Heat &Mass Transfer

AE-202

Energy and Exergy Analysis of Gas Turbine Power Plant and Method of improving its Efficiency

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We would also like to take this opportunity to thank our family for constantly believing in us and providing the morale to never give up. We would like to thank our parents, whose love and guidance are with us in whatever we pursue. They are the ultimate role models.

DECLARATION

This is to certify that Aaryan Saraswat (2k21/AE/02), Dev Peshawari (2k21/AE/18) batch 2K21/AE hereby declare that the project work entitle "Energy and Exergy Analysis of Gas Turbine Power Plant and Method of Improving its Efficiency" submitted to the Delhi Technological University, is a record of an original work done by us under the guidance of Professor R. S. Mishra, Department of Mechanical Engineering, Delhi Technological University.

The results embodied in this project have not been submitted to any other University or Institute. This is our original work and we have not copied from any other student's work or from any other sources except where due reference or acknowledgment is made explicitly.

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Chapter 1

1.1 Motivation

Our main motivation is to learn and understand the practical implementation of thermodynamic concepts that play an integral role in almost all industries.

Contemplating various types of Gas Turbine Power Plants and its application as Jet Engine Gas Turbine, Industrial Gas Turbine for Power Generation and Mechanical Drive, Microturbines, and many more.

1.2 Introduction

A gas turbine, also called a combustion turbine, is a type of continuous and internal combustion engine. The main elements common to all gas turbine engines are:

- an upstream rotating gas compressor
- a combustor
- a downstream turbine on the same shaft as the compressor.

The basic operation of the gas turbine is a Joule/Brayton cycle with air as the working fluid: atmospheric air flows through the compressor that brings it to higher pressure, energy is then added by spraying fuel into the air and igniting it so that the combustion generates a high-temperature flow; this high-temperature pressurized gas enters a turbine, producing a shaft work output in the process, used to drive the compressor. The Gas Turbine power plants are being used for producing electricity, operating airplanes and for various industrial applications such as refineries and petrochemical plants.

1.3 Advantages and Disadvantages

Advantages of gas turbine power plant:

- It is simple to design & construct compared to a Steam Power plant, since no boilers are required.
- Much smaller compared to the Steam Power plant of the same capacity.
- Lower operating costs.
- They are small in size, weigh less and have low initial cost per unit output.

- They are easy to install within short periods.
- They are quick-starting and smooth running.
- They offer flexibility by supplying electricity for power generation as well as by supplying compressed air for process needs.
- They are capable of using a range of liquid and gaseous fuels including synthetic fuels.
- They are subjected (put) to fewer environmental restrictions than other prime movers.
- Water consumption is less compared to steam power plants.

Disadvantages of gas turbine power plant:

- Problem of starting. Compressor needs to be operating thus an external source of power is necessary.
- Net output is low since greater power is used to drive the compressor.
- Temperature of combustion chamber is too high thus resulting in a lower life
- An electric motor or an I.C. an engine is necessary for starting the plant. The starting motor must bring the compressor well towards the operating speed. So, starting is not simple as in the case of other power plants.
- Gas turbine plants have less vibrations when compared with reciprocating engines of the same speed. However the high frequency noise from the compressor is objectionable.
- High temperatures impose severe restriction on the servicing conditions of the plant.
- Overall efficiency is low since two-thirds of the total power output is used for driving the compressor.

- The blades of the turbine require special cooling methods due to the severity of operating temperatures and pressures. In practice, the temperatures at the entry of the turbine are as high as 1100° C - 1260°C. Hence they should be made of special metals and alloys.
- They are incompatible with solid fuels.

1.4 Applications

- By far the most important application is in Aviation, where they provide the motive power for jet propulsion.
- In the field of Electric Power Generation, gas turbines must compete with steam

  Turbines in large central power stations and with diesel engines in smaller plants.
- In the Automobile Industry, it is possible to add heat recovery to the exhaust of turbine-compressor sets, which in turn produces high pressure steam for introduction into a condensing steam and these types of turbines are used in areas without significant quantities of cooling water

Chapter 2

2.1 Literature Review

M. M. Rahman, et. al. (July, 2011) [1]

Presented the parametric study of thermodynamic performance on gas turbine power plants. The variation of operating conditions (compression ratio, turbine inlet and exhaust temperature, air to fuel ratio, isentropic compressor and turbine efficiency, and ambient temperature) on the

performance of gas turbines (thermal efficiency, compressor work, power, specific fuel consumption, heat rate) were investigated.

