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Power Cycles

by

Engineering Software

Power Cycles

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Carnot Cycle

This section provides a Carnot Cycle analysis when the working fluid is air.

Analysis

In the presented Carnot Cycle analysis, only air is considered as the working fluid behaving as a perfect gas -- specific heat has a constant value. Ideal gas state equation is valid -- pv = RT.

Air enters a compressor at point 1 and it exits the compressor at point 2. Isentropic compression is considered with no entropy change. Air enters a heat exchanger -- heat addition -- at point 2 and it exits the heat exchanger at point 3. At a constant temperature, heat addition takes place. Air enters a turbine at point 3 and it exits the turbine at point 4. Isentropic expansion is considered with no entropy change. Air enters a heat exchanger -- heat rejection -- at point 4 and it exits the heat exchanger at point 1. At a constant temperature, heat rejection takes place. It should be mentioned that air at point 1 enters the compressor and the cycle is repeated.

Figure 1 presents a Carnot Cycle schematic layout.

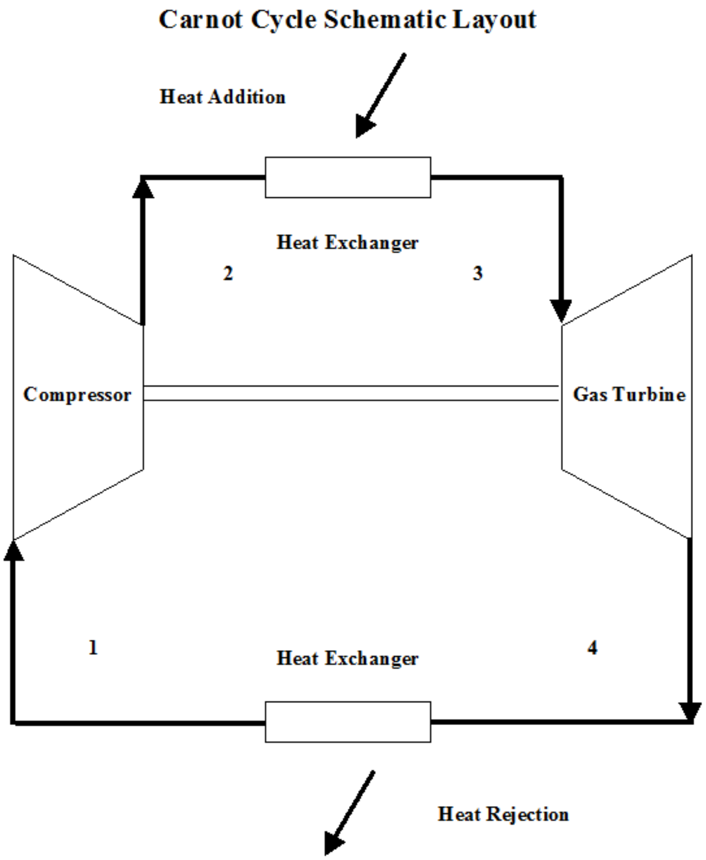

Figure 1 - Carnot Cycle Schematic Layout

Figure 2 presents a Carnot Cycle temperature vs entropy diagram.

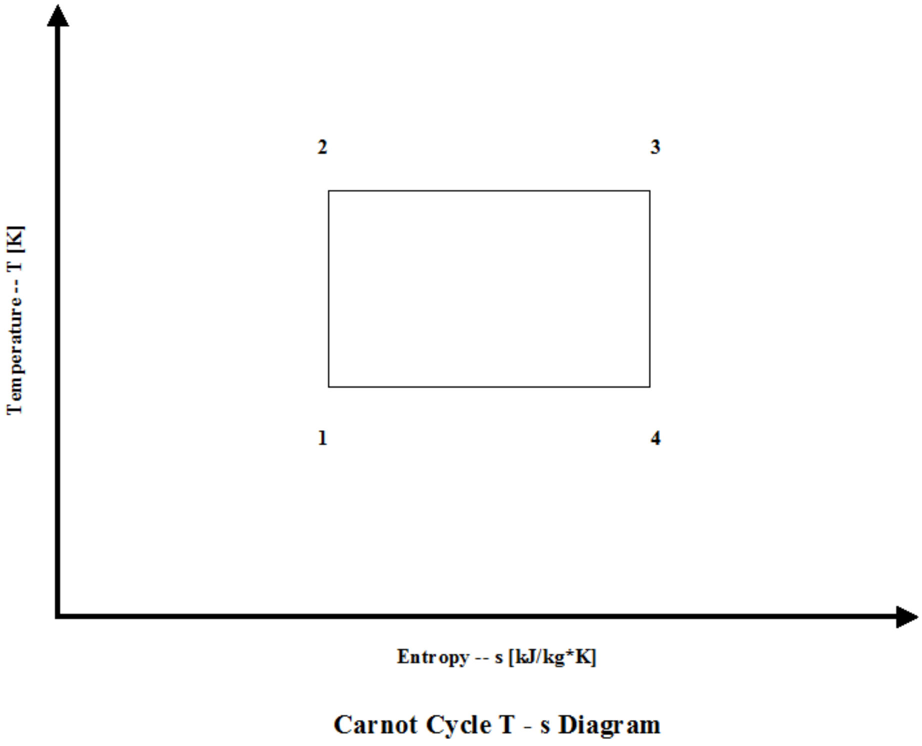

Figure 2 - Carnot Cycle Temperature vs Entropy Diagram

Figure 3 presents the Carnot Cycle efficiency as a function of the heat addition temperature. It should be noted that the inlet conditions are standard ambient conditions: temperature of 298 [K] and absolute pressure of 1 [atm].

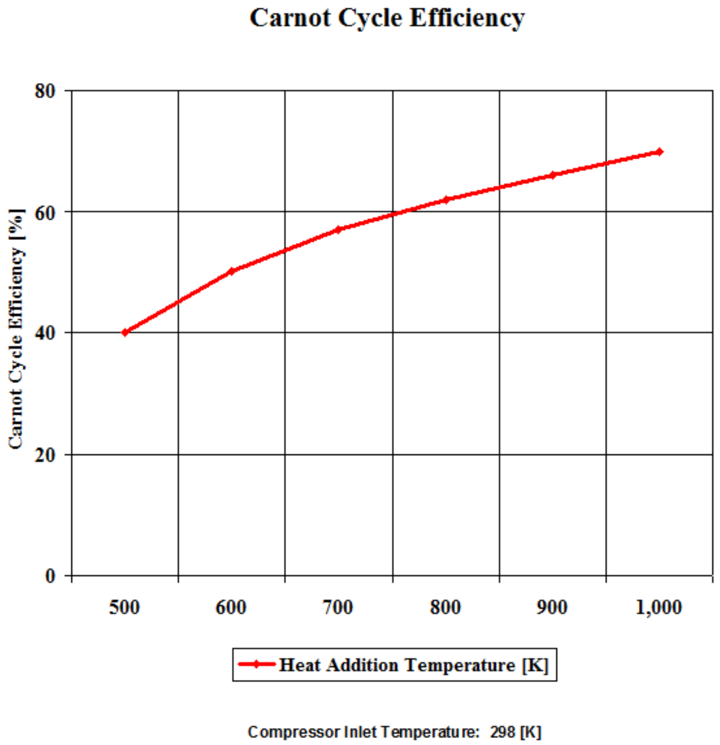

Figure 3 - Carnot Cycle Efficiency vs Heat Addition Temperature

Figure 4 presents the Carnot Cycle efficiency as a function of the heat rejection temperature. It should be noted that the turbine inlet temperature is at 800 [K].

