POWER PLANT ENGINEERING

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4. Gas Turbine Power Plants: Cycles and Materials

The aim of the lecture is get acquainted with the basic thermodynamic cycles of the gas turbine power plants

4. Gas Turbine Power Plants: Cycles and Materials

4.1 Overview of Gas Turbine Power Plants

Gas turbines have these advantages over steam plants:

- They are small in size, mass, and initial cost per unit output.
- Their delivery time is relatively short and they can be installed quickly.
- They are quick starting (as low as 10 s), often by remote control.
- They are smooth running
- They can use a wide variety of liquid and gaseous fuels including gasified coal and synthetic fuels
- They are subject to fewer environmental restrictions than other prime movers.

4.2 Classification of Gas Turbine Power Plants

Classification according to the design:

1. Industrial heavy-duty gas turbines:
   - used in electric utilities to deliver base load power
   - Pressure ratio ranges from 5:1 earlier units to 15 to 35:1 for modern units
   - Inlet temperature to the turbine reaches as high as 1371 ^0C
   - Rated range 3MW-480 MW with efficiency of 30-46%
   - Usually employ axial flow compressors and turbines
   - Advantages:
     - High availability
     - Long life
     - higher efficiencies
     - Significantly lower noise levels than aircraft-type gas turbines

3. Gas Tubine Power Plants: Cycles

2. Aircraft-derivative gas turbines:

- Mainly used in the gas transmission industry and as peaking units in power plants.
- Rated range from 2.5MW-50MW with efficiency of 35-45%
- Advantages:
  - Relatively low installation cost
  - They can easily be operated unattended by remote control

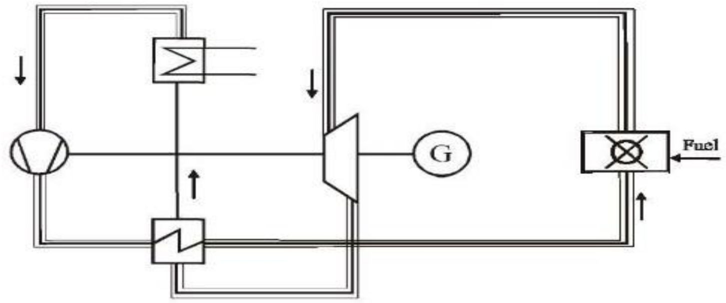
3. Medium-range gas turbines:

- These units are normally used on offshore platforms and in petrochemical plants for compressor drive trains
- Rated range 2.5MW-15MW with efficiencies usually less than 30%

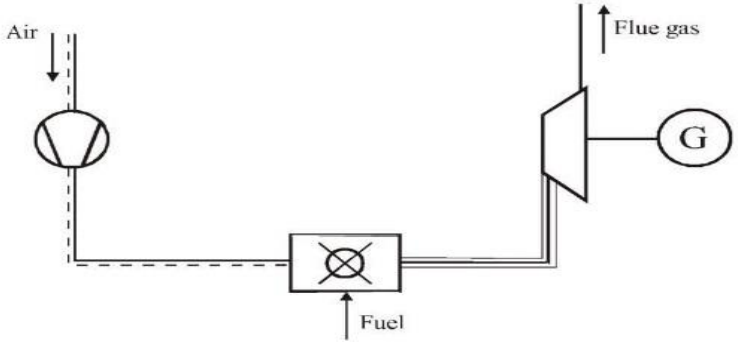
4. Small gas turbines:

- Rated below 2.5 MW with efficiency less than 25%
- They usually used centrifugal compressors and radial inflow turbines

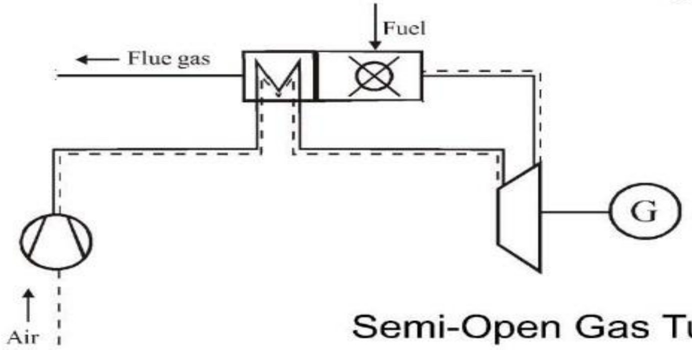
Another classification **can be based on the circuit of the working fluid**



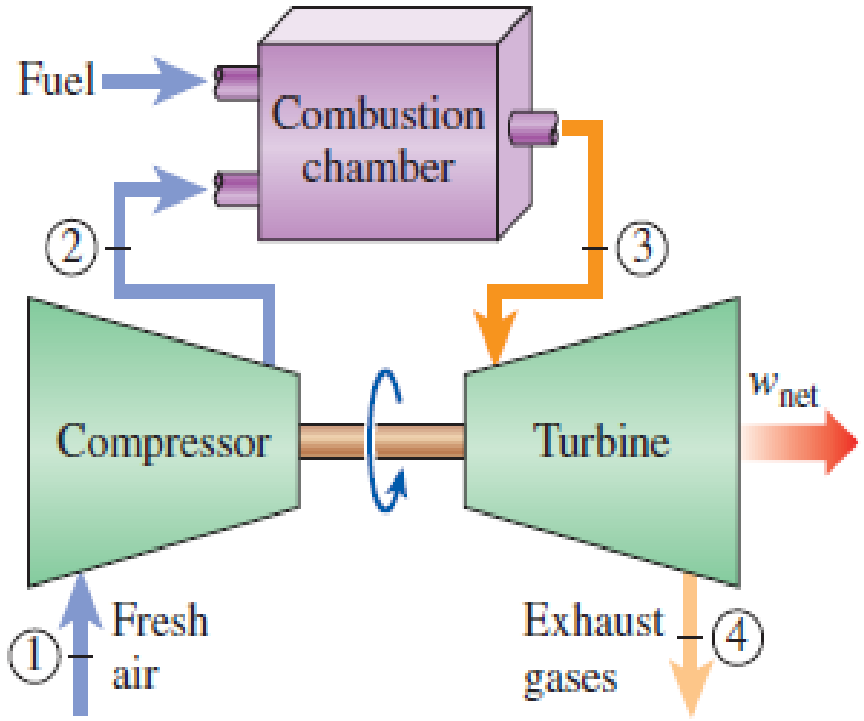
Closed Gas Turbine Cycle



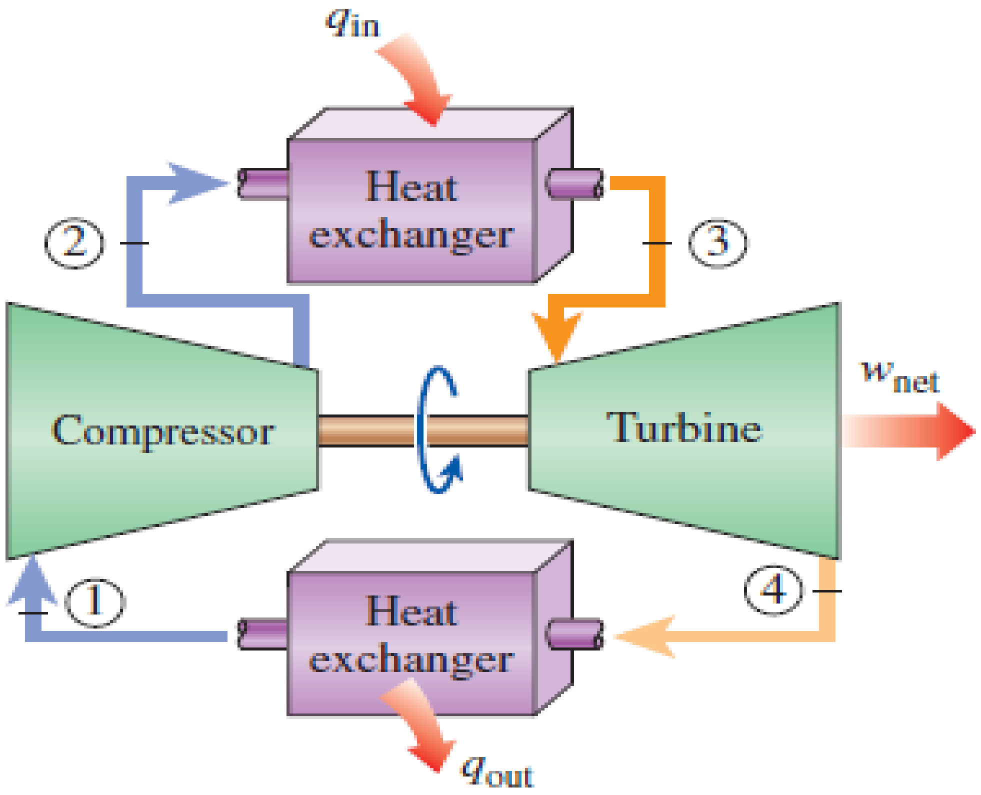
Open Gas Turbine **Cycle**



Semi-Open Gas Turbine **Cycle**



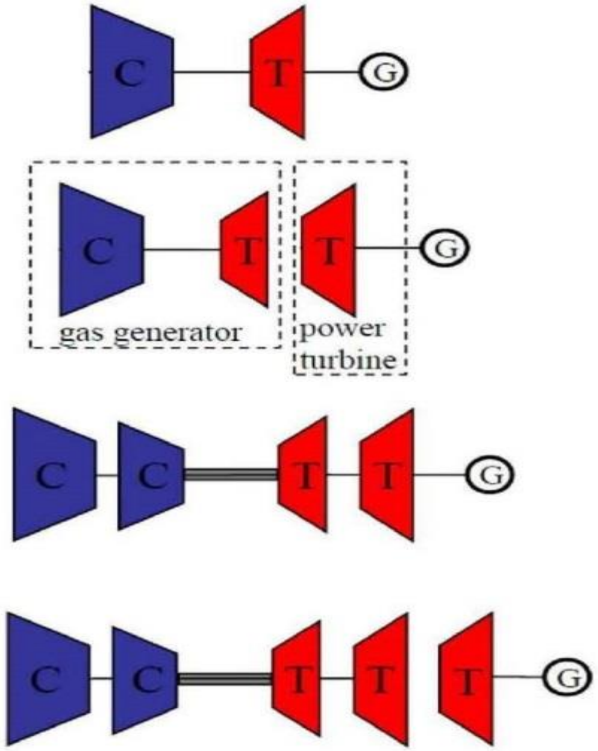
An open-cycle gas-turbine engine.



A closed-cycle gas-turbine engine.