Hiroshi Taniguchi et. al. (July 2000) [2]

Developed a thermodynamic process of power generation for a gas turbine, a steam turbine, and a compressor, supported by adiabatic expansion or compression processes. The possible inlet temperature of gas turbines has been increased to 1500 degrees C, and the inlet temperature of steam turbines also has been increased to 650 degrees C.

Upathumchard Ularee et. al.(2014) [3]

Studied options for waste heat recovery in a 700 MW gas-turbine combined cycle power plant in Thailand. Two approaches were considered: absorption chillers for space cooling and power production using the Organic Rankine cycle. These waste heat recovery alternatives aim to effectively utilize the two feasible waste heat streams at two different temperatures, which are the hot ventilation air from the combustion turbine rooms and the return lubrication oil from the power plant.

Ahmed Shahadha Hussein et. al.(2014) [4]

Developed a sophisticated method is from the earlier work and is used to calculate the cooling flow required for a nozzle guide vane (the first and second blade row) of a high temperature gas turbine, with given inlet gas temperature and coolant inlet temperature. Analyses of gas turbine plant performance, including the effects of turbine cooling, are also calculated.

Ankit Kumar et. al. (2017) [5]

Performed thermodynamic analysis of an open cycle gas turbine power plant. The mathematical formulation for the specific work and efficiency were derived and analyzed. The effect of operating parameters like the ambient temperature, relative humidity, compressor pressure ratio etc. of a gas turbine plant is studied

R. Espanani et. al.(2013) [6]

Presented that gas turbines are machines that work directly with ambient air, then anything that causes a change in the inlet air condition, affecting turbine efficiency. Relative humidity, mean sea level, and environment temperature have a direct effect on gas turbine efficiency.

Sharma Abhishek et. al. (2017) [7]

Reviewed research activities and studies carried out worldwide so far on gas turbine power plants. The historical development in chronological order was presented, followed by some fundamental features about the gas turbine. The paper mainly focused on the effect of different operating parameters and atmospheric conditions on the performance of gas turbine power plants. Research on gas turbines with evaporative coolers, heat exchangers and absorption chillers has also been assessed.

Lorena Giordano et. al. (2012) [8]

Proposed a unified approach for the analysis of different exhaust heat recovery techniques. This methodology is based on relationships of general validity, and on a characteristic plane for exhaust heat recovery, that indicates directly the performance obtainable with different recovery techniques, compared to a baseline non recovery plant. From the unified thermodynamic approach an innovative repowering scheme has been proposed.

K. Rama Krishna et. al. (July 2009) [9]

Studied how to improve the performance of a gas turbine plant. The most popular way is to lower intake air temperature to around 15°C (ISO) and 100 percent RH before entering the air compressor of a gas turbine (GT). This research proposes that use of alternative regenerator configuration (compared to the conventional regenerative cycle, the position of regenerator is kept immediately after the high pressure turbine), along with intake air cooling, helps increase the efficiency of gas turbine power plants. Cooling the air at the compressor inlet by using a spray cooler helps increase the density of air flowing into the plant, thereby increasing the specific power output. It also reduces the compressor-specific work input and hence increases the net output. The alternative regenerator configuration causes the air to enter into the combustion chamber at a higher temperature. The

increase in mean temperature of heat addition results in high efficiency for the gas turbine.

V. Tara Chand et. al. (2013) [10]

Presented how exergy analysis is carried out by conducting mass, energy and energy balance of each component in the gas turbine plant. Parametric analysis of the influence of various factors namely compression ratio (rp), compressor inlet air temperature (AT) and turbine inlet temperature (TIT) on irreversibilities of each and every component of a gas turbine plant is carried out. Exergy analysis revealed that most sensitive components in gas turbine plants were combustion chambers.

Awaludin Martin et. al. (July 2016) [11]

Described the performance of a 20 MW gas turbine power plant by using the exergy analysis and data from the plant's record books. The first and second laws of thermodynamics, as well as the mass and energy conservation law, were applied in each of the components. The results show that more exergy destruction occurred in the combustion chamber up to 71.03% or 21.98 MW. Meanwhile, the lowest exergy occured in the compressor at 12.33% or 3.15 MW. Thermal efficiency of the gas turbine power plant, according to the first law, was 33.77%, and exergy efficiency was 32.25%.

