).

1	**1.148**	**(482.67 - T6)**	=	**0.1063**	**2550.9**	2550.9
554.11	**-**	**1.148T6**	=	271.09		
	1.148 T6		=	283.02		
		T6	=	246.53 C		Correct.
		T6	=	247 C		*Textbook value.*



Steam Turbine.

Next we need to solve for the dryness of the steam in exiting the turbine into the condenser.
This will help us solve for the work produced of the turbine.
Turbine is adiabatic no heat added so the entropy should be the same down the vertical line.
The reason we have 2 entropies bs and b is because of the turbine efficiency.
Equating entropies sa = sbs. Dryness factor xbs at bs is (sa - sf)/(sfg). Then enthalpy at hbs = hf + (xbs)(hfg).

ha = 3272.0 kJ/kg
sa = 6.8525 kJ/kg K
sbs = sa
sbs-f = 0.423 kJ/kg K
sbs-fg = 8.052 kJ/kg K
xbs = 0.7985
hbs = hf + (xbs)(hfg)
hbs-f = 121.4 kJ/kg
hbs-fg = 2432.6 kJ/kg
hbs = 2063.82 kJ/kg

SUPERHEATED STEAM
v in m³/kg, u in kJ/kg, h in kJ/kg, s in kJ/(kg°K)
P= 40 bar

T	v	u	h	s
400	0.0734	2920	3213	6.769
450	0.08	3010	3330	6.936

SATURATED STEAM - PRESSURE TABLE

P bar	T °C	Spec. vol. m³/kg Sat. liq. v_f X1000	Sat. vap. v_g	Int. Ener. kJ/kg Sat. liq. u_f	Sat. vap. u_g	Enthalpy kJ/kg Sat. liq. h_f	Sat. vap. h_g	Entropy kJ/(kg°K) Sat. liq. s_f	Sat. vap. s_g
0.04	28.96	1.004	34.80	121.4	2415	121.4	2554	0.423	8.475

In the steam cycle there are plants doing work, steam turbine and pump, pump can be negligible compared to turbine.
So neglecting pump work, the steam turbine output is Wst = ws(ha - hbs)(η$_T$).
W-ST = **wst (ha - hbs) (ηT).**
wst 29.235 kg/s — Provided in problem data the steam flow rate in the steam turbine.
W-ST = 28,963 kW

Flow rates we have through the steps completed thus far:
1). Steam flow rate wst (through steam turbine in steam cycle)
2). In HRSG the steam flow rate ws.
3). Flow of gas set at 1 kg - per unit for supporting calculation in the combustion chamber.
4). Flow of air in gas cycle (calculated) f ie **wa**
So now we need to calculate the **mass flow rate of gas in the gas turbine.**

wg = wst / wa
wg = 275.1 kg/s — Mass flow rate of gas in gas turbine.

Compressor (gas cycle).

Air flow rate entering the compressor:
wa = (1 - wf)wg = (1 - f)wg

Flow of fuel f + air = Flow rate of gas = 1 kg/s
w(f + a) = 1, therefore wa = 1 - wf. These were done in the factor of flow rate of gas. Correct.
So now the actual flow rate of air is (1 - f)wg.
(1 - f) = 1 - 0.0162 = 0.9838
wa = 270.6 kg/s — This is the air flow rate entering the compressor.

Power Output (Gas and Steam Cycle).

Power output of Gas Turbine **W-GT = wg (cp-g)(T3 - T4) - wa (cp-a)(T2 - T1)**
Gas cycle power output = work done by steam turbine - work done by compressor.
Steam turbine generates work (pos), while compressor uses external energy to make it work (neg).

wg (cp-g)(T3 - T4) = 131,794 kW
wa (cp-a)(T2 - T1) = 78,237 kW
W-GT = 53,557 kW — Power output from the gas turbine cycle.

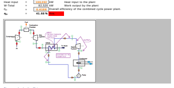
Total power output from the gas turbine W-Total= W-ST + W-GT.

W-ST = 28,963 kW
W-GT = 53,557 kW
W-Total = 82,520 **kW** Ans.

Fuel mass flow rate wf = f x wg = fuel rate per unit kg times mass flow rate of gas in the gas turbine.
wf = f x wg.
f = 0.0162 kg/s — Per unit kg calculation.
wg = 275.1 kg/s
wf = 4.459 kg/s

Efficiency of the **combined** cycle power plant.
The input to the combined cycle power plant is the heat supplied into the combustion chamber.
This heat is equivalent to the flow rate of fuel multiplied by the calorific value of the liquid Octane (C_8H_{18}).
Calorific value of liquid Octane in kJ/kg. **Heat input = wf x 44,430 kJ/kg.**

Heat input = 198,092 kW — Heat input to the plant
W-Total = 82,520 kW — Work output by the plant
η_O = 0.4166 — Overall efficiency of the combined cycle power plant.
η_O = **41.66%** Ans.



Steam cycle plant's efficiency.

See the figure above.
η_{st} = steam turbine output divided by the heat input = **((ha - hb)(Steam Turbine Eff- 0.82)) / (ha - he).**

η_{st} = 0.388
η_{st} = **38.8%** Ans.

Gas cycle plant's efficiency.
η_{gc} = **Total work produced by the gas cycle divided by the total heat generated in the combustor.**

η_{gc} = 53,557 / 198,092
η_{gc} = 0.2704
η_{gc} = **27.0%** Ans.

Comments: *As expected the steam cycle has higher efficiency than the gas cycle.*
*Ironic since we expect the **primary source cycle** to perform better than the **secondary receiving cycle**.*
*Secondary because the steam cycle receives its heat through a heat exchanger, which could be said in another way heat is **induced** in the secondary system (bottom cycle). May sound a little **electrical**. This for the gas-steam combination.*

Estimation of the heat loss coefficient x_L in the exhaust stack:
Note: This is a required calculation plus its a checking method on the previous overall efficiency.

x_L *is the heat loss cofficient. Refer to textbook on this topic for further indepth details.*
x_L: Heat Loss QL / (Heat input Q1).
QL the heat losses through the stack environment and Q1 the heat generated by the fuel flow mass.

QL = wg (cp-g) (T6 - To) .....where To is the environment temperature where the exhaust gas is released.
Q1 = Heat input into the gas cycle = wf multiplied by 'calorific value of Octane C8H18'.

To = 25 C — Ambient temperature out of stack.
(T6 -To) = 222 C
cp-g = 1.148
wg = 275.1
QL = 70,109 kW
Q1 = 198,092 kW
x_L = 0.354
x_L = **35.4%** Ans. — Heat loss coefficient estimation.

There are various overall efficiencies equations involving series and parallel combination cycles.
This plant layout based on the Ts figure is a series combination.
Figure below shows the GT-ST series combination. This helps interpreting the overall efficiency equation applied here.

The overall efficiency for this series combination of GT-ST cycles:
ηo = η1 + η2 - η1η2 -η2(xL).

η1	=	0.2704	efficiency of the gas cycle plant
η2	=	0.3884	efficiency of the steam cycle plant
xL	=	0.3539	lost heat coefficient
η_o	=	0.4163	Ans. Overall efficiency of the combined cycle power plant.
ηo	=	**41.63 %**	Ans. Overall efficiency of the combined cycle power plant.

The difference between the two methods is:

Δη₀	=	-0.03 %	Difference between (power output method) - (loss heat coefficient method).
Δη₀	=	0.03 %	Absolute value of the same above.

Comment: The results from both methods were very close since they differed by 0.03%.

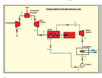
Part (b).

Exergy fluxes and irreversibilities (**Section 3.6.6 Exergy Analysis of Combined Cycle**):
See Exergy Notes as required.
Assume **Ψ = (ΔG₀ / ΔH₀) = 1.0401 + 0.1728 (h/c)**
where **(h/c)** is the mass ratio of hydrogen to carbon in the fuel.
ΔG₀ change in Gibbs function, and ΔH₀ is change in enthalpy.
Carbon C: 12
Hydrogen H: 1
Substitute C and H in (h/c) in equation above for C8H18. That is (1x18)/(18x1)
Ψ = 1.0401 + 0.1728 (0.188)
Ψ = 1.0725

We have the Helmholtz function F = U - TS, now we have the Gibbs function G defined by: = H - TS, and G = H - TS = U + pV - TS. Logically you would say may H = U + pV which is the basic expressions of enthalpy H! We have both enthalpy and entropy defining the Gibbs equation; G = H - TS. Therefore TΔS + G - H.

ΔH₀	=	wf x Net CV (NCV)	Here NCV is Fuel CV.
ΔH₀	=	198,092 kW	Note tutor: Ψ = (-ΔG₀) / (-ΔH₀) Negative sign cancels off.
ΔG₀	=	Ψ x ΔH₀	Actual individual term is with a negative sign.
ΔG₀	=	212,453 kW	
-ΔG₀	=	**212,453 kW**	Ans.
T₀ΔS₀	=	ΔG₀ - ΔH₀	Is equal to rate of exergy loss in combustion or reaction.
T₀ΔS₀	=	**14,362 kW**	Ans. Is the rate of exergy loss in combustion.

Next calculate the amounts of loss work or exergy destruction due to irreversibility in the major related plants:



Compressor.
Calculate: Loss Work OR Rate of Energy Dissipation OR Irreversibility.

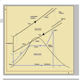
To	=	25	C	298	K
T1	=	25	C	298	K
T2	=	312.37	C	585.37	K
p1	=	1.00	bar		
p2	=	8.00	bar		
cp-a	=	1.006	kJ/kg K		
γ air	=	1.4			
Ra = cp-a(γ -1)/γ)					
wa	=	270.6	kg/s		

Ic = wa To(s2 - s1) ...with entropy the calculations use K instead of C please check units.
(s2 - s1) = cp-a(ln(T2/T1)) - **Ra** (ln(p2/p1))

Ra		0.2874
cp-a(ln(T2/T1))	=	0.6792
Ra (cp-a) (ln(p2/p1))	=	0.5977
(s2 - s1)	=	0.0815 kJ/kg K
Ic	=	**6573 kW** Ans. Irreversibility of compressor or exergy destruction in compressor.