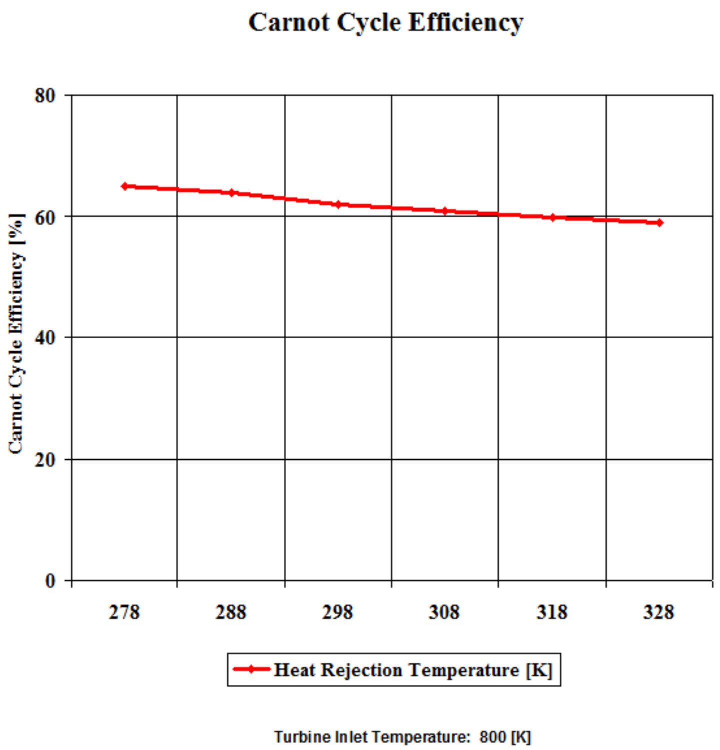

Figure 4 - Carnot Cycle Efficiency vs Heat Rejection Temperature

One can notice that the Carnot Cycle efficiency increases with an increase in the heat addition temperature when the heat rejection temperature does not change at all. One can notice that the Carnot Cycle efficiency decreases with an increase in the heat rejection temperature when the heat addition temperature does not change at all.

Assumptions

Working fluid is air. There is no friction. Compression and expansion are isentropic -- there is no entropy change. During heat addition and heat rejection, the air temperature does not change. Ideal gas state equation is valid -- pv = RT. Air behaves as a perfect gas -- specific heat has a constant value.

Governing Equations

$$T_2/T_1 = (p_2/p_1)^{(\varkappa-1)/\varkappa}$$

$$T_3/T_4 = (p_3/p_4)^{(\varkappa-1)/\varkappa}$$

$$\varkappa = c_p/c_v$$

$$c_p - c_v = R$$

$$pv = RT$$

$$q_h = T_2\Delta s = T_A\Delta s$$

$$q_l = T_1\Delta s = T_R\Delta s$$

$$\eta = 1 - T_1/T_2 = 1 - T_R/T_A$$

Input Data

T_1 = 298 [K]

p_1 = 1 [atm]

R = 0.2867 [kJ/kg*K]

c_p = 1.004 [kJ/kg*K]

$\varkappa$ = 1.4 [/]

Results

Carnot Cycle Efficiency vs Heat Addition Temperature
Heat Rejection Temperature = 298 [K]

Heat Addition Temperature [K]	Carnot Cycle Efficiency [%]
500	40.40
600	50.33
700	57.43
800	62.75
900	66.89
1,000	70.22

Carnot Cycle Efficiency vs Heat Rejection Temperature
Heat Addition Temperature = 800 [K]

Heat Rejection Temperature [K]	Carnot Cycle Efficiency [%]
278	65.25
288	64.00
298	62.75
308	61.50
318	60.25
328	59.00

Conclusions

The Carnot Cycle efficiency increases with an increase in the heat addition temperature when the heat rejection temperature does not change at all. Furthermore, the Carnot Cycle efficiency decreases with an increase in the heat rejection temperature when the heat addition temperature does not change at all. The Carnot Cycle efficiency is not dependent on the working fluid properties.

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Brayton Cycle for Power Application

This section provides a Brayton Cycle analysis when the working fluid is air.

Analysis

In the presented Brayton Cycle analysis, only air is considered as the working fluid behaving as a perfect gas -- specific heat has a constant value. Ideal gas state equation is valid -- pv = RT.

A gas turbine is a heat engine that uses a high temperature, high pressure gas as the working fluid. Combustion of a fuel in air is usually used to produce the needed temperatures and pressures in the gas turbine, which is why gas turbines are often referred to as combustion turbines. Expansion of the high temperature, high pressure working fluid takes place in the gas turbine. The gas turbine shaft rotation drives an electric generator and a compressor for the working fluid, air, used in the gas turbine combustor. Many gas turbines also use a heat exchanger called a recouperator to impart turbine exhaust heat into the combustor's air/fuel mixture. Gas turbines produce high quality heat that can be used to generate steam for combined heat and power and combined-cycle applications, significantly enhancing efficiency.

Air is compressed, isentropically, along line 1-2 by a compressor and it enters a combustor. At a constant pressure, combustion takes place -- fuel is added to the combustor and the air temperature raises. High temperature air exits the combustor at point 3. Then air enters a gas turbine where an isentropic expansion occurs, producing power. Air exits the gas turbine at point 4. It should be mentioned that air at point 1 enters the compressor and the cycle is repeated.

Figure 1 presents a Brayton Cycle schematic layout.