Fidelis Ibiang Abam et. al. (2012) [12]

Studied the effect of inlet air cooling system and components irreversibilities on the performance of an active 25MW gas turbine power plant was

investigated. The objective of this study was to establish the potential benefits of improving the performance of the current gas turbine plant into a more advanced cycle with high efficiency and power output through inlet air cooling.

Mohammad Hossein Ahmadi et. al. (2016) [13]

Aimed this study is to evaluate the performance of combined cycle power plants in full and part-load operations in terms of energy, exergy and environment. The environmental analysis is performed using a life cycle assessment method. Two different strategies are selected to control the gas turbine under part-load conditions, including Turbine Exhaust Temperature (TET) constant and Turbine Inlet Temperature (TIT) constant modes.

Results show that gas turbine has a higher efficiency in TET constant mode

Anmmar Mahmoud Ahmed et.al. (2016) [14]

Carried out exergy and energy analyses in each component to study the effect of compression ratio, ambient temperature, and load on energy losses and exergy destruction. The highest exergy destruction occurred in the combustion chamber and the lowest exergy destruction occurred in the compressor. This study reveals possible areas of focus to improve performance of the power plant in the future.

S.O. Oyedepo et. al. (April 2015) [15]

Presented thermodynamic analysis of the design and performance of eleven selected gas turbine power plants using the first and second laws of

thermodynamics concepts. Energy analysis showed that the combustion chamber and the turbine are the components having the highest proportion of energy loss in the plants. Energy loss in the combustion chamber and turbine varied from 33.31 to 39.95% and 30.83 to 35.24% respectively. The exergy analysis revealed that the combustion chamber is the most exergy destructive component compared to other cycle components. Exergy destruction in the combustion chamber varied from 86.05 to 94.67%.

A.Rezaei et al.2014) [16]

Studied options for waste heat recovery in a 700 MW gas-turbine combined cycle power plant in Thailand. Two approaches were considered: absorption chillers for space cooling and power production using the Organic Rankine cycle. These waste heat recovery alternatives aim to effectively utilize the two feasible waste heat streams at two different temperatures, which are the 8 hot ventilation air from the combustion turbine rooms and the return lubrication oil from the power plant.

M. H. Yilmaz et. al.(2014) [17]

Developed a sophisticated method is from the earlier work and is used to calculate the cooling flow required for a nozzle guide vane (the first and second blade row) of a high temperature gas turbine, with given inlet gas temperature and coolant inlet temperature. Analyses of gas turbine plant performance, including the effects of turbine cooling, are also calculated.

J. Yan (2017) [18]

Reviewed research activities and studies carried out worldwide so far on gas turbine power plants. The historical development in chronological order was presented, followed by some fundamental features of the gas turbine. The paper mainly focused on the effect of different operating parameters and atmospheric conditions on the performance of gas turbine power plants. Research on gas turbines with evaporative coolers, heat exchangers, and absorption chillers has also been assessed.

B. Dincer and A. Rosen [19]

Proposed a unified approach for the analysis of different exhaust heat recovery techniques. This methodology is based on relationships of general validity, and on a characteristic plane for exhaust heat recovery, that indicates directly the performance obtainable with different recovery techniques, compared to a baseline non-recovery plant. From the unified thermodynamic approach, an innovative repowering scheme has been proposed.

A.Ahmadi[20]

Presented that gas turbines are machines that work directly with ambient air, then anything that causes a change in the inlet air condition , affecting turbine efficiency. Relative humidity, mean sea level and environment temperature have a direct effect on gas turbine efficiency.

2.2 Conclusions

It is shown from air standard analysis that there are diminishing returns on efficiency as combustion temperature is increased; for real gases there appears to be a limit on this maximum temperature for maximum thermal efficiency. The compression ratio, ambient temperature, air to fuel ratio as well as the isentropic efficiencies can strongly influence thermal efficiency. In

addition, the thermal efficiency and power output decreases linearly with increase of the ambient temperature and air to fuel ratio.

2.3 Research Gap

Already extensive research to the best of our knowledge has been done, still we think there is a scope for further exploration to the increase of efficiency through the cooling method. Although vaporization and many other scientific and thermodynamic methods have already been used, we strongly believe that there's a possibility to explore other methods of Turbine Inlet Temperature (TIT) Cooling.

2.4 Objectives

- Energy Analysis and understanding the factors on which the performance is dependent.
- Exergy Analysis and understanding the factors on which the performance is dependent.
- Methods to increase efficiency to obtain higher output with the same available resources.

