TOPIC 3
CHAPTER 9 : PART 11

BRAYTON CYCLE –
THE IDEAL CYCLE FOR GAS TURBINE

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INTRODUCTION

A **gas turbine** is an engine that discharges a fast moving jet of fluid to generate thrust in accordance with Newton's third law of motion. This broad definition of jet engines includes turbojets, turbofans, rockets and ramjets and water jets, but in common usage, the term generally refers to a gas turbine used to produce a jet of high speed exhaust gases for special propulsive purposes.

F-15 Eagle is powered by two Pratt & Whitney F100 axial-flow turbofan engines

F-15 Eagle engine is tested at Robins Air Force Base, Georgia, USA

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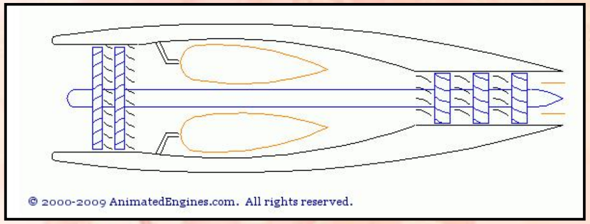
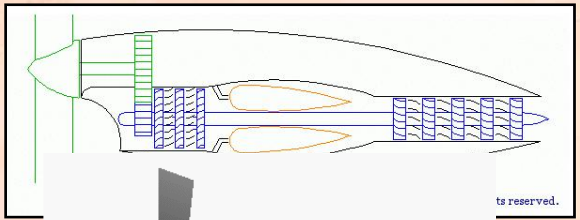
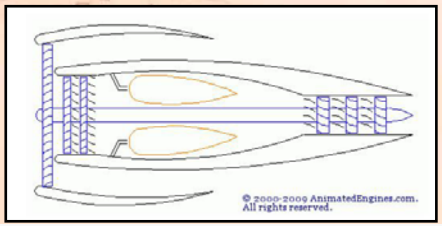
TYPES OF GAS TURBINE

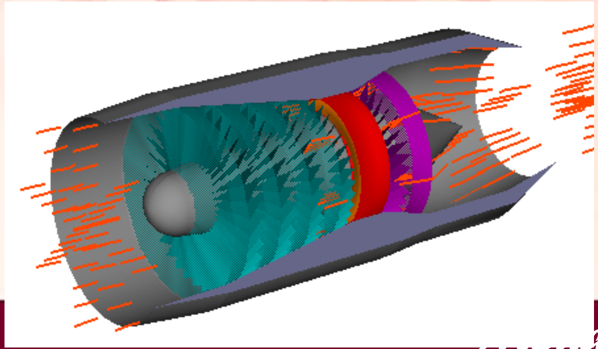
Gas Turbine

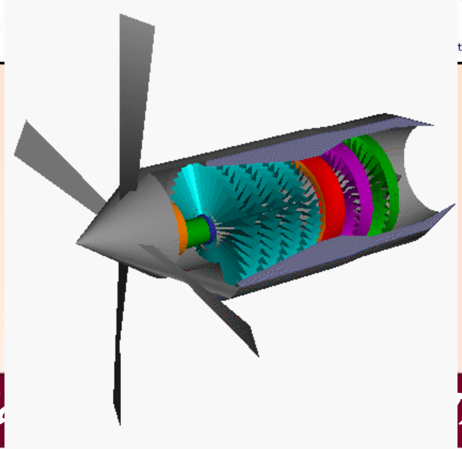
Turbojet	**Turboprop**	**Turbofan**
The combustion gasses flow through the nozzle generating 100% thrust and drive a turbine shaft.	Most of the gas pressure drives the turbine. Shaft drives a propeller that creates the majority of the thrust	The gas pressure drives the turbine. Turbine shaft drives an external fan. Both gasses and fan create the thrust

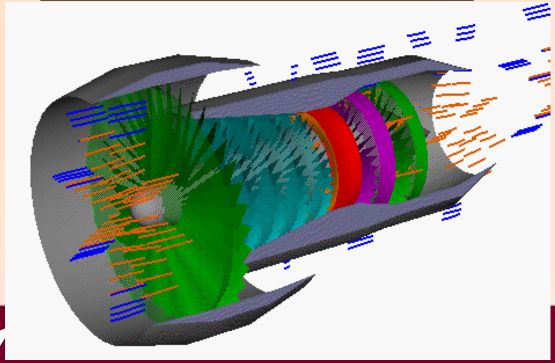












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INTRODUCTION

So why does the M-1 tank use a 1,500 horsepower gas turbine engine instead of a diesel engine?

Advantages of Gas Turbines

- Great **power-to-weight ratio** compared to reciprocating engines. i.e. the amount of power you get out of the engine compared to the weight of the engine itself is very good.
- **Smaller** than their reciprocating counterparts of the same power

Disadvantages of Jet Engines

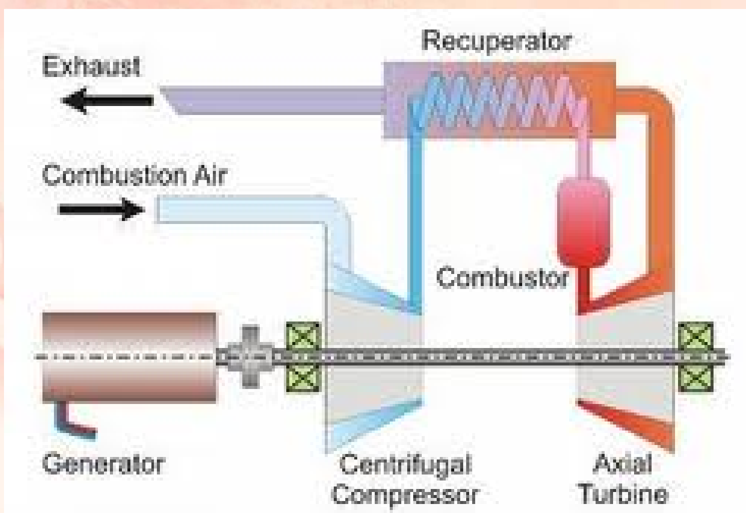
- Compared to a reciprocating engine of the same size, gas turbines are **expensive** - because of the high spin and operating temperatures, designing and manufacturing gas turbines is a tough problem
- Gas turbines use more fuel when they are idling, and they prefer a constant rather than a fluctuating load.

THE USE OF GAS TURBINE

- Aircraft propulsion system
- Electric power generation
- Marine vehicle propulsion
- Combined-cycle power plant (with steam power plant)
- Tanks

F-15 Eagle

F-15 Eagle



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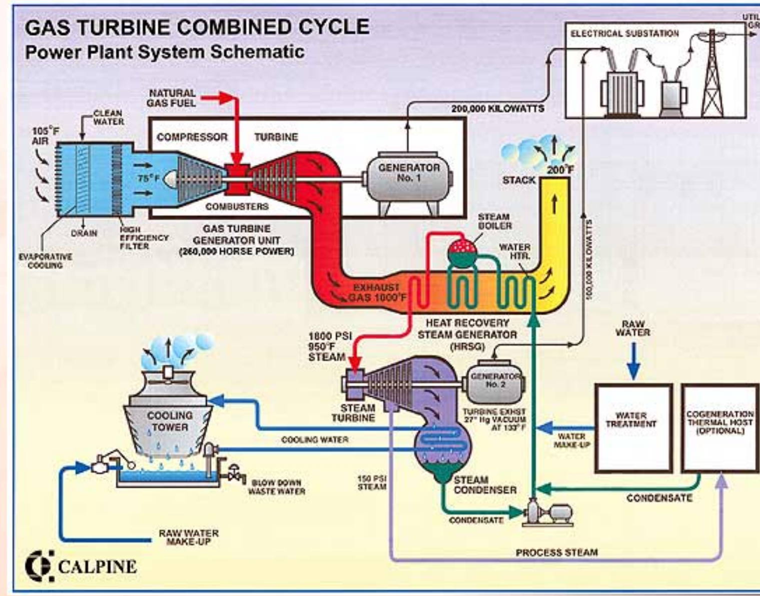
THE USE OF GAS TURBINE


***Naval Vessel - Iroquois*-class destroyers**



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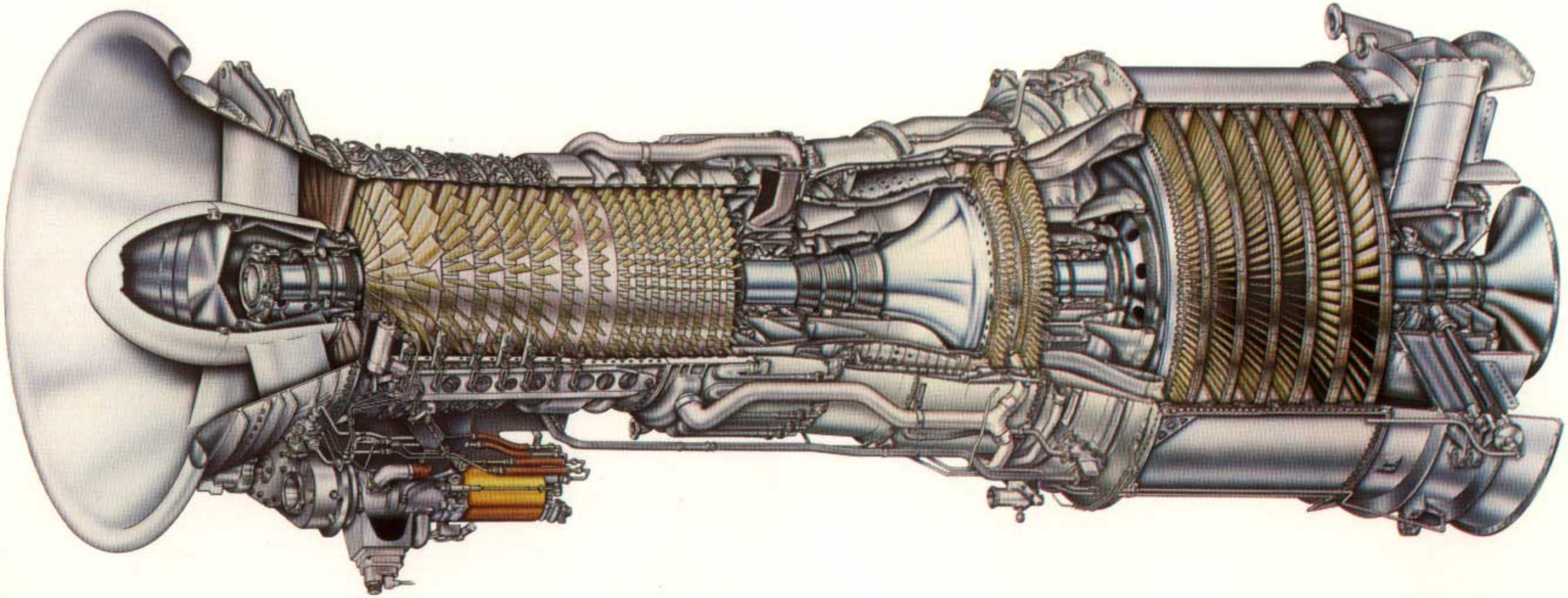
GAS TURBINE POWER PLANT



General Electric LM2500 Gas Turbine

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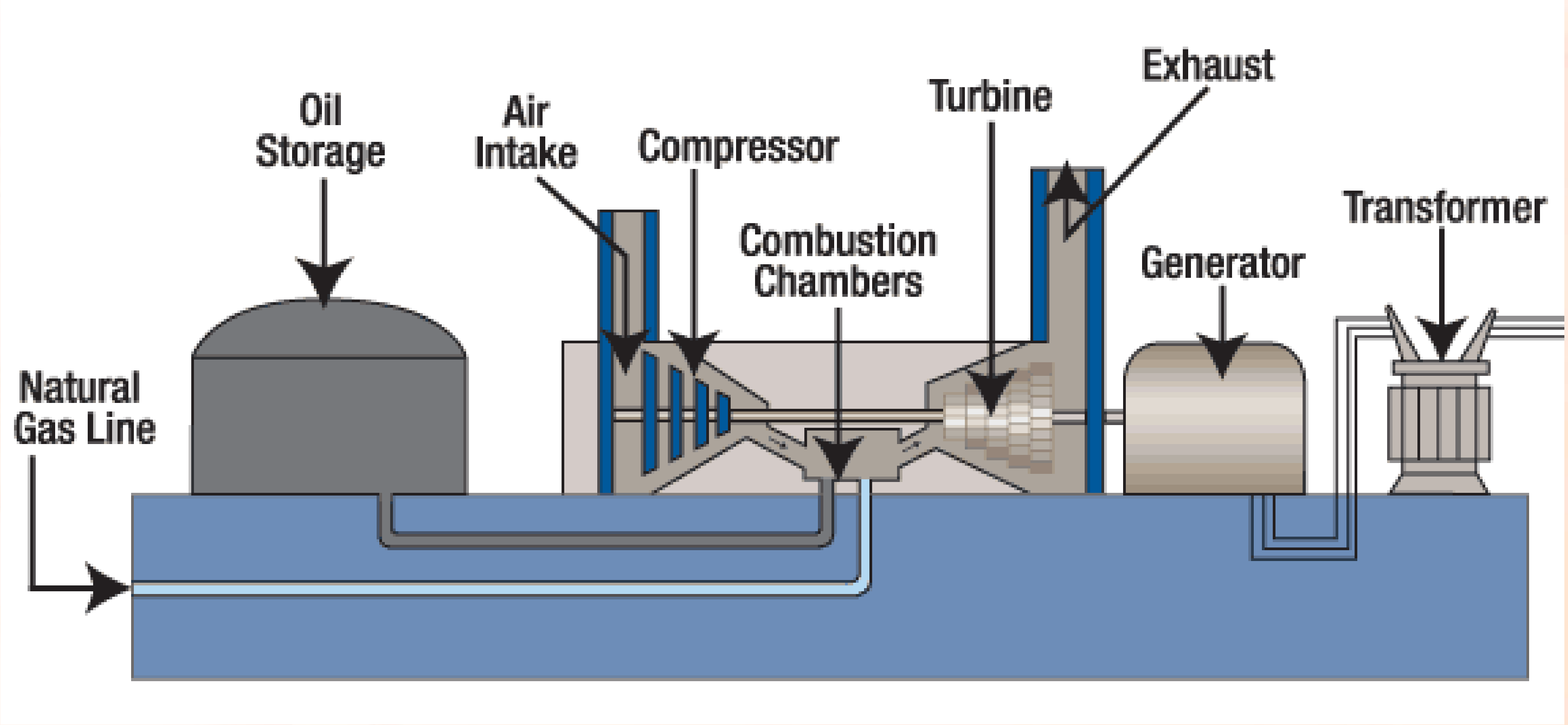
GAS TURBINE POWER PLANT