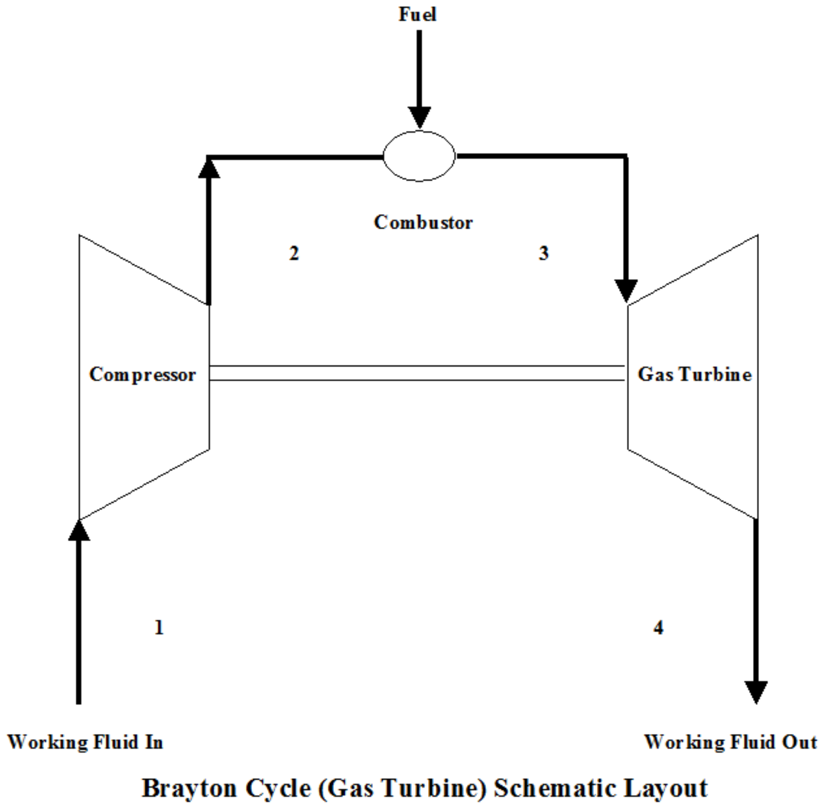

Figure 1 - Brayton Cycle Schematic Layout

Figure 2 presents a Brayton Cycle temperature vs entropy digram.

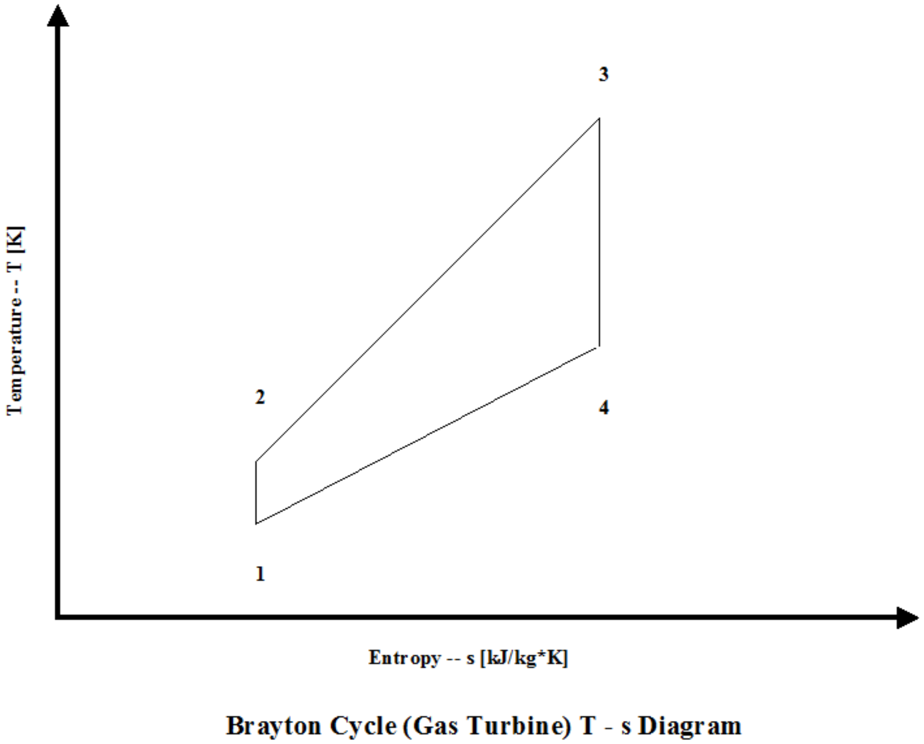

Figure 2 - Brayton Cycle Temperature vs Entropy Diagram

It should be pointed out that this material deals with the open Brayton Cycle. The way how the T - s diagram is presented, it describes a closed Brayton Cycle -- this would require a heat exchanger after point 4 where the working fluid would be cooled down to point 1 and the cycle repeats. Therefore, the T - s diagram is presented as a closed Brayton Cycle to allow easier understanding and derivation of the Brayton Cycle thermal efficiency -- heat addition and heat rejection.

The gas turbine and compressor are connected by shaft so the considerable amount of work done on the gas turbine is used to power the compressor.

It can be noticed from the T - s diagram that the work done on the gas turbine is greater than the work necessary to power the compressor -- constant pressure lines in the T - s diagram diverge by going to the right side (entropy wise).

Figure 3 presents the Brayton Cycle efficiency as a function of the compression ratio. It should be noted that the inlet conditions are standard ambient conditions: temperature of 298 [K] and absolute pressure of 1 [atm].

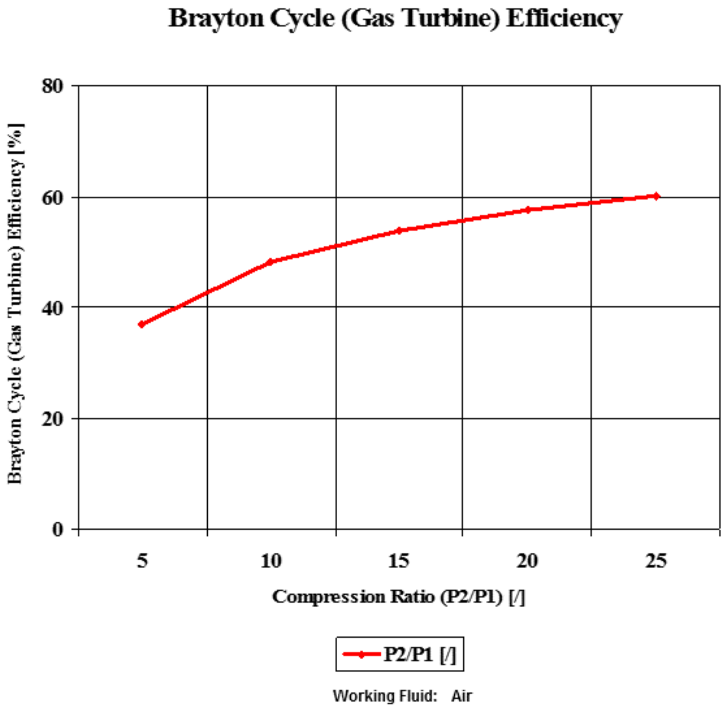

Figure 3 - Brayton Cycle Efficiency

Here, two general performance trends are considered. First, impact of the gas turbine inlet temperature and compression ratio on the Brayton Cycle specific power output and second, impact of the working fluid mass flow rate on the Brayton Cycle power output for a fixed gas turbine inlet temperature.

Figure 4 presents the results of the first performance trend, while Figure 5 presents the results of the second trend.