Chapter 3

3.1 Construction of Gas Turbine

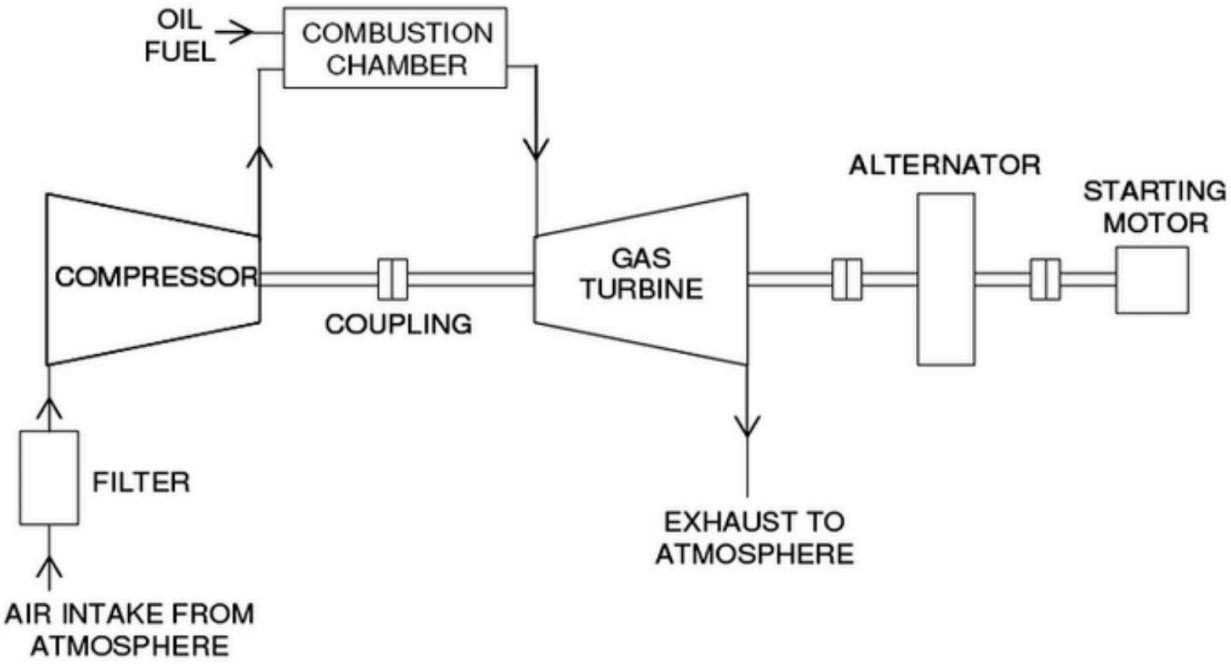


Fig 3.1

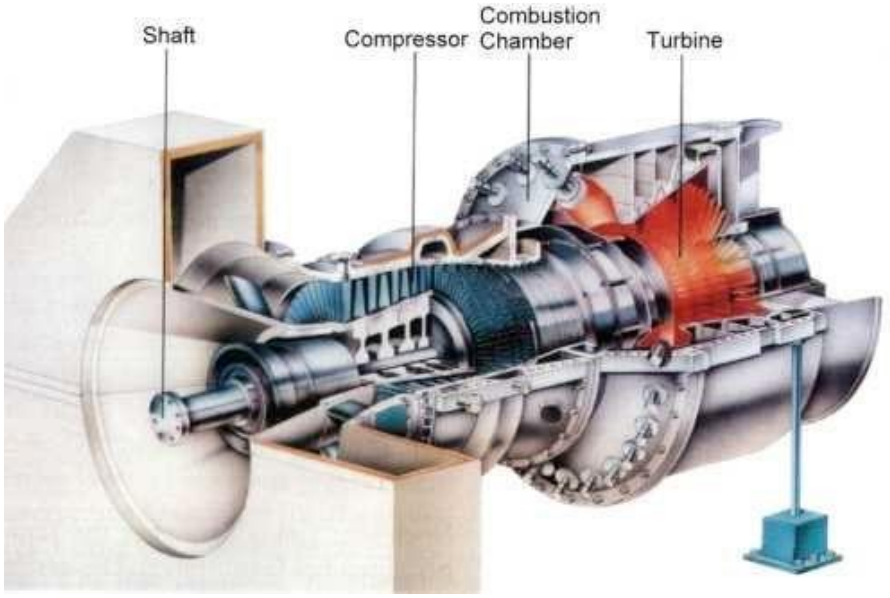


Fig 3.2

3.2 Components of Gas Turbine

- **The compressor** draws air into the engine, pressurizes it, and feeds it to the combustion chamber at speeds of hundreds of miles per hour.
- **A combustor** is a component or area of a gas turbine or scramjet engine where combustion takes place. It is also known as a burner, **combustion chamber** or flame holder.
- An **alternator** is a rotating machine which can produce electricity only when it rotates.
- A **turbine** is a device that harnesses the kinetic energy of some fluid - such as water, steam, air, or combustion gases - and turns this into the rotational motion of the device itself.

3.2 Working of Gas Turbine

In an ideal gas turbine, gases undergo four thermodynamic processes: isentropic compression, isobaric (constant pressure) combustion, an isentropic expansion, and heat rejection. Together, these make up the Brayton cycle. In a real gas turbine, mechanical energy is changed irreversibly (due to internal friction and turbulence) into pressure and thermal energy when the gas is compressed. Heat is added in the combustion chamber and the specific volume of the gas increases, accompanied by a slight loss in pressure. During expansion through the stator and rotor passages