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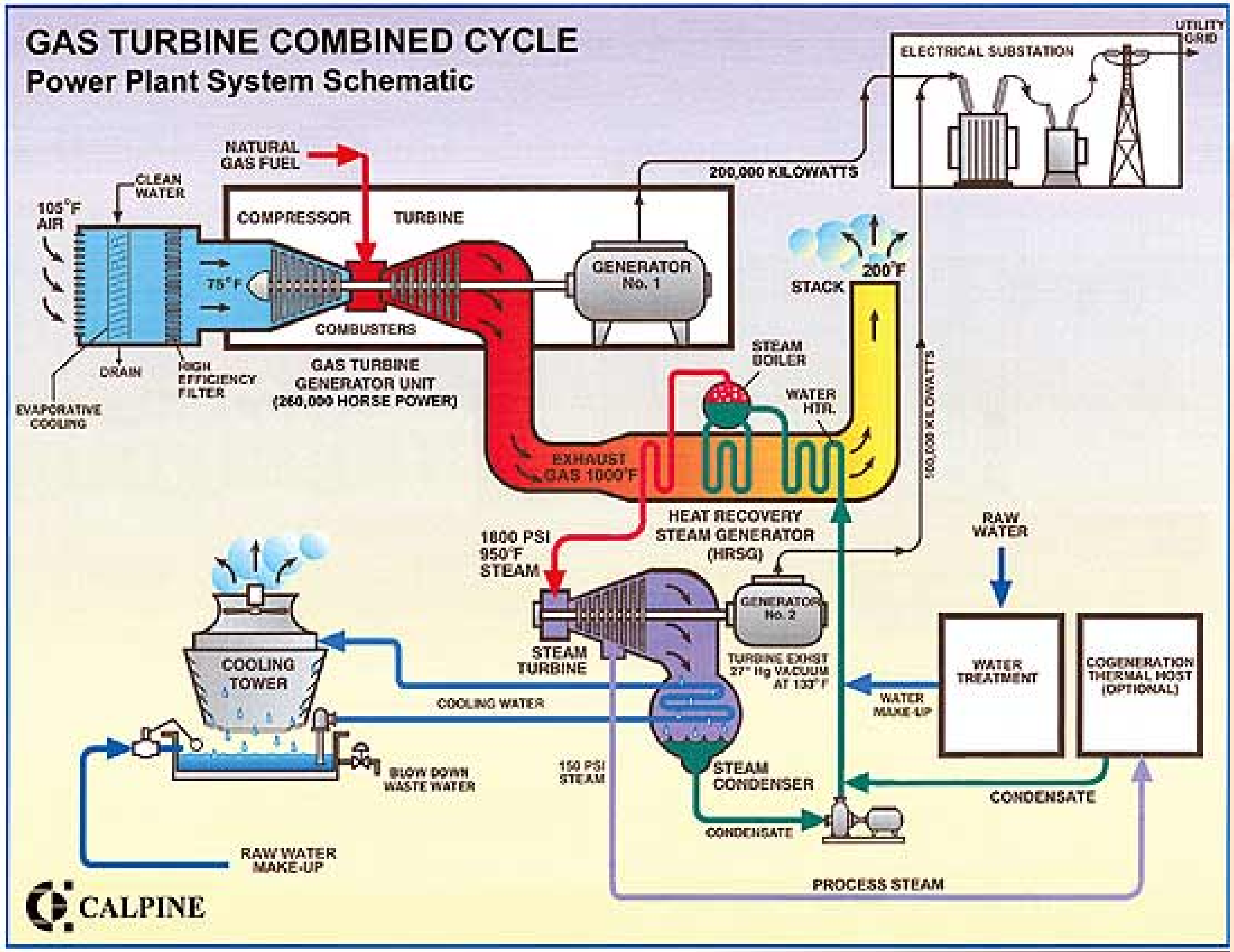
GAS TURBINE POWER PLANT



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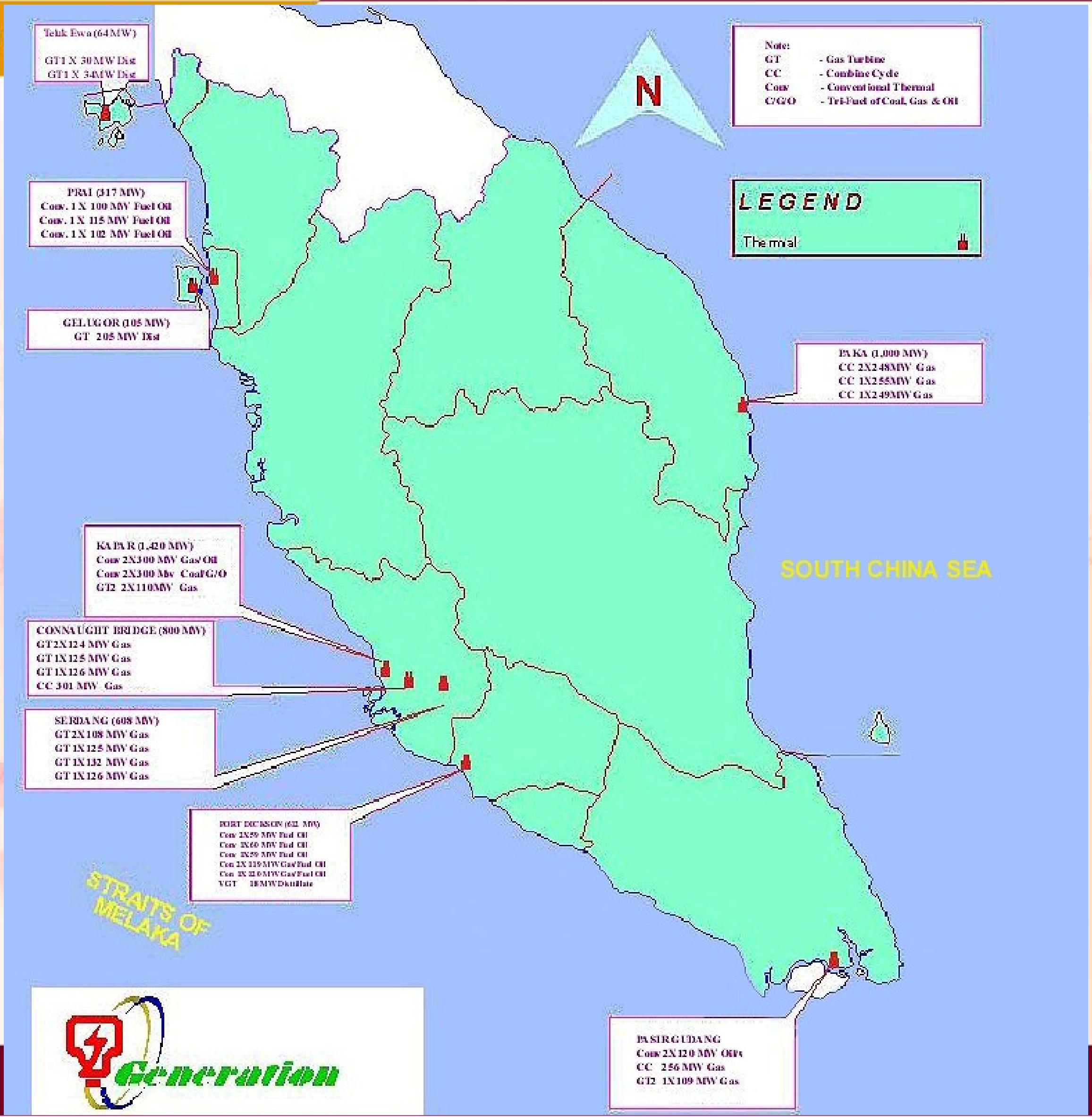
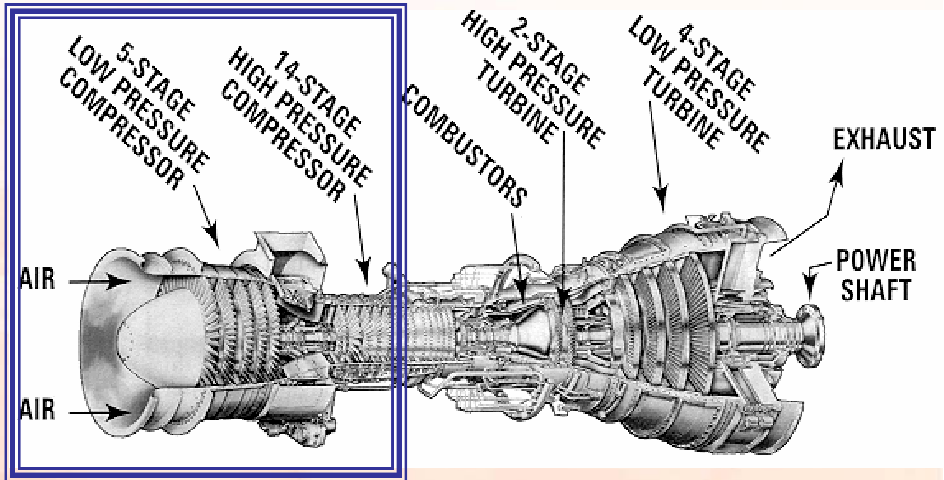
MAJOR POWER PLANTS IN MALAYSIA



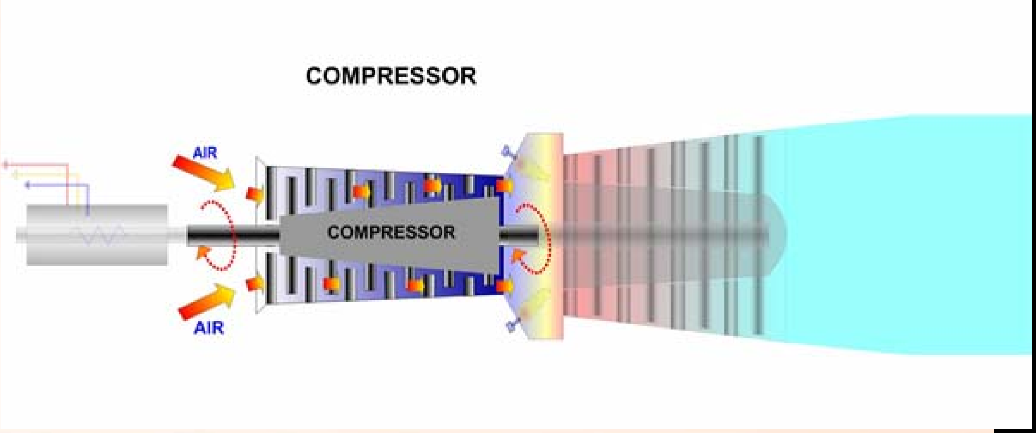
Go to list of gas turbine in Malaysia

Main Components of Gas Turbine Power Plant



1. Compressor

- The compressor sucks in air form the atmosphere and compresses it to pressures in the range of 15 to 20 bar.
- The compressor consists of a number of rows of blades mounted on a shaft.
- The shaft is connected and rotates along with the main gas turbine.



