Lecture 9: Gas Power Systems – Cont….

Course: MECH-422 – Power Plants

Instructor: Kashif Liaqat

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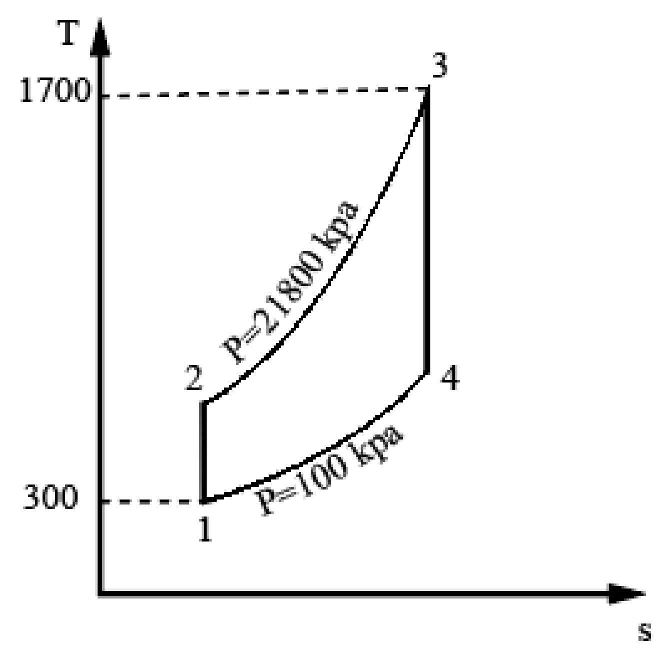
BUITEMS – DEPARTMENT OF MECHANICAL ENGINEERING

Example 9.1

Currently, one of the world's largest and most efficient gas turbines is model 9HA.02 introduced by General Electric (GE) in 2014 with the power production of 470 MW and the efficiency of 41.5%. The compressor pressure ratio is 21.8. The turbine inlet temperature (TIT) is 1700 K. Air enters the cycle with the mass flow rate of 978 kg/s at the temperature of 300 K and the pressure of 100 kPa. Assuming that the gas turbine operates as an air-standard Brayton cycle, determine the compressor and turbine outlet temperatures (in K), the energy transfer per second in each component (in kW), the net power generation (in kW), the back work ratio, and the efficiency of the cycle (a) using the air property table data and (b) assuming constant specific heats for air. Compare your results from parts (a) and (b) with the same data provided by the manufacturer.



Comparison of the results of parts a and b in Example 9.1

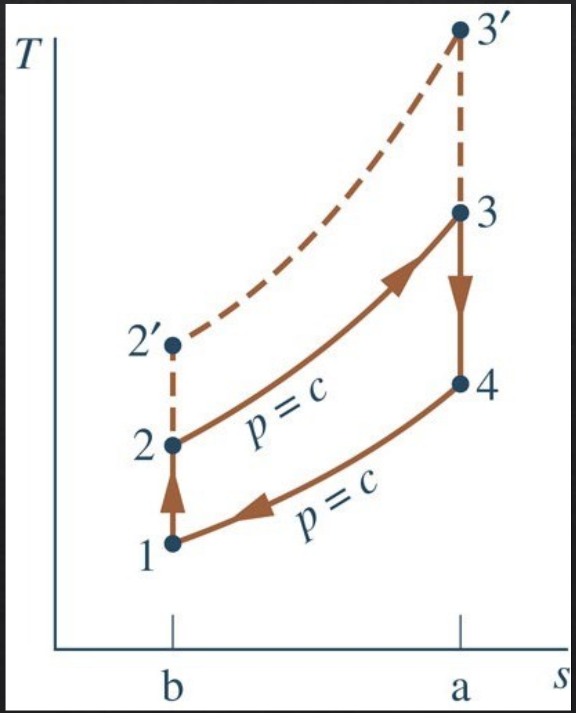
Parameter	Air-Standard Brayton Cycle	Cold Air-Standard Brayton Cycle
T_2 (K)	709	724
T_4 (K)	797	705
$\dot{Q}_L$ (MW)	507	397
$\dot{Q}_H$ (MW)	1132	959
$\dot{W}_{Comp}$ (MW)	413	416
$\dot{W}_{Turb}$ (MW)	1038	977
$\dot{W}_{Net}$ (MW)	625	561
BWR (%)	39.8	42.6
η_{Th} (%)	55.2	58.5