in the turbine, irreversible energy transformation once again occurs. Fresh air is taken in, in place of the heat rejection. If the engine has a power turbine added to drive an industrial generator or a helicopter rotor, the exit pressure will be as close to the entry pressure as possible with only enough energy left to overcome the pressure losses in the exhaust ducting and expel the exhaust. For a turboprop engine there will be a particular balance between propeller power and jet thrust which gives the most economical operation. In a turbojet engine only enough pressure and energy is extracted from the flow to drive the compressor and other components.

Chapter 4

4.1 Thermodynamic Processes

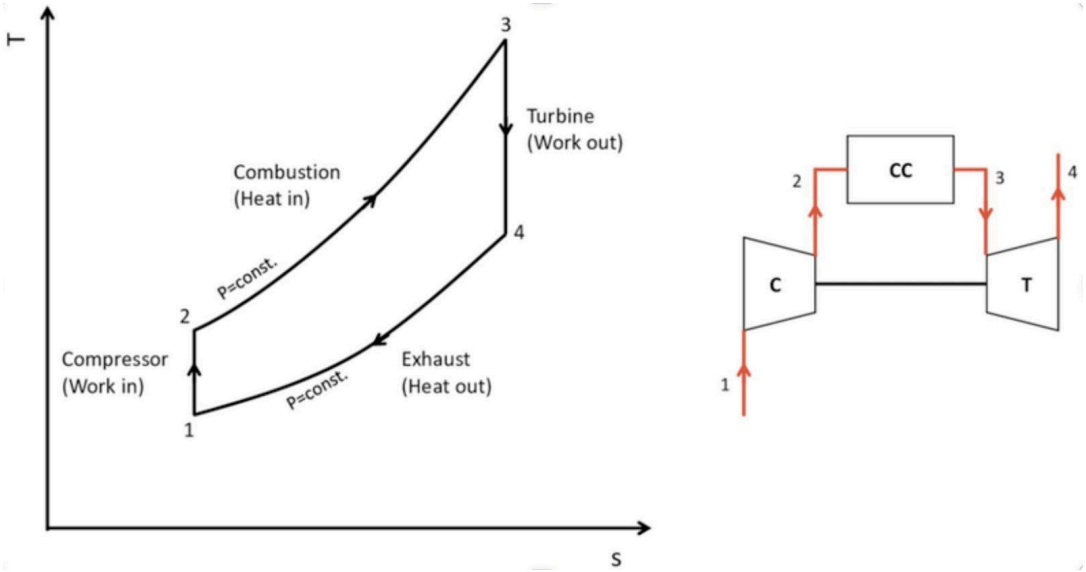


Fig 4.1

The basic operation of the gas turbine is a Brayton cycle (or joule cycle) with air as the working fluid : atmospheric air flows through the compressor that brings it to higher pressure ; energy is then added by spraying fuel into the air and igniting it so that the combustion generates a high-temperature flow ; this high-temperature pressurized gas enters a turbine, producing a shaft work output in the process, used to drive the compressor. The Gas Turbine power plants are being used for producing electricity, operating airplanes and for various industrial applications such as refineries and petrochemical plants.