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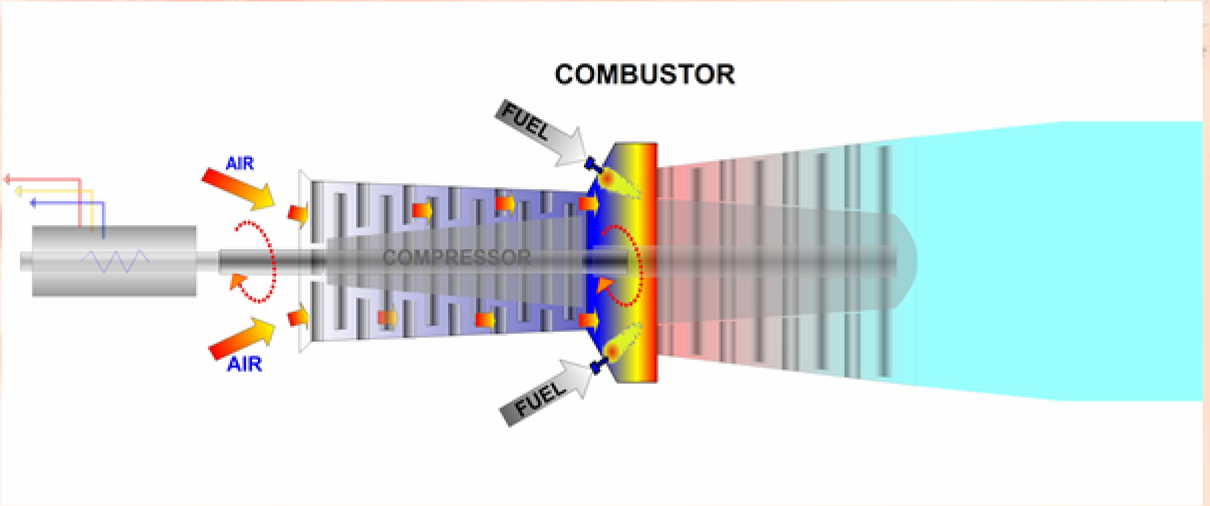
Main Components of Gas Turbine Power Plant

2. Combustor

- This is an annular chamber where the fuel burns and is similar to the furnace in a boiler.
- The hot gases in the range of 1400 to 1500 ° C leave the chamber with high energy levels.
- The chamber and the subsequent sections are made of special alloys and designs that can withstand this high temperature



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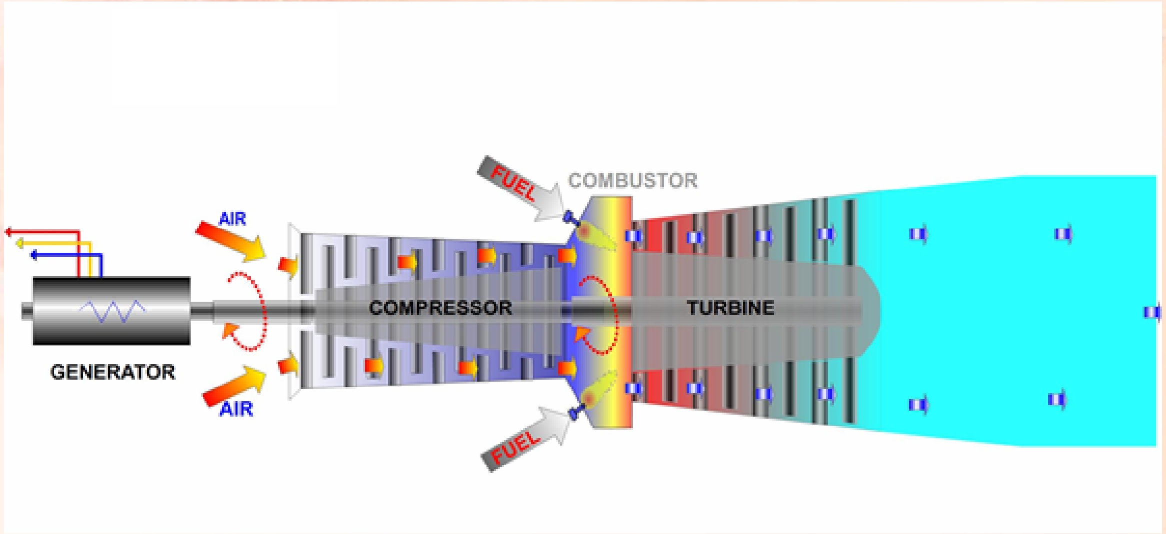
Main Components of Gas Turbine Power Plant

3. Turbine

- The turbine does the main work of energy conversion.
- The turbine portion also consists of rows of blades fixed to the shaft. The kinetic energy of the hot gases impacting on the blades rotates the blades and the shaft.
- The gas temperature leaving the Turbine is in the range of **500 to 550 ° C**.
- The gas turbine shaft connects to the generator to produce electric power.



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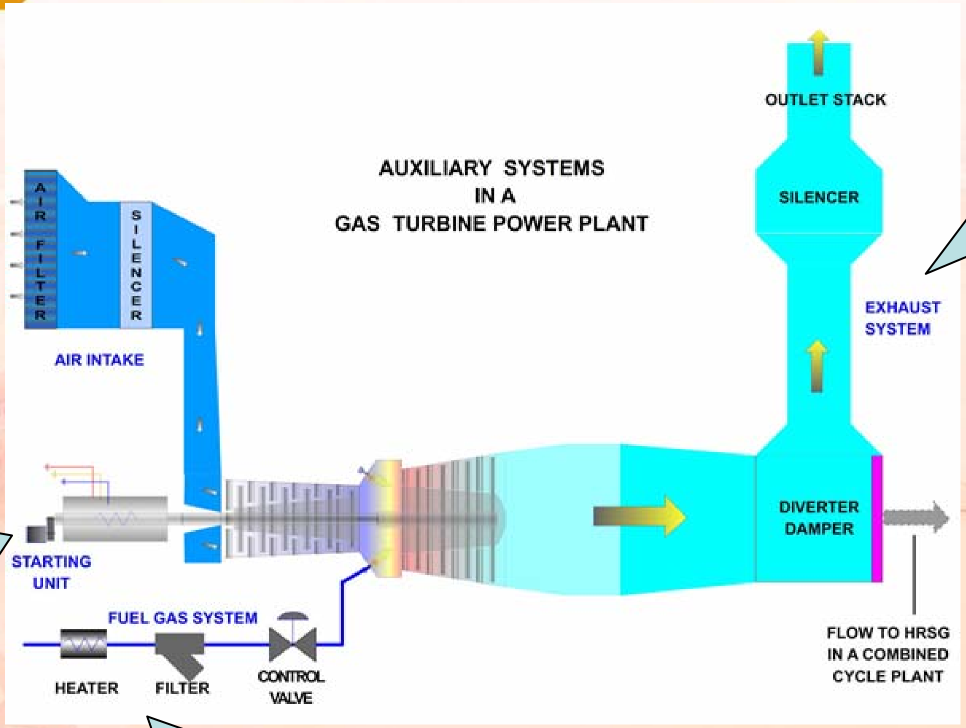
Auxiliary Components of Gas Turbine Power Plant

Air Intake System provides clean air into the compressor

Starting system provides the initial momentum for the Gas Turbine to reach the operating speed. This is similar to the starter motor of your car



Exhaust system discharges the hot gases to a level which is safe for the people and the environment

The Fuel system prepares a clean fuel for burning in the combustor. Gas Turbines normally burn Natural gas but can also fire diesel or distillate fuels

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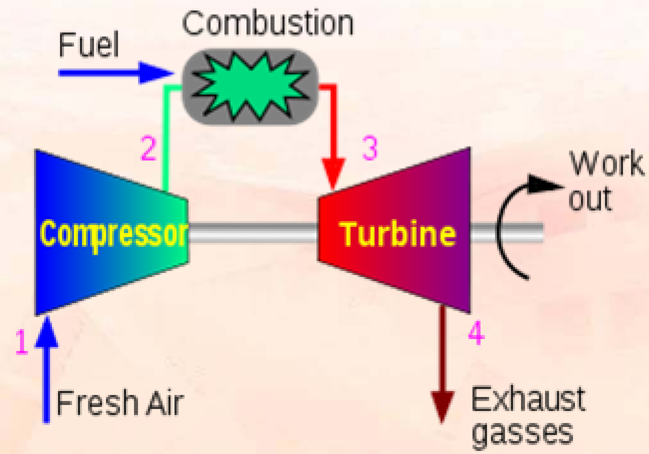
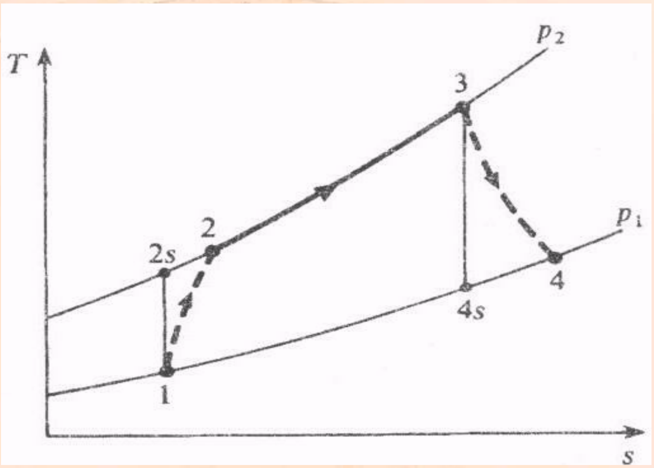
How a Gas Turbine Works?

- Fresh air at ambient conditions is drawn into the compressor, its temperature and pressure are raised.
- The high-pressure air proceeds into the combustion chamber, the fuel is burned at constant pressure.
- The resulting high-temperature gases then enter the turbine and expand to the atmospheric pressure while producing power.
- The exhaust gases leaving the turbine are thrown out (not re-circulated), causing the cycle to be classified as an open cycle.





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Air Standard Cycle

Why?

The actual cycle :

- Difficult to analyze due to the presence of complicating effects, such as friction.
- The working fluid remains a gas throughout the entire cycle, involves chemical analysis, causes more complicated analysis.
- The working fluid does not undergo a complete thermodynamic cycle, it is thrown out at the end of the cycle (as exhaust gases) instead of being returned to the initial state.
- Working on an open cycle.

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The Air Standard Assumptions

1. The working fluid is air, continuously circulates in a closed loop and behaves as an ideal gas.
2. All processes are internally reversible.
3. The combustion process is replaced by a heat-addition process from an external source.
4. The exhaust gas 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, 300 K. This assumption is called **cold-air-standard assumption**

$$\textit{For isentropic} \rightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\textit{Variable specific heat} \rightarrow \frac{P_2}{P_1} = \frac{P_{r2}}{P_{r1}}$$

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ACTUAL VS BRAYTON CYCLE

- The compression and expansion processes remain the same
- The combustion process is replaced by a constant-pressure heat-addition from an external source
- The exhaust process is replaced by a constant-pressure heat-rejection process to the ambient air.
- The ideal cycle that the working fluid undergoes this closed loop is the **Brayton cycle**.

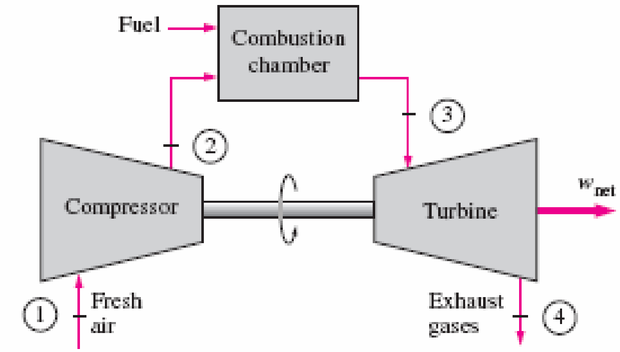

FIGURE 9–29
An open-cycle gas-turbine engine.

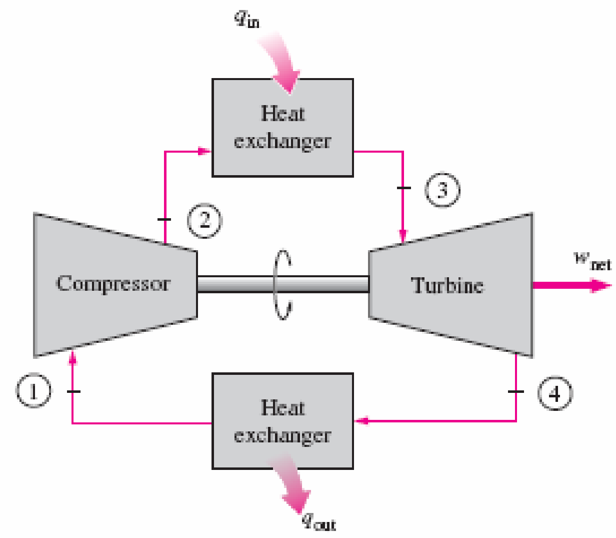

FIGURE 9–30
A closed-cycle gas-turbine engine.

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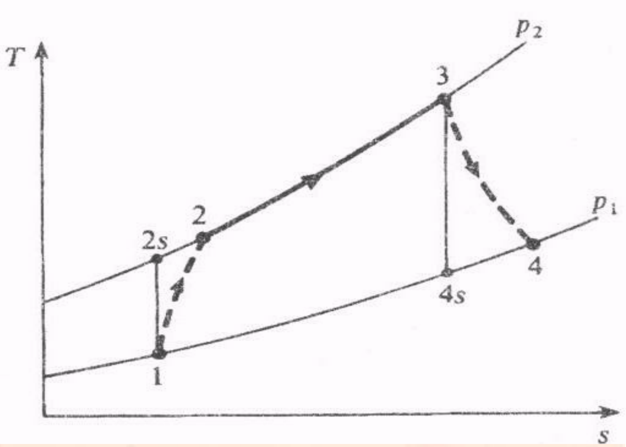
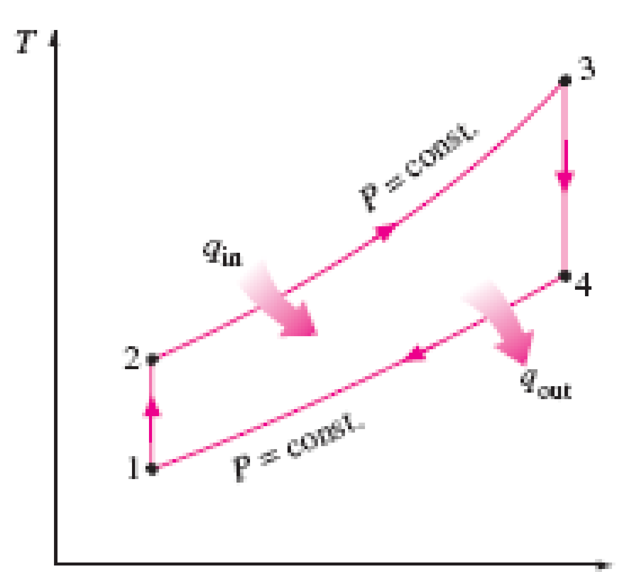
ACTUAL VS BRAYTON CYCLE



The Brayton cycle consists of four internally reversible processes:

- Process 1-2: isentropic compression (in a compressor)
- Process 2-3: constant-pressure heat-addition through a heat exchanger
- Process 3-4: isentropic expansion (in a turbine)
- Process 4-1: constant-pressure heat-rejection through a heat exchanger