Comparison of selected parameters of the air-standard Brayton cycle and the cold air-standard Brayton cycle with the actual gas turbine specifications in Example 9.1

Parameter	Net Power Output (MW)	Efficiency (%)	Exhaust Temperature (K)	Exhaust Energy (MW)
Actual gas turbine	470	41.5	892	712
Air-standard Brayton cycle	625 (33%)	55.2 (33%)	797 (−11%)	507 (−29%)
Cold air-standard Brayton cycle	561 (19%)	58.5 (40%)	705 (−21%)	397 (−44%)

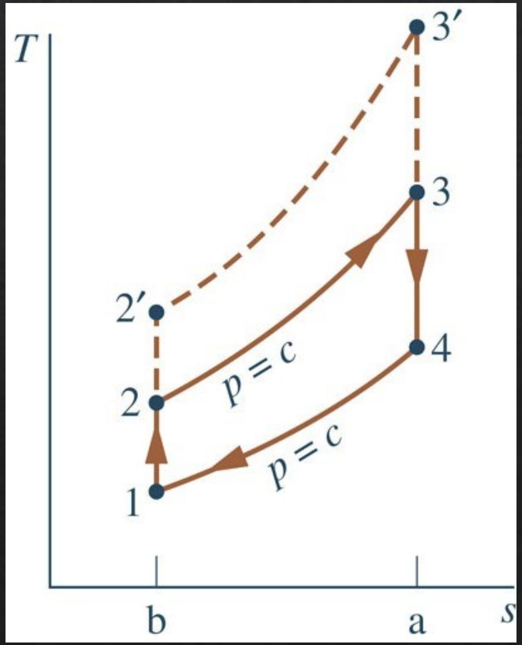
Effects of Compressor Pressure Ratio on Brayton Cycle Performance (1 of 7)

► That the compressor pressure ratio, p_2/p_1, is an important operating parameter for gas turbines is brought out simply by the following discussions centering on the T-s diagram:



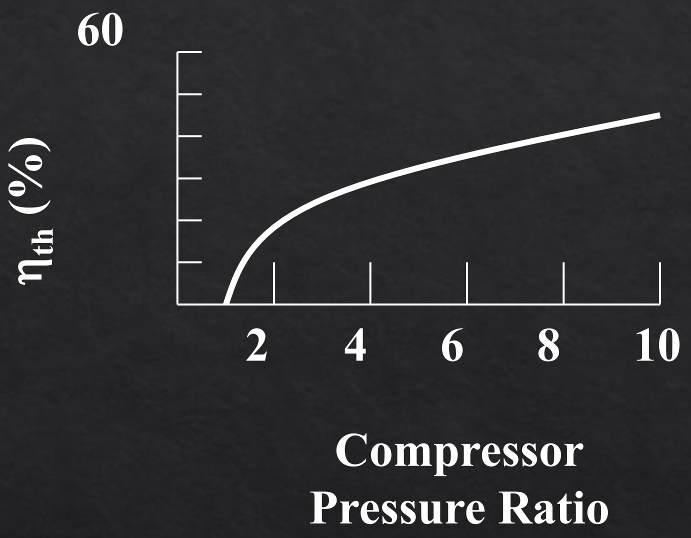
Effects of Compressor Pressure Ratio on Brayton Cycle Performance (2 of 7)

- Increasing the compressor pressure ratio from p_2/p_1 to $p_{2'}/p_1$ changes the cycle from **1-2-3-4-1** to **1-2′-3′-4-1**.
  - Since the average temperature of heat addition is greater in cycle **1-2′-3′-4-1**, and both cycles have the same heat rejection process, cycle **1-2′-3′-4-1** has the greater thermal efficiency.