3. Gas Tubine Power Plants: Cycles

Classification based on the number of shafts



1-shaft gas turbine (all GTs > 100 MW but also smaller)

2-shaft gas turbine

2-shaft gas turbine

3-shaft gas turbine

4.3 The Gas Turbine Power Cycle

- The actual gas power cycles are rather complex. To reduce the analysis to a manageable level, we utilize the following approximations, commonly known as the **air-standard** assumptions:

   1. The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.
   2. All the processes that make up the cycle are internally reversible.
   3. The combustion process is replaced by a heat-addition process from an external source.
   4. The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.
   5. air has constant specific heats whose values are determined at *room temperature* (25°C)

- A cycle for which the air-standard assumptions are applicable is frequently as an **air-standard cycle**.

Development of Gas Turbines

The gas turbine has experienced phenomenal progress and growth since its first successful development in the 1930s. The early gas turbines built in the 1940s and even 1950s had simple-cycle efficiencies of about 17 percent because of the low compressor and turbine efficiencies and low turbine inlet temperatures due to metallurgical limitations of those times. Therefore, gas turbines found only limited use despite their versatility and their ability to burn a variety of fuels. The efforts to improve the cycle efficiency concentrated in three areas:

Development of Gas Turbines

1. **Increasing the turbine inlet (or firing) temperatures**

This has been the primary approach taken to improve gas-turbine efficiency. The turbine inlet temperatures have increased steadily from about 540 C (1000 F) in the 1940s to 1425 C (2600 F) and even higher today. These increases were made possible by the development of new materials and the innovative cooling techniques for the critical components such as coating the turbine blades with ceramic layers and cooling the blades with the discharge air from the compressor. Maintaining high turbine inlet temperatures with an air-cooling technique requires the combustion temperature to be higher to compensate for the cooling effect of the cooling air. However, higher combustion temperatures increase the amount of nitrogen oxides (NO_x), which are responsible for the formation of ozone at ground level and smog. Using steam as the coolant allowed an increase in the turbine inlet temperatures by 200 F without an increase in the combustion temperature. Steam is also a much more effective heat transfer medium than air.

Development of Gas Turbines

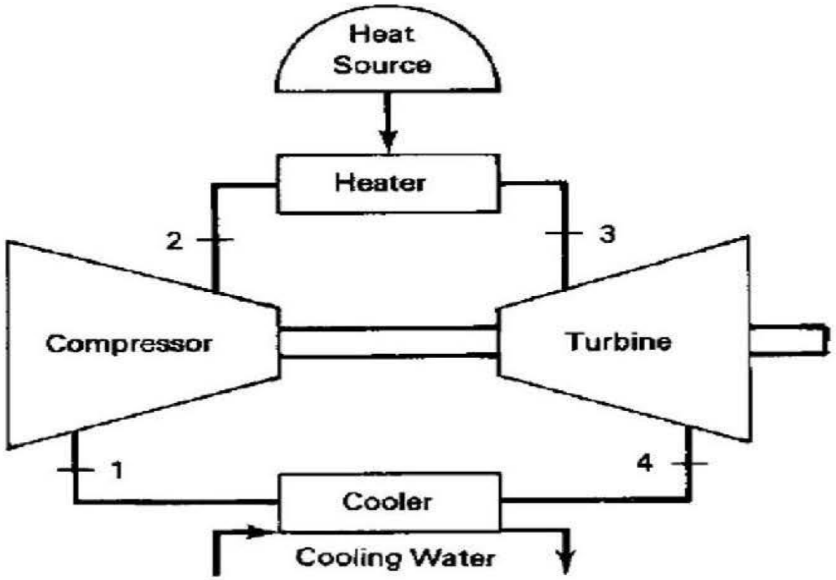
2. Increasing the efficiencies of turbo machinery components

The performance of early turbines suffered greatly from the inefficiencies of turbines and compressors. However, the advent of computers and advanced techniques for computer-aided design made it possible to design these components aerodynamically with minimal losses. The increased efficiencies of the turbines and compressors resulted in a significant increase in the cycle efficiency.

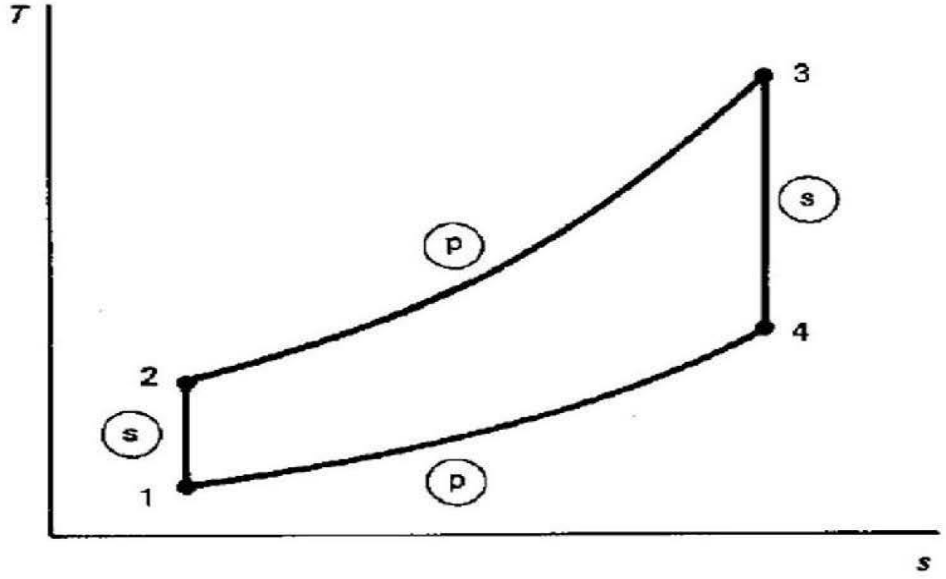
Development of Gas Turbines

3. Adding modifications to the basic cycle The simple-cycle efficiencies of early gas turbines were practically doubled by incorporating intercooling, regeneration (or recuperation), and reheating. These improvements, of course, come at the expense of increased initial and operation costs, and they cannot be justified unless the decrease in fuel costs offsets the increase in other costs. The relatively low fuel prices, the general desire in the industry to minimize installation costs, and the tremendous increase in the simple-cycle efficiency to about 40 percent left little desire for opting for these modifications.

The Gas Turbine Power Cycle (Joule-Brayton cycle)

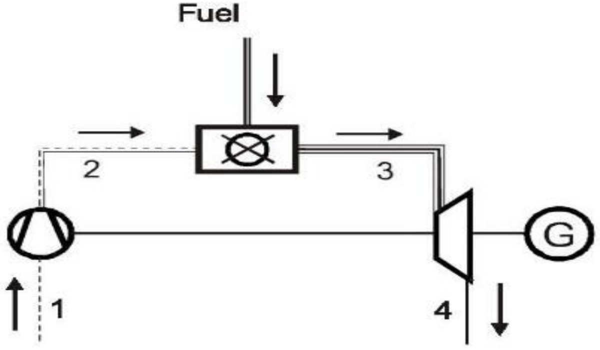


(a) Flow Diagram

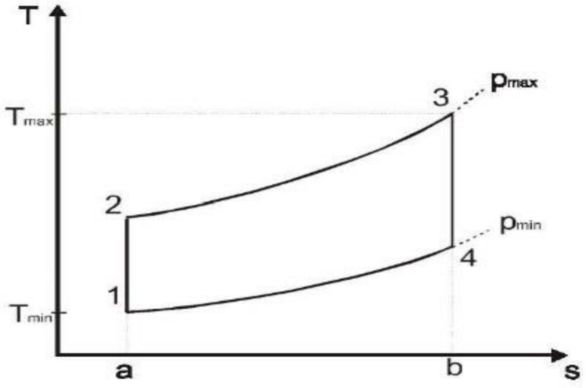


(b) Temperature-Entropy Diagram

4.4 The Open-Circuit Brayton Process



Heat cycle diagram



T-s diagram

The cycle consists of the following process:

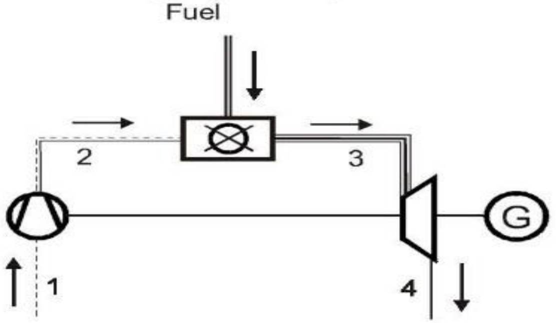
1 → 2 Air is drawn at atmospheric pressure (state 1) is compressed isentropically in the compressor

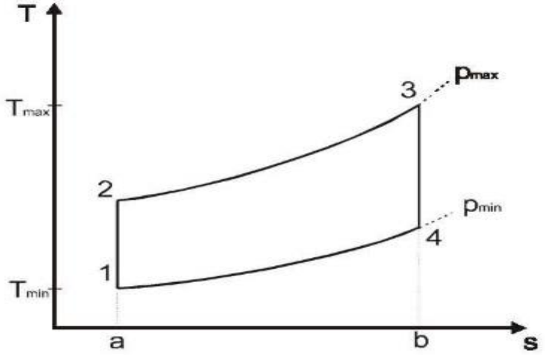
2 → 3 Constant pressure (isobaric) burning of the injected fuel spray in the combustion chamber to provide high pressure and high temperature air at state point 3

3 → 4 Isentropic expansion of the products of combustion in the gas turbine providing the energy output to drive the compressor anc the coupled generator

4.5 Analysis of the Joule-Brayton cycle

The Joule-Brayton cycle
- is the basic thermodynamic cycle of gas turbine power plants
- the ideal cycle consists of:
  - two reversible aidabatic (isentropic)proce sses
  - two constant pressure processes



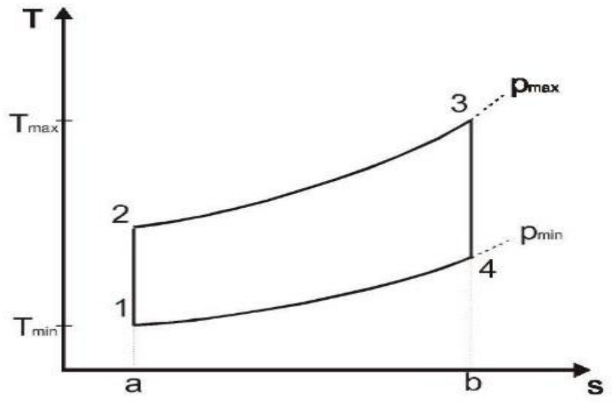


The ideal Joule-Brayton cycle consists of:

1 → 2	isentropic compression in the compressor
2 → 3	isobaric heat supply in the combustion chamber
3 → 4	isentropic expansion in the expansion turbine
4 →1	isobaric heat dissipatic n to the atmosphere