(a) T-s diagram

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BRAYTON CYCLE – THE ANALYSIS

- All 4 processes of the Brayton cycle are executed in steady flow devices, thus, they should be analyzed as steady-flow processes.
- By neglecting the changes in kinetic and potential energies, the energy balance for a steady-flow process can be expressed, on a unit mass basis, as:

$$\sum q - \sum w = \Delta h$$

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{exit} - h_{inlet} = c_p (T_{exit} - T_{inlet})$$

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BRAYTON CYCLE – THE ANALYSIS

The energy balance for each process of the Brayton cycle can be expressed, on a unit mass basis, as:

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

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

$$w_{tur} = w_{34} = h_3 - h_4 = c_p\left(T_3 - T_4\right)$$

$$w_{com} = w_{12} = h_2 - h_1 = c_p\left(T_2 - T_1\right)$$

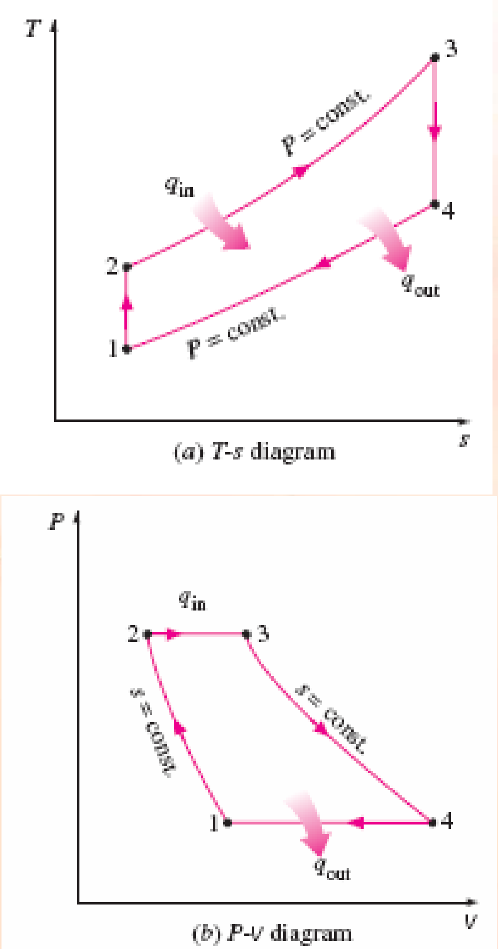

FIGURE 9–31

T-s and *P-v* diagrams for the ideal Brayton cycle.

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BRAYTON CYCLE – THE ANALYSIS

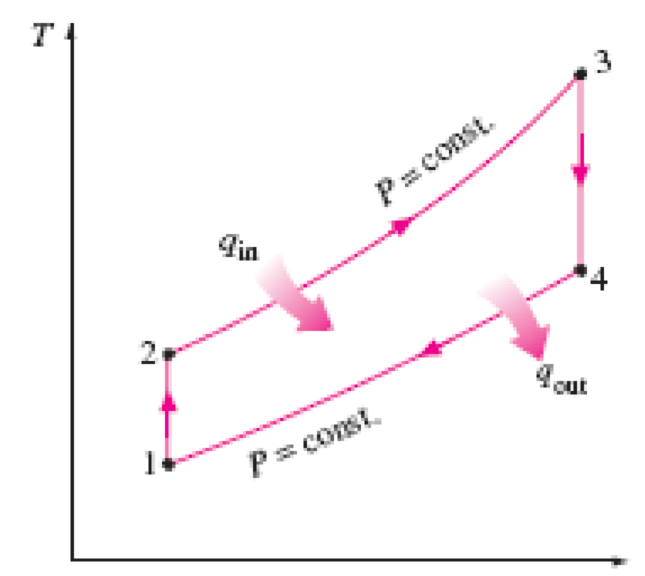
The first-law of thermodynamic states that, for a closed system undergoing a cycle, the net work output is equal to net heat input i.e. $w_{net} = q_{in} - q_{out}$

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

For isentropic processes, 1-2 and 3-4

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = \left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} = \frac{T_3}{T_4}$$

$$\Rightarrow (P_2 = P_3 \;\; and \;\; P_4 = P_1)$$



(a) T-s diagram

Since $P_2 = P_3$ and $P_4 = P_1$, thus

$$\frac{p_2}{p_1} = \frac{p_3}{p_4} = r_p = pressure\ ratio$$

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Thus,

$$T_2 = T_1 r_p^{\frac{k-1}{k}} \quad \text{and} \quad T_3 = T_4 r_p^{\frac{k-1}{k}}$$

Substituting into the thermal efficiency equation,

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

Also,

$$\frac{(T_4 - T_1)}{(T_3 - T_2)} = \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} = \frac{T_1}{T_2} \quad \Rightarrow \text{Thus, } \eta_{th} = 1 - \frac{T_1}{T_2}$$

Note: Only valid for ideal Brayton cycle – under the cold air-standard assumptions

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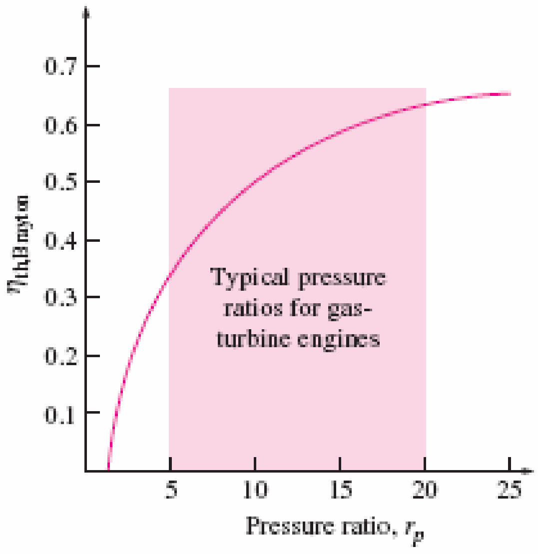
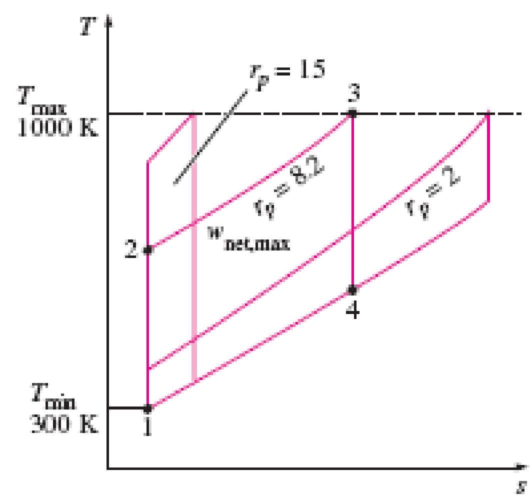
Parameters Affecting Thermal Efficiency

- The thermal efficiency of Brayton cycle depends on the pressure ratio, r_p of the gas turbine and the specific heat ratio, k of the working fluid.

- The thermal efficiency increases with both of these parameters, which is also the case for actual gas turbines.



- For the fixed turbine inlet temperature, T_3, the net work output increases with the r_P, reaches a maximum at $r_p = (T_{max} / T_{min})^{k/[2(k-1)]}$ and then starts to decrease
- In most common designs, the pressure ration of gas turbines ranges from 11 to 16.



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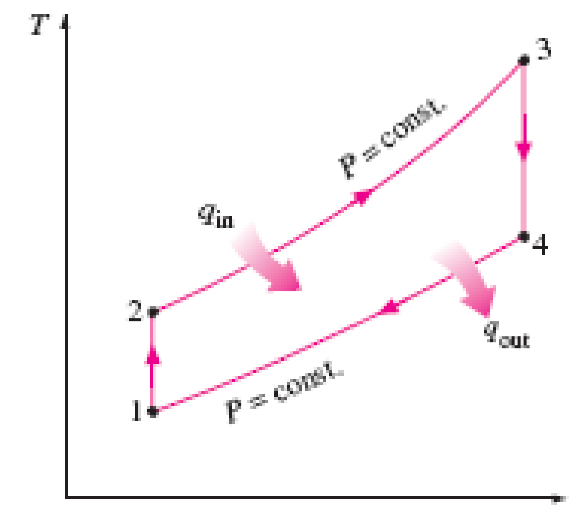
WORK RATIO

Work Ratio, r_w (air-standard assumptions) is defined as

$$r_w = \frac{w_{net}}{w_{turbine}} = \frac{w_{34} - w_{12}}{w_{34}} = \frac{c_p\left(T_3 - T_4\right) - c_p\left(T_2 - T_1\right)}{c_p\left(T_3 - T_4\right)} = 1 - \frac{\left(T_2 - T_1\right)}{\left(T_3 - T_4\right)}$$

We know that, $T_2 = T_1 . r_p^{\frac{k-1}{k}}$ and $T_4 = \dfrac{T_3}{r_p^{\frac{k-1}{k}}}$



(a) T-s diagram

Therefore,

$$r_w = 1 - \frac{T_1\left(r_p^{\frac{k-1}{k}} - 1\right)}{T_3\left(1 - \dfrac{1}{r_p^{\frac{k-1}{k}}}\right)} = 1 - \frac{T_1\left(r_p^{\frac{k-1}{k}} - 1\right) . r_p^{\frac{k-1}{k}}}{T_3\left(r_p^{\frac{k-1}{k}} - 1\right)}$$

$$= 1 - \frac{T_1}{T_3} . r_p^{\frac{k-1}{k}}$$

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BACK WORK RATIO

- BWR is defined as the ratio of compressor work to the turbine work

$$r_{bw} = \frac{w_{comp}}{w_{turbine}} = \frac{w_{12}}{w_{34}} = \frac{c_p(T_2 - T_1)}{c_p(T_3 - T_4)}$$

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

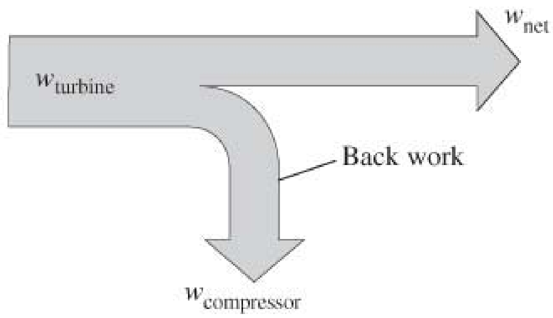

FIGURE 9–34
The fraction of the turbine work used to drive the compressor is called the back work ratio.

- The BWR in gas turbine power plant is very high, normally one-half of turbine work output is used to drive the compressor
- Thus required a larger turbine

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EXAMPLE 9-5 Pg 507

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.

Assumptions : Steady operating conditions, kinetic and potential energy changes are negligible

Analysis : The variation od specific heats with temperature is to be considered

a) The air temperature at the compressor and turbine exits are determined from isentropic relations

Process 1-2 : Isentropic compression

$$T_1 = 300K \rightarrow h_1 = 300.19 \text{ kJ/kg} , P_{r1} = 1.386$$

$$P_{r2} = \left(\frac{P_2}{P_1}\right) P_{r1} = (8)(1.386) = 11.09 \rightarrow T_2 = 540K$$

$$h_2 = 544.35 \text{ kJ/kg}$$

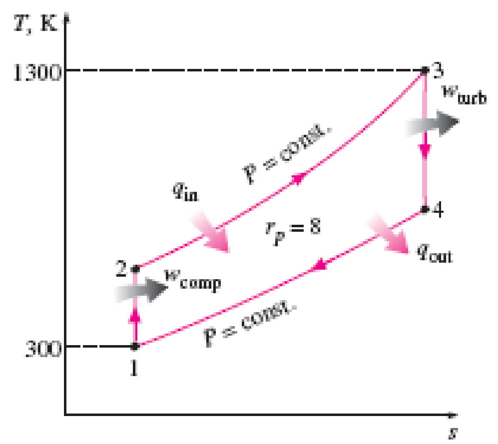

FIGURE 9–35
T-s diagram for the Brayton cycle discussed in Example 9–5.

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EXAMPLE 9-5 Pg 507

Process 3-4 : Isentropic expansion

$$T_3 = 1300K \rightarrow h_3 = 1395.97 \text{ kJ/kg}, P_{r3} = 330.9$$

$$P_{r4} = \left(\frac{P_4}{P_3}\right) P_{r3} = \left(\frac{1}{8}\right)(330.9) = 41.36 \rightarrow T_4 = 770K$$

$$h_4 = 789.37 \text{ kJ/kg}$$

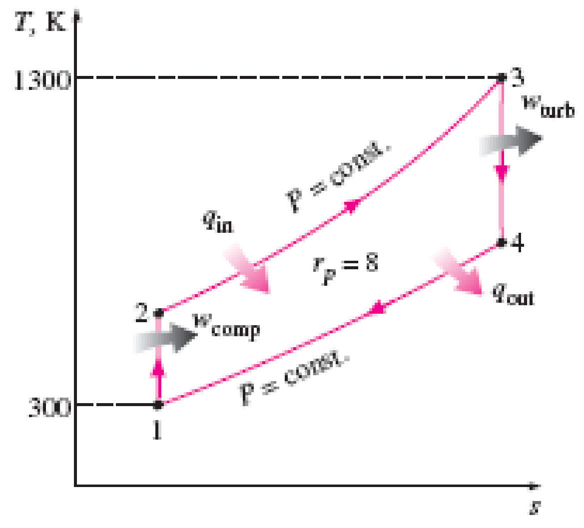

FIGURE 9–35

T-s diagram for the Brayton cycle discussed in Example 9–5.