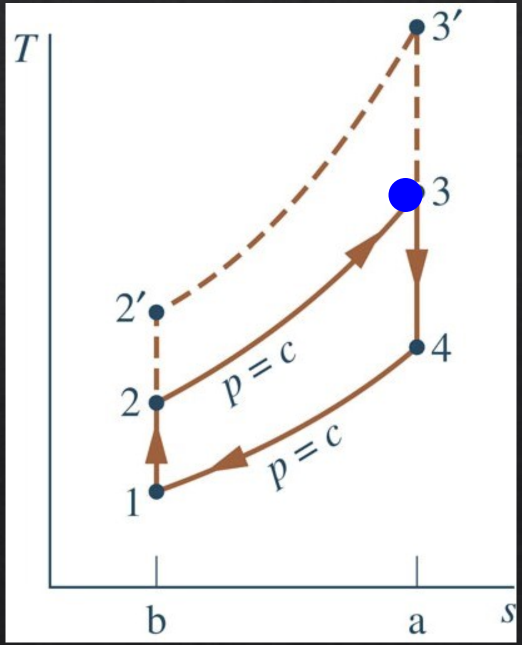
Effects of Compressor Pressure Ratio on Brayton Cycle Performance (3 of 7)

- Accordingly, the Brayton cycle thermal efficiency increases as the compressor pressure ratio increases.



Effects of Compressor Pressure Ratio on Brayton Cycle Performance (4 of 7)

- The turbine inlet temperature also increases with increasing compressor ratio – from T_3 to $T_{3'}$.



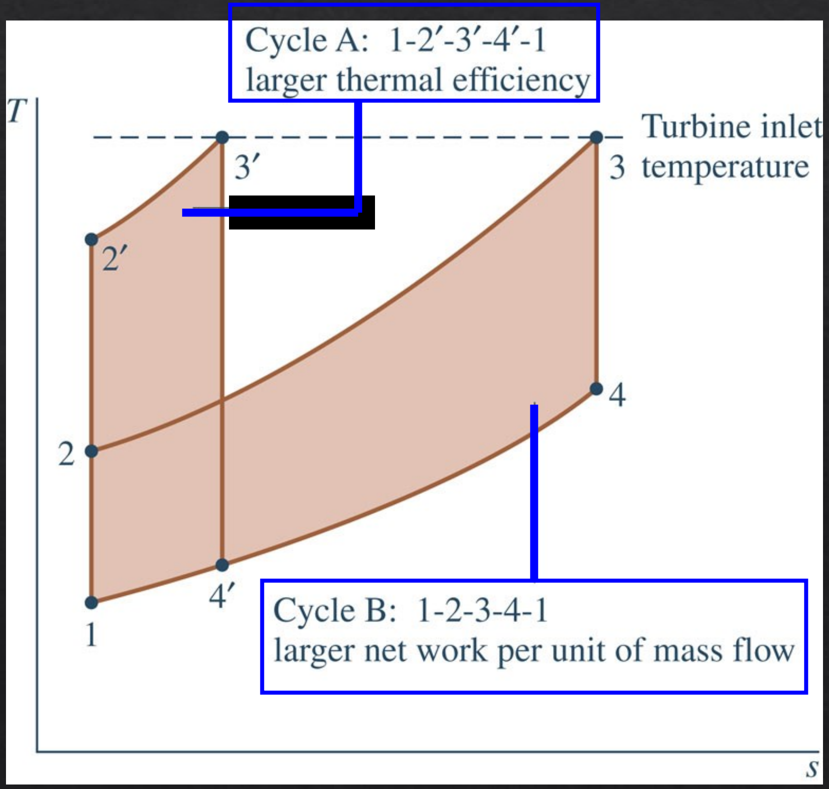
Effects of Compressor Pressure Ratio on Brayton Cycle Performance (5 of 7)

- ►However, there is a ***limit*** on the maximum temperature at the turbine inlet imposed by metallurgical considerations of the turbine blades.
- ►Let's consider the effect of increasing compressor pressure ratio on Brayton cycle performance when the turbine inlet temperature is held constant.
- ►This is investigated using the T-s diagram as presented next.

Effects of Compressor Pressure Ratio on Brayton Cycle Performance (6 of 7)

► The figure shows the ***T-s*** diagrams of two ideal Brayton cycles having the same turbine inlet temperature but different compressor pressure ratios.