3. Gas Tubine Power Plants: Cycles

Cycle analysis



max. pressure ratio $r_p = \frac{p_{\max}}{p_{\min}} = \frac{p_3}{p_4} = \frac{p_2}{p_1}$

max. temperature ratio $\tau = \frac{T_{\max}}{T_{\min}} = \frac{T_3}{T_1}$

Isentropic compression or expansion of an ideal gas
(a polytropic process with k=1.4)

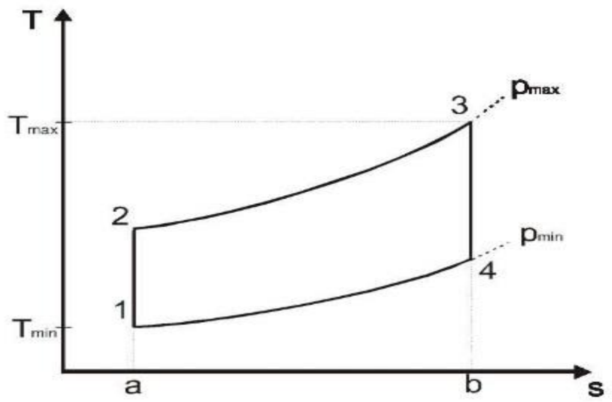
$$p \cdot v^{\kappa} = \text{const.} \quad \text{and} \quad \kappa = \frac{c_p}{c_v}$$

Thus, $$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\kappa-1}{\kappa}} = \frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\kappa-1}{\kappa}}$$

or $$\frac{p_3}{p_1} = r_p = \left(\frac{T_3}{T_1}\right)^{k/k-1} = (\tau)^{k/k-1}$$

Hence, $$\frac{T_4}{T_1} = \frac{T_3}{T_2} \quad or \quad \frac{T_1}{T_4} = \frac{T_2}{T_3}$$

3. Gas Tubine Power Plants: Cycles



The ***specific work output*** from the turbine:

$$w_t = h_3 - h_4 = c_p(T_3 - T_4)$$

The ***specific work input*** to the compressor:

$$w_c = h_2 - h_1 = c_p(T_2 - T_1)$$

The quantity of ***heat suplied*** to the cycle:

$$q_{in} = q_{23} = h_3 - h_2 = c_p(T_3 - T_2)$$

The quantity of ***heat rejected*** by the cycle:

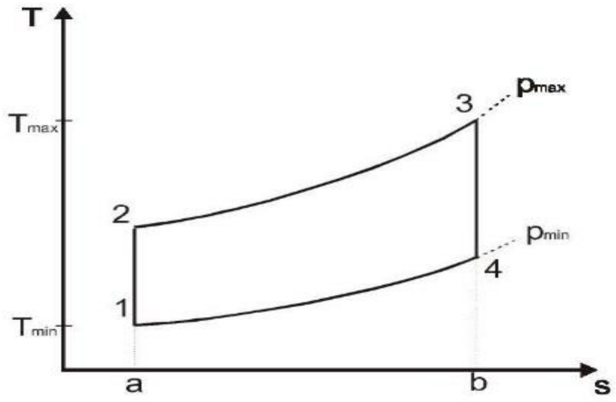
$$q_{out} = q_{41} = h_4 - h_1 = c_p(T_4 - T_1)$$

The ***net work output*** of the cycle:

$$w_{net} = w_t - w_c = c_p\,[(T_3\text{-}T_4)\text{-}(T_2\text{-}T_1)]$$

3. Gas Tubine Power Plants: Cycles

Now the ***Thermal Efficiency*** is given by:



$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{w_t - w_C}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{\left(\frac{T_4}{T_1} - 1\right)T_1}{\left(\frac{T_3}{T_2} - 1\right)T_2}$$

with $\frac{T_4}{T_1} = \frac{T_3}{T_2}$ and from the isentropic relations,

we can rewrite the *efficiency* as:

$$\eta_{th} = 1 - \frac{T_1}{T_2} = 1 - \frac{T_4}{T_3} = 1 - \left(\frac{p_1}{p_2}\right)^{\frac{\kappa-1}{\kappa}} = 1 - \left(\frac{p_4}{p_3}\right)^{\frac{\kappa-1}{\kappa}} = 1 - \left(\frac{1}{r_p}\right)^{\frac{\kappa-1}{\kappa}}$$

Thus, the thermal efficiency of the Brayton cycle is a function of the isentropic pressure ratio. And the higher the pressure ratio the higher the efficiency of the ideal cycle. However, the maximum pressure ratio and therefore the maximum temperature that can be attained in a Brayton cycle is fixed by material considerations.

The ***work ratio*** is defined as the ratio of the net work output to the turbine work

$$work\ ratio = \frac{w_{net}}{w_t} = \frac{w_t - w_c}{w_t} = 1 - \frac{w_c}{w_t}$$

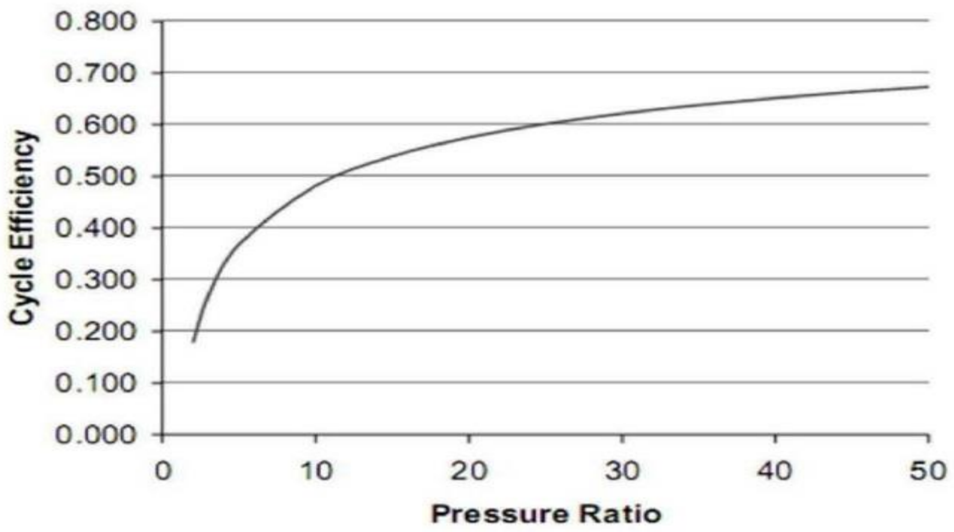
$$= 1 - \left(\frac{T_2 - T_1}{T_3 - T_4}\right) = 1 - \frac{T_1}{T_4}\left(\frac{\frac{T_2}{T_1} - 1}{\frac{T_3}{T_4} - 1}\right) = 1 - \frac{T_1}{T_4}\left(\frac{r_p^{\frac{k-1}{k}} - 1}{r_p^{\frac{k-1}{k}} - 1}\right) = 1 - \frac{T_1}{T_4}$$

In terms of the pressure ratio and the minimum and maximum temperatures in the cycle, the work ratio of the Brayton cycle is:

$$work\ ratic = 1 - \frac{T_{\min}}{T_{\max}}\left(r_p^{\frac{k-1}{k}}\right)$$

Thus, for a high work ratio (and less susceptibility to the irreversibilities in the compressor and turbine) the isentropic temperature ratio (*i.e.*, the pressure ratio) should be low

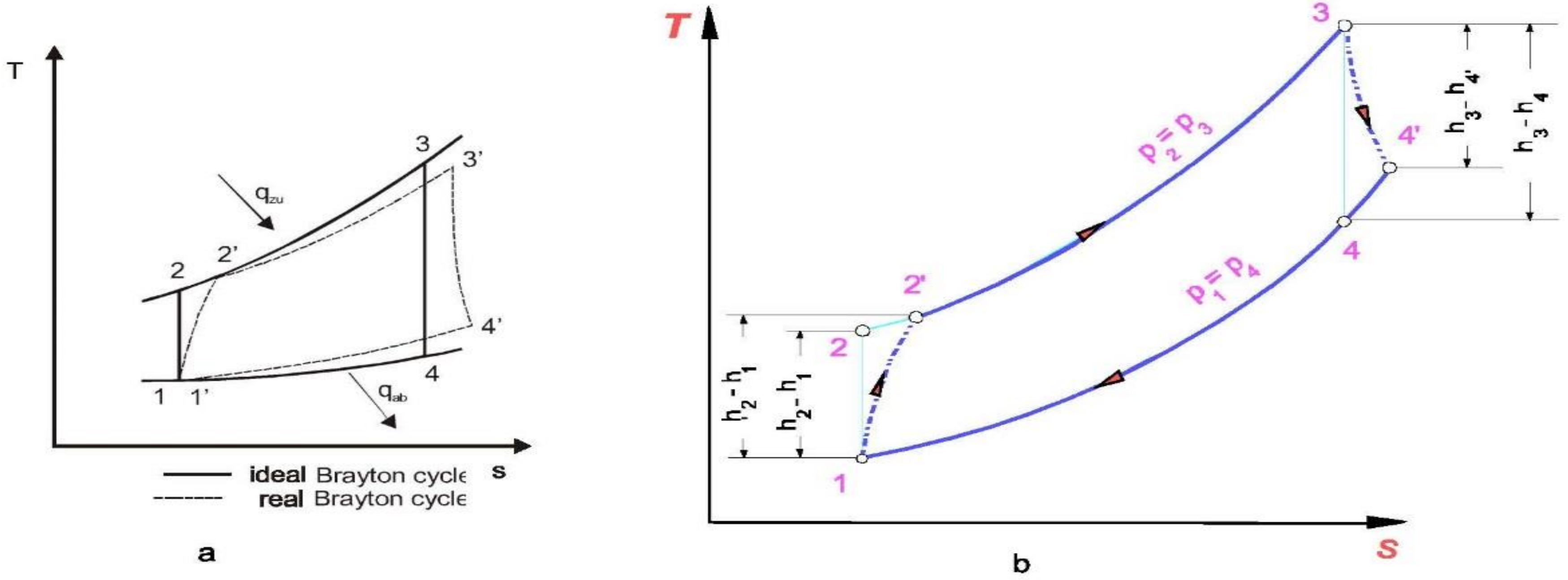
Variation of Joule-Brayton cycle efficiency with pressure ratio



4.6 Actual Joule-Brayton Cycle

- The turbine and compressor are not isentropic
- There is a pressure loss in the combustor
- There is pressure drops in the flow passages, heater and cooler
- *cp* and k vary with temperature.
- *cp* for combustion products is higher than *cp* for air
- Combustion occurs internally instead of heat being supplied externally
- The addition of fuel in the combustor increases the mass flow rate in the turbine
- The turbine exhaust is released to the atmosphere and there is no cooler