(b) The backwork ratio

$$w_{comp} = h_2 - h_1 = 544.35 - 300.19 = 244.16 \text{ kJ/kg}$$

$$w_{turb} = h_3 - h_4 = 1395.97 - 789.37 = 606.60 \text{ kJ/kg}$$

$$r_{bw} = \frac{w_{comp}}{w_{turb}} = \frac{244.16}{606.60} = 0.403$$

Note : 40.3% of turbine output is used to drive the compressor

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EXAMPLE 9-5 Pg 507

(c) The thermal efficiency

$$q_{in} = q_{23} = h_3 - h_2 = 1395.97 - 544.35 = 851.62 \text{ kJ/kg}$$

$$w_{net} = w_{turb} - w_{comp} = 606.60 - 244.16 = 362.4 \text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{362.40}{851.62} = 0.426 \text{ or } 42.6\%$$

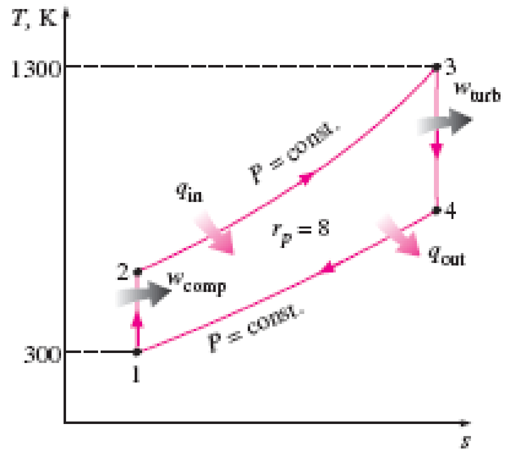

FIGURE 9–35
T-s diagram for the Brayton cycle discussed in Example 9–5.

Discussion Under the cold-air-standard assumptions (constant specific heat values at room temperature), the thermal efficiency would be, from Eq. 9–17,

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

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DEVIATION OF ACTUAL GAS TURBINE FROM IDEALIZED ONES

The differences between actual gas turbine and ideal Brayton cycle :

- Pressure drop during the heat-addition and heat rejection processes
- Larger actual work input to the compressor
- The actual work output from the turbine is less because of irriversibilities

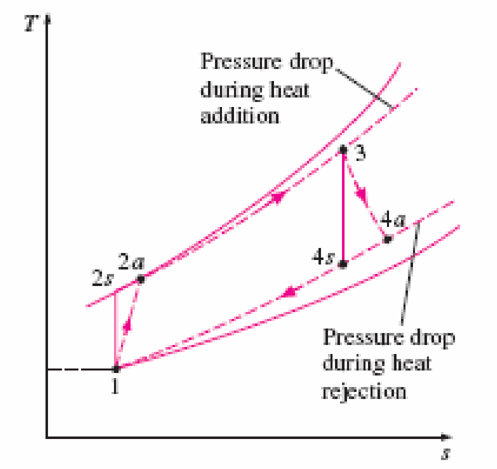

FIGURE 9–36

The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

Isentropic efficiency of compressor

$$\eta_c = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

Isentropic efficiency of turbine

$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

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EXAMPLE 9-6 Pg 509

Assuming a compressor efficiency of 80 percent and a turbine efficiency of 85 percent, determine (a) the back ratio (b) the thermal efficiency (c) the turbine exit temperature of the gas turbine cycle discussed in Example 9-5.

a) The back work ratio

$$w_{comp} = \frac{w_s}{\eta_c} = \frac{244.16}{0.80} = 305.20 \text{ kJ/kg}$$

$$w_{turb} = \eta_T w_s = (0.85)(606.60) = 515.61 \text{ kJ/k}$$

$$r_{bw} = \frac{w_{comp}}{w_{turb}} = \frac{305.20}{515.61} = 0.592 \text{ or } 59.2\%$$

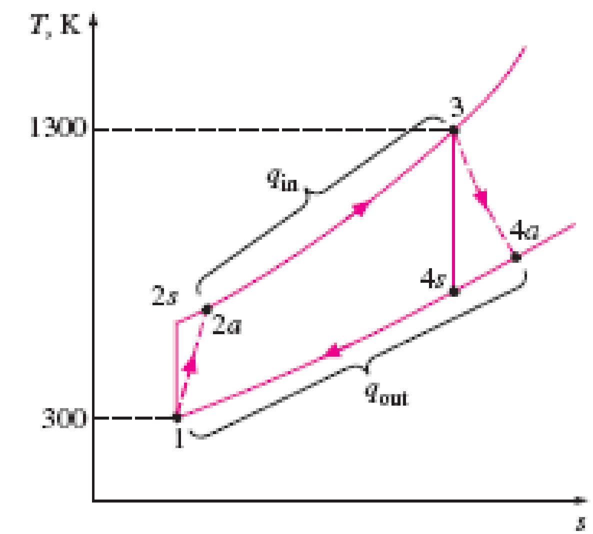

FIGURE 9–37

T-s diagram of the gas-turbine cycle discussed in Example 9–6.

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EXAMPLE 9-6 Pg 509

b) The thermal efficiency

$$w_{comp} = h_{2a} - h_1 \Rightarrow h_{2a}$$

$$h_{2a} = h_1 + w_{comp}$$

$$= 300.19 + 305.20$$

$$= 605.39 \quad (\text{and } T_{2a} = 598K \rightarrow \text{Table A-17})$$

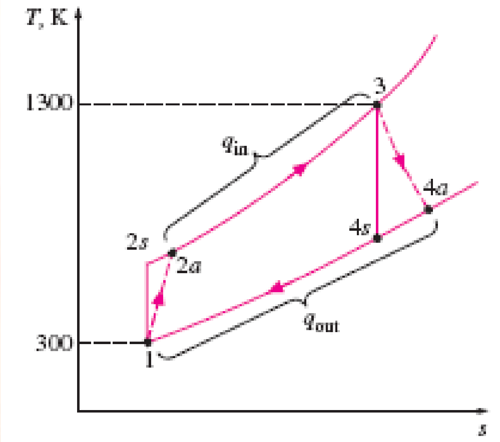

FIGURE 9–37
T-s diagram of the gas-turbine cycle discussed in Example 9–6.

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

$$w_{net} = w_{turb} - w_{comp} = 515.61 - 305.20 = 210.41 \text{ kJ/kg}$$

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

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EXAMPLE 9-6 Pg 509

c) The air temperature at the turbine exit, T_{4a}

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

From Table A-17, **T_{4a} = 853 K**

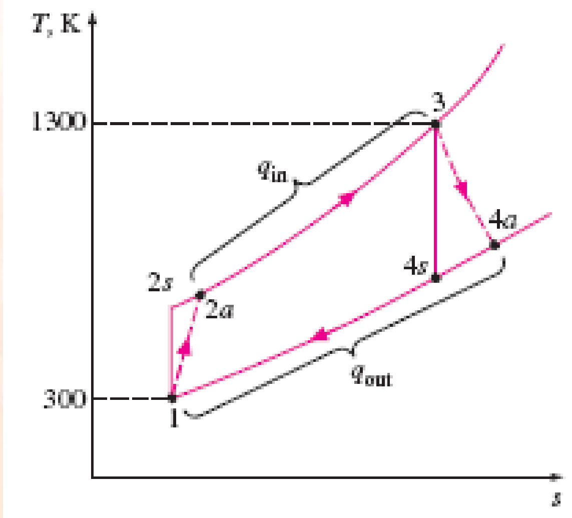

FIGURE 9–37

T-s diagram of the gas-turbine cycle discussed in Example 9–6.

ASSIGNMENT 5

9–89/90 (page 540)

Air enters the compressor of a gas-turbine engine at 300 K and 100 kPa, where it is compressed to 700 kPa and 580 K. Heat is transferred to air in the amount of 950 kJ/kg before it enters the turbine.

For a turbine efficiency of 86 percent, determine:

(a) the fraction of turbine work output used to drive the compressor,

(b) the thermal efficiency.

Assume:

(a) variable specific heats for air.

(b) constant specific heats at 300 K.

IMPROVEMENTS OF GAS TURBINE'S PERFORMANCE

The early gas turbines (1940s to 1959s) found only limited use despite their versatility and their ability to burn a variety of fuels, because its thermal efficiency was only about 17%. Efforts to improve the cycle efficiency are concentrated in three areas:

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

   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.

2. **Increasing the efficiencies of turbo-machinery components (turbines, compressors).**

   The advent of computers and advanced techniques for computer-aided design made it possible to design these components aerodynamically with minimal losses.

3. **Adding modifications to the basic cycle (inter-cooling, regeneration or recuperation, and reheating).**

   The simple-cycle efficiencies of early gas turbines were practically doubled by incorporating inter-cooling, regeneration (or recuperation), and reheating.

BRAYTON CYCLE WITH REGENERATION

- Temperature of the exhaust gas is **higher** than the temperature of the air leaving the compressor.
- The air leaving the compressor can be pre-heated by the hot exhaust gases in a **counter-flow** heat exchanger (a *regenerator* or *recuperator*) – a process called **regeneration.**
- The thermal efficiency of the Brayton cycle **increases** due to regeneration since less fuel is used for the same work output.

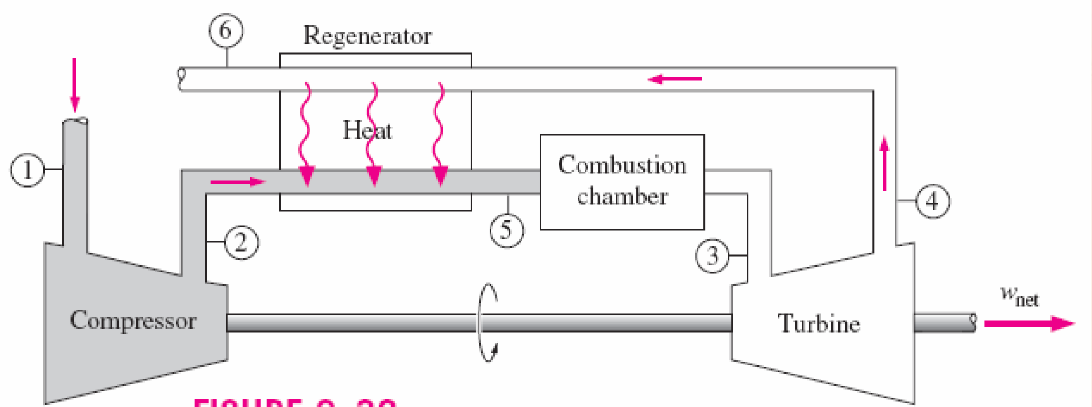

FIGURE 9–38

A gas-turbine engine with regenerator.

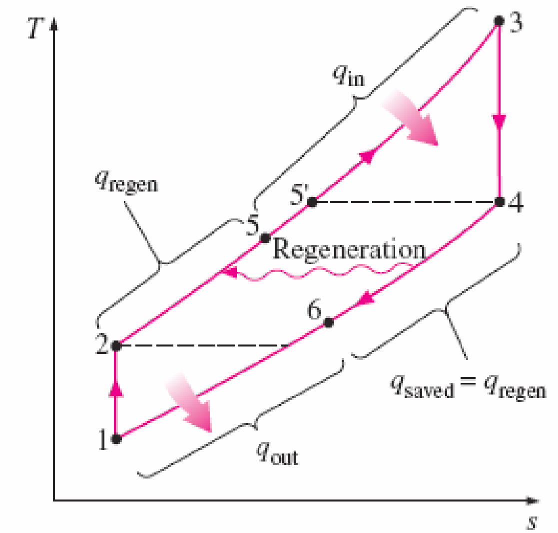

FIGURE 9–39

T-*s* diagram of a Brayton cycle with regeneration.

Note:

The use of a regenerator is recommended only when the turbine exhaust temperature is higher than the compressor exit temperature.

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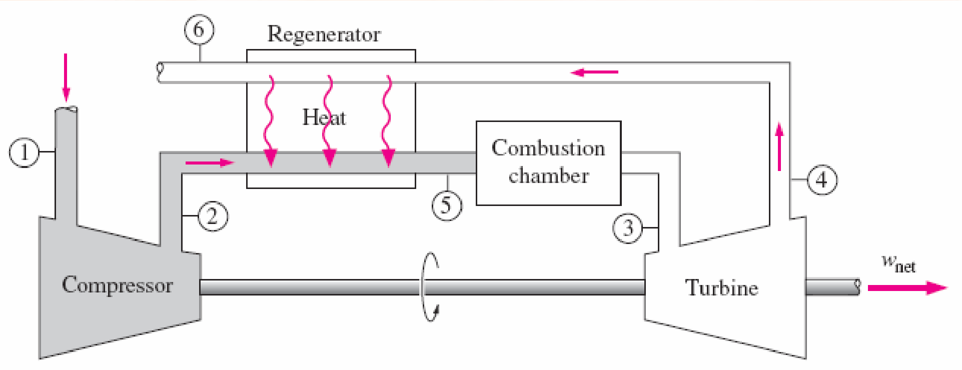
BRAYTON CYCLE WITH REGENERATION

Effectiveness of the Regenerator

Assuming the regenerator is well insulated and changes in kinetic and potential energies are negligible, the **actual** and **maximum heat transfers** from the exhaust gases to the air can be expressed as

$$q_{regen,act} = h_5 - h_2$$

$$q_{regen,max} = h_{5'} - h_2 = h_4 - h_2$$



Effectiveness of the regenerator,

$$\varepsilon = \frac{q_{regen,act}}{q_{regen,max}} = \frac{h_5 - h_2}{h_4 - h_2}$$

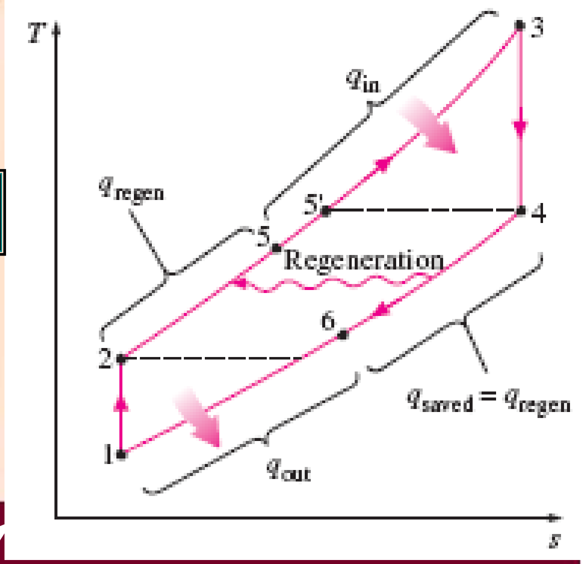
Effectiveness under **cold-air standard** assumptions,

$$\varepsilon = \frac{T_5 - T_2}{T_4 - T_2}$$

Note : If ε = 100%, $q_{regen,act} = q_{regen,max}$

Thermal efficiency under **cold-air standard** assumptions,

$$\eta_{th,regen} = 1 - \left(\frac{T_1}{T_3}\right)\left(r_p\right)^{(k-1)/k}$$



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EXAMPLE 9-7 Pg 512

Determine the thermal efficiency of the gas-turbine described in Example 9–6 if a regenerator having an effectiveness of 80 percent is installed.