Process 1-2 Isentropic Compression

Isentropic compression (compression in a compressor) – The working gas (e.g. helium) is compressed adiabatically from state 1 to state 2 by the compressor (usually an axial-flow compressor). The surroundings do work on the gas, increasing its internal energy (temperature) and compressing it (increasing its pressure). On the other hand the entropy remains unchanged. The work required for the compressor is given by

$$W_C = H_2 - H_1 .$$

Process 2-3 Isobaric Heat Addition

Isobaric heat addition (in a heat exchanger) – In this phase (between state 2 and state 3) there is a constant-pressure heat transfer to the gas from an external source, since the chamber is open to flow in and out. In an open ideal Brayton cycle, the compressed air then runs through a combustion chamber, where fuel is burned and air or another medium is heated (2 → 3). It is a constant-pressure process, since the chamber is open to flow in and out. The net heat added is given by

$$Q_{add} = H_3 - H_2$$

Process 3-4 Isentropic Expansion

Isentropic expansion (expansion in a turbine) – The compressed and heated gas expands adiabatically from state 3 to state 4 in a turbine. The gas does work on the surroundings (blades of the turbine) and loses an amount of internal energy equal to the work that leaves the system. Again the entropy remains unchanged. The work done by the turbine is given by

$$W_T = H_4 - H_3 .$$

Process 4-1 Isobaric Heat Rejection

Isobaric heat rejection (in a heat exchanger) – In this phase the cycle completes by a constant-pressure process in which heat is rejected from the gas. The working gas temperature drops from point 4 to point 1. The net heat rejected is given by

$$Q_{re} = H_4 - H_1$$

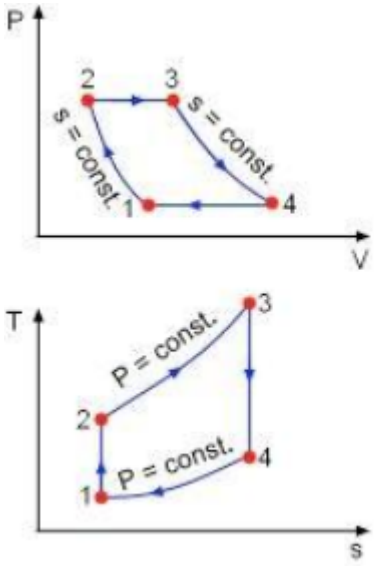


Fig 4.2

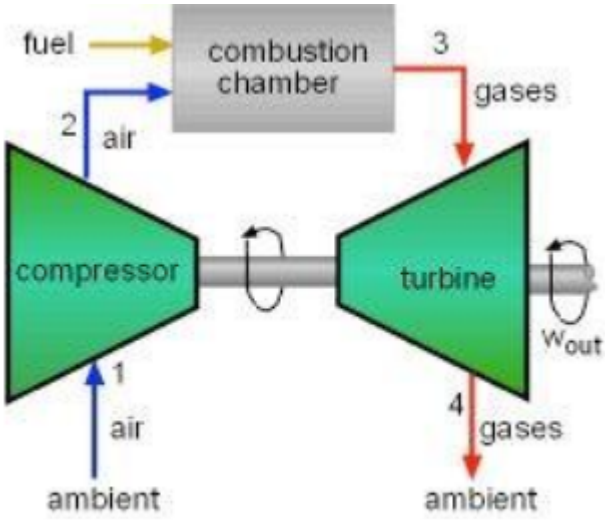


Fig 4.3

The P-v and T-s diagrams of an ideal Brayton cycle are shown on the left. In an ideal Brayton cycle, heat is added to the cycle at a constant pressure process (process 2-3).

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

Heat is rejected at a constant pressure process (process 4 -1).

$$q_{out} = h \quad _4 - h \quad _1 = c$$

$$_P(T_4 - T \quad _1)$$

4.2 Energy Analysis (1st Law Efficiency)

Then the thermal efficiency of the ideal Brayton cycle under the cold air-standard assumption is given as

$$\eta_{th,Brayton} = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{P_1}{P_2}\right)^{(k-1)/k}$$
$$= 1 - \frac{1}{r_P^{(k-1)/k}}$$

Process 1-2 and process 3-4 are isentropic processes, and $P_2 = P_3$ and $P_4 = P_1$ thus,

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

$$\frac{T_1}{T_2} = \frac{T_4}{T_3} \rightarrow T_4/T_1 = T_3/T_2$$

Considering all the relations above, the thermal efficiency becomes,

$$\eta_{th,Brayton} = 1 - \frac{c_P(T_4 - T_1)}{c_P(T_3 - T_2)}$$
$$= 1 - \frac{T_1}{T_2}\frac{T_4/T_1 - 1}{T_3/T_2 - 1}$$

Where $r_P = P_2/P_1$ is the pressure ratio and k is the specific heat ratio.