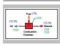
Combustor.
Calculate: Loss Work OR Rate of Energy Dissipation OR Irreversibility.

Since the equation used here are relatively involved. Notes are briefly presented to assist in understanding the solution.



Subscripts **P, R, A,** and **F** are **products, reactants, air,** and **fuel**.
I'comb = T₀ [(S₂)₃ - (S₁)₁] ... Outlet of combustor is the product subscript 3, and reactant at inlet is subscript 2.
Reactants R consist of fuel F and air A.
So we set,
(S₁)₁ = (S₁)₂ + (S₁)₂
Lets substitute into the equation for I'comb
I'comb = T₀ [(S₂)₃ - ((S₁)₂ + (S₁)₂)]

Reactants = Product.
S-reactants < S-product. Products at higher temperature.
C₈H₁₈ + O₂ = CO₂ + H₂O.
Fuel + Air = Gasses + Water.
Fuel at 0, and Air at 2 to 0 = Gasses at 3 to 0. Water vapour neglected.



Simple case, entropy of **gasses (product)** is ((S₂)₃ - (S₂)₂) + (S₂)₂.
The **net change** in entropy ((S₂)₃ - (S₂)₂) added to the **initial entropy** (S₂)₂ gives the final entropy at S3.
The initial entropy not canceled because air has CO2 from compressor passed through combustor added to the product.
Like wise for air it has net change and initial entropy, because air too contains CO2 at 0 and 2.
Lets say there are **two contributors** of CO2 to the product; **air** and the **combustor**.
So, main source of CO2 was from the combustion with additional CO2 passed through by air; (S₂)₂ and (S₁)₂.
Fuel a reactant exist at 0, and a product at 3, its processed in combustion between 2 and 3. So this is S(₁)₂.
You have a better explanation you are welcome to provide!
I'comb = To { [(S₂)₃ - (S₂)₂] + (S₂)₂ - [((S₁)₂ - (S₁)₂) + (S₁)₂ + (S₁)₂] }

ΔS₀ = (S₂)₃ - [(S₁)₂ + (S₁)₂] ...all at the initial condition or location in layout subscript 0.
By substitution and rearranging:
I'comb = To { [(S₂)₃ - (S₂)₂] - [(S₁)₂ - (S₁)₂] - Δ(S₀) }
I'comb = T₀ [wg (cp-g) ln(T₃/T₀) - wg (cp-g)(Rg) ln(p₃/p₀)] - [wg (cp-g) ln(T₂/T₀) - wa (cp-a)(Ra) ln(p₂/p₀)] + (ΔS₀) }
where T₀(ΔS₀) = ΔG₀ - ΔH₀. This was already calculated.
The third term wg (cp-g) ln(T₂/T₀), has wg because its subscript 2 ie entering the combustor that's gas flow.
The fourth term wa (Ra) ln(p₂/p₀), has wa because its subscript 0 ie entering compressor that's air flow.
Explanation on the I-combustor equation above provided in the notes below from Exergy Notes, specific to the pressure term in the square parenthesis.

A2 - A1 = E2 - E1 - p₀(V2 - V1) - T₀(S2 - S1).
A2 - A1 = { ∫(1 - (T₀/T_o) dQ) - [W₁₋₂ - p₀(V2 - V1)] - { T₀ S_gen }
(Δ Exergy) = (Exergy transfer w/heat) - (Exergy transfer w/work) - (exergy destruction)
Next the same equation in the form of a rate equation.
dA/T = Σⱼ [(1 - (T₀/Tⱼ)] dQ̇ⱼ - [W - p₀dV/dt)] - [İ]
where İ = T₀ S_gen
dA/T --> Rate of change of exergy.
Σⱼ [(1 - (T/T₀)] dQ --> Rate of energy transfer with heat Q at the boundary (i.e. σ) where the instantaneous temperature is Tⱼ
[W - p₀dV/dt)] --> Rate of exergy transfer as work where dV/ dT is the rate of change of system volume.
[İ] => Rate of exergy loss due to irreversibility (= T₀ S_gen).
For an isolated system, the exergy balance ΔA = -I.
Since I>0, the only processes allowed by the 2nd law are those for which the exergy of the isolated system decreases. In other words, the exergy of an isolated system can never increase. It is the counterpart of the entropy principle which status that the entropy of an isolated system can never decrease.
- From Basic and Applied Thermodynamics 2nd Ed., by P.K. Nag, Tata-McGrawHill.

There is no work done by the combustor so **Wcomb = 0.**

I'comb = T₀ [wg (cp-g) ln(T₃/T₀) - wg (cp-g)(Rg) ln(p₃/p₀)] - [wg (cp-g) ln(T₂/T₀) - wa (cp-a)(Ra) ln(p₂/p₀)] + (ΔS₀) }

Setup the variables data from above:

T3	=	1173 K
T2	=	585.37 K
T0	=	298.00 K
p3	=	7.76 bar
p2	=	8.00 bar
p0	=	1.00 bar
cp-a	=	1.006 kJ/kg K
cp-g	=	1.148 kJ/kg K
γ-a	=	1.40
γ-g	=	1.333
wa	=	270.6 kg/s
wg	=	275.1 kg/s
ηc	=	0.85
ηt	=	0.88
Ra	=	0.287 As textbook->(γ-1)/(γ)....no efficiency applied
Rg	=	0.250 As textbook->(γ-1)/(γ)....no efficiency applied

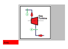
T0(wg (cp-g) ln(T3/T0))	=	128,952	
T0(wg (cp-g)(Rg) ln(p3/p0)	=	48,171	
Subtract	=	80,781	kW
T0(wa (cp-a) ln(T2/T0))	=	54,776	
T0(wa (cp-a)(Ra) ln(p2/p0)	=	48,492	
Subtract	=	-6,284	kW
T0 (ΔS0)	=	14,362	kW Calculated prior.
Total	=	88,858	kW
I'comb	=	**88,858 kW**	Ans.

Gas Turbine.

Rate of energy dissipation in the gas turbine or lost work.

I'GT = wg To (s4 - s3).
(s4 -s3) = cp-g ln (T4/T3) - cp-g Rg ln (p4/p3).

Values to input:



To	=	298.00	K
T4	=	755.67	K
T3	=	1173	K
p4	=	1.05	bar
p3	=	7.76	bar
cp-g	=	1.148	kJ/kg K
wg	=	275.1	kg/s
Rg	=	0.250	
(s4 -s3)	=	0.0688	kJ/kg K
I'GT	=	5643	kW Ans.

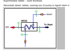
HRSG.

Rate of energy dissipation or lost work in the heat recovery steam generator.

I'HRSG = To ΔS = To[ws (sₐ - sₑ) + wg(s₆ -s₅)].

sₐ at 40 bar, and 425C = 6.853 kJ/kg k Superheated tables - used textbook.
sₑ at 0.04 bar, and 170.4C (liquid) = 2.046 kJ/kg k Saturated steam tables, coming out of pump in liquid state s-f, tabs.
sₐ - sₑ = 4.807 kJ/kg k

s₆ - s₅ = cp-g ln(T6/T4) - cpg Rg ln(p6/p4)

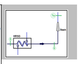


T6	=	246.53 C	519.53 K
p6	=	1.05 bar	
p4	=	1 bar	
cp-g ln(T6/T4)	=	-0.4301 kJ/kg K	
cpg Rg ln(p6/p4)	=	0.0140 kJ/kg K	
subtract	=	-0.4161 kJ/kg K	
s₆ - s₄	=	-0.4161 kJ/kg K	
ws	=	29.2350 kg/s	
To ws (sₐ - sₑ)	=	41879 kW	
To wg (s₆ -s₄)	=	-34114 kW	
I'HRSG	=	**7764** kW	Ans.

Steam Turbine.

[illegible]

[illegible]



[illegible]

To	=	298	K		25	C
T6	=	519.52	K		246.53	C
po	=	1	bar			
p6 = p1	=	1.05	bar			
(wg) (To)	=	81977	(cp-g)(T6 - To))	=	254	
(cp-g)(ln(T6/To)	=	0.6381	R(ex-gas)(ln(p6/po)	=	0.01219	
Stack(exhaust losses)	=	######	kW <-- WRONG	We are getting losses greater than input!		

Using either K or C scale the results were extremely huge - WRONG.
This formula was presented in a journal paper in 2011. It was writen: Applying the perfect gas behavior for the change in enthalpy and specific entropy from the exhaust state to the surroundings the rate of exergy loss with the exhaust gas is written as:
Stack(exhaust losses) = (mass gas flow rate) (To) [(cp-g(T6 - To)) - ((cp-g)(ln(T6/To) - R(ex-gas)(ln(p6/po)
So, this may feel or look accurate but it may be over emphasised!

Continuing to our textbook method.
Instead, we return to the textbook and inject the same starting idea of the impact from both enthalpy and entropy for the final exergy losses. Applying the equation below, part for **enthalpy** and **entropy** ie **1st and 2nd laws** respectively.

I'exhaust = wg cp-g [(T6 - To) - To ln (T6/To)].

wg cp-g (T6 - To)...for this term use the **Celcius scale** - 1st law (energy).
See previous calculations like the HRSG, one side gases the other steam.
wg cp-g (To ln (T6/To)) ...use the **Kelvin scale** - 2nd law (exergy). Both resulting in same units kW.

T6	=	246.53	C	519.53 K
To	=	25	C	298.00 K

wg cp-g (T6 - To)	=	69960 kW	
(wg cp-g To)ln (T6/To))	=	52309 kW	
I'exhaust	=	17651 kW	Ans. Textbook answer 17,760 kW.

Exergy Balance Tabulations.

The calculations made for exergy input, exergy losses, and power output (all in kW) are tabulated below.