3. Gas Tubine Power Plants: Cycles



T-s diagram of Brayton gas turbine cycle (a) actual and (b) neglecting the pressure drop terms

Assuming constant specific heat capacities, the actual compression work and the actual expansion work may be defined in terms of the isentropic efficiencies of the compressor and turbine as:

The ***Compressor Efficiency*** is:

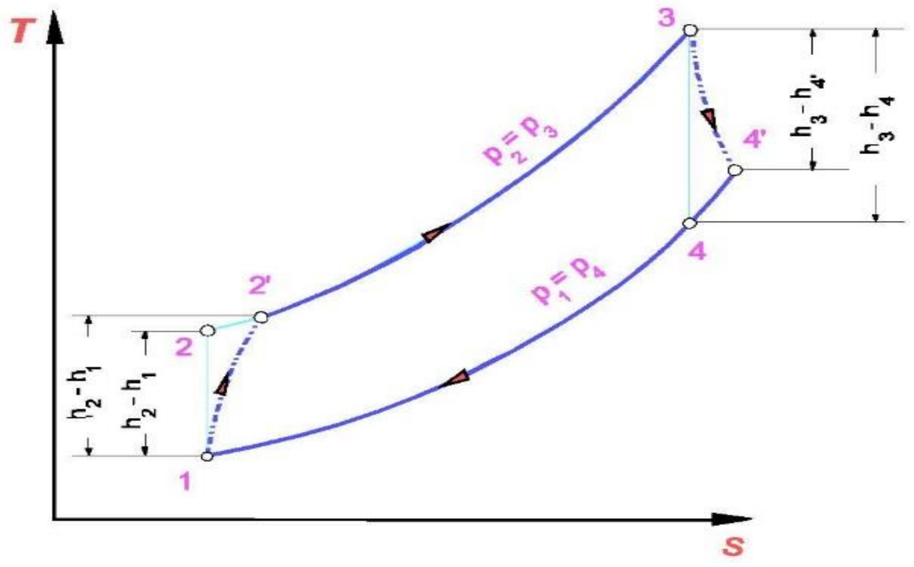
$$\eta_c = \frac{isentropic\ work}{actual\ work} = \frac{h_2 - h_1}{h_{2'} - h_1}$$

Thus; $$w_c = \frac{c_p (T_2 - T_1)}{\eta_c}$$

The ***Turbine Efficiency*** is:

$$\eta_t = \frac{actual\ work}{isentropic\ work} = \frac{h_{3'} - h_{4'}}{h_3 - h_4}$$

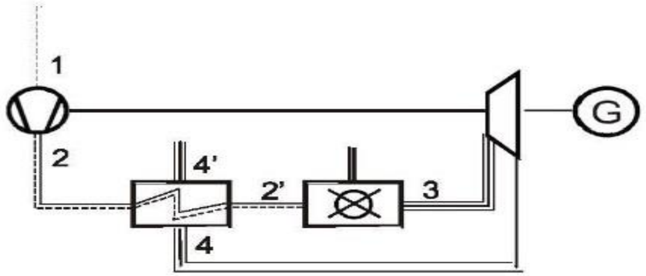
Thus; $$w_t = \eta_t \cdot c_p (T_3 - T_4)$$

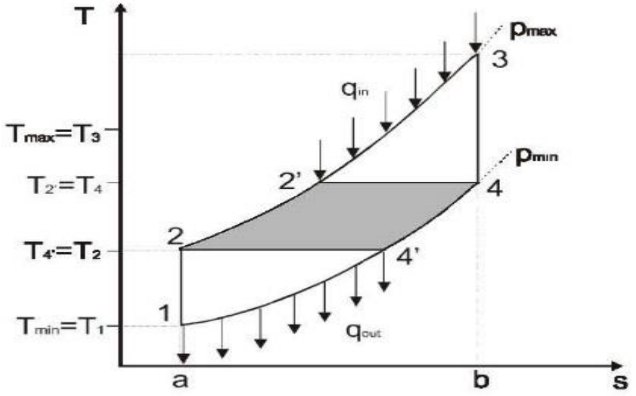


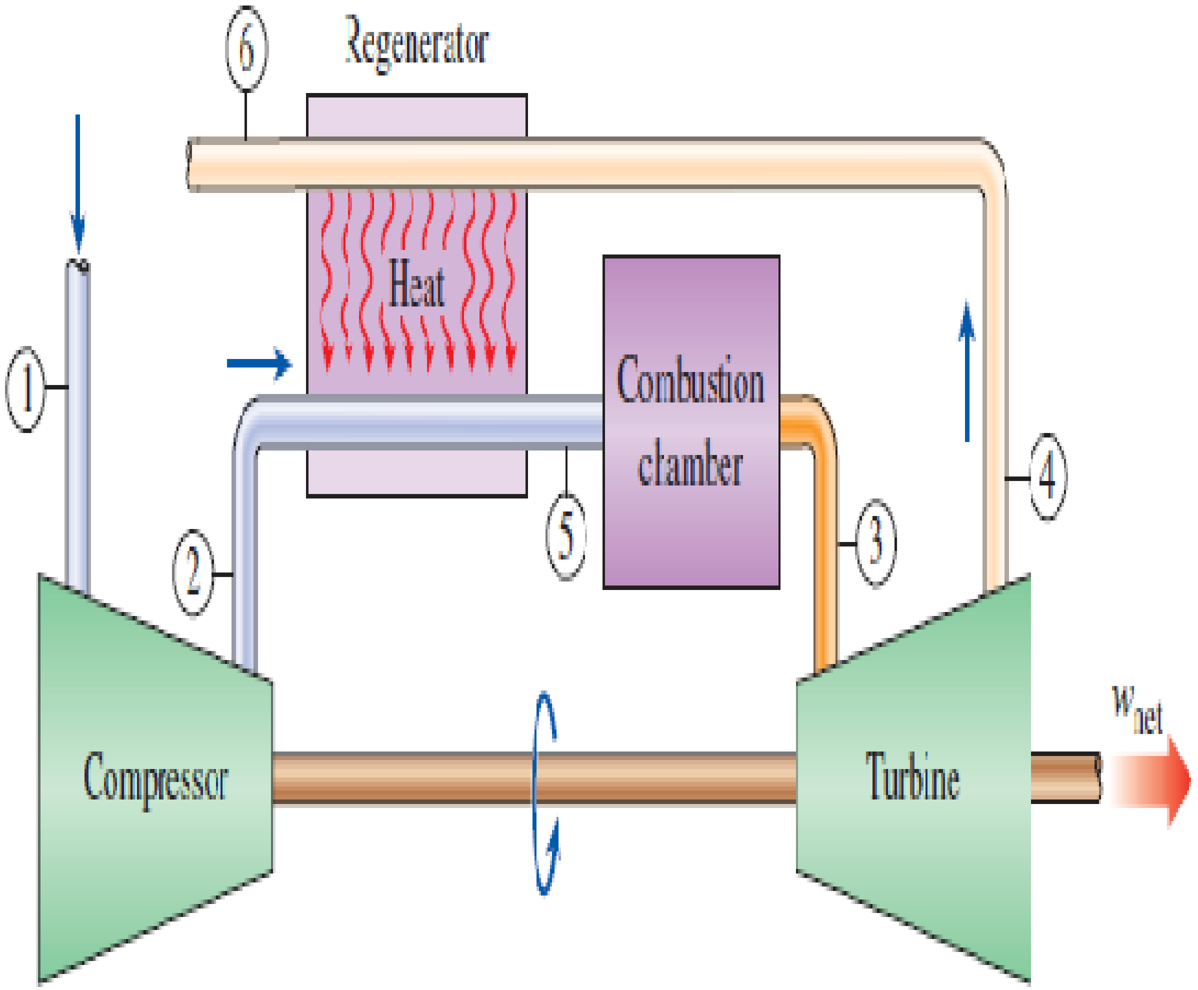
4.7 Brayton Cycle with Regeneration

-The temperature of the exhaust gases leaving the turbine is much higher than the compressed air leaving the compressor

- Increasing the temperature of the compressed air before it enters the combustion chamber, thus
  - the thermal efficiency of the gas-turbine may be improved
  - reduces the quantity of fuel that need be burnt in the combustion chamber to attain the required turbine output
  - the net work output of this cycle doesn't change

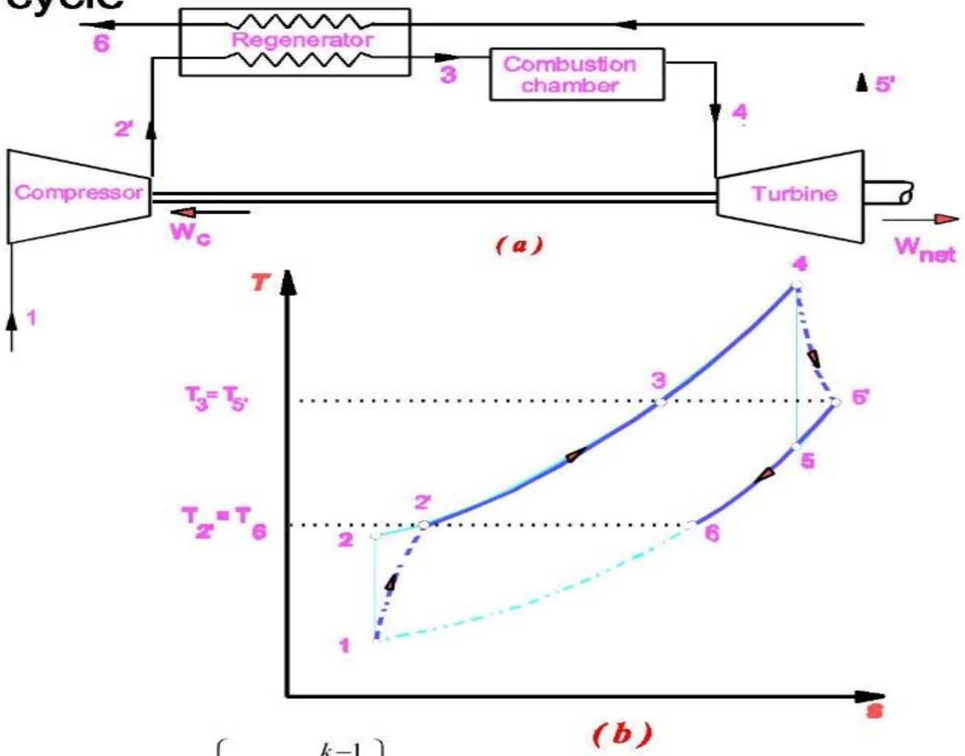






A gas-turbine engine with regenerator.