Analysis The *T*-*s* diagram of the cycle is shown in Fig. 9–41. We first determine the enthalpy of the air at the exit of the regenerator, using the definition of effectiveness:

$$\epsilon = \frac{h_5 - h_{2a}}{h_{4a} - h_{2a}}$$

$$0.80 = \frac{(h_5 - 605.39)\text{ kJ/kg}}{(880.36 - 605.39)\text{ kJ/kg}} \rightarrow h_5 = 825.37\text{ kJ/kg}$$

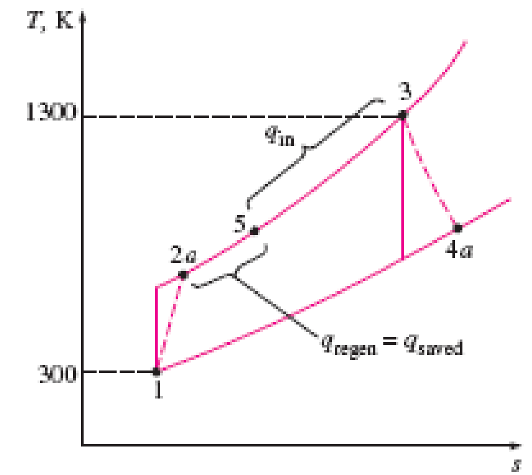

FIGURE 9–41

T-s diagram of the regenerative Brayton cycle described in Example 9–7.

Thus,

$$q_{in} = h_3 - h_5 = (1395.97 - 825.37)\text{ kJ/kg} = 570.60\text{ kJ/kg}$$

This represents a savings of 220.0 kJ/kg from the heat input requirements. The addition of a regenerator (assumed to be frictionless) does not affect the net work output. Thus,

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{210.41\text{ kJ/kg}}{570.60\text{ kJ/kg}} = 0.369\text{ or }36.9\%$$

Note : η_{th} has gone up from 26.6% to 36.9%

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Prob. 9-110 Pg 542

Air enters the compressor of a regenerative gas turbine engine at 310 K and 100 kPa, where it is compressed to 900 kPa and 650 K. The generator has an effectiveness of 80 percent and the air enters the turbine at 1400 K. For a turbine efficiency of 90 percent, determine:

a) The amount of heat transfer in the generator
b) The thermal efficiency
c) Assume variable specifics heats for air.

Answer : 193 kJ/kg , 40.0%

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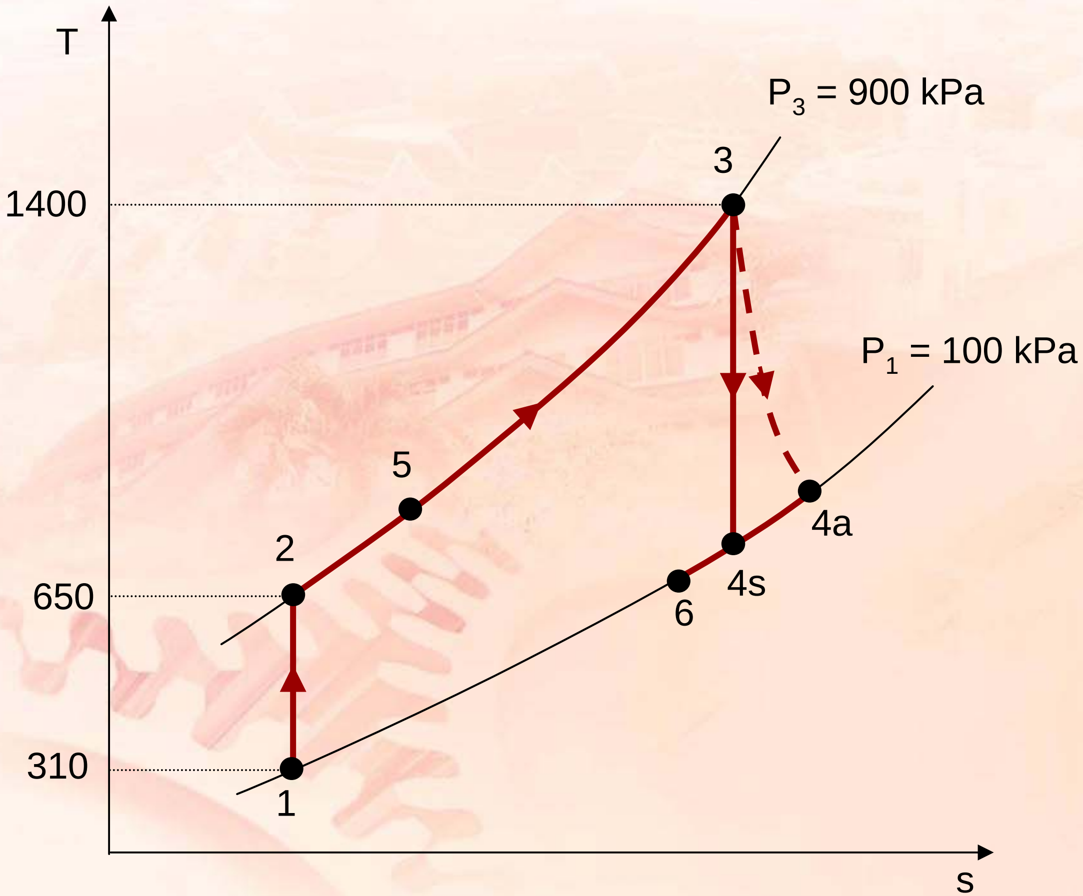
Prob. 9-110 Pg 542



$$\varepsilon = \frac{h_5 - h_2}{h_{4a} - h_2} = 0.72$$

$$\eta_T = \frac{h_3 - h_{4a}}{h_3 - h_{4s}} = 0.86$$

$$q_{gen} = h_5 - h_2$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{w_{turb} - w_{comp}}{q_{in}}$$

BRAYTON CYCLE WITH INTERCOOLING, REHEATING, & REGENERATION

The **net work output** of a gas-turbine cycle can be increased by either:

a) decreasing the compressor work, or
b) increasing the turbine work, or
c) both.

The compressor work input can be decreased by carrying out the compression process in stages and cooling the gas in between, using **multistage compression** with **intercooling**.

The work output of a turbine can be increased by expanding the gas in stages and reheating it in between, utilizing a **multistage expansion** with **reheating.**

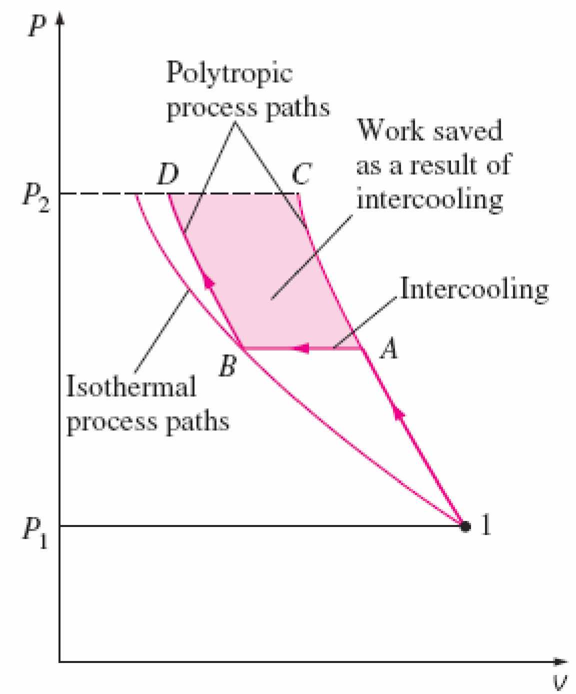


FIGURE 9–42

Comparison of work inputs to a single-stage compressor (1*AC*) and a two-stage compressor with intercooling (1*ABD*).

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BRAYTON CYCLE WITH INTERCOOLING, REHEATING, & REGENERATION

Physical arrangement of an **ideal two-stage** gas-turbine cycle with intercooling, reheating, and regeneration

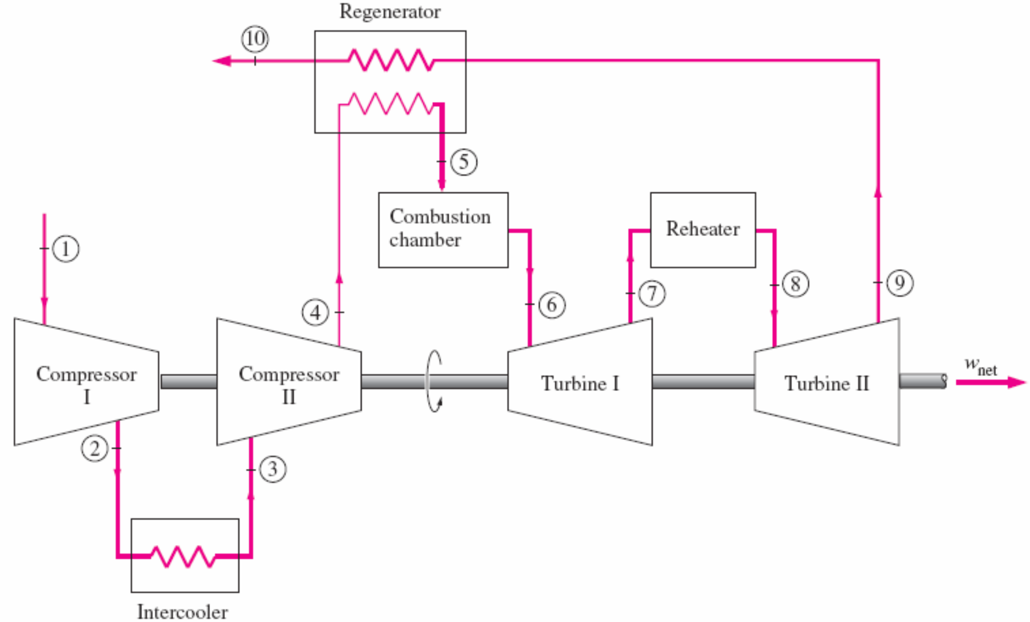
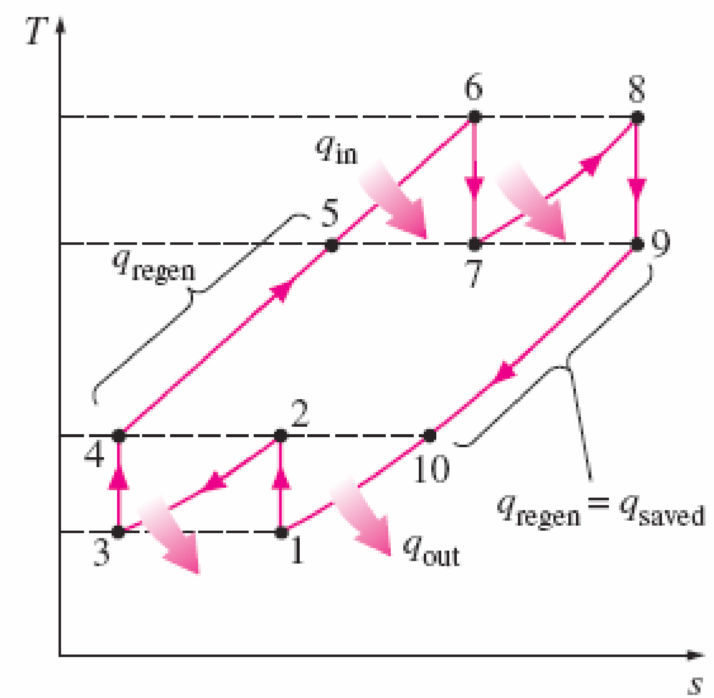


FIGURE 9–43

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



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BRAYTON CYCLE WITH INTERCOOLING, REHEATING, & REGENERATION

Conditions for Best Performance

The work input to a two-stage compressor is **minimized** when equal pressure ratios are maintained across each stage. This procedure also maximizes the turbine work output.Thus, for best performance,

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad \textbf{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9}$$

- Intercooling and reheating always **decreases** thermal efficiency unless are accompanied by **regeneration.**
- Therefore, intercooling and reheating are always **used in conjunction** with regeneration.