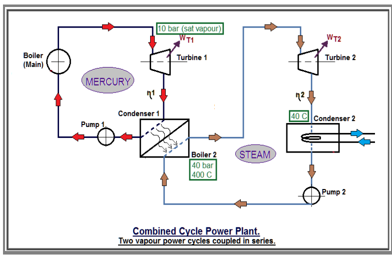
Exergy input (kW)		Power/Exergy output (kW)		Exergy losses/destruction (kW)		Losses numerical order by position	% Total Loss
Total rate of exergy input **-ΔGo**	212,453	W^s_{GT}	53,557	Compressor	6,573	4	5.0%
		W^s_{ST}	28,963	Combustor	88,858	1	66.9%
				Gas turbine	5,643	6	4.3%
				HRSG	7,764	3	5.8%
				Steam turbine	6,267	5	4.7%
				Exhaust gases	17,651	2	13.3%
						Total -->	100.0%
Total	212,453	Total	82,520	Total	132,757		

Power/Exergy output + Exergy losses (destruction)	215,277 kW	
Exergy input	212,453 kW	
% of Exergy output + losses OVER Exergy input	1.0133	101.33 %
Exergy output and losses is > Exergy input by	0.0133	**1.33 %** Ans.

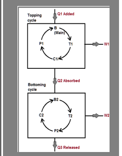
2nd Law Efficiency OR Exergetic Efficiency = (Exergy output) / (Exergy input).
η2nd law = 0.3884 **38.84 %** Ans.

End of a combined cycle power plant exergy analysis example. Any errors or omissions apologies in advance.

Source: Combined cycle power generation, Power Plant Engineering, 4th edition. PK Nag. TaTaMcGrawHill. Example Problems



Combined Cycle Power Plant.
Two vapour power cycles coupled in series.

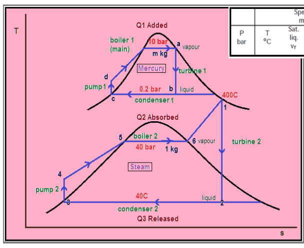


Problem 1:

A mercury cycle is superposed on the steam cycle operating between the boiler outlet condition at 40 bar, 400C and the condenser temperature of 40C. The heat released by mercury condensing at 0.2 bar is used to impart the latent heat of vaporisation to the water in the steam cycle. Mercury enters the mercury turbine as saturated vapour at 10 bar. Compute: (a) kg of mercury circulated per kg of water, and (b). The efficiency of the combined cycle.

The property values of saturated mercury are given below:

		kJ / kg		kJ / kg K		m^3 / kg	
p (bar)	t (deg C)	hf	hg	sf	sg	vf	vg
10	515.5	72.23	363	0.1478	0.5167	8.09E-05	0.0333
0.2	277.3	38.35	336.55	0.0967	0.6385	7.74E-05	1.163



		Spec. vol. m^3/kg		Int. Ener. kJ/kg		Enthalpy kJ/kg		Entropy kJ/(kg°K)	
P bar	T °C	Sat. liq. v_f	Sat. vap. v_g	Sat. liq. u_f	Sat. vap. u_g	Sat. liq. h_f	Sat. vap. h_g	Sat. liq. s_f	Sat. vap. s_g

Solution:

First apply the mercury and steam tables to get values of enthalpy and entropy.

Mercury cycle:

p-boiler1: 10 bar
T-boiler1: 515.5 C
h-a: 363 kJ/kg Vapour state
s-a: 0.5167 Vapour state enter T1 *Entropy vertical on T-s diagram; x- axis the same entropy.*
s-b: 0.5167 liquid state leaving T1 *The system is turbine and what enters and leaves has the same entropy.*
Change of state (vapour to liquid) so there is a drying factor x?
x b:? b is at 0.2 bar gas-vapour state, leaving the mercury turbine, on leaving there is expansion,.......
s-b = sf(liquid) +xb(sg(vapour/gas)-sf) ie = sf + xb(sfg).
b-sf = 0.0967
b-sg = 0.6385
0.5167 = 0.0967 + x-b 0.5418
x-b: = 0.7752
h-b = hf@0.2 bar + (x-b)(hg - hf)
hf-0.2bar = 38.35
hg-0.2bar = 336.55
h-b: = 269.51 kJ/kg
h-c: = 38.35 kJ/kg *h-c is the enthalpy at 0.2 bar liquid state, exiting the condenser is all liquid.*
P-mercury: Pump inlet pressure is 0.2 bar, outlet pressure is 10 bar.
The total enthalpy is that which enters the pump and that increment entropy created by the pump working on the fluid

Enthalpy at h-c (liquid ph sf): 38.35
Enthalphy inc. by condenser pump
Specific vol saturated liquid: 7.74E-05
Pressure difference: 9.8 bar
Efficiency of pump (ideal): 100
Increase in enthalpy: 7.59E-02
We assume no change in enthalpy since the change is so small at 0.075851
h-d: = 38.43 kJ/kg *Text book answer 38.35 neglecting pump work. Minor table differences - decimal plac*

Steam cycle:

p-boiler 2: 40 bar
T-boiler 2: 400 C

h1: 3213 kJ/kg
h1 is super heated steam at 400C.
h2 drop down to saturated steam.
There is a dry quality **x** involved, from 1 to 2.
s1: 6.769 kJ/kg
s2 = s1: 6.769 kJ/kg *In same turbine*
s2 = sf +(x-s2)(sg-sf)
sf - 40C: 0.5725
sg - 40C: 8.2570
x-s2: 0.8064

SUPERHEATED STEAM
v in m³-kg, u in kJ/kg, h in kJ/kg, s in kJ-(kg°K)
P= 40 bar

T	v	u	h	s
400	0.0734	2920	3213	6.769

SATURATED STEAM - TEMPERATURE TABLE

		Spec. vol. m^3-kg		Int. Ener. kJ/kg		Enthalpy kJ/kg		Entropy kJ-(kg°K)	
T °C	P bar	Sat. liq. v_f	Sat. vap. v_g	Sat. liq. u_f	Sat. vap. u_g	Sat. liq. h_f	Sat. vap. h_g	Sat. liq. s_f	Sat. vap. s_g
40	0.0738	1.0076	19.52	167.5	2430	167.5	2574	0.5725	8.257

Continuing to calculate h2? At 40C saturated steam temperature table.
hf-40C: 167.5
sg-40C: 2574
h2: 2108 kJ/kg h here is part liquid and vapour.
h3 at 40C: 167.50 kJ/kg h here is all liquid, exiting the condenser.
h4? Is coming out of the pump all liquid and with possible increase in enthalpy from the pump working on the fluid.
Enthalpy at h3 (liquid ph sf): 167.5
Enthalphy inc. by cond pump
Specific vol saturated liquid: 1.0076 Sat. Steam pressure at 40C: 0.0738
Pressure difference: 39.9262 *40 - 0.0738*
Efficiency of pump (ideal): 100
Increase in enthalpy: 4.02 Multiply by 10^-3 to the specific volume of saturated liquid vf
h4: 171.52 kJ/kg *Txtbk ans: 171.6 kJ/kg.*
See table to right for h5 and h7

h5: 1087 kJ/kg Liquid state entering boiler
h6: 2801 kJ/kg Vapour state leaving boiler

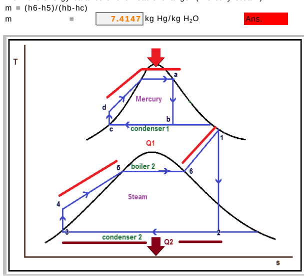
SATURATED STEAM - PRESSURE TABLE

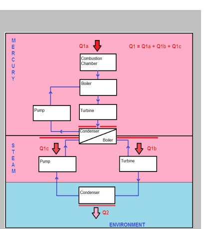
		Spec. vol. m^3-kg		Int. Ener. kJ/kg		Enthalpy kJ/kg		Entropy kJ-(kg°K)	
P bar	T °C	Sat. liq. v_f X1000	Sat. vap. v_g	Sat. liq. u_f	Sat. vap. u_g	Sat. liq. h_f	Sat. vap. h_g	Sat. liq. s_f	Sat. vap. s_g
40	250.4	1.252	0.050	1082	2602	1087	2801	2.796	6.070

Recap:

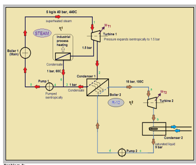
h-a	363
h-b	270
h-c	38.35
h-d	38.43
h1	3213
h2	2108
h3	167.5
h4	171.52
h5	1087
h6	2801

a).
Set m = the mass of mercury (top) circulated per kg of steam (bottom).
From the energy balance of the heat exchanger (mercury-steam):
m = (h6-h5)/(hb-hc)
m = 7.4147 kg Hg/kg H_2O Ans.





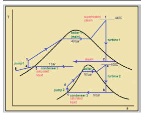
b).
Calculate the heat Q1 composed of the red path shown in figure above to the right:
Q1 = m kg(ha - hd) + 1kg(h1 - h6) + 1kg(h5 - h4)
Q1a = 2406.62 kJ/kg *Combustor Chamber highest source of heat energy.*
Q1b = 412.00 kJ/kg *Turbine lowest source of heat energy reflected by enthalpy value.*
Q1c = 915.48 kJ/kg *Pump the 2nd highest of heat this because of the work if does reflected in heat.*
Q1 = 3734 kJ/kg *Total heat supplied from the system to condenser, this is the input.*
Q2 = (h2 - h3)
Q2 = 1941 kJ/kg *Condenser expels heat to the environment in this case the output.*
Efficiency h of the combined cycle: 1 - (Q2/Q1).
Efficiency = 0.4803
Efficiency = **48 %** Ans.