The actual gas-turbine cycle is different from the ideal Brayton cycle since there are irreversibilities. Hence, in an actual gas-turbine cycle, the compressor consumes more work and the turbine produces less work than that of the ideal Brayton cycle. The irreversibilities in an actual compressor and an actual turbine can be considered by using the adiabatic efficiencies of the compressor and turbine. They are:

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

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

4.3 Exergy Analysis

The steady state exergy balance equation for a control volume according to (Kotas,1995) is

$$\dot{E}x = \sum_j \left(1 - \frac{T_0}{Tj}\right)\dot{Q}_j + \dot{W}_{CV} + \sum_i \dot{m}_i e_i - \sum_e \dot{m}_e e_e \qquad (1)$$

The subscripts i refer to conditions at the inlet, e and j exits of control volume boundaries, and o is the reference state. The components' energy balance is carried out using equation (1).

$$\dot{E}x_{c_1} = \dot{m}_a\, c_p^h (T_1 - T_0) - T_0\, \dot{m}_a (s_1 - s_0) \qquad (2)$$

$$\text{and } s_1 - s_0 = c_p^s \ln\left(\frac{T_1}{T_0}\right) - R \ln\left(\frac{p_1}{p_0}\right) \qquad (3)$$

Where $E_{xC1}$is the exergy rate of the air entering to the compressor unit.

$$\dot{E}x_{c_2} = \dot{m}_a\, c_p^h (T_2 - T_0) - T_0\, \dot{m}_a (s_2 - s_0) \qquad (4)$$

$$\text{and } s_2 - s_0 = c_p^s \ln\left(\frac{T_2}{T_0}\right) - R \ln\left(\frac{p_2}{p_0}\right) \qquad (5)$$

Where E_{xc2} is the exergy rate of the air leaving the compressor unit

$$\dot{W}c = \dot{m}_{air}\, c_p^h (T_2 - T_1) \qquad (6)$$

W_c is the power input rate to the compressor,

$$\dot{I}c = \dot{W}c + \dot{E}x_1 - \dot{E}x_2 \qquad (7)$$

I_c is the total exergy destruction rate in the compressor, T_1 and T_2 are the compressor inlet and exit temperatures, respectively; P_1 and P_2 are the compressor inlet and exit pressures, respectively, c^h_p and c^s_p are mean molar isobaric heat capacity for evaluating enthalpy and entropy changes. They are expressed in equations (8) and (9).

$$c_p^h = \frac{1}{T - T_0} \int_{T_0}^{T} \bar{c}_p dT \qquad (8)$$

$$c_p^s = \frac{1}{\ln\left(\frac{T}{T_0}\right)} \int_{T_0}^{T} \bar{c}_p dT \qquad (9)$$

C_p is obtained in a polynomial form as a function of temperature as given in equation(10)

$$\bar{c}_p(T) = a + \bar{b}^2 T + cT^2 + dT^3 \qquad (10)$$

The values of a, b, c and d are constant characteristics of the gas considered and obtained from standard tables.The combustion process is stage 2-3 from figure 1. The exergy flow rate in the combustion chamber is obtained from equation (1). In the combustion chamber, the shaft work is zero and equation (1) reduces to:

$$\sum_i \left(\dot{E}xf \right) i + \sum_{in} \dot{E}x_2 = \sum_{out} \dot{E}x_3 + \dot{I}_{cc} \qquad (10)$$

I_{cc} is the combustion chamber irreversibility..E xfis the fuel exergy rate which is expressed in equation (11)

$$\dot{E}xf = \dot{E}x_{ph} + \dot{E}x_{ch} = \dot{m}_{fuel}\left(c_p^h\left(T_5 - T_0\right) - T_0\left[c_p^s \ln\left(\frac{T_5}{T_0}\right) - R\ln\left(\frac{p_5}{p_0}\right)\right] + \sum_i x_i e_{i0} + RT_0 \sum_i \ln \gamma_i x_i\right)$$