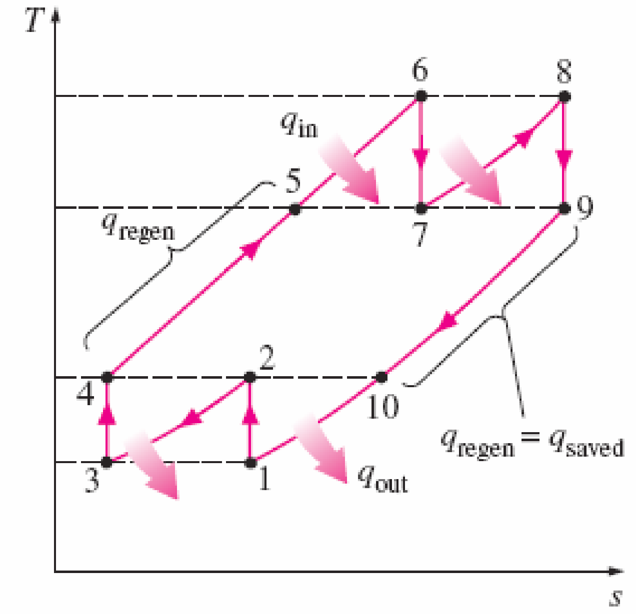

FIGURE 9–44

T-s diagram of an ideal gas-turbine cycle with intercooling, reheating, and regeneration.

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EXAMPLE 9-8 Pg 515

An ideal gas-turbine cycle with two stages of compression and two stages of expansion has an overall pressure ratio of 8. Air enters each stage of the compressor at 300 K and each stage of the turbine at 1300 K. Determine the back work ratio and the thermal efficiency of this gas-turbine cycle, assuming (*a*) no regenerators and (*b*) an ideal regenerator with 100 percent effectiveness. Compare the results with those obtained in Example 9–5.

Assumptions 1 Steady operating conditions exist. 2 The air-standard assumptions are applicable. 3 Kinetic and potential energy changes are negligible.

Analysis The *T-s* diagram of the ideal gas-turbine cycle described is shown in Fig. 9–46. We note that the cycle involves two stages of expansion, two stages of compression, and regeneration.

For two-stage compression and expansion, the work input is minimized and the work output is maximized when both stages of the compressor and the turbine have the same pressure ratio. Thus,

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} = \sqrt{8} = 2.83 \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9} = \sqrt{8} = 2.83$$

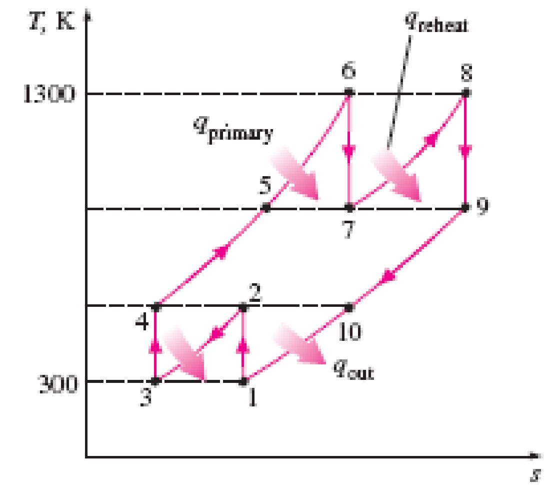

FIGURE 9–46

T-s diagram of the gas-turbine cycle discussed in Example 9–8.

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EXAMPLE 9-8 Pg 515

Air enters each stage of the compressor at the same temperature, and each stage has the same isentropic efficiency (100 percent in this case). Therefore, the temperature (and enthalpy) of the air at the exit of each compression stage will be the same. A similar argument can be given for the turbine. Thus,

At inlets: $T_1 = T_3, \quad h_1 = h_3 \quad \text{and} \quad T_6 = T_8, \quad h_6 = h_8$

At exits: $T_2 = T_4, \quad h_2 = h_4 \quad \text{and} \quad T_7 = T_9, \quad h_7 = h_9$

(*a*) In the absence of any regeneration, the back work ratio and the thermal efficiency are determined by using data from Table A–17 as follows:

$$T_1 = 300\text{ K} \rightarrow h_1 = 300.19\text{ kJ/kg}$$

$$P_{r1} = 1.386$$

$$P_{r2} = \frac{P_2}{P_1}P_{r1} = \sqrt{8}(1.386) = 3.92 \rightarrow T_2 = 403.3\text{ K}$$

$$h_2 = 404.31\text{ kJ/kg}$$

$$T_6 = 1300\text{ K} \rightarrow h_6 = 1395.97\text{ kJ/kg}$$

$$P_{r6} = 330.9$$

$$P_{r7} = \frac{P_7}{P_6}P_{r6} = \frac{1}{\sqrt{8}}(330.9) = 117.0 \rightarrow T_7 = 1006.4\text{ K}$$

$$h_7 = 1053.33\text{ kJ/kg}$$

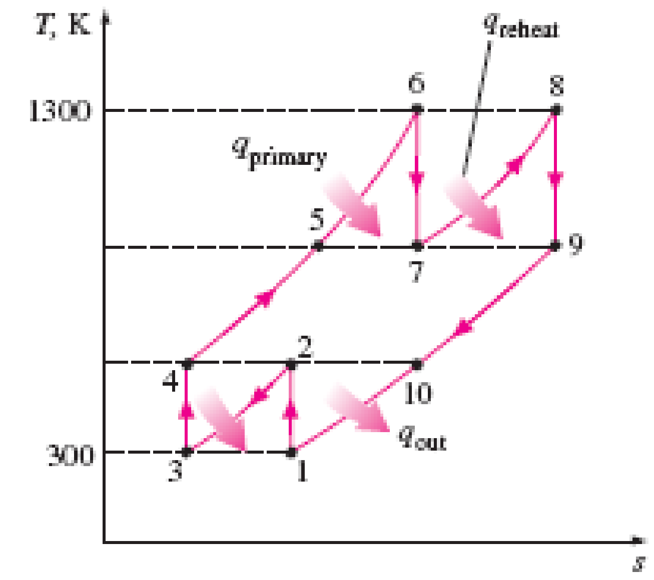

FIGURE 9–46

T-s diagram of the gas-turbine cycle discussed in Example 9–8.

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EXAMPLE 9-8 Pg 515

Then

$$w_{\text{comp,in}} = 2(w_{\text{comp,in,I}}) = 2(h_2 - h_1) = 2(404.31 - 300.19) = 208.24 \text{ kJ/kg}$$

$$w_{\text{turb,out}} = 2(w_{\text{turb,out,I}}) = 2(h_6 - h_7) = 2(1395.97 - 1053.33) = 685.28 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{turb,out}} - w_{\text{comp,in}} = 685.28 - 208.24 = 477.04 \text{ kJ/kg}$$

$$q_{\text{in}} = q_{\text{primary}} + q_{\text{reheat}} = (h_6 - h_4) + (h_8 - h_7)$$

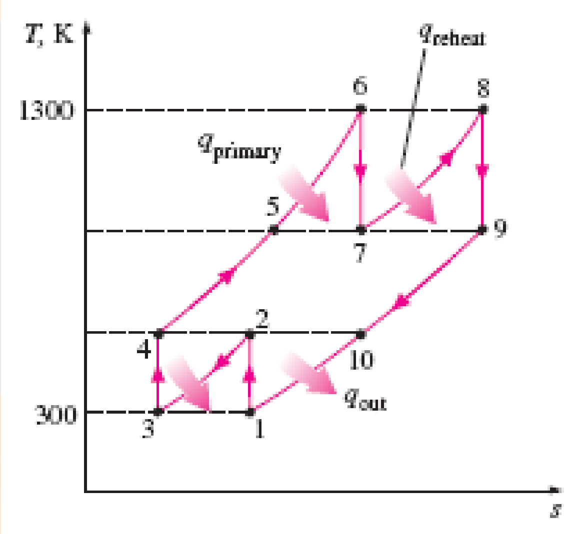
$$= (1395.97 - 404.31) + (1395.97 - 1053.33) = 1334.30 \text{ kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{comp,in}}}{w_{\text{turb,out}}} = \frac{208.24 \text{ kJ/kg}}{685.28 \text{ kJ/kg}} = \mathbf{0.304 \text{ or } 30.4\%}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{477.04 \text{ kJ/kg}}{1334.30 \text{ kJ/kg}} = \mathbf{0.358 \text{ or } 35.8\%}$$



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EXAMPLE 9-8 Pg 515

(*b*) The addition of an ideal regenerator (no pressure drops, 100 percent effectiveness) does not affect the compressor work and the turbine work. Therefore, the net work output and the back work ratio of an ideal gas-turbine cycle are identical whether there is a regenerator or not. A regenerator, however, reduces the heat input requirements by preheating the air leaving the compressor, using the hot exhaust gases. In an ideal regenerator, the compressed air is heated to the turbine exit temperature T_9 before it enters the combustion chamber. Thus, under the air-standard assumptions, $h_5 = h_7 = h_9$.

The heat input and the thermal efficiency in this case are

$$q_{in} = q_{primary} + q_{reheat} = (h_6 - h_5) + (h_8 - h_7)$$
$$= (1395.97 - 1053.33) + (1395.97 - 1053.33) = 685.28 \text{ kJ/kg}$$

and

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{477.04 \text{ kJ/kg}}{685.28 \text{ kJ/kg}} = 0.696 \text{ or } 69.6\%$$

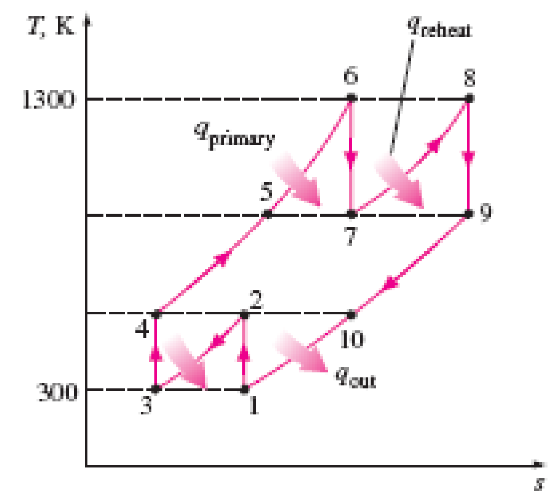

FIGURE 9–46
T-s diagram of the gas-turbine cycle discussed in Example 9–8.

Prob. 9–121 (page 543)

Consider an ideal gas-turbine cycle with **two stages** of compression and two stages of expansion. The pressure ratio across each stage of the compressor and turbine is 3. The air enters each stage of the compressor at 300 K and each stage of the turbine at 1200 K. Determine:

a) the **back work ratio**, and

b) the **thermal efficiency** of the cycle

assuming:

1. no regenerator is used, and
2. a regenerator with **75 percent** effectiveness is used.

Use a **variable specific heats** assumption.

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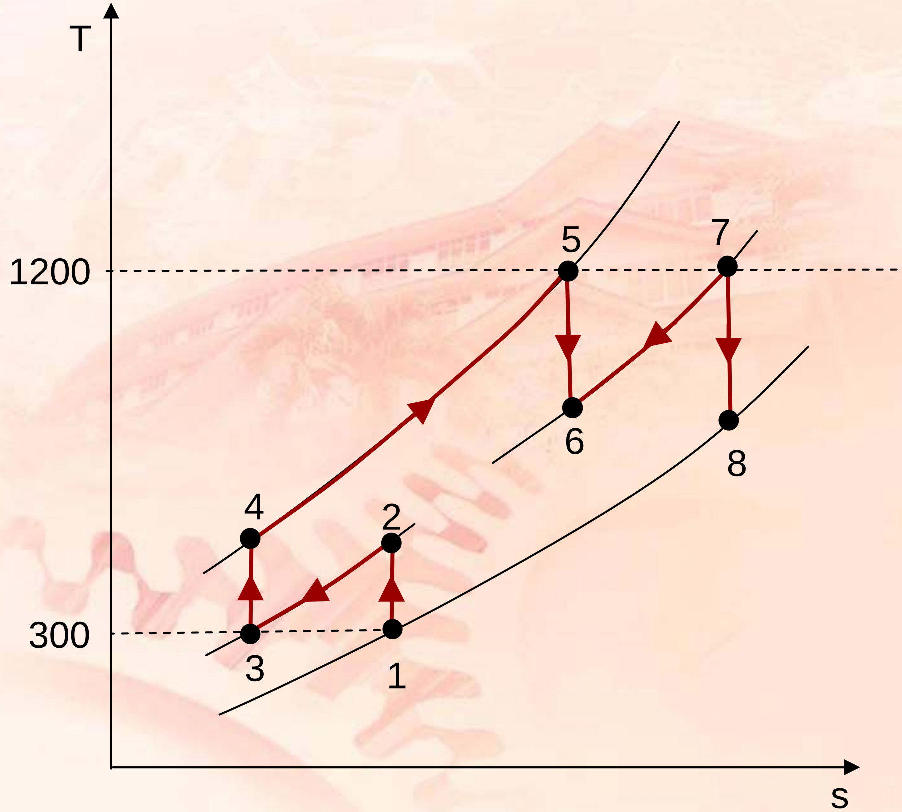
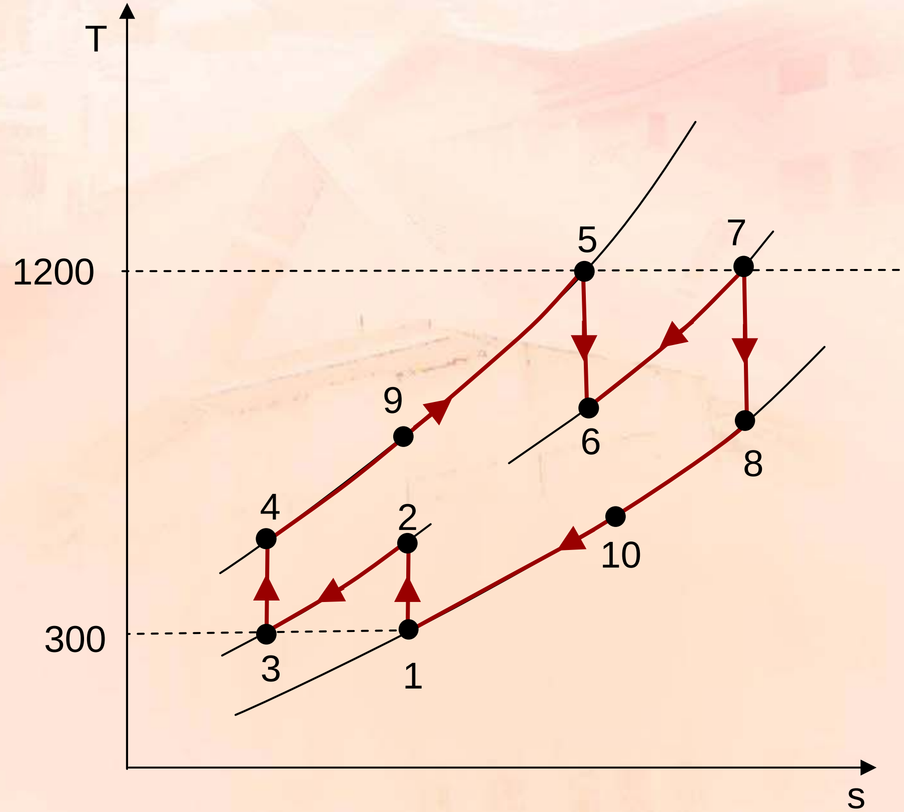
Prob. 9–124 (page 556)