Problem 2:

In a **cogeneration** binary cycle, superheated steam enters the turbine with a mass flow rate of **5 kg/s** at 40 bar, 440C, and expands isentropically to 1.5 bar.
Half of the flow is extracted at 1.5 bar and used for **industrial process heating.**
Remaining steam passes through a heat exchanger which serves as the boiler of the Refrigerant-12 cycle and the condenser for the steam cycle.
The condensate leaves the heat exchanger as saturated liquid at 1 bar, where it is combined with the return flow from the industrial process at 60C and 1 bar, before being pumped isentropically to the steam generator.
The Refrigerant-12 cycle is an ideal Rankine cycle with refrigerant entering the turbine at 16 bar, 100C and saturated liquid leaving the condenser at 9 bar.
Determine:
a). The rate of heat transfer in the steam generator.
b). The net power output of the binary cycle.
c). The rate of heat transfer to the industrial process.

Solution:



See figure above for steam and R-12 cycle.

Steam cycle:
Using the same method for calculating the values from tables in ExProblem 1, we will skip some of these steps.

h1	=	3307.1 kJ/kg	Superheated steam
s1	=	6.7690 kJ/kg K	Superheated steam
s2	=	6.7690 kJ/kg K	Dryness factor at s2, s2 = s1
h2	=	2570.8 kJ/kg	Steam (liquid+vapour), using dryness factor to calculate h2
h3	=	417.46 kJ/kg	hf at 1 bar saturated steam table
h6	=	251.21 kJ/kg	return flow from industrial heating at 60C entering pump1
hf-6:	=	251.130 kJ/kg	
vf-6:	=	1.0172E-03 m3/kg	vf x 10^-3. Specific volume of liquid water is very small as always.
p	=	1 bar	
p-60C	=	0.1994 bar	
Efficiency	=	100	Entering the pump and leaving the efficiency is 100% ideal case.

SATURATED STEAM - TEMPERATURE TABLE

T °C	P bar	Spec. vol. m³/kg Sat. liq. v_f X1000	Spec. vol. m³/kg Sat. vap. v_g	Int. Ener. kJ/kg Sat. liq. u_f	Int. Ener. kJ/kg Sat. vap. u_g	Enthalpy kJ/kg Sat. liq. h_f	Enthalpy kJ/kg Sat. vap. h_g	Entropy kJ/(kg°K) Sat. liq. s_f	Entropy kJ/(kg°K) Sat. vap. s_g
60	0.1994	1.017	7.671	251.1	2457	251.1	2610	0.8311	7.910

Piping 3 and 6 or streams 3 and 6 mix before entering the pump thru stream 4.
The mixing is at the same temperature 60C and no increase or added heat Q so this is adiabatic mixing.

Energy balance of streams 3 and 6:

m'3h3 + m'6h6 = (m'3 + m'6)h4 The symbol apostrophe ' here represents rate.
h4 is the entering enthalpy to the pump.
We need to solve for h4, the other variables are known.
Super heated steam flows at the rate of 5 kg/s into the turbine.
We assume this is the flow rate from the steam generator (main bolier 1)
This flow rate thus is set by pump 1
We are told the flow rate to the industrial process is half that to the turbine inlet.
Flow rates for h3 and h6 add to 5 kg/s and each stream is at half this rate at 2.5kg/s.

m'3	=	2.5	kg/s
m'6	=	2.5	kg/s
h3	=	417.46	kg/kJ
h6	=	251.21	kg/kJ
h4	=	334.34	kg/kJ

h5 calculate it similar to h6:
We are not given the temperature of stream 4 the mix of stream 3 and 6
We assume this temperatre is at **80C**. The mix of stream 3 at 60C and stream 6 at some temperature higher than 60C.

h5	=	338.349 kJ/kg	return flow industrial at 60C entering pump1
hf-4:	=	334.336 kJ/kg	hf at point 4.
vf-4:	=	1.0290E-03 m3/kg	vf x 10^-3. Specific volume of liquid water is very small always.
p-5	=	40 bar	
p-4	=	1 bar	
Efficiency	=	100	Entering the pump and leaving the efficiency is 100% ideal case.

SATURATED STEAM - TEMPERATURE TABLE

T °C	P bar	Spec. vol. m³/kg Sat. liq. v_f X1000	Spec. vol. m³/kg Sat. vap. v_g	Int. Ener. kJ/kg Sat. liq. u_f	Int. Ener. kJ/kg Sat. vap. u_g	Enthalpy kJ/kg Sat. liq. h_f	Enthalpy kJ/kg Sat. vap. h_g	Entropy kJ/(kg°K) Sat. liq. s_f	Entropy kJ/(kg°K) Sat. vap. s_g
80	0.4739	1.029	3.407	334.8	2482	334.9	2644	1.075	7.612

R-12 cycle:

P=1.5 MPa T °C	v m³/kg	h kJ/kg	u kJ/kg	s kJ/(kg·K)	P=2 MPa T °C	v m³/kg	h kJ/kg	u kJ/kg	s kJ/(kg·K)
100	0.014463	242.64	220.95	0.7722	100	0.010038	236.94	216.86	0.7408

Thru the table above by interpolation we get the values below at 16 Bar (1.6 MPa)

ha	=	241.58 kJ/kg	
s-a	=	0.7656 kJ/kg K	
s-b	=	0.7656 kJ/kg K	s-b = s-a.
hc	=	71.93 kJ/kg	Close enough to the R-12 table below at 9 bar (0.9MPa) for hc.

P_{sat} MPa	T °C	v_f m³/kg	v_g m³/kg	v_{fg} m³/kg	h_f kJ/kg	h_g kJ/kg	h_{fg} kJ/kg	u_f kJ/kg	u_g kJ/kg	u_{fg} kJ/kg	s_f kJ/(kg K)	s_g kJ/(kg K)	s_{fg} kJ/(kg K)
0.900	37.58	0.000792	0.019327	0.018536	72.10	202.36	130.25	71.39	184.96	113.57	0.2640	0.6832	0.4192

Next we need to calculate hb:
s-b = sf(liquid) +xb(sg(vapour/gas)-sf) ie = sf + xb(sfg).

sf-b	=	0.2640 kJ/kg K	
sg-b	=	0.6832 kJ/kg K	
x-b	=	1.197	

h-b = hf@9 bar + (x-b)(hg - hf)

hf-9bar	=	72.10 kJ/kg	
hg-9bar	=	202.36 kJ/kg	
h-b:	=	227.96 kJ/kg	Close to the textbook example, table values not shown in txbk only hb's value.
h-b:	=	229.43 kJ/kg	We use txtbook value here.

Next we need to calculate h-d:

h-d	=	72.654 kJ/kg	Txtbook value 72.48 close enough we use txtbook value.
hc-f:	=	72.100 kJ/kg	At 9 bar (0.9 Mpa) see table above. Enthalpy of c for the liquid (f).
hc-g:	=	202.360 kJ/kg	
vc-f:	=	0.000792 m3/kg	vf x 10^-3. Specific volume of liquid water is very small always.
p-a	=	16 bar	
p-b	=	9 bar	
Efficiency	=	100	Entering the pump and leaving the efficiency is 100% ideal case.
h-d:	=	72.48 kJ/kg	

a). Rate of heat transfer in the steam generator (main boiler 1):
Q' = m1'(h1 - h5) ...this expression is explained in the Exergy Example Problems Sheet.

m1'	=	5 kg/s	
Q'	=	14,844 kW	Ans.

b). Total power output:

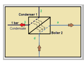
Network of the steam cycle:

Wnet = m-t'(h1 - h2) - m-p'(h5 - h4) = W-turbine - W-pump.
Mass flow rate in the turbine and pump is the same 5 kg/s, m-t' = m-p' = 5 kg/s.
Only work done is by the turbine and pump in the steam cycle.

W-turbine1	=	3681.5 kW	Work done by turbine generates electrical power (+)
W-pump1	=	20.07 kW	Pumping fluid consumes work (-)
W-net	=	3661.43 kW	

Network of the R-12 cycle:

We need to solve for the flow rate of R-12, so we use the energy balance expression.



m'3 (h2 - h3) = m'R-12(h-a - h-d)

m'R-12 = 31.84 kg/s

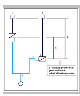
Now to work done by turbine (+) and pump 2 (-):
Wnet-R-12:
Wnet-R-12: m'-t2(ha - hb) - m'-p2(hd - hc) .....hd>hc since hd is after the fluid had work done on it.
m'-t2 = m'-p2 = m'R-12 = 31.84 kg/s

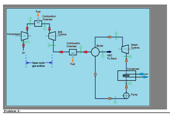
W-Turb2	=	386.80 kW
W-Pump2	=	17.51 kW
Wnet R-12:		369.29 kW

Total power output:
Work total = Net work stem cycle + Net work R-12 cycle.
Power output = 4030.72 kW Ans.

c). Rate of heat transfer to the industrial process (the objective of cogeneration power and heating here):
Q'Heating = m'-process (h2 - h6) .....see figure on process heating below.

m'process	=	2.5 kg/s	flow rate thru piping h6 is the flow leaving the industrial heating process.
Q'H	=	5799 kW	Ans. See figure below for h2 - h6 explanation.

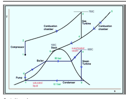




Problem 3:

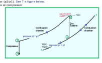
In a **combined gas turbine - steam turbine** power plant, the exhaust gas from the open cycle gas turbine is the supply gas to the steam generator of the steam cycle at which additional fuel is burned in the gas.
The **pressure ratio** for the **gas turbine is 7.5**, the air inlet temperature is 15 C and the maximum cycle temperature is 750C.
Combustion of additional fuel raises the gas temperature to 750C and the gas leaves the steam generator at 100C.
The steam is supplied to the turbine at **50 bar, 600C** and the condenser pressure is **0.1 bar**.
The total power output of the plant is 200 MW.
The **calorific value** of the **fuel burned** is **43.3 MJ/kg.**
Neglecting the effect of the mass flow rate of fuel on the air flow, determine:
a). The flow rates of air and steam required?
b). The power outputs of the gas turbine and steam turbine?
c). The thermal efficiency of the combined plant?
d). The air fuel ratio?
Take cp = 1.11 kJ/kg K and γ (Greek Gamma) = 1.33 for **combustion gasses**.
Take cp = 1.005 kJ/kg K and γ = 1.4 for **air**.
Neglect pump work.