Efficiency of the Brayton Regenerative cycle



The ***thermal ratio*** of the heat exchanger - regenerator - is defined as the ratio of the temperature rise of the air to the maximum temperature difference available. That is

$$Thermal\ ratio = \frac{T_3 - T_{2'}}{T_{5'} - T_{2'}}$$

The ***thermal efficiency*** of the regenerative cycle is given by:

$$\eta_{th} = \frac{\eta_t T_4 \left\{1 - \left[\frac{1}{r_p}\right]^{\frac{k-1}{k}}\right\} - \frac{T_1}{\eta_c}\left\{r_p^{\frac{k-1}{k}} - 1\right\}}{T_4 - T_3}$$

For ideal regenerator, $T_3 = T_{5'}$, thus

$$T_4 - T_3 = T_4 - T_{5'} = \eta_t (T_4 - T_5)$$

$$= \eta_t T_4 \left(1 - \frac{T_5}{T_4}\right) \quad and \quad \frac{T_5}{T_4} = \left(\frac{1}{r_p}\right)^{\frac{k-1}{k}}$$

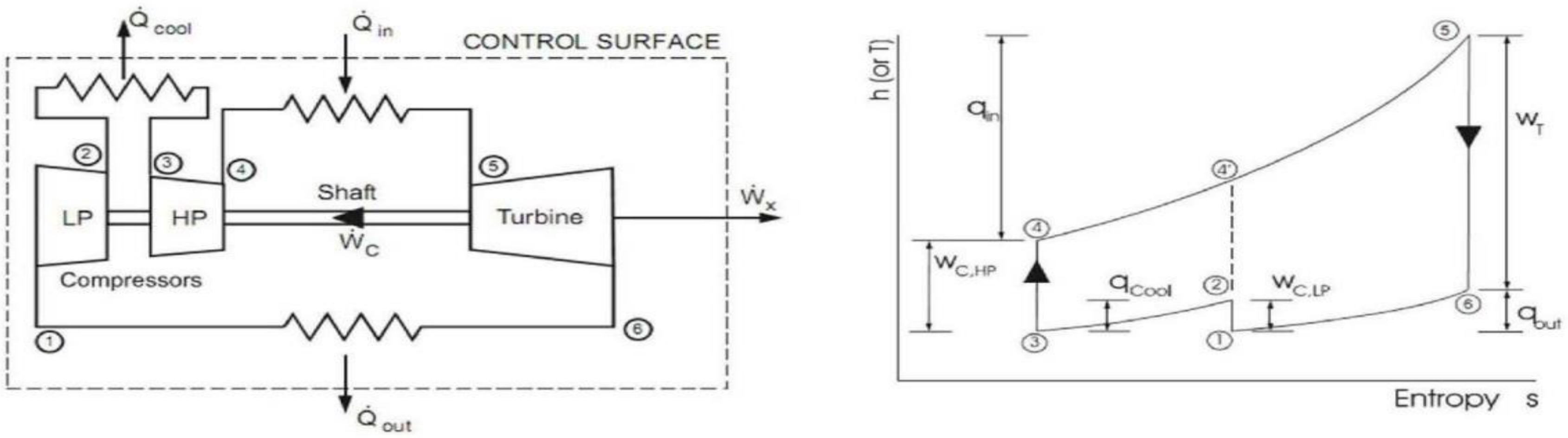
Thus, $\eta_{th} = 1 - \left\{\frac{T_1}{T_4} \cdot \frac{r_p^{\frac{k-1}{k}}}{\eta_t \eta_c}\right\}$

Note-: η_{th} decrease with r_p, b/c $Q_{5'6}$ decrease with r_p

4.8 Brayton Cycle with Intercooling

Even though regeneration improves the thermal efficiency of a gas-turbine cycle, no improvement is observed on the work ratio. The work ratio of a gas-turbine may be improved either by reducing the work input to drive the compressor or by increasing the work output from the turbine.

For a given pressure ratio, the work input to the compressor is reduced when compression is performed in stages with intercoolers between the stages



The closed circuit gas turbine with 1 stage of intercooling

3. Gas Tubine Power Plants: Cycles

In the ideal air-standard cycle with intercooling,

- The intercooler operates at constant pressure (p3 = p2)

- Exit temperature of intercooler is equal to temperature at inlet to first compressor (T3 = T1).

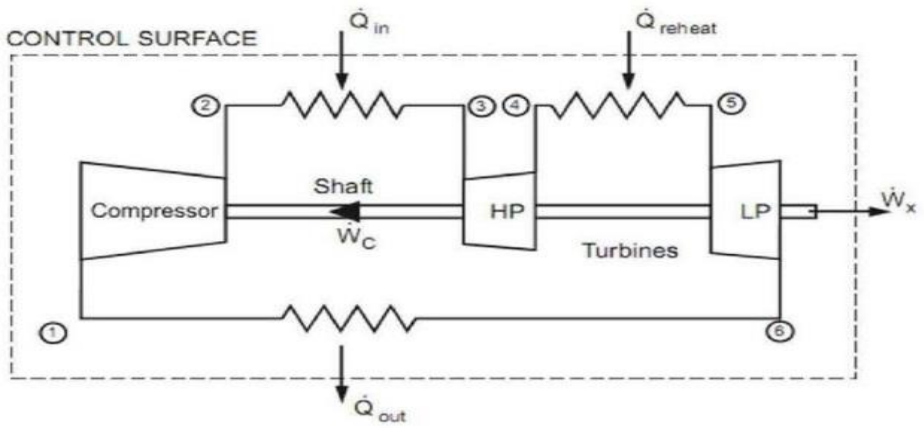
Hence, intercooling

- reduces the efficiency
- increases work output
- enables cooler (and hence less) air to be used to cool the turbine

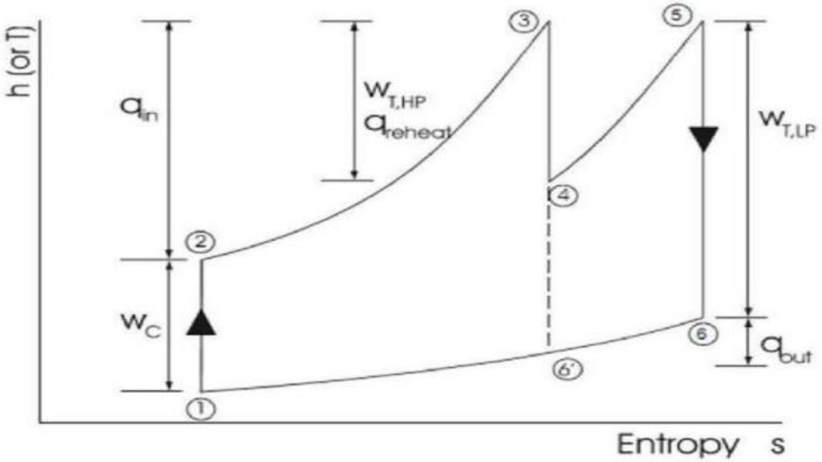
the work required to compress in a steady flow device can be reduced by compressing in stages

- cooling the gas reduces the specific volume and in turn the work required for compression
- by itself compression with intercooling does not provide a significant increase in the efficiency of a gas turbine because the temperature at the combustor inlet would require additional heat transfer to achieve the desired turbine inlet temperature
- but the lower temperature at the compressor exit enhances the potential for regeneration i.e. a larger ΔT across the heat exchanger

4.9 Brayton Cycle with Reheater



Closed circuit gas turbine with reheat



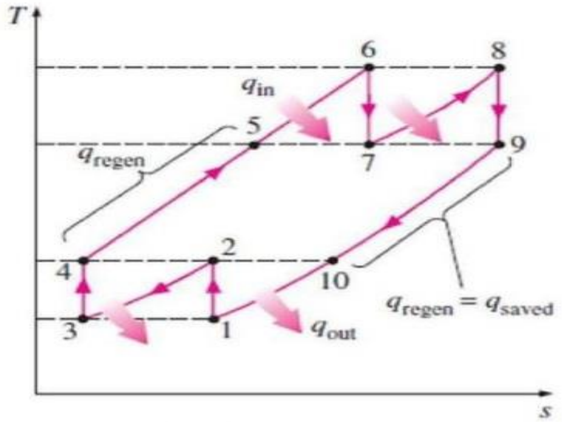
In the ideal air-standard reheat cycle,

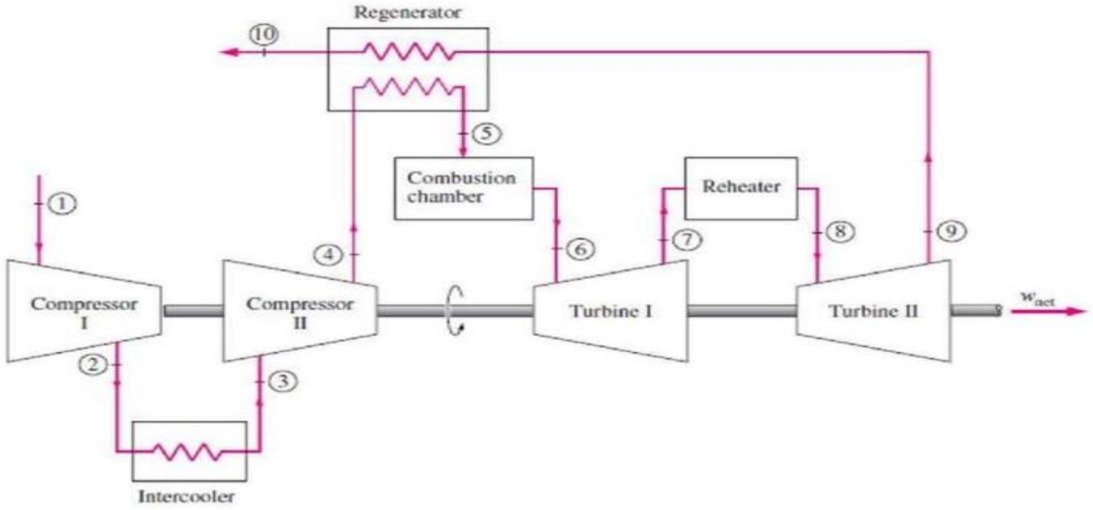
- reheat occurs at constant pressure (p4 = p5)
- exit temperature of reheater is equal to temperature at inlet to first turbine (T5 = T3).