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} = \frac{P_5}{P_6} = \frac{P_7}{P_8} = 3$$





Q1 FINAL EXAM APRIL 2010

(b) A Brayton cycle with regeneration using air as the working fluid has a pressure ratio of 8. The air temperatures at entry to the compressor and at entry to the turbine are 310 K and 1200 K, respectively. Assume an isentropic efficiency of 80 % for the compressor, 85 % for the turbine, and the regenerator effectiveness of 70 %. Accounting for the variation of specific heats with temperature, determine:

(i) the air temperature at the turbine exit, (^{o}C)

(ii) the net power output if 100 kg/s of air flows through the cycle, (kW)

(iii) the thermal efficiency, (%)

(18 marks)

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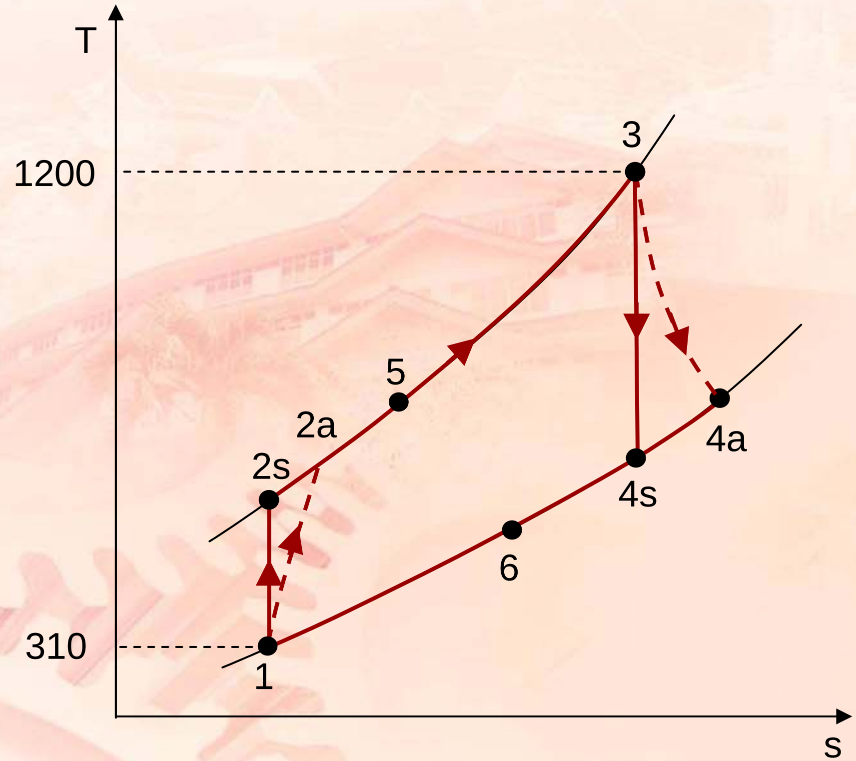
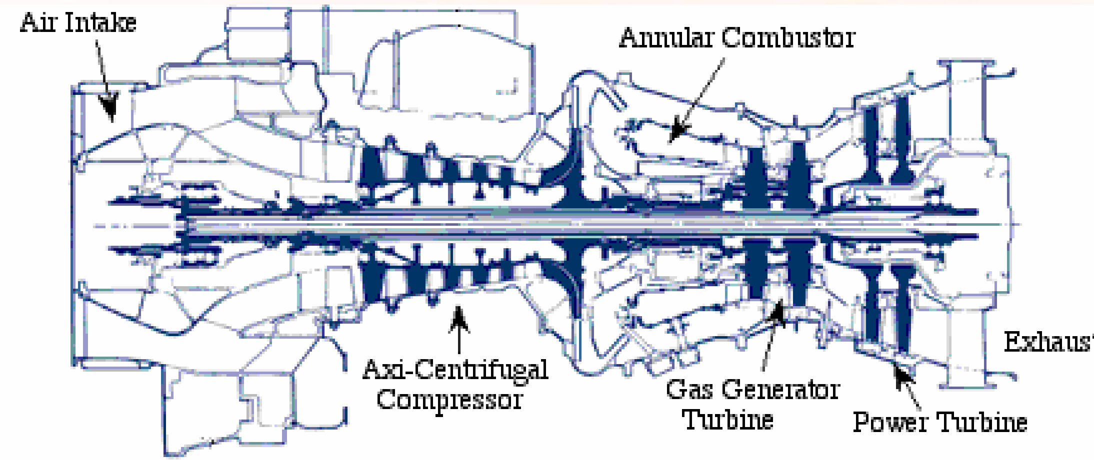


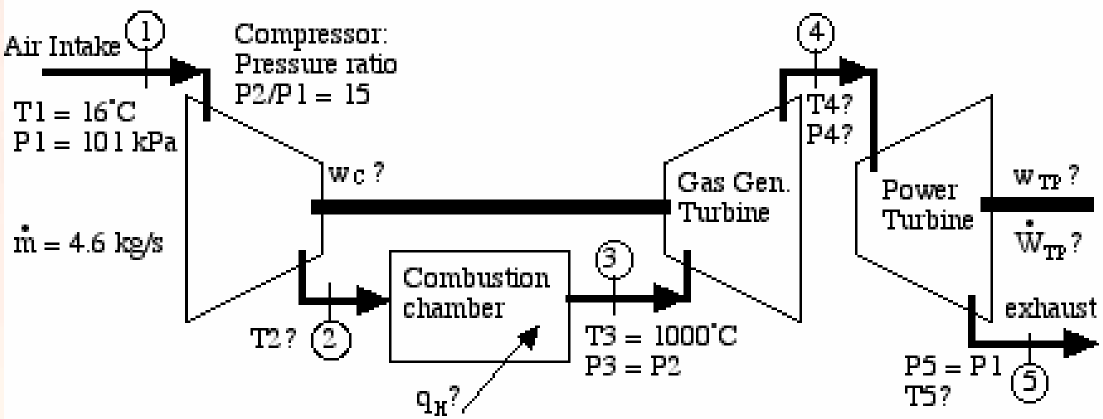
$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = 0.80$$

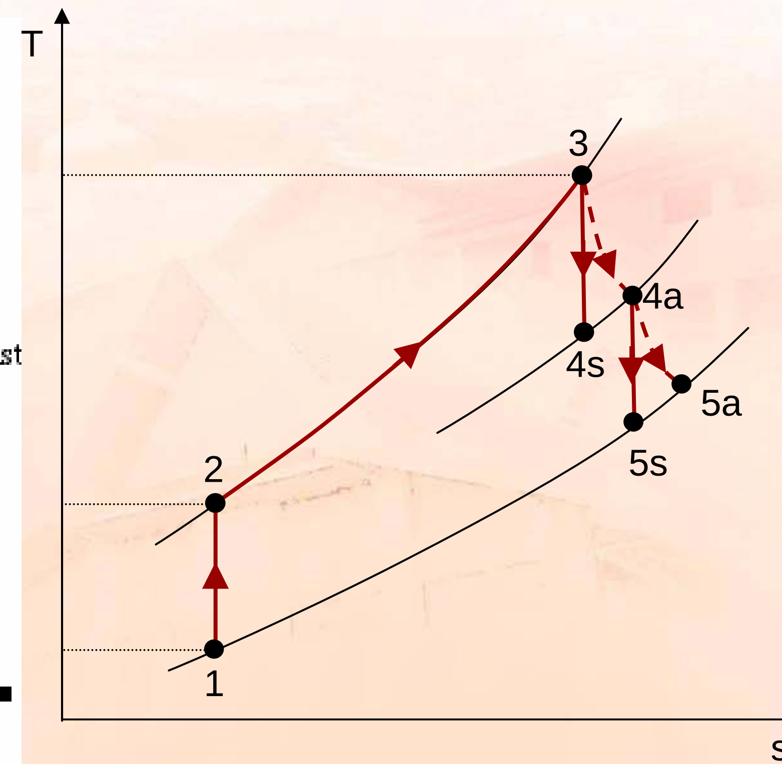
$$\eta_T = \frac{h_3 - h_{4a}}{h_3 - h_{4s}} = 0.85$$

$$\varepsilon = \frac{h_5 - h_{2a}}{h_{4a} - h_{2a}} = 0.70$$

Two-Stage Expansion







$$w_{comp} = w_{turb,HP}$$

$$w_{net} = w_{turb,LP}$$

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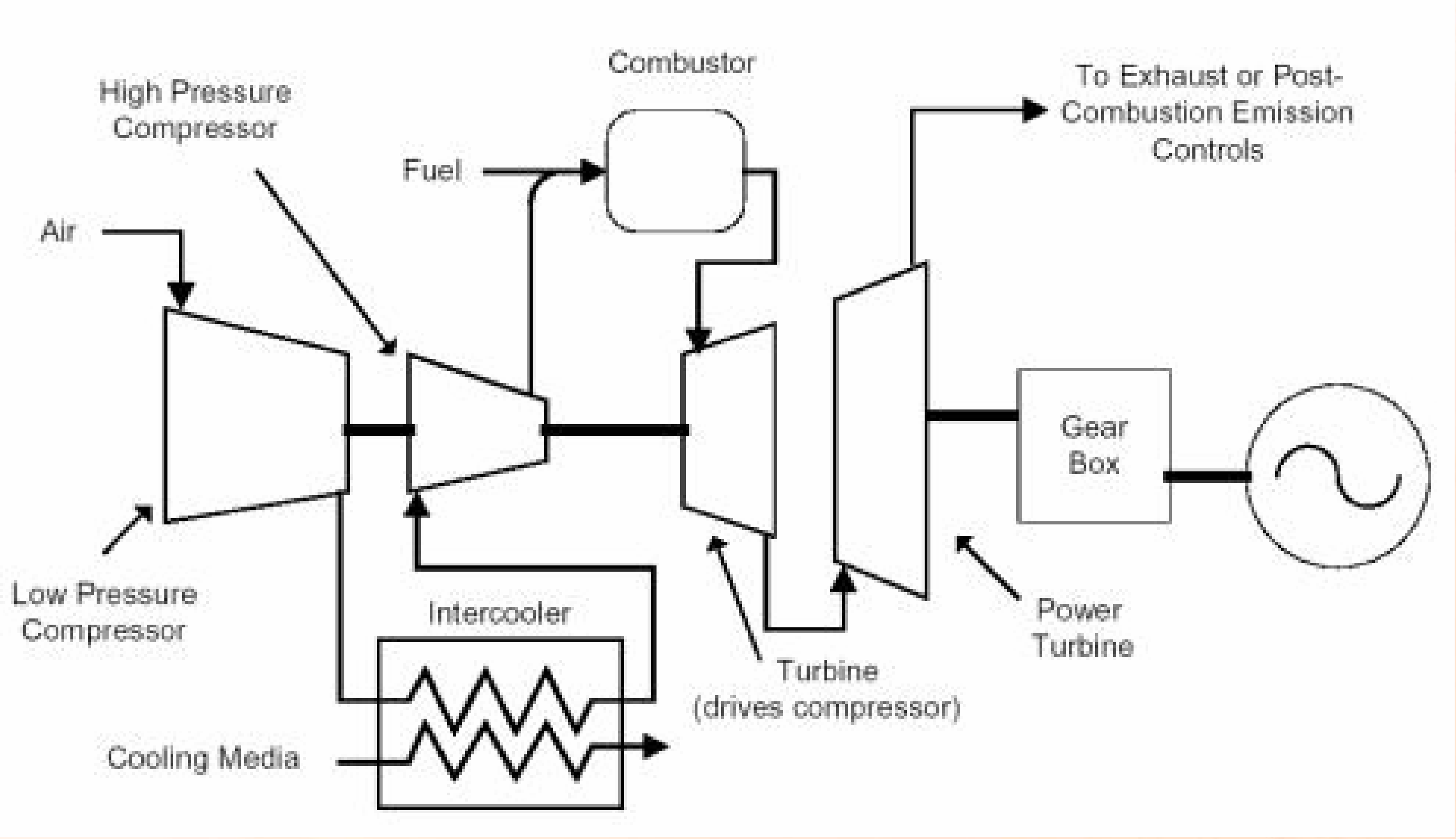
Two-Stage Compression, Two-stage expansion



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