Solution:



Gas turbine <u>cycle</u>:
<u>Compressor:</u>

p-ratio-GT:		7.5	Pressure ratio of gas turbine cycle same for compressor (p2/p1). See T-s figure below.
Air inlet temp T1:		15 C	288 K Air inlet temperature at compressor
γ air:		1.4	
Equation: (T2/T1) = (p2/p1)^(γ-1)/γ			
(γ-1)/γ air	=	0.286	
(p2/p1)	=	7.5	
T1	=	288 K	*See numbering in T-s figure at right.*
T2	=	512.165 K	Air outlet temperature at compressor



<u>Gas Turbine:</u>

Max GT cycle temp T3:		750 C	1023 K
γ combustion gas:		1.33	
Equation: (T3/T4) = (p3/p4)^(γ-1)/γ			
(γ-1)/γ gas	=	0.248	
(p4/p3)	=	7.5	See Ts figure, ratio of p2/p1 = p3/p4
T3	=	1023 K	*See numbering in T-s figure at right.*
T4	=	620.52 K	Gas outlet temperature at GT
T4		347.52 C	

Steam turbine <u>cycle</u>:

60 bar,600C	=	3658	kJ/kg	
40 bar,600C	=	3674	kJ/kg	
ha:	=	3666	Using the steam table below	
ha:	=	**3670**	We will use the textbook Mollier Chart approximate value presented for the solution.	

SUPERHEATED STEAM
v in m³/kg, u in kJ/kg, h in kJ/kg, s in kJ/(kg°K)

	P = 40 bar				P = 60 bar			
T	v	u	h	s	v	u	h	s
600	0.0988	3279	3674	7.369	0.0653	3267	3658	7.168

P 40 bar, s: 7.369 kJ/kg K
P 60 bar, s: 7.168 kJ/kg K
P 50 bar, s: 7.2685 kJ/kg K

SATURATED STEAM - PRESSURE TABLE

		Spec. vol. m³/kg		Int. Ener. kJ/kg		Enthalpy kJ/kg		Entropy kJ/(kg°K)	
P bar	T °C	Sat. liq. v_f X1000	Sat. vap. v_g	Sat. liq. u_f	Sat. vap. u_g	Sat. liq. h_f	Sat. vap. h_g	Sat. liq. s_f	Sat. vap. s_g
0.1	45.8	1.010	14.68	191.8	2438	191.8	2585	0.649	8.150

Using table above for s values.

7.2685	=	0.649	+ x 7.501
x dryness	=	0.8825	Apply this dryness or quality (x) for enthalpy calculation next.
hf:	191.8		
hg:	2585		
hb	=	2304	
hb	=	**2305**	Textbook has 2305 we will use the approximate Mollier chart instead.
hc:	=	191.8	Exiting the condenser entering the pump its all saturated liquid.
hc:	=	**192**	Textbook uses 192, we will too.
hd:	=	hc	Pump work is negligible so the enthalpy exiting the pump is same as entering.
hd:	=	**192**	kJ/kg

<u>Net work done in Gas Cycle:</u>

cp-g	=	1.11	kJ/kg K	Specific heat of gas
cp-a	=	1.005	kJ/kg K	Specific heat of air

Wnet(Gas cycle) = Work done by GT - Work done by Compressor
Wnet(Gcycle) = **m'air** (cp-g)(T3 - T4) - **m'air** (cp-a)(T2 - T1)
..apostrophe to represent flow rate, so mass flow rate of air **m'air**
Through the expression above we solve for mass flow rate of air (m'a). **The air contains the source of gasses for combustion.**

Wnet(Gcycle)	=	m'a 446.75	-	m'a 225.29
Wnet(Gcycle)	=	m'a 221.47	Gas cycle	

<u>Net work done in Steam Cycle:</u>

Wnet(Gas cycle) = Work done by GT - Work done by Compressor
Wnet(Steam cycle) = **m's** (ha - hb) - pump work is negligible 0.
Through the expression above we solve for mass flow rate of air m'steam.

Wnet(Scycle)	=	m's 1365.00	-	0
Wnet(Scycle)	=	m's **1365**	Steam cycle - Scycle.	

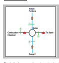
<u>Plant total work:</u>
The problem statement provides us with a plant total power output of 200MW.
The power plant's output is work done to provide electrical power to the value of 200MW.
So plant work output is equal to 200 MW.

Wnet Gas Cycle + Wnet Steam Cycle = 200 MW.

We use the expression above to set the equation below.

1365 m's + 221.47 m'a = 200 MW

<u>Boiler for steam:</u>



The boiler for generating steam is a heat exchanger.
Expression for the mass flow rate in the steam generator heat exchanger:
m'a (cp-g)(T5 - T6) = m's (cp-a)(ha - hd)

T5	=	750	C	1023	K
T6	=	100	C	373	K
(T5 - T6)	=	650	K		
(ha - hb)	=	3478	K		

(m'a)	**1.11**	650	=	**(m's) 3478**
(m'a)	721.5		=	**(m's) 3478**
		(m's) / (m'a)	=	0.2074
		(m's)	=	**(m'a) 0.2074**

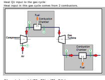
Returning to the net work steam cycle we substitute m's into the expression:

1365 m's		+	**221.47 m'a**	=	**200 MW**
1365	x	**0.2074 (m'a)**	+	**221.47 m'a**	= **200 MW**
504.63 (m'a)	=	**200 MW**			
(m'a)	=	0.39633 Mg/s			
(m'a)	=	396.33 **kg/s**	Mass flow rate of air	Ans.	
(m's)	=	0.2074	x	396.3278 kg/s	
(m's)	=	82.22 kg/s	Mass flow rate of steam	Ans.	

Returning to gas and steam turbine expressions:

Wnet(GTcycle)	=	m'a **221.47**	
Wnet(GTcycle)	=	87,773.87 kW	
Wnet(GTcycle)	=	87.77 MW	Ans.
Wnet(Scycle)	=	m's **1365**	
Wnet(Scycle)	=	####### kW	
Wnet(Scycle)	=	112.23 MW	Ans.

Heat Q1 input to the gas cycle:
Heat input in this gas cycle comes from 2 combustors.



Q1 = m'a (cp-g) ((T3 - T2) + (T5 - T4))

m'a (cp-g)	=	439.924
(T3 - T2)	=	510.835
(T5 - T4)	=	402.481
Q1	=	401789.44 kW
Q1	=	401.79 MW

Thermal efficiency of the combined power plant:
ηthermal = Work net / Q1.
Work net of the power plant is the electrical power output 200 MW.
Input heat to the combined power plant is Q1.

Output power	=	200.00	MW
Input heat Q1	=	401.79	MW
ηthermal	=	0.49777	
ηthermal	=	**50 %**	Ans.

<u>The air fuel ratio:</u>
The combustion chamber requires air and fuel for combustion.

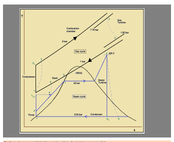
The calorific value of the fuel burned: **43.3** MJ/kg **43300** kJ/kg
Q1 the total heat input = mass flow rate of fuel m'f x calorific value of fuel.
m'f in other words is the fuel burning rate.
Q1 = m'f x (43.4 x 10^3) kJ/kg.

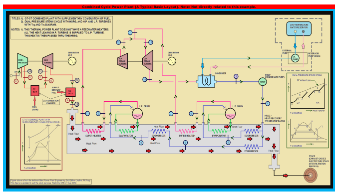
Fuel burning rate = Heat generated by gasses in the two combustion chambers. Remember: Air contains the source of gasses for combustion.
Q1 = (m'f) x (43.3 x 10^3) = (m'a) x (cp-g) x ((T3 - T2) + (T5 - T4))
(m'f) / (m'a) = [(cp-g) x ((T3 - T2) + (T5 - T4))] / [(43.3 x 10^3)]

(T3 - T2)	=	510.835
(T5 - T4)	=	402.481
Total	=	913.316

Air to fuel ratio:

(m'f) / (m'a)	=	42.71	Ans.





The figure above was provided in the example problem. No plant layout was provided.
We will work through each equipment's analysis, and then connect them to simple combined cycle process.
This will allow us to put together a simple plant layout for this example. A plant layout is highly suggested to solve the problem.

Problem 4: *Main example on combined cycle power generation.*

The following data refer to a combined cycle power plant:

Compressor			
Inlet condition of air to compressor	:		
- air pressure	:	1 bar	
- air temperature	:	298 K	25 C
Pressure ratio of compressor	:	8	
Combustion chamber			
Pressure drop in the combustion chamber	:	3% %	
Gas turbine			
Maximum gas temperature at inlet to the gas turbine	:	1173 K	900 C
HSRG			
Feedwater temperature to the HRSG	:	443.4 K	
Pressure drop of gas in the HRSG	:	0.05 bar	5 kPa
Steam turbine			
Condition of steam at inlet to the steam turbine			
- steam pressure	:	40 bar	
- steam temperature	:	698 K	425 C
Steam flow			
Steam flow rate	:	29.235 kg/s	
Specific heats			
Specific heat of air	:	1.006 kJ/kg K	
Specific heat ratio of air	:	1.4	
Specific heat of gas	:	1.148 kJ/kg K	
specific heat ratio of gas	:	1.333	
Efficiencies			
Efficiency of compressor	:	0.88	
Efficiency of gas turbine	:	0.88	
Efficiency of steam turbine	:	0.82	
Calorific value			
Calorific value of liquid octane (C_8H_{18}) used as fuel	:	44.43 MJ/kg	

Assume **$\Psi = (\Delta G_0 / \Delta H_0) = 1.0401 + 0.1728\ (h/c)$**
where **(h/c)** is the mass ratio of hydrogen to carbon in the fuel.

a). Determine the total power output and the overall efficiency of the combined cycle plant and estimate the "lost heat" coefficient x_L.

b). Give an energy balance of the plant and estimate the exergetic efficiency.