- ► **Cycle A** has the greater compressor pressure ratio and thus the greater thermal efficiency.
- ► **Cycle B** has the larger enclosed area and thus the greater net work developed per unit of mass flow.
- ► For **Cycle A** to develop the same net power as **Cycle B**, a larger mass flow rate would be required, and this might dictate a larger system.

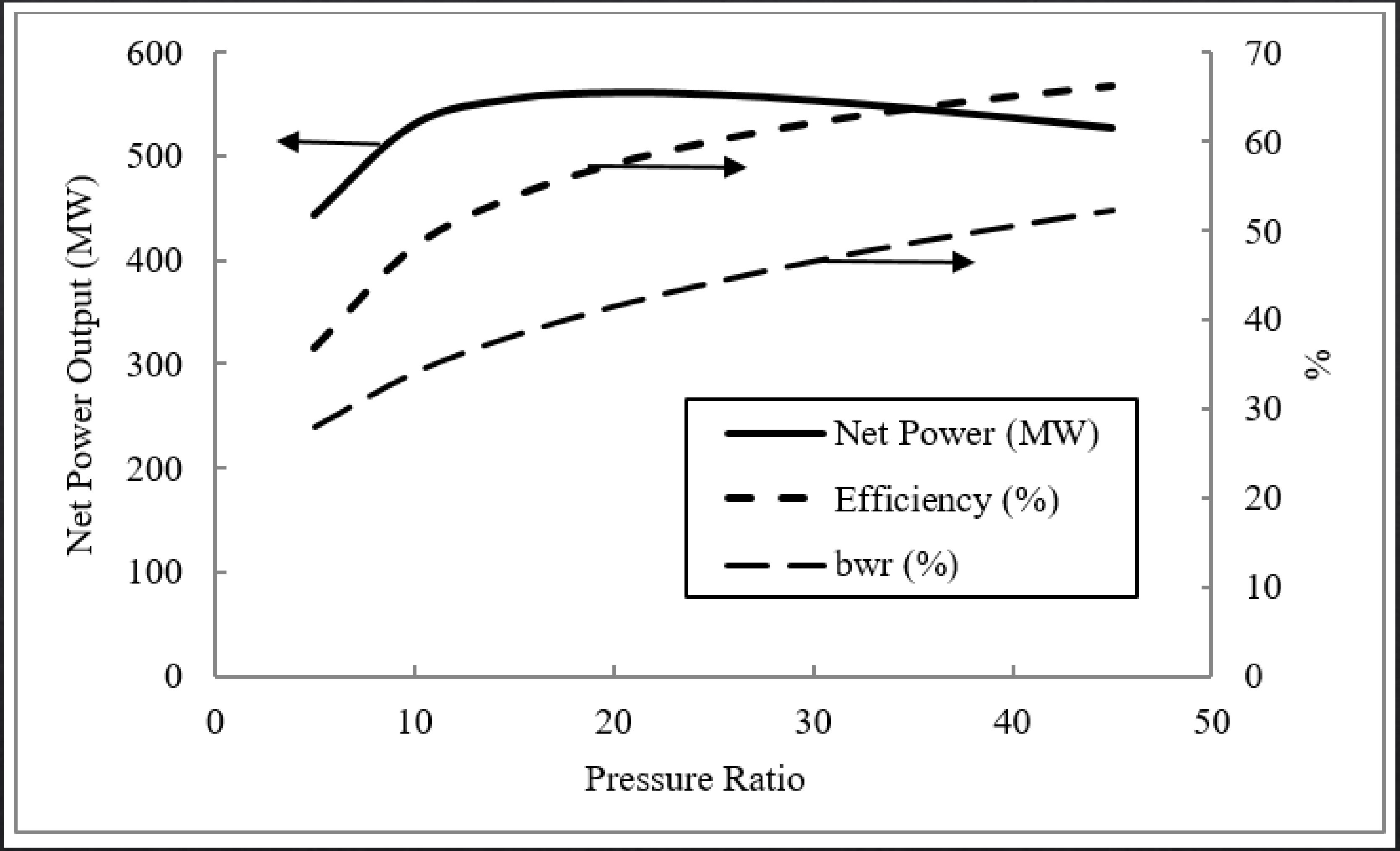


Effects of Compressor Pressure Ratio on Brayton Cycle Performance (7 of 7)

- Accordingly, for turbine-powered vehicles, where size and weight are constrained, it may be desirable to operate near the compressor pressure ratio for greater net work per unit of mass flow and not the pressure ratio for greater thermal efficiency.