Hence, reheating

- reduces the efficiency
- requires increased number of high temperature components
- increases work output (area of the diagram in the T-s diagram)

4.10 Brayton with Intercooled, Reheated and Regeneration



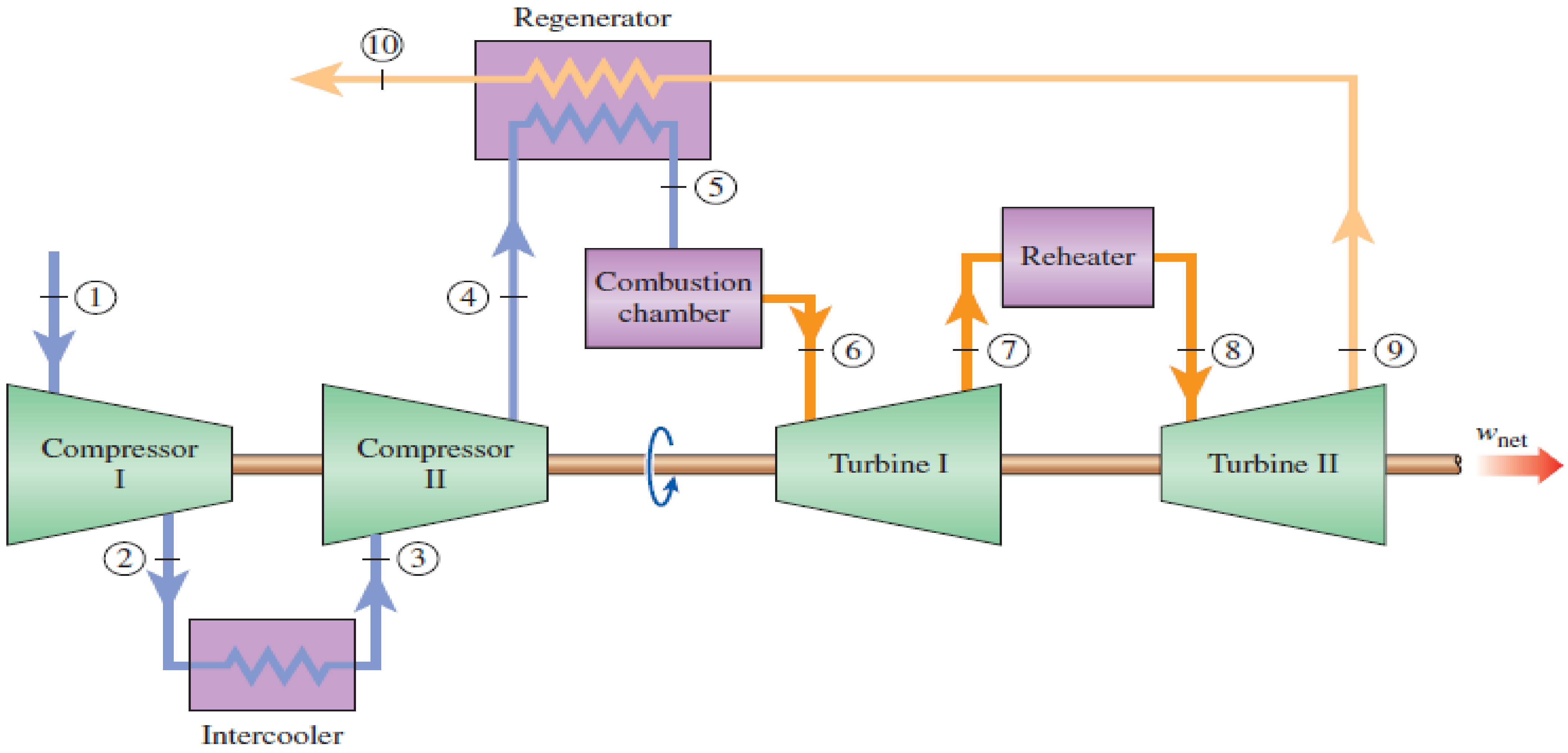


We see that

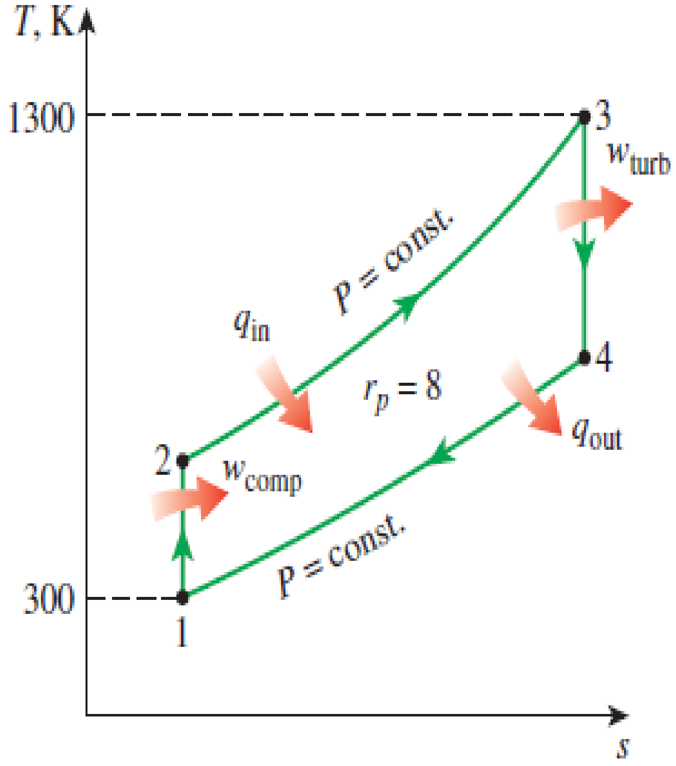
- Intercooling lowers the final compressor delivery temperature
- Reheating raises the final turbine exit temperature
- Recuperation can now transfer far more heat

As a result

- the cycle efficiency increases still further and we still have the increased work output



A gas-turbine engine with two-stage compression with intercooling, two-stage expansion with reheating, and regeneration.



T-s diagram for the Brayton cycle discussed in Example .

EXAMPLE The Simple Ideal Brayton Cycle

A gas-turbine power plant operating on an ideal Brayton cycle has a pressure ratio of 8. The gas temperature is 300 K at the compressor inlet and 1300 K at the turbine inlet. Utilizing the air-standard assumptions, determine (*a*) the gas temperature at the exits of the compressor and the turbine, (*b*) the back work ratio, and (*c*) the thermal efficiency.

SOLUTION A power plant operating on the ideal Brayton cycle is considered. The compressor and turbine exit temperatures, back work ratio, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** The variation of specific heats with temperature is to be considered.

Analysis The *T-s* diagram of the ideal Brayton cycle described is shown in Fig. . We note that the components involved in the Brayton cycle are steady-flow devices.

(*a*) The air temperatures at the compressor and turbine exits are determined from isentropic relations:

Process 1–2 (isentropic compression of an ideal gas):

$$T_1 = 300\ \text{K} \longrightarrow h_1 = 300.19\ \text{kJ/kg}$$
$$P_{r1} = 1.386$$
$$P_{r2} = \frac{P_2}{P_1} P_{r1} = (8)(1.386) = 11.09 \longrightarrow T_2 = 540\ \text{K} \qquad \text{(at compressor exit)}$$
$$h_2 = 544.35\ \text{kJ/kg}$$

Process 3–4 (isentropic expansion of an ideal gas):

$$T_3 = 1300\ \text{K} \longrightarrow h_3 = 1395.97\ \text{kJ/kg}$$
$$P_{r3} = 330.9$$
$$P_{r4} = \frac{P_4}{P_3} P_{r3} = \left(\frac{1}{8}\right)(330.9) = 41.36 \longrightarrow T_4 = 770\ \text{K} \qquad \text{(at turbine exit)}$$
$$h_4 = 789.37\ \text{kJ/kg}$$

(*b*) To find the back work ratio, we need to find the work input to the compressor and the work output of the turbine:

$$w_{\text{comp,in}} = h_2 - h_1 = 544.35 - 300.19 = 244.16\ \text{kJ/kg}$$
$$w_{\text{turb,out}} = h_3 - h_4 = 1395.97 - 789.37 = 606.60\ \text{kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{comp,in}}}{w_{\text{turb,out}}} = \frac{244.16\ \text{kJ/kg}}{606.60\ \text{kJ/kg}} = 0.403$$

That is, 40.3 percent of the turbine work output is used just to drive the compressor.

(*c*) The thermal efficiency of the cycle is the ratio of the net power output to the total heat input:

$$q_{in} = h_3 - h_2 = 1395.97 - 544.35 = 851.62 \text{ kJ/kg}$$
$$w_{net} = w_{out} - w_{in} = 606.60 - 244.16 = 362.4 \text{ kJ/kg}$$

Thus,

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{362.4 \text{ kJ/kg}}{851.62 \text{ kJ/kg}} = \mathbf{0.426 \text{ or } 42.6\%}$$

The thermal efficiency could also be determined from

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

where

$$q_{out} = h_4 - h_1 = 789.37 - 300.19 = 489.2 \text{ kJ/kg}$$

Discussion Under the cold-air-standard assumptions (constant specific heat values at room temperature), the thermal efficiency would be,

$$\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}} = 1 - \frac{1}{8^{(1.4-1)/1.4}} = 0.448 \text{ or } 44.8\%$$

which is sufficiently close to the value obtained by accounting for the variation of specific heats with temperature.

- **Back Work Ratio**

The fraction of the **work** produced by the turbine that is consumed by the compressor.

EXAMPLE An Actual Gas-Turbine Cycle

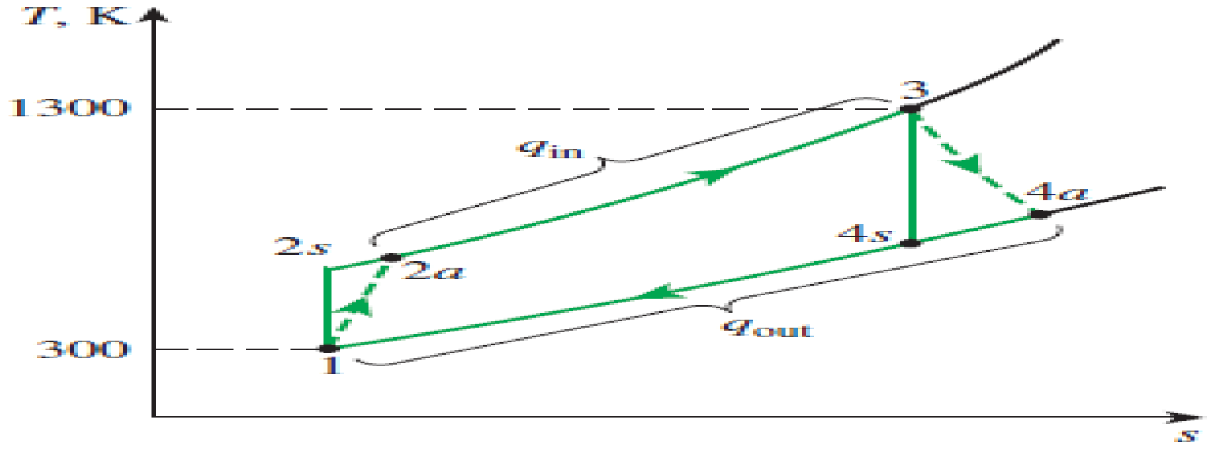
Assuming a compressor efficiency of 80 percent and a turbine efficiency of 85 percent, determine (*a*) the back work ratio, (*b*) the thermal efficiency, and (*c*) the turbine exit temperature of the gas-turbine cycle discussed in last example

SOLUTION The Brayton cycle discussed in last example is reconsidered. For specified turbine and compressor efficiencies, the back work ratio, the thermal efficiency, and the turbine exit temperature are to be determined.