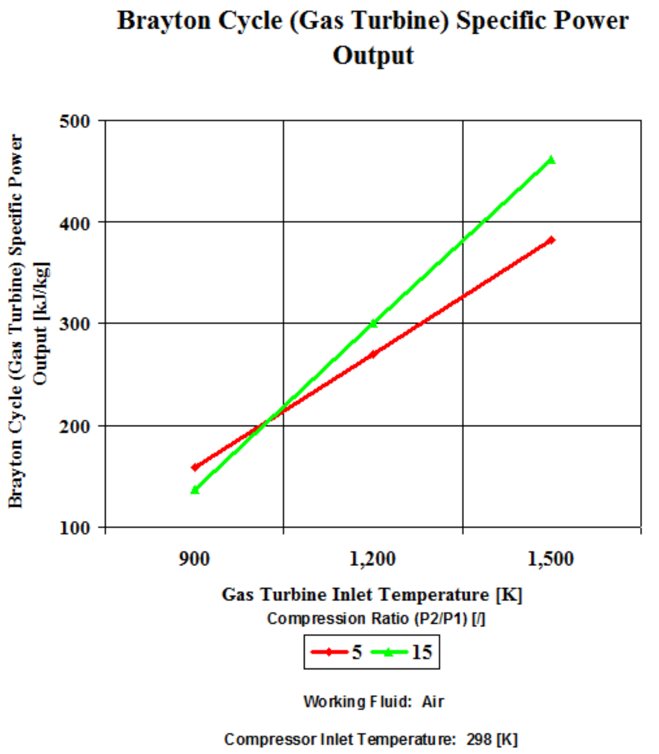

Figure 4 - Brayton Cycle Specific Power Output

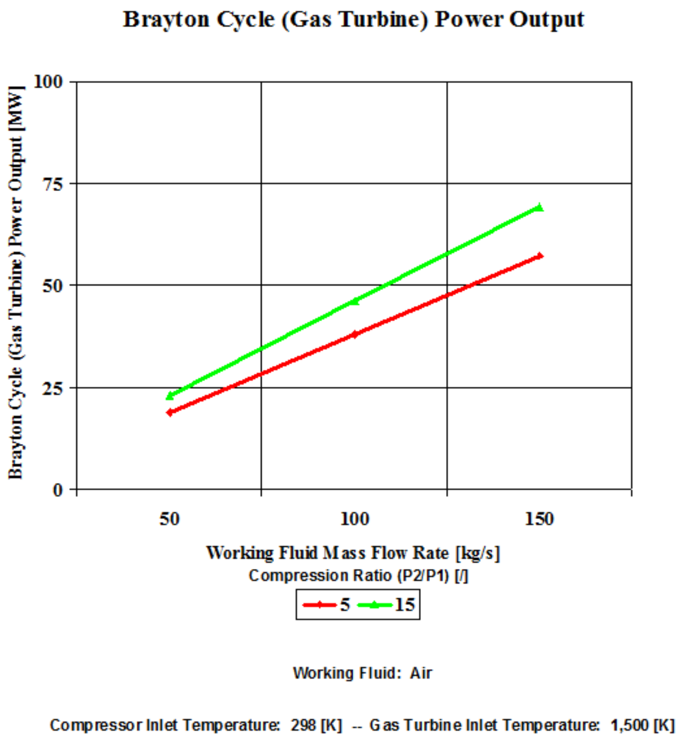

Figure 5 - Brayton Cycle Power Output

One can notice that the Brayton Cycle efficiency increases with an increase in the compression ratio values. One can notice that the Brayton Cycle specific power output increases with an increase in the gas turbine inlet temperature. Furthermore, the increase is greater for the higher compression ratio values.

One can notice that the Brayton Cycle power output increases with an increase in the working fluid mass flow rate for a fixed gas turbine inlet temperature value. The increase is greater for the higher compression ratio values.

Assumptions

Working fluid is air. There is no friction. Compression and expansion processes are reversible and adiabatic – isentropic. Ideal gas state equation is valid -- pv = RT. Air behaves as a perfect gas -- specific heat has a constant value.

Governing Equations

$$T_2/T_1 = (p_2/p_1)^{(\varkappa-1/)\varkappa}$$

$$p_2/p_1 = (T_2/T_1)^{\varkappa/(\varkappa-1)}$$

$$T_3/T_4 = (p_3/p_4)^{(\varkappa-1/)\varkappa}$$

$$p_3/p_4 = (T_3/T_4)^{\varkappa/(\varkappa-1)}$$

$$\varkappa = c_p/c_v$$

$$c_p - c_v = R$$

$$pv = RT$$

$$w = q_h - q_l$$

$$q_h = c_p(T_3 - T_2)$$

$$q_l = c_p(T_4 - T_1)$$

$$w = c_p(T_3 - T_2) - c_p(T_4 - T_1)$$

$$W = (c_p(T_3 - T_2) - c_p(T_4 - T_1))m$$

$$\eta = 1 - 1/r_p^{(\varkappa-1)/\varkappa}$$

$$r_p = p_2/p_1$$

Power Cycles

Input Data

T_1 = 298 [K]

p_1 = 1 [atm]

T_3 = 900, 1,200 and 1,500 [K]

p_3 = 5, 10, 15, 20 and 25 [atm]

R = 0.2867 [kJ/kg*K]

c_p = 1.004 [kJ/kg*K]

$\varkappa$ = 1.4 [/]

m = 50, 100 and 150 [kg/s]

Results

Brayton Cycle Efficiency vs Compression Ratio

Compression Ratio [/]	Brayton Cycle Efficiency [%]
5	36.92
10	48.22
15	53.87
20	57.53
25	60.16

Specific Power Output vs Compression Ratio for a few Gas Turbine Inlet Temperature Values

Specific Power Output [kW/kg/s]	**Gas Turbine Inlet Temperature [K]**		
Compression Ratio [/]	900	1,200	1,500
5	159	270	381
15	137	300	462

Power Output vs Compression Ratio for a few Mass Flow Rates
Gas Turbine Inlet Temperature = 1,500 [/]

Power Output [MW]	**Mass Flow Rate [kg/s]**		
Compression Ratio [/]	50	100	150
5	19.1	38.1	57.2
15	23.1	46.2	69.3

Conclusions

The Brayton Cycle efficiency depends on the compression ratio and working fluid properties. The efficiency increases with an increase in the compression ratio values. Also, the efficiency increases with the higher value for $\varkappa$, which is a ratio of gas specific heat values (c_p/c_v).

The Brayton Cycle specific power output increases with an increase in the gas turbine inlet temperature. Furthermore, the increase is greater for the higher compression ratio values. The Brayton Cycle power output increases with an increase in the working fluid mass flow rate for a fixed gas turbine inlet temperature value. The increase is greater for the higher compression ratio values.

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Otto Cycle

This section provides an Otto Cycle analysis when the working fluid is air.

Analysis

In the presented Otto Cycle analysis, only air is considered as the working fluid behaving as a perfect gas -- specific heat has a constant value. Ideal gas state equation is valid -- pv = RT.