Efficiency, net power output, and back work ratio of the Brayton cycle as a function of the compressor pressure ratio in Example 9.1

Pressure Ratio	T_2 (K)	T_4 (K)	Q_L (MW)	Q_H (MW)	W_{Comp} (MW)	W_{Turb} (MW)	W_{Net} (MW)	BWR (%)	η_{Th} (%)
5	475	1073	759	1203	172	615	443	27.9	36.9
10	579	881	570	1101	274	805	530	34.1	48.2
15	650	784	475	1031	344	899	555	38.3	53.9
20	706	722	415	976	399	960	561	41.5	57.5
25	753	678	371	930	444	1004	559	44.3	60.1
30	793	643	337	891	484	1038	554	46.6	62.2
35	828	616	310	856	519	1065	546	48.7	63.8
40	861	593	287	824	551	1087	537	50.6	65.1



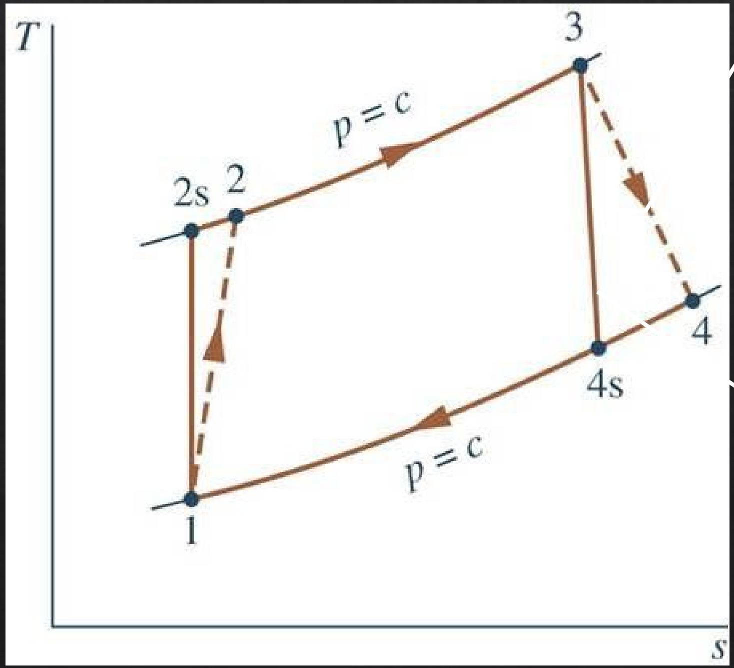
Variation of efficiency, net power output, and back work ratio as a function of compressor pressure ratio in Example 9.1

Gas Turbine Power Plant Irreversibility (1 of 3)

- The most significant irreversibility by far is the ***irreversibility of combustion***.
- ***Irreversibilities*** related to ***flow through the turbine and compressor*** also significantly impact gas turbine performance.
- They act to
  - ***decrease*** the work developed by the turbine and
  - ***increase*** the work required by the compressor,
  - thereby ***decreasing*** the net work of the power plant.

Gas Turbine Power Plant Irreversibility (2 of 3)

►**Isentropic turbine efficiency**, accounts for the effects of irreversibilities within the turbine in terms of actual and isentropic turbine work, each per unit of mass flowing through the turbine.



work developed in the actual expansion from turbine inlet state to the turbine exit pressure

$$\eta_t = \frac{(\dot{W}_t / \dot{m})}{(\dot{W}_t / \dot{m})_s} = \frac{(h_3 - h_4)}{(h_3 - h_{4s})}$$

work developed in an isentropic expansion from turbine inlet state to exit pressure