Solution:

Comments:
This problem does not provide a plant layout figure, we assume the given data is adequate to construct the layout.
There is only one combustion chamber, NOT two as in the previous example problem.
The textbook provides a Ts figure. We relate the Ts figure to the power plant process.
We will provide a figure for each major equipment.
We could assume the top cycle, (gas), is an open cycle. Ts shows 1 and 2 are connected. However, objective is to solve the problem the better we can. One major equipment at a time.
The Ts figure shows the output from the stack number 6 connected to number 1, this does not imply a closed cycle.
The stack gasses will not be sucked through the compressor because they are not combustible.
The connection to 1 and 6 imply they are connected to the outside air pressure and temperature.

Part (a).

Gas Turbine Plant.

Compressor.

p1		1 bar		
p2		8 bar	Compressor outlet pressure 8 bar, top cycle top of curve.	
Air inlet temp **T1**:		298 K	25 C	Air inlet temperature at compressor.
γ air:		1.4	Specific heat ratio of air	

Equation: (T2/T1) = (p2/p1)^(γ-1)/(γη_C)

(γ-1)/γ air	=	0.325	
(p2/p1)	=	8	
(T2/T1)	=	1.9642	
T2	=	585.37 K	Temperature exiting compressor.
T2	=	312.37 C	In Celsius.

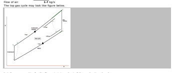
Combustor.

Pressure loss in CC		3%	
Pressure loss:		0.24 bar	
P3 ?		7.76 bar	Pressure entering GT? Is P3 after losses taken out at P2

Flow rate of combustion gas and fuel? Set to the following:

Flow rate gas	1 kg/s	1 kg/s used in previous example OR Exergy problems for **per unit mass.**
Flow rate fuel	f kg/s	

In combustor fuel and air flowing through with gasses released. **Flow of fuel f + air = Flow rate of gas = 1 kg/s**
w(f + a) = 1, therefore wa = 1 - wf.
Flow of gas: 1 kg/s
Flow of air: **1-f** kg/s
The top gas cycle may look like figure below.

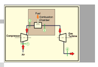


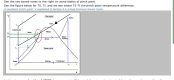
Lets form an equation for the flow rate into and out of the combustion chamber.
Apostrophe used to represent rate, example mass flow rate of gas m'-g.

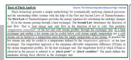
f x CV (Calorific value of fuel) = m'-g (cp-g) (T3 - T4) - m'-f (cp-a) (T2 - T1)
f x CV (Calorific value of fuel) = **m'-g (cp-g) (T3 - T4) - (1 - f) (cp-a) (T2 - T1)**
Plugin and solve for f. Use deg C for temperature values since CV is in MJ/kg, no K unit here.

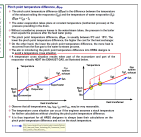
CV =	44.43 **MJ/kg**	**44430 kJ/kg**			
T3:	900.00 C	**T4-->T1:**	**25** C	*T0 is ambient condition, T1 = T0. Inlet air temperature.*	
T2:	312.37 C	**T1:**	**25** C	*T4 above eventually is exhausted to T0.*	

(f)	44430	=	1	1.148	875.00	-	(1 - f)	1.006	287.37
44430(f)		=		1004.500		-	(1 - f)		289.090
44430(f)		=		1004.500		-	289.090	-	289.090(f)









Through interpolation the enthalpy at 425c at 40 bar is 3271.5. Textbook rounded to 3272.

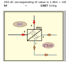
ha	=	**3272.0** kJ/kg

SATURATED STEAM - TEMPERATURE TABLE

T °C	P bar	Spec. vol. m³/kg Sat. liq. v_f X1000	Spec. vol. Sat. vap. v_g	Int. Ener. kJ/kg Sat. liq. u_f	Int. Ener. Sat. vap. u_g	Enthalpy kJ/kg Sat. liq. h_f	Enthalpy Sat. vap. h_g	Entropy kJ/(kg°K) Sat. liq. s_f	Entropy Sat. vap. s_g
250	39.73	1.251	0.05	1080	2603	1085.3	2802	2.790	6.073
260	46.88	1.275	0.042	1128	2600	1134.4	2797	2.884	6.002

Notice on the saturated steam tables at 250C the pressure is 39.73 bar. Our pressure from d-a is 40 bar.
From 250 to 260 the pressure P bar increases and so does the enthalpy hf.
We choose the hf column because the working fluid coming from the pump into the HRSG is in liquid state.
250.4C corresponding hf value is 1.964 + 1085.3 = 1087.2. Textbook rounded to 1087.

hf	=	**1087** kJ/kg



Applying energy balance to the HRSG plant, shown above with 4-5, and f-a marked on the figure.
(w-g)(cp-g)(T4 - T5) = (w-s)(cp-a)(ha - hf)

Specific heat of air	=	1.006 kJ/kg K
Specific heat of gas	=	1.148 kJ/kg K
Flow rate of gas	=	1.000 kg/s

What are we doing here? Trying to solve for the flow rate of steam, **(w-s)**, on the steam side in the HRSG plant.

	(w-g)(cp-g)(T4 - T5)		=	**(w-s)(ha - hf)** .....**Energy balance.** Note: RHS enthalpy no cp.				
1.000	1.148	202.272	=	(w-s)	2185.0			
Mass flow rate of steam **w-s:**			=	0.1063 kg/s		in HRSG	Textbook rounded off to	0.106 kg/s

Next we need to solve for T6.

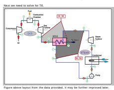


Figure above layout from the data provided, it may be further improved later.

From previous notes and examples, we know **m1' w-1 (T1 - T2) = m2' w-2 (Ta - Tb)** in general.
An energy balance need to be constructed to **capture T6.**
We have calculated T5, and from the Ts graph we can see how to construct this with the layout above.
If we can find the enthalpy **he** in the steam cycle then that would correspong to **h6** on the gas cycle.
Using the steam table calculations we may arrive to enthalpy **he**.
This means working through the steam cycle plant, similar to past examples from steam turbine through pump.
We are given Te in the problem data this is **170.4 deg C**! This would be saturated steam.
How do we find the enthalpy **he**?
he is between pump and HRSG

Te	=	**170.4** C	**pe**	=	40 bar

We know at 170.4 is past boiling point too low for superheated its saturated steam. Leaving pump mostly is liquid.

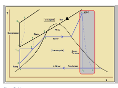
SATURATED STEAM - TEMPERATURE TABLE

T °C	P bar	Spec. vol. m³/kg Sat. liq. v_f X1000	Spec. vol. Sat. vap. v_g	Int. Ener. kJ/kg Sat. liq. u_f	Int. Ener. Sat. vap. u_g	Enthalpy kJ/kg Sat. liq. h_f	Enthalpy Sat. vap. h_g	Entropy kJ/(kg°K) Sat. liq. s_f	Entropy Sat. vap. s_g
170	7.916	1.114	0.243	718.3	2576	719.2	2769	2.042	6.666
180	10.02	1.127	0.194	762.1	2584	763.2	2778	2.140	6.586

he	=	720.96 kJ/kg	Through interpolation of the table above.
he-txbook	=	721.1 kJ/kg	Slight different value, method is the same.
Difference	=	*0.14 kJ/kg*	
he	=	**721.1** kJ/kg	We will use the textbook value for he.

Allow for total heat transfer in the HRSG
This heat transfer results with the stack temperature T6.
T6 thus is the temperature of the gasses leaving the HRSG and entering the stack.
Applying energy balance between (T4 - T6) and (ha - he)
(w-g)(cp-g)(T4 - T6) = (w-s)(ha - he)
Notice the match between (T6 - T4) and (ha - he). So we have an **energy balance** here.

	(w-g)(cp-g)(T4 - T6)		=	**(w-s)(ha - he)**		(RHS remember no cp-a on the enthalpy side).
1	**1.148**	(482.67 - T6)	=	**0.1063**	**2550.9**	2550.9
554.11	-	**1.148T6**	=	271.09		
	1.148 T6		=	283.02		
		T6	=	246.53 C		Correct.
		T6	=	**247** C		*Textbook value.*



Steam Turbine.

Next we need to solve for the dryness of the steam in exiting the turbine into the condenser.
This will help us solve for the work produced of the turbine.
Turbine is adiabatic no heat added so the entropy should be the same down the vertical line.
The reason we have 2 entropies bs and b is because of the turbine efficiency.
Equating entropies sa = sbs. Dryness factor xbs at bs is (sa - sf)/(sfg). Then enthalpy at hbs = hf + (xbs)(hfg).

ha	=	3272.0 kJ/kg
sa	=	6.8525 kJ/kg K
sbs = sa		
sbs-f	=	0.423 kJ/kg K
sbs-fg	=	8.052 kJ/kg K
xbs	=	0.7985
hbs = hf + (xbs)(hfg)		
hbs-f	=	121.4 kJ/kg
hbs-fg	=	2432.6 kJ/kg
hbs	=	2063.82 kJ/kg

SUPERHEATED STEAM
v in m³/kg, u in kJ/kg, h in kJ/kg, s in kJ/(kg°K)
P= 40 bar

T	v	u	h	s
400	0.0734	2920	3213	6.769
450	0.08	3010	3330	6.936

SATURATED STEAM - PRESSURE TABLE

P bar	T °C	Spec. vol. m³/kg Sat. liq. v_f X1000	Spec. vol. Sat. vap. v_g	Int. Ener. kJ/kg Sat. liq. u_f	Int. Ener. Sat. vap. u_g	Enthalpy kJ/kg Sat. liq. h_f	Enthalpy Sat. vap. h_g	Entropy kJ/(kg°K) Sat. liq. s_f	Entropy Sat. vap. s_g
0.04	28.96	1.004	34.80	121.4	2415	121.4	2554	0.423	8.475

In the steam cycle there are plants doing work, steam turbine and pump, pump can be negligible compared to turbine.
So neglecting pump work, the steam turbine output is Wst = ws(ha - hbs)(η_T).