Analysis (*a*) The *T-s* diagram of the cycle is shown in Fig. below. The actual compressor work and turbine work are determined by using the definitions of compressor and turbine efficiencies,

Compressor: $$w_{\text{comp,in}} = \frac{w_s}{\eta_C} = \frac{244.16 \text{ kJ/kg}}{0.80} = 305.20 \text{ kJ/kg}$$

Turbine: $$w_{\text{turb,out}} = \eta_T w_s = (0.85)(606.60 \text{ kJ/kg}) = 515.61 \text{ kJ/kg}$$



T-s diagram of the gas-turbine cycle

Thus,

$$r_{bw} = \frac{w_{comp,in}}{w_{turb,out}} = \frac{305.20 \text{ kJ/kg}}{515.61 \text{ kJ/kg}} = \mathbf{0.592}$$

That is, the compressor is now consuming 59.2 percent of the work produced by the turbine (up from 40.3 percent). This increase is due to the irreversibilities that occur within the compressor and the turbine.
(*b*) In this case, air leaves the compressor at a higher temperature and enthalpy, which are determined to be

$$\begin{aligned} w_{comp,in} = h_{2a} - h_1 \rightarrow h_{2a} &= h_1 + w_{comp,in} \\ &= 300.19 + 305.20 \\ &= 605.39 \text{ kJ/kg} \quad (\text{and } T_{2a} = 598 \text{ K}) \end{aligned}$$

Thus,

$$q_{in} = h_3 - h_{2a} = 1395.97 - 605.39 = 790.58 \text{ kJ/kg}$$

$$w_{net} = w_{out} - w_{in} = 515.61 - 305.20 = 210.41 \text{ kJ/kg}$$

and

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{210.41 \text{ kJ/kg}}{790.58 \text{ kJ/kg}} = \mathbf{0.266 \text{ or } 26.6\%}$$

That is, the irreversibilities occurring within the turbine and compressor caused the thermal efficiency of the gas turbine cycle to drop from 42.6 to 26.6 percent. This example shows how sensitive the performance of a gas-turbine power plant is to the efficiencies of the compressor and the turbine. In fact, gas-turbine efficiencies did not reach competitive values until significant improvements were made in the design of gas turbines and compressors.

(*c*) The air temperature at the turbine exit is determined from an energy balance on the turbine:

$$w_{\text{turb,out}} = h_3 - h_{4a} \rightarrow h_{4a} = h_3 - w_{\text{turb,out}}$$
$$= 1395.97 - 515.61$$
$$= 880.36\ \text{kJ/kg}$$

Then, from Table A–17,

$$T_{4a} = \mathbf{853\ K}$$

Discussion The temperature at turbine exit is considerably higher than that at the compressor exit (T_{2a} = 598 K), which suggests the use of regeneration to reduce fuel cost.

TABLE A–17

Ideal-gas properties of air

T K	h kJ/kg	P_r	u kJ/kg	v_r	s° kJ/kg·K	T K	h kJ/kg	P_r	u kJ/kg	v_r	s° kJ/kg·K
200	199.97	0.3363	142.56	1707.0	1.29559	580	586.04	14.38	419.55	115.7	2.37348
210	209.97	0.3987	149.69	1512.0	1.34444	590	596.52	15.31	427.15	110.6	2.39140
220	219.97	0.4690	156.82	1346.0	1.39105	600	607.02	16.28	434.78	105.8	2.40902
230	230.02	0.5477	164.00	1205.0	1.43557	610	617.53	17.30	442.42	101.2	2.42644
240	240.02	0.6355	171.13	1084.0	1.47824	620	628.07	18.36	450.09	96.92	2.44356
250	250.05	0.7329	178.28	979.0	1.51917	630	638.63	19.84	457.78	92.84	2.46048
260	260.09	0.8405	185.45	887.8	1.55848	640	649.22	20.64	465.50	88.99	2.47716
270	270.11	0.9590	192.60	808.0	1.59634	650	659.84	21.86	473.25	85.34	2.49364
280	280.13	1.0889	199.75	738.0	1.63279	660	670.47	23.13	481.01	81.89	2.50985
285	285.14	1.1584	203.33	706.1	1.65055	670	681.14	24.46	488.81	78.61	2.52589
290	290.16	1.2311	206.91	676.1	1.66802	680	691.82	25.85	496.62	75.50	2.54175
295	295.17	1.3068	210.49	647.9	1.68515	690	702.52	27.29	504.45	72.56	2.55731
298	298.18	1.3543	212.64	631.9	1.69528	700	713.27	28.80	512.33	69.76	2.57277
300	300.19	1.3860	214.07	621.2	1.70203	710	724.04	30.38	520.23	67.07	2.58810
305	305.22	1.4686	217.67	596.0	1.71865	720	734.82	32.02	528.14	64.53	2.60319
310	310.24	1.5546	221.25	572.3	1.73498	730	745.62	33.72	536.07	62.13	2.61803
315	315.27	1.6442	224.85	549.8	1.75106	740	756.44	35.50	544.02	59.82	2.63280
320	320.29	1.7375	228.42	528.6	1.76690	750	767.29	37.35	551.99	57.63	2.64737
325	325.31	1.8345	232.02	508.4	1.78249	760	778.18	39.27	560.01	55.54	2.66176
330	330.34	1.9352	235.61	489.4	1.79783	780	800.03	43.35	576.12	51.64	2.69013
340	340.42	2.149	242.82	454.1	1.82790	800	821.95	47.75	592.30	48.08	2.71787
350	350.49	2.379	250.02	422.2	1.85708	820	843.98	52.59	608.59	44.84	2.74504
360	360.58	2.626	257.24	393.4	1.88543	840	866.08	57.60	624.95	41.85	2.77170
370	370.67	2.892	264.46	367.2	1.91313	860	888.27	63.09	641.40	39.12	2.79783
380	380.77	3.176	271.69	343.4	1.94001	880	910.56	68.98	657.95	36.61	2.82344
390	390.88	3.481	278.93	321.5	1.96633	900	932.93	75.29	674.58	34.31	2.84856
400	400.98	3.806	286.16	301.6	1.99194	920	955.38	82.05	691.28	32.18	2.87324
410	411.12	4.153	293.43	283.3	2.01699	940	977.92	89.28	708.08	30.22	2.89748
420	421.26	4.522	300.69	266.6	2.04142	960	1000.55	97.00	725.02	28.40	2.92128
430	431.43	4.915	307.99	251.1	2.06533	980	1023.25	105.2	741.98	26.73	2.94468
440	441.61	5.332	315.30	236.8	2.08870	1000	1046.04	114.0	758.94	25.17	2.96770
450	451.80	5.775	322.62	223.6	2.11161	1020	1068.89	123.4	776.10	23.72	2.99034
460	462.02	6.245	329.97	211.4	2.13407	1040	1091.85	133.3	793.36	23.29	3.01260
470	472.24	6.742	337.32	200.1	2.15604	1060	1114.86	143.9	810.62	21.14	3.03449
480	482.49	7.268	344.70	189.5	2.17760	1080	1137.89	155.2	827.88	19.98	3.05608
490	492.74	7.824	352.08	179.7	2.19876	1100	1161.07	167.1	845.33	18.896	3.07732
500	503.02	8.411	359.49	170.6	2.21952	1120	1184.28	179.7	862.79	17.886	3.09825
510	513.32	9.031	366.92	162.1	2.23993	1140	1207.57	193.1	880.35	16.946	3.11883
520	523.63	9.684	374.36	154.1	2.25997	1160	1230.92	207.2	897.91	16.064	3.13916
530	533.98	10.37	381.84	146.7	2.27967	1180	1254.34	222.2	915.57	15.241	3.15916
540	544.35	11.10	389.34	139.7	2.29906	1200	1277.79	238.0	933.33	14.470	3.17888
550	555.74	11.86	396.86	133.1	2.31809	1220	1301.31	254.7	951.09	13.747	3.19834
560	565.17	12.66	404.42	127.0	2.33685	1240	1324.93	272.3	968.95	13.069	3.21751
570	575.59	13.50	411.97	121.2	2.35531						

TABLE A–17

Ideal-gas properties of air (*Concluded*)

T K	h kJ/kg	P_r	u kJ/kg	v_r	s° kJ/kg·K	T K	h kJ/kg	P_r	u kJ/kg	v_r	s° kJ/kg·K
1260	1348.55	290.8	986.90	12.435	3.23638	1600	1757.57	791.2	1298.30	5.804	3.52364
1280	1372.24	310.4	1004.76	11.835	3.25510	1620	1782.00	834.1	1316.96	5.574	3.53879
1300	1395.97	330.9	1022.82	11.275	3.27345	1640	1806.46	878.9	1335.72	5.355	3.55381
1320	1419.76	352.5	1040.88	10.747	3.29160	1660	1830.96	925.6	1354.48	5.147	3.56867
1340	1443.60	375.3	1058.94	10.247	3.30959	1680	1855.50	974.2	1373.24	4.949	3.58335
1360	1467.49	399.1	1077.10	9.780	3.32724	1700	1880.1	1025	1392.7	4.761	3.5979
1380	1491.44	424.2	1095.26	9.337	3.34474	1750	1941.6	1161	1439.8	4.328	3.6336
1400	1515.42	450.5	1113.52	8.919	3.36200	1800	2003.3	1310	1487.2	3.994	3.6684
1420	1539.44	478.0	1131.77	8.526	3.37901	1850	2065.3	1475	1534.9	3.601	3.7023
1440	1563.51	506.9	1150.13	8.153	3.39586	1900	2127.4	1655	1582.6	3.295	3.7354
1460	1587.63	537.1	1168.49	7.801	3.41247	1950	2189.7	1852	1630.6	3.022	3.7677
1480	1611.79	568.8	1186.95	7.468	3.42892	2000	2252.1	2068	1678.7	2.776	3.7994
1500	1635.97	601.9	1205.41	7.152	3.44516	2050	2314.6	2303	1726.8	2.555	3.8303
1520	1660.23	636.5	1223.87	6.854	3.46120	2100	2377.7	2559	1775.3	2.356	3.8605
1540	1684.51	672.8	1242.43	6.569	3.47712	2150	2440.3	2837	1823.8	2.175	3.8901
1560	1708.82	710.5	1260.99	6.301	3.49276	2200	2503.2	3138	1872.4	2.012	3.9191
1580	1733.17	750.0	1279.65	6.046	3.50829	2250	2566.4	3464	1921.3	1.864	3.9474

Note: The properties P_r (relative pressure) and v_r (relative specific volume) are dimensionless quantities used in the analysis of isentropic processes, and should not be confused with the properties pressure and specific volume.