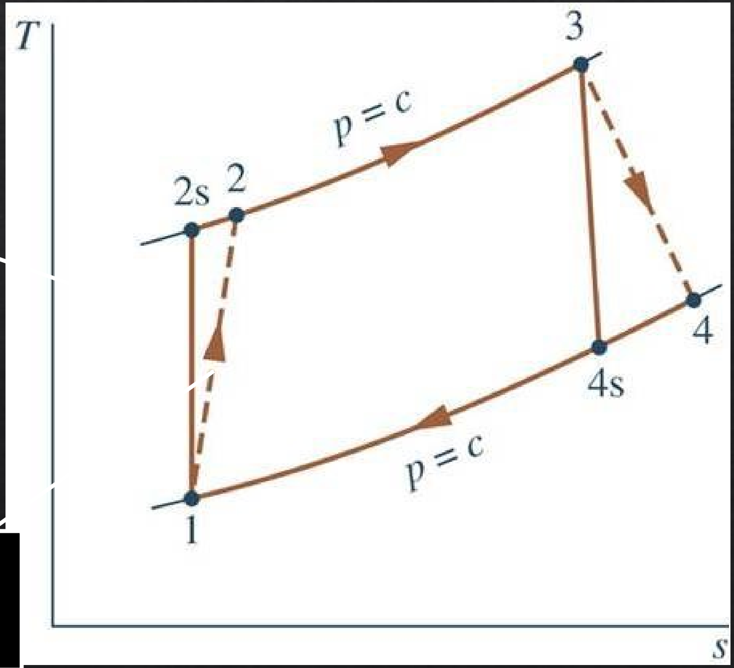
Gas Turbine Power Plant Irreversibility (3 of 3)

►**Isentropic compressor efficiency**, accounts for the effects of irreversibilities within the compressor in terms of actual and isentropic compressor work input, each per unit of mass flowing through the compressor.

work input for an isentropic process from compressor inlet state to exit pressure

$$\eta_c = \frac{(\dot{W}_c / \dot{m})_s}{(\dot{W}_c / \dot{m})} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)}$$

work input for the actual process from compressor inlet state to the compressor exit pressure



Example 9.5

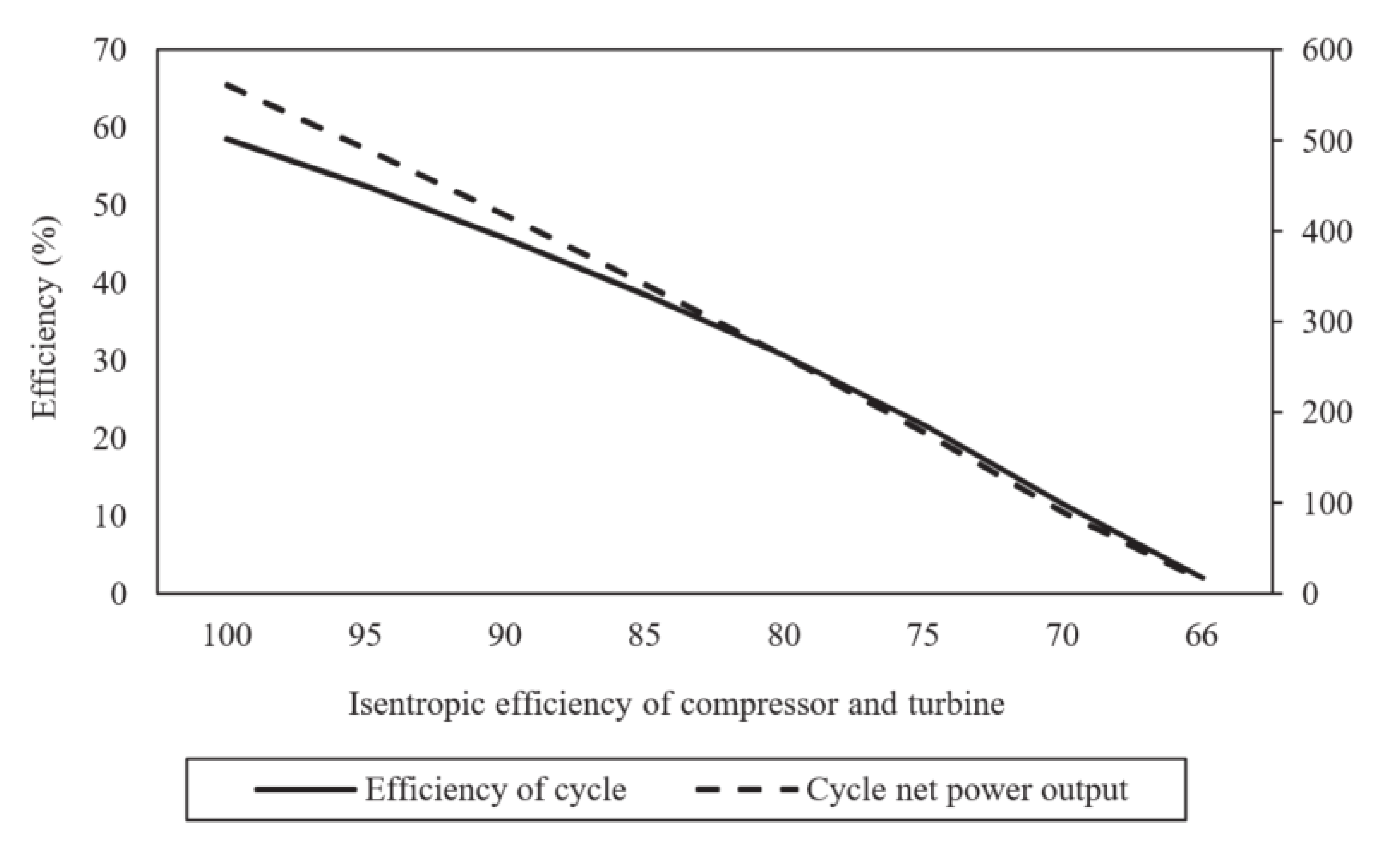
Reconsider Example 9.1, but this time instead of the isentropic compression and expansion processes, the compressor and turbine have the isentropic efficiency of 90%. Compare the results with the results from Example 9.1.