W-ST	=	**wst (ha - hbs) (ηT).**	
wst		29.235 kg/s	Provided in problem data the steam flow rate in the steam turbine.
W-ST	=	28,963 kW	

Flow rates we have through the steps completed thus far:
1). Steam flow rate wst (through steam turbine in steam cycle)
2). In HRSG the steam flow rate ws.
3). Flow of gas set at 1 kg - per unit for supporting calculation in the combustion chamber.
4). Flow of air in gas cycle (calculated) f ie **wa**
So now we need to calculate the **mass flow rate of gas in the gas turbine.**

wg = wst / wa

wg	=	275.1 kg/s	Mass flow rate of gas in gas turbine.

Compressor (gas cycle).

Air flow rate entering the compressor:
wa = (1 - wf)wg = (1 - f)wg

Flow of fuel f + air = Flow rate of gas = 1 kg/s
w(f + a) = 1, therefore wa = 1 - wf. These were done in the factor of flow rate of gas. Correct.
So now the actual flow rate of air is (1 - f)wg.

(1 - f)	=	1	-	0.0162	=	0.9838
wa	=	270.6 kg/s	This is the air flow rate entering the compressor.			

Power Output (Gas and Steam Cycle).

Power output of Gas Turbine **W-GT = wg (cp-g)(T3 - T4) - wa (cp-a)(T2 - T1)**
Gas cycle power output = work done by steam turbine - work done by compressor.
Steam turbine generates work (pos), while compressor uses external energy to make it work (neg).

wg (cp-g)(T3 - T4)	=	131,794 kW	
wa (cp-a)(T2 - T1)	=	78,237 kW	
W-GT	=	53,557 kW	Power output from the gas turbine cycle.

Total power output from the gas turbine W-Total= W-ST + W-GT.

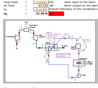
W-ST	=	28,963 kW	
W-GT	=	53,557 kW	
W-Total	=	**82,520 kW**	Ans.

Fuel mass flow rate wf = f x wg = fuel rate per unit kg times mass flow rate of gas in the gas turbine.
wf = f x wg.

f	=	0.0162 kg/s	Per unit kg calculation.
wg	=	275.1 kg/s	
wf	=	4.459 kg/s	

Efficiency of the **combined** cycle power plant.
The input to the combined cycle power plant is the heat supplied into the combustion chamber.
This heat is equivalent to the flow rate of fuel multiplied by the calorific value of the liquid Octane (C_8H_{18}).
Calorific value of liquid Octane in kJ/kg. **Heat input = wf x 44,430 kJ/kg.**

Heat input	=	198,092 kW	Heat input to the plant
W-Total	=	82,520 kW	Work output by the plant
η_O	=	0.4166	Overall efficiency of the combined cycle power plant.
η_O	=	**41.66 %**	Ans.



Steam cycle plant's efficiency.

See the figure above.
η_{st} = steam turbine output divided by the heat input = **((ha - hb)(Steam Turbine Eff- 0.82)) / (ha - he).**

η_{st}	=	0.388	
η_{st}	=	**38.8 %**	Ans.

Gas cycle plant's efficiency.
η_{gt} = **Total work produced by the gas cycle divided by the total heat generated in the combustor.**

η_{gt}	=	53,557	/	198,092
η_{gt}	=	0.2704		
η_{gt}	=	**27.0 %**	Ans.	

Comments: *As expected the steam cycle has higher efficiency than the gas cycle.*
*Ironic since we expect the **primary source cycle** to perform better than the **secondary receiving cycle**.*
*Secondary because the steam cycle receives its heat through a heat exchanger, which could be said in another way heat is **induced** in the secondary system (bottom cycle). May sound a little **electrical**. This for the gas-steam combination.*

Estimation of the heat loss coefficient x_L in the exhaust stack:
Note: This is a required calculation plus its a checking method on the previous overall efficiency.

x_L is the heat loss cofficient. *Refer to textbook on this topic for further indepth details.*
x_L: Heat Loss QL / (Heat input Q1).
QL the heat losses through the stack environment and Q1 the heat generated by the fuel flow mass.

QL = wg (cp-g) (T6 - To) .....where To is the environment temperature where the exhaust gas is released.
Q1 = Heat input into the gas cycle = wf multiplied by 'calorific value of Octane C8H18'.

To	=	25 C	Ambient temperature out of stack.
(T6 -To)	=	222 C	
cp-g	=	1.148	
wg	=	275.1	
QL	=	70,109 kW	
Q1	=	198,092 kW	
x_L	=	0.354	
x_L	=	**35.4 %**	Ans. Heat loss coefficient estimation.

There are various overall efficiencies equations involving series and parallel combination cycles.
This plant layout based on the Ts figure is a series combination.
Figure below shows the GT-ST series combination. This helps interpretating the overall efficiency equation applied here.

The overall efficiency for this series combination of GT-ST cycles:
ηo = η1 + η2 - η1η2 -η2(xL).

η1	=	0.2704	efficiency of the gas cycle plant
η2	=	0.3884	efficiency of the steam cycle plant
xL	=	0.3539	lost heat coefficient
ηo	=	0.4163	Ans. Overall efficiency of the combined cycle power plant.
ηo	=	**41.63 %**	Ans. Overall efficiency of the combined cycle power plant.

The difference between the two methods is:

Δηo	=	-0.03 %	Difference between (power output method) - (loss heat coefficient method).
Δηo	=	0.03 %	Absolute value of the same above.

Comment: The results from both methods were very close since they differed by 0.03%.

Part (b).

Exergy fluxes and irreversibilities **(Section 3.6.6 Exergy Analysis of Combined Cycle)**:
See Exergy Notes as required.
Assume **Ψ = (ΔG₀/ΔH₀) = 1.0401 + 0.1728 (h/c)**
where **(h/c)** is the mass ratio of hydrogen to carbon in the fuel.
ΔG_0 change in Gibbs function, and ΔH_0 is change in enthalpy.

Carbon C:	12
Hydrogen H:	1

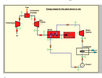
Substitute C and H in (h/c) in equation above for C8H18. That is (1x18)/(18x1)

Ψ = 1.0401 + 0.1728 (0.188)
Ψ = 1.0725

We have the Helmholtz function F = U - TS, now we have the Gibbs function G defined by: = H - TS,
and G = H - TS = U + pV - TS.
Logically you would say may H = U + pV which is the basic expressions of enthalpy H!
We have both enthalpy and entropy defining the Gibbs equation; G = H - TS. Therefore TΔS + G - H.

ΔH_o	=	wf x Net CV (NCV)	Here NCV is Fuel CV.
ΔH_o	=	198,092 kW	*Note txbk: Ψ = (-ΔG₀)/(-ΔH₀). Negative sign cancels off.*
ΔG_o	=	Ψ x ΔH₀	*Actual individual term is with a negative sign.*
ΔG_o	=	212,453 kW	
-ΔG₀	=	**212,453** kW	Ans.
$T_o\Delta S_o$	=	ΔG₀ - ΔH₀	Is equal to rate of exergy loss in combustion or reaction.
$T_o\Delta S_o$	=	**14,362** kW	Ans. Is the rate of exergy loss in combustion.

Next calculate the amounts of loss work or exergy destruction due to irreversibility in the major related plants:



Compressor.

Calculate: Loss Work OR Rate of Energy Dissipation OR Irreversibility.

To	=	25	C	298	K
T1	=	25	C	298	K
T2	=	312.37	C	585.37	K
p1	=	1.00	bar		
p2	=	8.00	bar		
cp-a	=	1.006	kJ/kg K		
γ air	=	1.4			

Ra = cp-a((γ -1)/γ)
wa = 270.6 kg/s

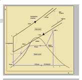
Ic = wa To(s2 - s1) ...with entropy the calculations use K instead of C please check units.
(s2 - s1) = cp-a(ln(T2/T1) - Ra (ln(p2/p1))

Ra	=	0.2874	
cp-a(ln(T2/T1))	=	0.6792	
Ra (cp-a) (ln(p2/p1))	=	0.5977	
(s2 - s1)	=	0.0815	kJ/kg K
Ic	=	**6573** kW	Ans. Irreversibility of compressor or exergy destruction in compressor.

Combustor.

Calculate: Loss Work OR Rate of Energy Dissipation OR Irreversibility.

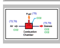
Since the equation used here are relatively involved. Notes are briefly presented to assist in understanding the solution.



Subscripts **P, R, A,** and **F** are **products, reactants, air, and fuel**.
$I'_{comb} = T_0 [(S_P)_3 - (S_R)_2]$ ... Outlet of combustor is the product subscript 3, and reactant at inlet is subscript 2.
Reactants R consist of fuel F and air A.
So we set,
$(S_R)_2 = (S_A)_2 + (S_F)_0$
Lets substitute into the equation for I'_{comb}.

$$I'_{comb} = T_0 [(S_P)_3 - ((S_A)_2 + (S_F)_0)]$$

Reactants → Product.
S-reactants < S-product. *Products at higher temperature.*
$C_8H_{18} + O_2 = CO_2 + H_2O$.
Fuel + Air = Gasses + Water.
Fuel at 0, and Air at 2 to 0 = Gasses at 3 to 0. *Water vapour neglected.*



Simple case, entropy of **gasses (product)** is $((S_p)_3 - (S_p)_0) + (S_p)_0$.
The **net change** in entropy $((S_p)_3 - (S_p)_0)$ added to the **initial entropy** $(S_p)_0$ gives the final entropy at S3.
The initial entropy not canceled because air has CO2 from compressor passed through combustor added to the product.
Like wise for air it has net change and initial entropy, because air has contains CO2 at 0 and 2.
Lets say there are **two contributors** of CO2 to the product; **air** and the **combustor**.
So, main source of CO2 was from the combustion with additional CO2 passed through by air; $(S_P)_0$ and $(S_A)_0$.
Fuel a reactant exist at 0, and a product at 3, its processed in combustion between 2 and 3. So this is $S_{(F)_0}$.
You have a better explanation you are welcome to provide!
$I'comb = T_0 \{ [(S_P)_3 - (S_P)_0] + (S_P)_0 - [((S_A)_2 - (S_A)_0) + (S_A)_0 + (S_F)_0] \}$

$\Delta S_0 = (S_P)_0 - [(S_F)_0 + (S_A)_0]$ ...all at the initial condition or location in layout subscript 0.