Air enters a cylinder at point 1 when compression starts and it ends at point 2. Isentropic compression is considered with no entropy change. Heat addition starts at point 2 and it ends at point 3. At a constant volume, combustion takes place (fuel is added to the cylinder and the air temperature raises) and/or heat gets added to air. Expansion starts at point 3 and it ends at point 4. Isentropic expansion is considered with no entropy change. Air heat rejection starts at point 4 and it ends at point 1. At a constant volume, air gets cooled and the working fluid temperature decreases. It should be mentioned that air at point 1 enters the compression process again and the cycle is repeated.

Figure 1 presents an Otto Cycle pressure vs volume diagram.

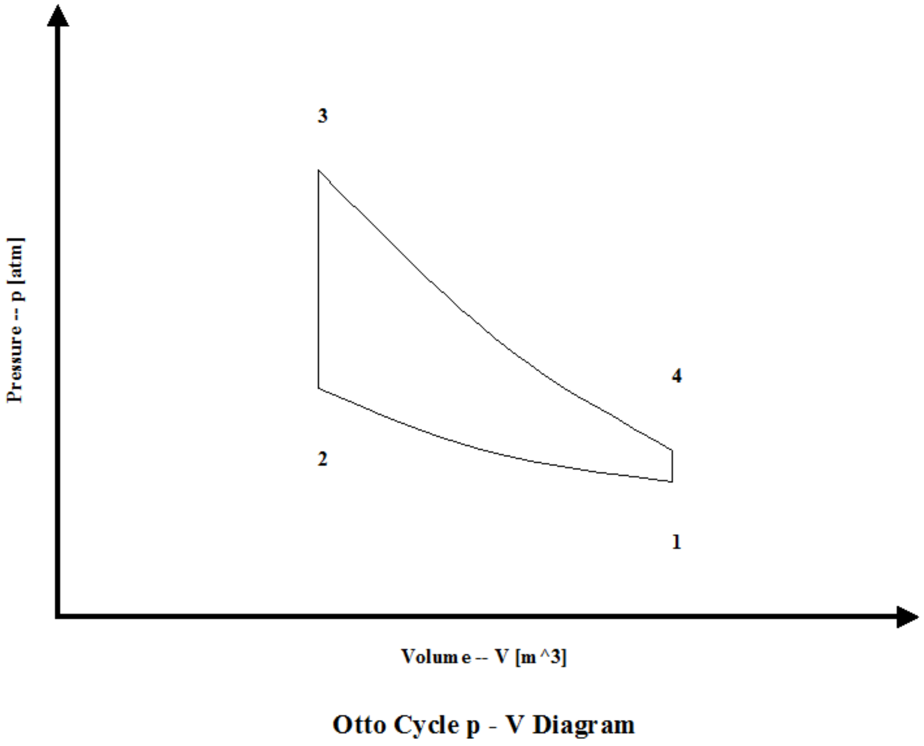

Figure 1 - Otto Cycle Pressure vs Volume Diagram

Figure 2 presents an Otto Cycle temperature vs entropy diagram.

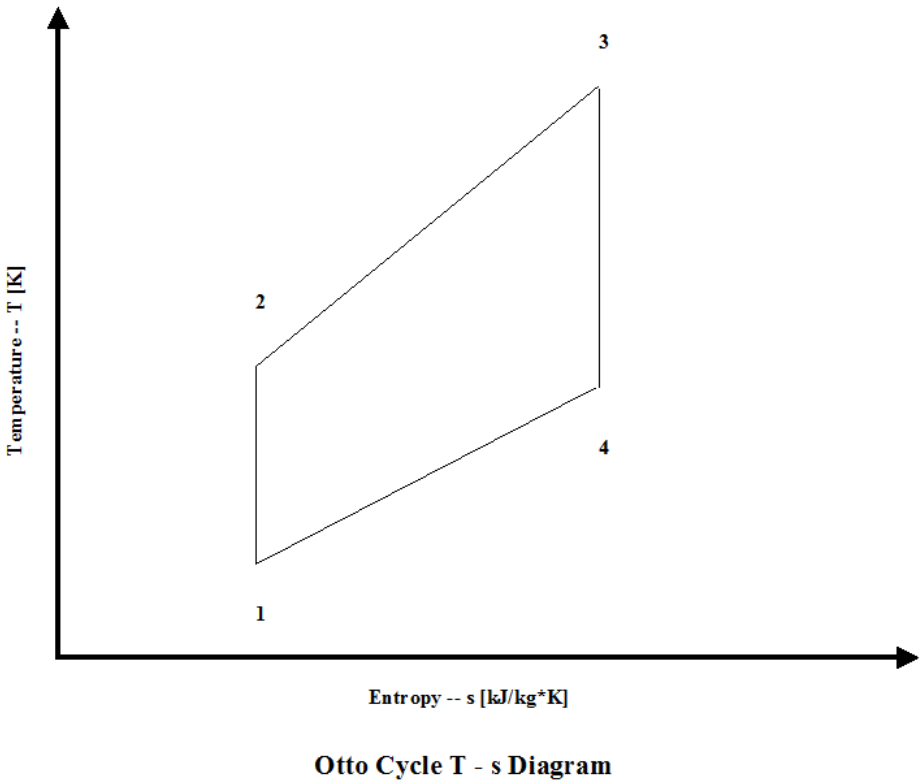

Figure 2 - Otto Cycle Temperature vs Entropy Diagram

Figure 3 presents the Otto Cycle efficiency as a function of the compression ratio. It should be noted that the inlet conditions are standard ambient conditions: temperature of 298 [K] and absolute pressure of 1 [atm].

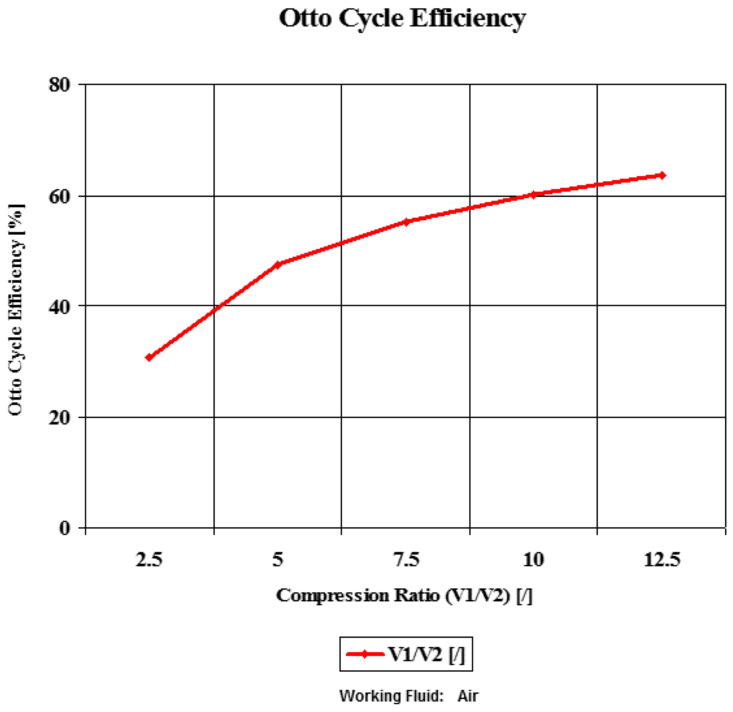

Figure 3 - Otto Cycle Efficiency

Figure 4 presents the Otto Cycle power output as a function of the combustion temperature and compression ratio. It should be noted that the number of revolutions is 60 [1/s] for given geometry of the four cylinder and four stroke Otto engine.