Source of Data: Kenneth Wark, *Thermodynamics,* 4th ed. (New York: McGraw-Hill, 1983), pp. 785–86, table A–5. Originally published in J. H. Keenan and J. Kaye, *Gas Tables* (New York: John Wiley & Sons, 1948).

4.11 Gas and Steam Combined Power Cycle (GSCC)

In practice, for steady flow devices or cycles,

- maximum delivery temperature from combustion chamber/boiler ≈ 1800 K
- minimum temperature in steady flow ≈ Tambient+10 K ≈ 300 K

Therefore, $\eta_{carnot} = 1 - \frac{T_{\min}}{T_{\max}} = 1 - \frac{300}{1800} = 0.83 \quad >> \eta_{Rankine} \,\&\, \eta_{Brayton}$

However,

- no practical device or cycle exists that can achieve this for large scale power generation so we,
- combine cycles which operate over narrower temperature ranges

Working Principle of the Combined Cycle Power Plant

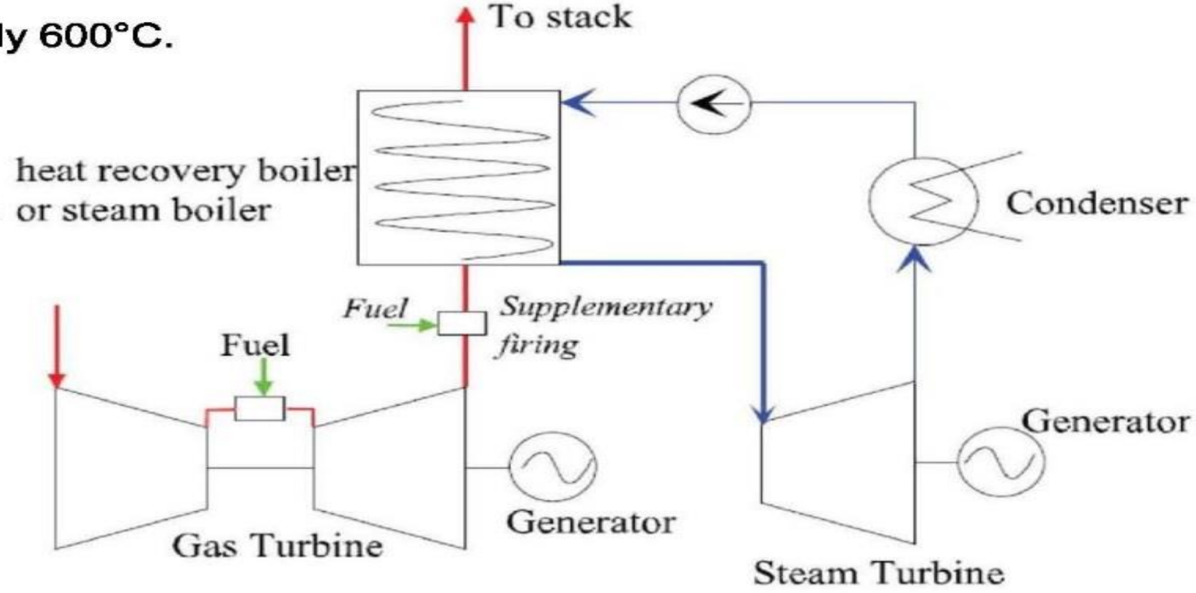
Typical working conditions of a combined cycle power plant

Gas turbines:

- Heat input (combustion) at up to 1500°C,
- Exhaust temperature typically 600°C.

Steam cycles:

- Heat input, around 550°C
- Heat rejection, around 35°C



Video

Cogeneration

Cogeneration is the simultaneous generation of electricity and steam (or heat) in a single powerplant. It has long been used by industries and municipalities that need process steam (or heat) as well as electricity. Examples are chemical industries, paper mills, and places that use district heating. Cogeneration is not usually used by large utilities which tend to produce electricity only. Cogeneration is advisable for industries and municipalities if they can produce electricity cheaper, or more conveniently, than that brought from a utility.

From an energy resource point of view, cogeneration is beneficial only if it saves primary energy when compared with separate generation of electricity and steam (or heat). The *cogeneration plant efficiency* η_{co} is given by

$$\eta_{co} = \frac{E + \Delta H_s}{Q_A} \tag{2-29}$$

where E = electric energy generated

Video

ΔH_s = heat energy, or heat energy in process steam

= (enthalpy of steam entering the process)

− (enthalpy of process condensate returning to plant)

Q_A = heat added to plant (in coal, nuclear fuel, etc.)

For separate generation of electricity and steam, the heat added per unit *total* energy output is

$$\frac{e}{\eta_e} + \frac{(1-e)}{\eta_h}$$

where e = electrical fraction of total energy output $= \dfrac{E}{(E+\Delta H_s)}$

η_e = electric plant efficiency

η_h = steam (or heat) generator efficiency

The *combined efficiency* η_c *for separate generation* is therefore given by

$$\eta_c = \frac{1}{(e/\eta_e) + [(1-e)/\eta_h]} \qquad (2\text{-}30)$$

and cogeneration is beneficial if the efficiency of the cogeneration plant Eq. (2-29) exceeds that of separate generation, Eq. (2-30).

Types of Cogeneration

There are two broad categories of cogeneration:

1. *The topping cycle*, in which primary heat at the higher temperature end of the Rankine cycle is used to generate high-pressure and -temperature steam and electricity in the usual manner. Dependng on process requirements, process steam at low-pressure and temperature is either (*a*) extracted from the turbine at an intermediate stage, much as for feedwater heating, or (*b*) taken at the turbine exhaust,

in which case it is called a *back pressure* turbine. Process steam pressure requirements vary widely, between 0.5 and 40 bar.

2. *The bottoming cycle*, in which primary heat is used at high temperature directly for process requirements. An example is the high-temperature cement kiln. The process low-grade (low temperature and availability) waste heat is then used to generate electricity, obviously at low efficiency. The bottoming cycle thus has a combined efficiency that most certainly lies below that given by Eq. (2-30), and therefore is of little thermodynamic or economic interest.

Only the topping cycle, therefore, can provide true savings in primary energy. In addition, most process applications require low grade (temperature, availability) steam. Such steam is conveniently produced in a topping cycle. There are several arrangements for cogeneration in a topping cycle. Some are:

(*a*) Steam-electric powerplant with a back-pressure turbine.
(*b*) Steam-electric powerplant with steam extraction from a condensing turbine (Fig. 2-27).
(*c*) Gas-turbine powerplant with a heat-recovery boiler (using the gas turbine exhaust to generate steam).
(*d*) Combined steam-gas-turbine-cycle powerplant . The steam turbine is either of the back-pressure type (*a*) or of the extraction-condensing type (*b*), above.

The most suitable electric-to-heat generation ratios vary from type to type. The back-pressure steam turbine plant (*a*) is most suitable only when the electric demand is low compared with the heat demand. The combined-cycle plant (*d*) is most suitable only when the electric demand is high, about comparable to the heat demand or higher, though its range is wider with an extraction-condensing steam turbine than with a back-pressure turbine. The gas-turbine cycle (*c*) lies in between. Only the extraction-condensing plant (*b*) is suitable over a wide range of ratios.

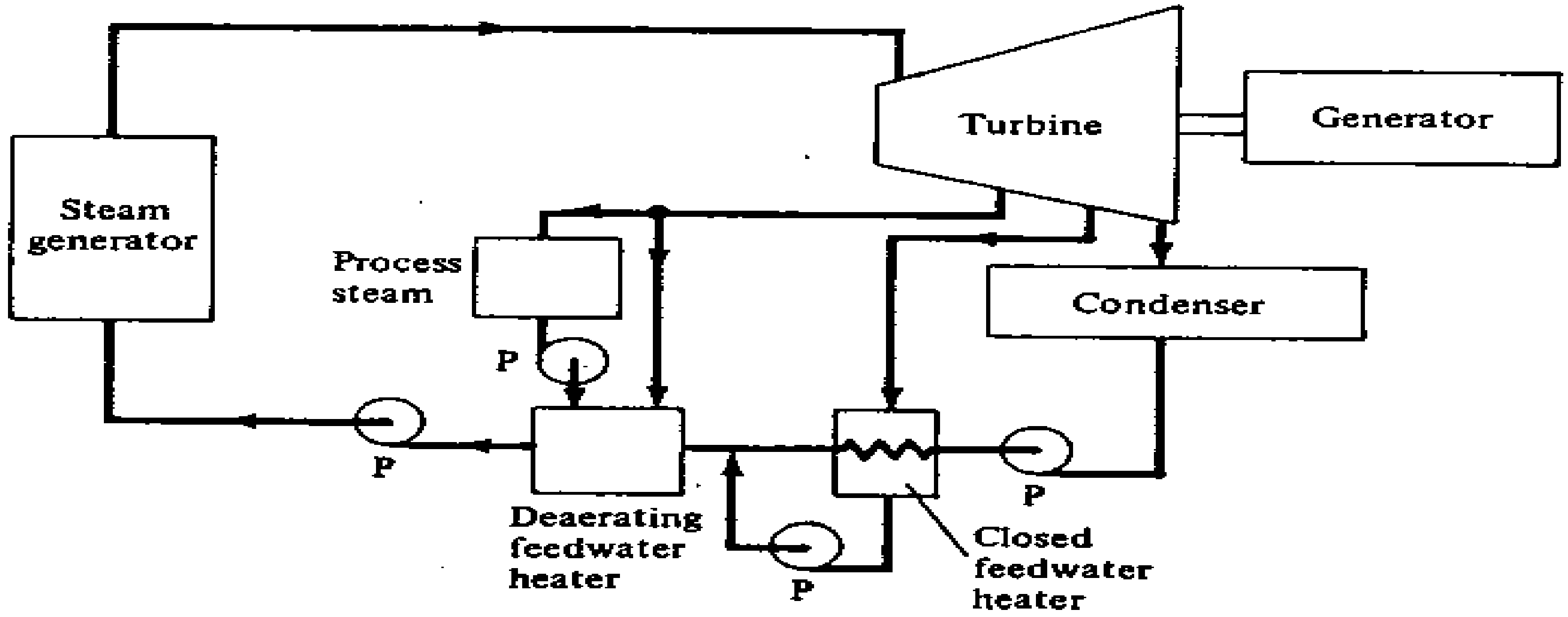


Figure 2-27 Schematic of basic cogeneration plant with extraction-condensing turbine.