By substitution and rearranging:
$I'comb = T_0 \{ [(S_P)_3 - (S_P)_0] - [(S_A)_2 - (S_A)_0] , \Delta(S_0) \}$
$I'_{comb} = T_0 [\{ wg (cp\text{-}g) \ln(T_3/T_0) - wg (cp\text{-}g)(Rg) \ln(p_3/p_0) \} - \{ wg (cp\text{-}g) \ln(T_2/T_0) - wa (cp\text{-}a)(Ra) \ln(p_2/p_0) \} + (\Delta S_0)]$
where $T_0(\Delta S_0) = \Delta G_0 - \Delta H_0$. This was already calculated.
The third term **wg (cp-g) ln(T₂/T₀)**, *has* **wg** *because its subscript 2 ie entering the combustor that's gas flow.*
The fourth term **wa (Ra) ln(p₂/p₀)**, *has wa because its subscript 0 ie entering compressor that's air flow.*
Explanation on the I-combustor equation above provided in the notes below from Exergy Notes, specific to the pressure term in the square parenthesis.

A2 - A1 = E2 - E1 - p₀(V2 - V1) - T₀(S2 - S1).
A2 - A1 = { ∫₁²(1 - (T₀/T₀) δQ } - [W₍₁₂₎ - p₀(V2 - V1)] - { T₀ S_gen }
(Δ Exergy) = (Exergy transfer w/heat) - (Exergy transfer w/work) - (Exergy destruction)
Next the same equation in the form of a rate equation.
dA/T = Σⱼ [(1 - (T₀/Tⱼ)] dQⱼ - [W - p₀dV/dT)] - [İ]
where İ = T₀ S_gen
dA/T --> Rate of change of exergy.
Σⱼ[(1 - (T₀/Tⱼ)] dQ --> Rate of exergy transfer with heat Q at the boundary (i.e. y) where the instantaneous temperature is Tⱼ
[W - p₀dV/dT)] --> Rate of exergy transfer as work where dV/ dT is the rate of change of system volume.
[İ] --> Rate of exergy loss due to irreversibility (= T₀ S_gen).
For an isolated system, the exergy balance ΔA = - I.
Since I>0, the only processes allowed by the 2nd law are those for which the exergy of the isolated system decreases. In other words, the exergy of an isolated system can never increase. It is the counterpart of the entropy principle which states that the entropy of an isolated system can never decrease.
- From Basic and Applied Thermodynamics 2nd Ed., by P.K. Nag. Tata-McGrawHill.

There is no work done by the combustor so $W_{comb} = 0$.

$I'_{comb} = T_0 [\{ wg (cp\text{-}g) \ln(T_3/T_0) - wg (cp\text{-}g)(Rg) \ln(p_3/p_0) \} - \{ wa (cp\text{-}a) \ln(T_2/T_0) - wa (cp\text{-}a)(Ra) \ln(p_2/p_0) \} + (\Delta S_0)]$

Setup the variables data from above:

T3	=	1173 K	
T2	=	585.37 K	
T0	=	298.00 K	
p3	=	7.76 bar	
p2	=	8.00 bar	
p0	=	1.00 bar	
cp-a	=	1.006 kJ/kg K	
cp-g	=	1.148 kJ/kg K	
γ-a	=	1.40	
γ-g	=	1.333	
wa	=	270.6 kg/s	
wg	=	275.1 kg/s	
ηc	=	0.88	
ηt	=	0.88	
Ra	=	0.287	As textbook-->(γ-1)/(γ)...no efficiency applied
Rg	=	0.250	As textbook-->(γ-1)/(γ)...no efficiency applied

T0(wg (cp-g) ln(T3/T0))	=	128,952	
T0(wg (cp-g)(Rg) ln(p3/p0)	=	48,171	
Subtract	=	80,781	kW
T0(wa (cp-a) ln(T2/T0))	=	54,776	
T0(wa (cp-a)(Ra) ln(p2/p0))	=	48,492	
Subtract	=	6,284	kW
T0 (ΔS0)	=	14,362	kW Calculated prior.
Total	=	88,858	kW
I'comb	=	**88,858 kW**	Ans.

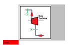
Gas Turbine.

Rate of energy dissipation in the gas turbine or lost work.

I'GT = wg To (s4 - s3).
(s4 -s3) = cp-g ln (T4/T3) - cp-g Rg ln (p4/p3).

Values to input:

To	=	298.00	K
T4	=	755.67	K
T3	=	1173	K
p4	=	1.05	bar
p3	=	7.76	bar
cp-g	=	1.148	kJ/kg K
wg	=	275.1	kg/s
Rg	=	0.250	



(s4 -s3)	=	0.0688	kJ/kg K
I'GT	=	5643	kW Ans.

HRSG.

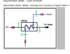
Rate of energy dissipation or lost work in the heat recovery steam generator.

I'HRSG = To ΔS = To[ws (sₐ - sₑ) + wg(s₆ -s₄)].

sₐ at 40 bar, and 425C	=	6.853 kJ/kg k	Superheated tables - used textbook.
sₑ at 0.04 bar, and 170.4C (liquid)	=	2.046 kJ/kg k	Saturated steam tables, coming out of pump in liquid state 6-f, table.
sₐ - sₑ	=	4.807	kJ/kg k

s₆ - s₄ = cp-g ln(T6 /T4) - cpg Rg ln(p6 /p4)

T6	=	246.53 C	519.53 K
p6	=	1.05 bar	
p4	=	1 bar	
cp-g ln(T6/T4)	=	-0.4301	kJ/kg K
cpg Rg ln(p6/p4)	=	0.0140	kJ/kg K
subtract	=	-0.4161	kJ/kg K
s₆ - s₄	=	-0.4161	kJ/kg K
ws	=	29.2350	kg/s
To ws (sₐ - sₑ)	=	41878	kW
To wg (s₆ -s₄)	=	-34114	kW
I'HRSG	=	7764	kW Ans.



Steam Turbine.

Rate of energy dissipation or lost work in the heat recovery steam generator.

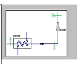
I'steam = ws (sb - sa)To.

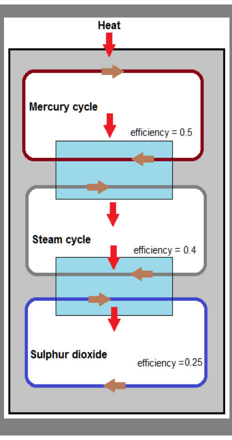
sₐ at 40 bar, and 425C	=	6.853 kJ/kg k	Superheated tables
How do we find sₕ? Follow these steps the expressions or equations you already know.			
hₐ at 40 bar, and 425C	=	3272 kJ/kg	Superheated tables
hₕ calculated prior	=	2063.82 kJ/kg	
hₐ - hₕ = ηₜ(hₐ - hₕₛ). See Ts figure.			
ηₜ	=	0.82	
ηT(ha - hbs)	=	990.7	
Find the dryness x?			
hₕ = hₐ - ηₜ(hₐ - hₕₛ).			
hₕ	=	2281	kJ/kg

SATURATED STEAM - PRESSURE TABLE

p bar	T °C	Spec. vol. m³/kg Sat. liq.	Spec. vol. Sat. vap.	Int. Ener. kJ/kg Sat. liq.	Int. Ener. Sat. vap.	Enthalpy kJ/kg Sat. liq.	Enthalpy Evap.	Enthalpy Sat. vap.	Entropy kJ/kg K Sat. liq.	Entropy Sat. vap.
0.04	28.96	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

From the steam table based on pressure above and the Ts figure to the top right,
hb = hbf + x-b (hg - hf).





Problem 5:

For a mercury-steam-sulphur dioxide cycle, the heat rejected in the mercury cycle is given to the steam cycle and the heat rejected by the steam cycle is utlised in the SO2 (sulphur dioxide) cycle.
If the efficiencies of the mercury, steam, and SO2 cycles are 0.5, 0.4, and 0.25, respectively, find the overall efficiency of the composite cycle.

Solution:

From the notes of the textbook Power Plant Engineering 4th edition, the efficiency for multiple series combined cycle power plant is given as,
$\eta = 1 - (1 - \eta_1)(1 - \eta_2)(1 - \eta_3)$.

η_1	=	0.50 mercury
η_2	=	0.40 steam
η_3	=	0.25 sulphur dioxide
η	=	0.775
η	=	**77.5%** Ans.

End of chapter 3 examples on combined cycle power generation.

Textbook: Power Plant Engineering. 4th edition. PK Nag. TaTaMcGrawHill.

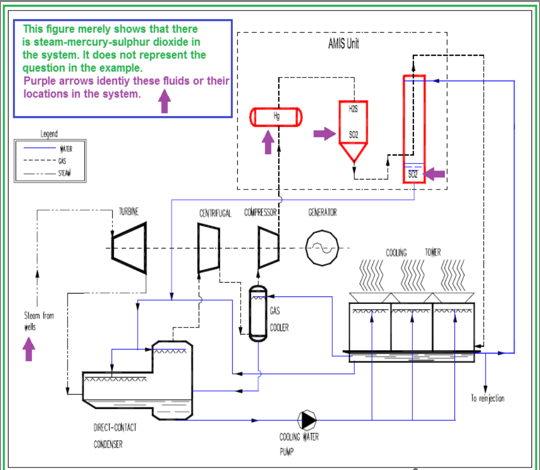


Figure 2. Simplified scheme of a standard geothermal power plant equipped with AMIS® abatement system

Greening of Geothermal Power: An Innovative Technology for Abatement of Hydrogen Sulphide and Mercury Emission

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Proceedings World Geothermal Congress 2005
Antalya, Turkey, 24-29 April 2005