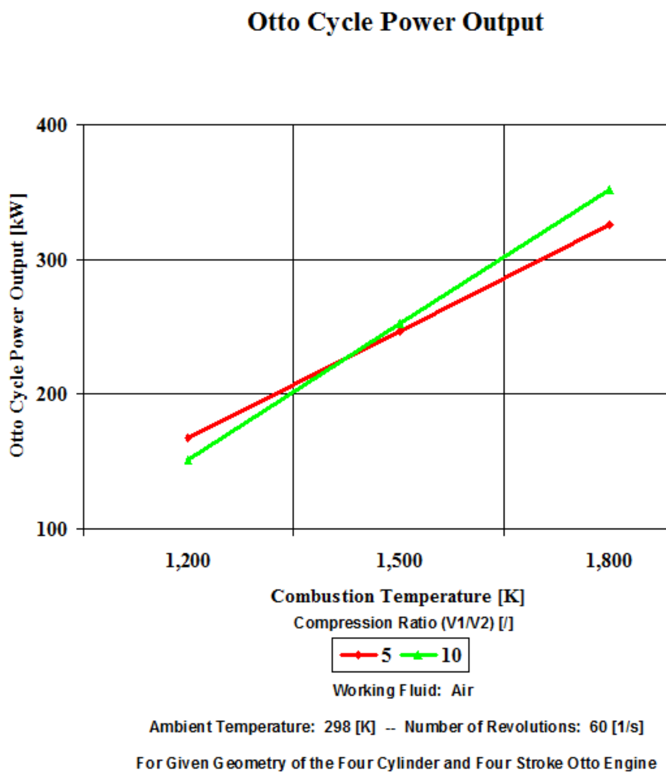

Figure 4 - Otto Cycle Power Output

One can notice that the Otto Cycle efficiency increases with an increase in the compression ratio values. One can notice that the Otto Cycle power output increases with an increase in the combustion temperature. The Otto Cycle power output is greater for the higher compression ratio values.

Assumptions

Working fluid is air. There is no friction. Compression and expansion are isentropic -- there is no entropy change. Ideal gas state equation is valid -- pv = RT. Air behaves as a perfect gas -- specific heat has a constant value.

Power Cycles

Governing Equations

$$T_2/T_1 = (V_1/V_2)^{(\varkappa-1)}$$

$$V_1/V_2 = (T_2/T_1)^{1/(\varkappa-1)}$$

$$T_3/T_4 = (V_4/V_3)^{(\varkappa-1)}$$

$$V_4/V_3 = (T_3/T_4)^{1/(\varkappa-1)}$$

$$\varkappa = c_p/c_v$$

$$c_p - c_v = R$$

$$pv = RT$$

$$w = q_h - q_l$$

$$q_h = c_v(T_3 - T_2)$$

$$q_l = c_v(T_4 - T_1)$$

$$w = c_v(T_3 - T_2) - c_v(T_4 - T_1)$$

$$W = (c_v(T_3 - T_2) - c_v(T_4 - T_1))m$$

$$\eta = 1 - 1/\varepsilon^{(\varkappa-1)}$$

$$\varepsilon = V_1/V_2$$

Power Cycles

Input Data

$T_1 = 298$ [K]

$p_1 = 1$ [atm]

$T_3 = 1{,}200, 1{,}500$ and $1{,}800$ [K]

$\varepsilon = 2.5, 5, 7.5, 10$ and 12.5 [/]

$R = 0.2867$ [kJ/kg*K]

$c_p = 1.004$ [kJ/kg*K]

$\varkappa = 1.4$ [/]

Results

Otto Cycle Efficiency vs Compression Ratio

Compression Ratio [/]	**Otto Cycle Efficiency [%]**
2.5	30.69
5	47.47
7.5	55.33
10	60.19
12.5	63.59

Otto Cycle Power Output

Power Output [kW]	**Combustion Temperature [K]**		
Compression Ratio [/]	1,200	1,500	1,800
5	167.1	246.4	325.6
10	151.2	251.7	352.2

Conclusions

The Otto Cycle efficiency increases with an increase in the compression ratio values. Also, the Otto Cycle power output increases with an increase in the combustion temperature. The Otto Cycle power output is greater for the higher compression ratio values.

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Diesel Cycle

This section provides a Diesel Cycle analysis when the working fluid is air.

Analysis

In the presented Diesel Cycle analysis, only air is considered as the working fluid behaving as a perfect gas -- specific heat has a constant value. Ideal gas state equation is valid -- pv = RT.

Air enters a cylinder at point 1 when compression starts and it ends at point 2. Isentropic compression is considered with no entropy change. Heat addition starts at point 2 and it ends at point 3. At a constant pressure, combustion takes place (fuel is added to the cylinder and the air temperature raises) and/or heat gets added to air. Expansion starts at point 3 and it ends at point 4. Isentropic expansion is considered with no entropy change. Air heat rejection starts at point 4 and it ends at point 1. At a constant volume, air gets cooled and the working fluid temperature decreases. It should be mentioned that air at point 1 enters the compression process again and the cycle is repeated.

Figure 1 presents a Diesel Cycle pressure vs volume diagram.

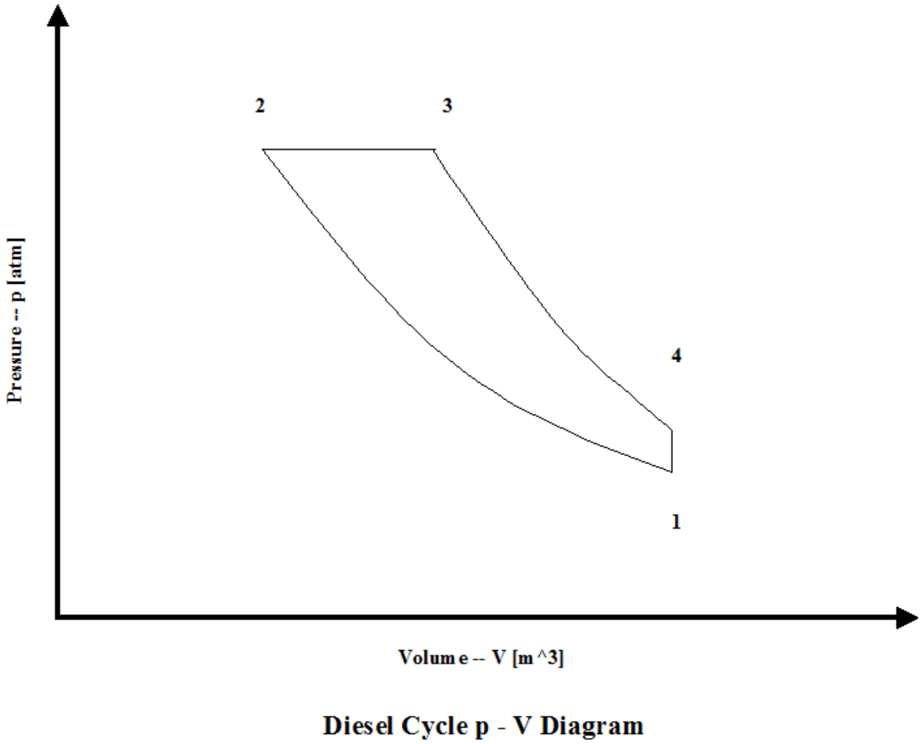

Figure 1 - Diesel Cycle Pressure vs Volume Diagram

Figure 2 presents a Diesel Cycle temperature vs entropy diagram.