Comparison of the results for the air-standard and cold air-standard analysis with the compressor and turbine efficiency of 90% and the air-standard analysis with the isentropic compression and expansion

Parameter	Air-Standard Brayton Cycle with Isentropic Compression and Expansion Processes	Air-Standard Brayton Cycle with $\eta_{Comp} = \eta_{Turb} = 90\%$	Cold Air-Standard Brayton Cycle with $\eta_{Comp} = \eta_{Turb} = 90\%$
T_2 (K)	709	752	771
T_4 (K)	797	892	804
$\dot{Q}_L$ (MW)	507	611	495
$\dot{Q}_H$ (MW)	1132	1086	912
$\dot{W}_{Comp}$ (MW)	413	459	462
$\dot{W}_{Turb}$ (MW)	1038	935	879
$\dot{W}_{Net}$ (MW)	625	475	417
BWR (%)	39.8	49.2	52.6
η_{Th} (%)	55.2	43.8	45.7

Comparison of results of selected parameters for the actual gas turbine, non-ideal Brayton cycle, and ideal Brayton cycle

Parameter	Net Power Output (MW)	Efficiency (%)	Exhaust Temperature (K)	Exhaust Energy (MW)
Actual gas turbine	470	41.5	892	712
Air-standard Brayton cycle – isentropic processes	625 (33%)	55.2 (33%)	797 (−11%)	507 (−9%)
Air-standard Brayton cycle with $\eta_{Comp} = \eta_{Turb} = 90\%$	475 (1%)	43.8 (6%)	892 (0%)	611 (−14%)



Efficiency and net power output of nonideal Brayton cycle as a function of the isentropic efficiencies of the compressor and turbine (cold air standard analyses)

Gas Turbine Power Plant Loss

- The exhaust gas temperature of a ***simple*** gas turbine is typically well above the ambient temperature. Thus, the exhaust gas has considerable thermodynamic utility (exergy) that would be irrevocably *lost* were the gas discharged directly to the ambient.
- ***Regenerative*** gas turbines and gas turbine-based ***combined cycles*** aim to avoid such a significant loss by using the hot exhaust gas cost-effectively.

End of Lecture!