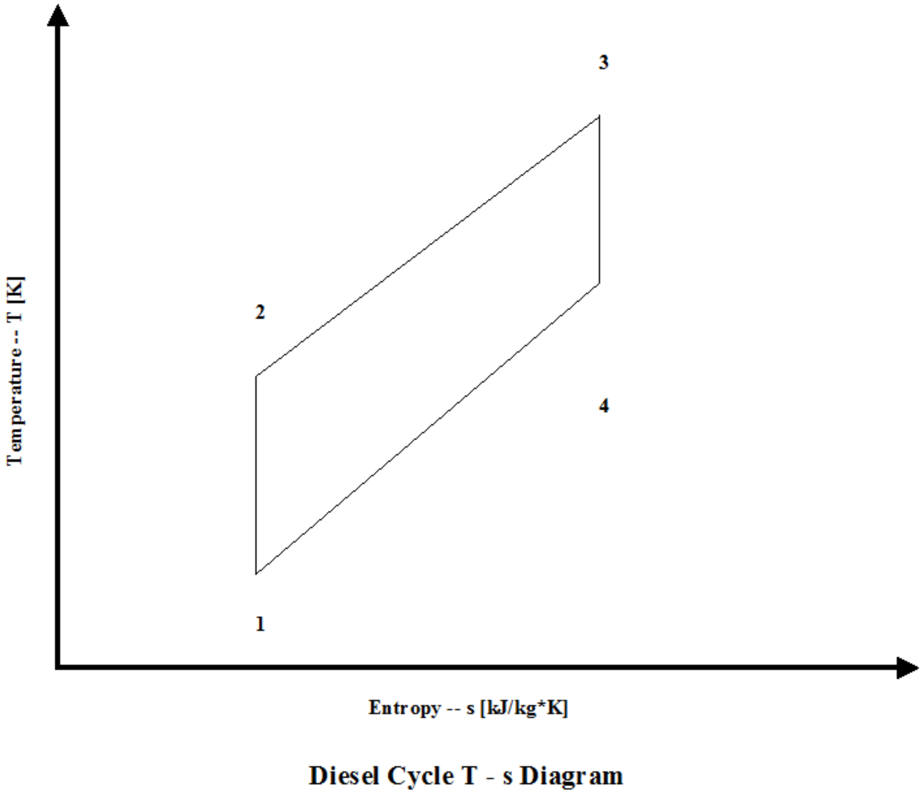

Figure 2 - Diesel Cycle Temperature vs Entropy Diagram

Figure 3 presents the Diesel Cycle efficiency as a function of the compression ratio and cut off ratio. It should be noted that the inlet conditions are standard ambient conditions: temperature of 298 [K] and absolute pressure of 1 [atm].

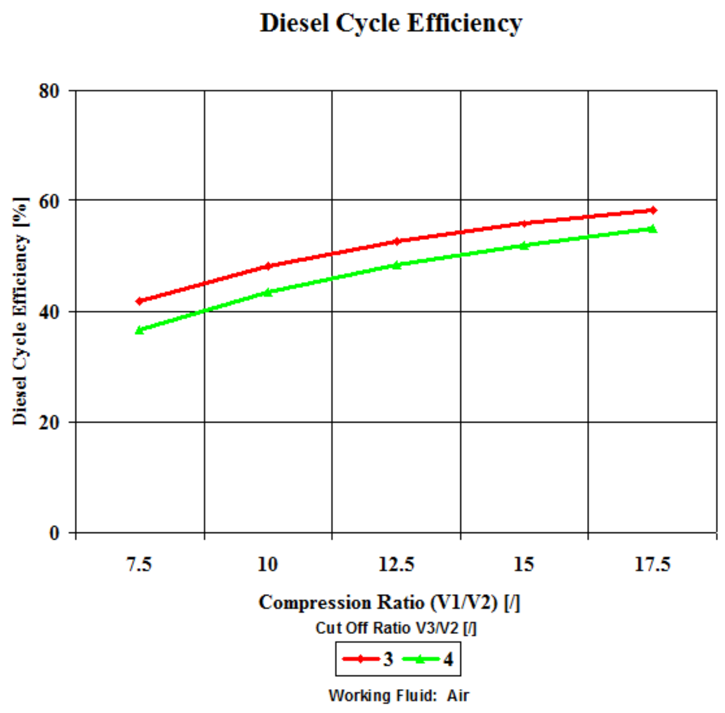

Figure 3 - Diesel Cycle Efficiency

Figure 4 presents the Diesel Cycle efficiency as a function of the compression ratio and combustion temperature.

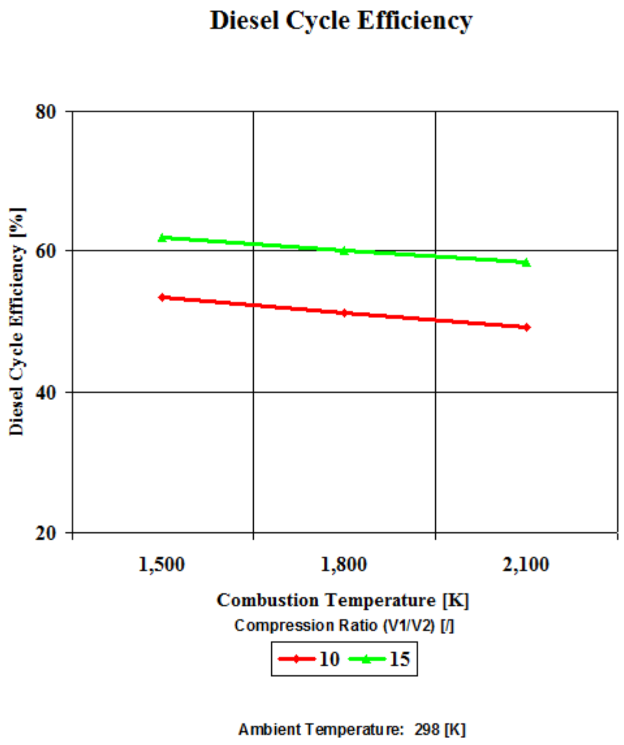

Figure 4 - Diesel Cycle Efficiency

Figure 5 presents the Diesel Cycle cut off ratio as a function of the combustion temperature and compression ratio.

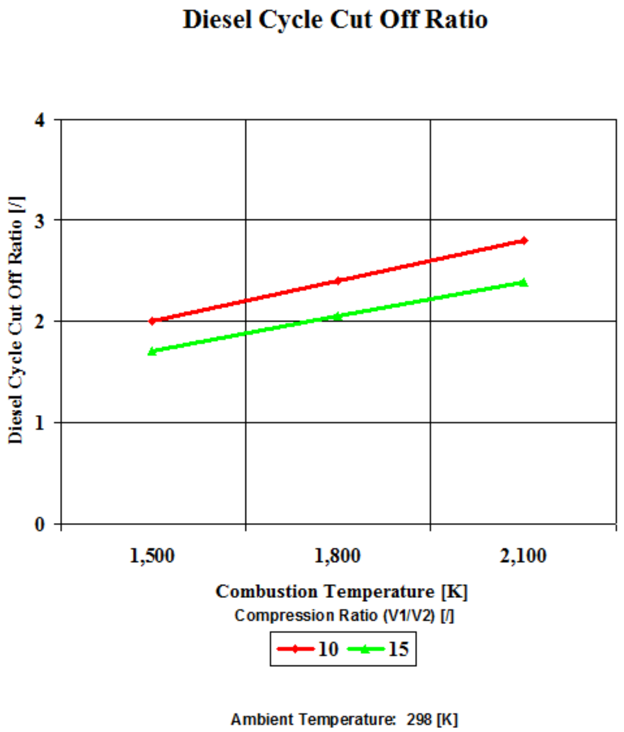

Figure 5 - Diesel Cycle Cut Off Ratio

Figure 6 presents the Diesel Cycle power output as a function the combustion temperature and compression ratio. It should be noted that the number of revolutions is 60 [1/s] for given geometry of the four cylinder and four stroke Diesel engine.

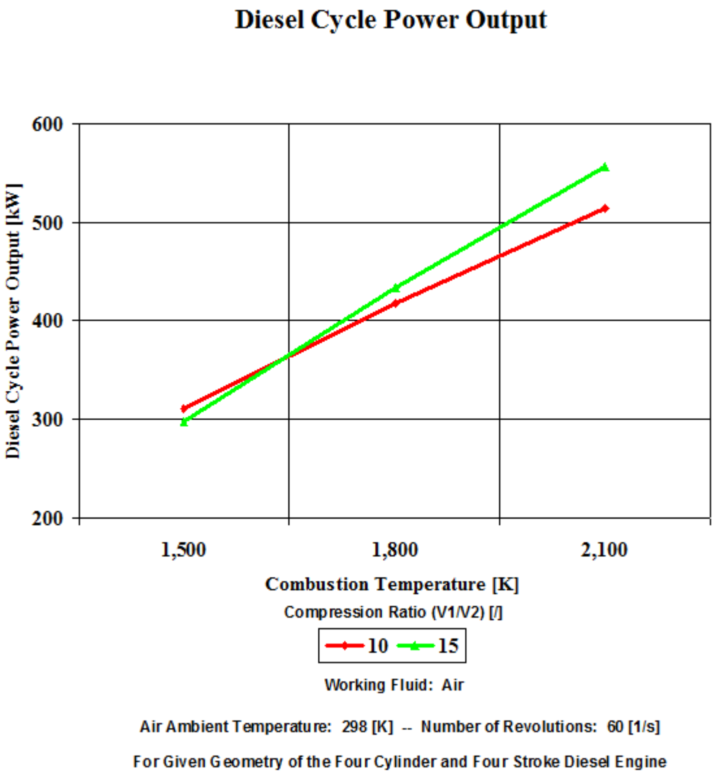

Figure 6 - Diesel Cycle Power Output

One can notice that the Diesel Cycle efficiency increases with an increase in the compression ratio and a decrease of the cut off ratio values. One can notice that the Diesel Cycle power output increases with an increase in the compression ratio and combustion temperature values.

Assumptions

Working fluid is air. There is no friction. Compression and expansion are isentropic -- there is no entropy change. Ideal gas state equation is valid -- pv = RT. Air behaves as a perfect gas -- specific heat has a constant value.

Governing Equations

$$T_2/T_1 = (V_1/V_2)^{(\varkappa-1)}$$

$$V_1/V_2 = (T_2/T_1)^{1/(\varkappa-1)}$$

$$T_3/T_4 = (V_4/V_3)^{(\varkappa-1)}$$

$$V_4/V_3 = (T_3/T_4)^{1/(\varkappa-1)}$$

$$\varkappa = c_p/c_v$$

$$c_p - c_v = R$$

$$pv = RT$$

$$w = q_h - q_l$$

$$q_{h\,=}\, c_p(T_3 - T_2)$$

$$q_{l\,=}\, c_v(T_4 - T_1)$$

$$w = c_p(T_3 - T_2) - c_v(T_4 - T_1)$$

$$W = (c_p(T_3 - T_2) - c_v(T_4 - T_1))m$$

$$\eta = 1 - (\varphi^{\varkappa} - 1)/(\varkappa\varepsilon^{(\varkappa-1)}(\varphi - 1))$$

$$\varepsilon = V_1/V_2$$

$$\varphi = V_3/V_2$$

Input Data

$T_1 = 298$ [K]

$p_1 = 1$ [atm]

$T_3 = 1{,}500$, 1,800 and 2,100 [K]

$\varepsilon = 7.5$, 10, 12.5, 15 and 17.5 [/]

$\varphi = 3$ and 4 [/]

$R = 0.2867$ [kJ/kg*K]

$c_p = 1.004$ [kJ/kg*K]

$\varkappa = 1.4$ [/]

Results

Diesel Cycle Efficiency

Diesel Cycle Efficiency [%]	**Compression Ratio [/]**				
Cut Off Ratio [/]	7.5	10	12.5	15	17.5
3	41.69	48.03	52.46	55.81	58.45
4	36.57	43.46	48.29	51.93	54.80

Diesel Cycle Efficiency

Diesel Cycle Efficiency [%]	**Combustion Temperature [K]**		
Compression Ratio [/]	1,500	1,800	2,100
10	53.39	51.12	49.20
15	61.94	60.12	58.49

Diesel Cycle Cut Off Ratio

Cut Off Ratio [/]	**Combustion Temperature [K]**		
Compression Ratio [/]	1,500	1,800	2,100
10	2.00	2.40	2.80
15	1.70	2.05	2.39

Diesel Cycle Power Output

Power Output [kW]	**Combustion Temperature [K]**		
Compression Ratio [/]	1,500	1,800	2,100
10	311	417	514
15	297	433	557

Conclusions

The Diesel Cycle efficiency increases with an increase in the compression ratio and a decrease in the cut off ratio values. Also, the Diesel Cycle power output increases with an increase in the compression ratio and combustion temperature values.

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