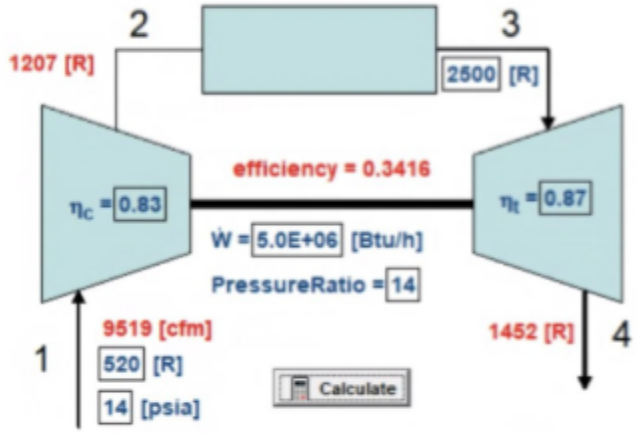
Chapter 5

5.1 Results (Energy Analysis)

For the study we used the following parameters:

efficiency	0.3553
eta_c	0.83
eta_t	0.87
h2_id	244.3
h4_id	313.4
h[1]	114.8
h[2]	270.8
h[3]	645.9
h[4]	356.6
m_dot	37519
PressureRatio	14
P[1]	14
P[2]	196
P[3]	196
P[4]	14
q_23	375.1
s2_id	1.337
s4_id	1.578
s[1]	1.337
s[3]	1.578
T[1]	520
T[2]	1118
T[3]	2500
T[4]	1452
V_dot	7943
w_c	-156
W_dot	5.0E+06
w_net	133.3
w_t	289.3

Fig 5.1



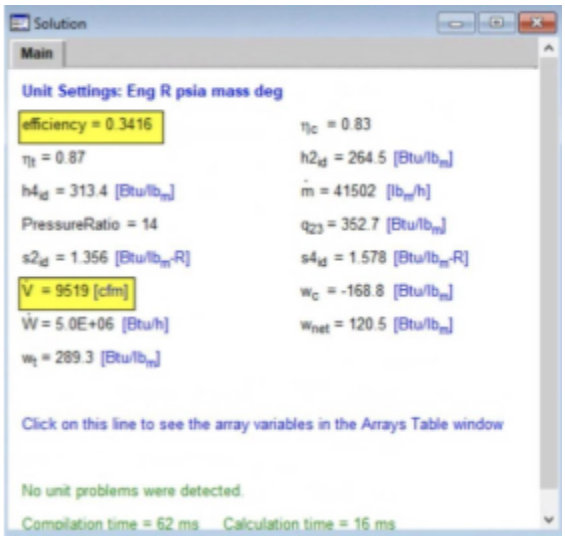


Fig 5.2

Fig 5.3

We obtained the following results:

We received an efficiency of ***34.16%*** with our model and the above-taken values.

5.2 Thermodynamic Analysis

An exergy analysis of a GT topping cycle is presented in this section. The effects of ambient temperature, compression ratio, and TIT on power output, exergy destruction rate, and exergy efficiency are obtained from the exergy (second law) analysis.

With an increase in ambient temperature from 0°C to 50°C the exergy destruction rate of the compressor, combustion chamber, and gas turbine decreases. The maximum exergy destruction takes place in the combustion chamber followed by the gas turbine.

There is a significant change in the exergy efficiency of the combustion chamber with temperature. As the ambient temperature increases, the exergy efficiency decreases. This is due to the rise in temperature the fuel flow rate decreases and thus the total fuel exergy also reduces. Thus the overall exergy destruction and exergy efficiency of the combustion chamber reduces with the rise in temperature. At the same time, the exergy efficiency of gas turbines increases with the rise in ambient temperature from 0°C to 30°C from 92.76% to 92.92% and thereafter by a further increase in temperature efficiency reduces as the gas turbine load reduces with the rise in temperature from the rated capacity. The exergy destruction rate of the gas turbine reduces with the rise in ambient temperature. As the temperature rises the gas turbine load reduces and the turbine exit temperature increases. Total exergy at the gas turbine inlet and outlet reduces, resulting in a reduction in the gas turbine exergy destruction rate.

The effect of TIT and compression ratio on power output and overall exergy efficiency of the gas cycle is reference temperature. Exergy efficiency and power output of the gas cycle increase with the increase in TIT and decrease with a decrease in compression ratio. An increase in the TIT leads to an increase in the GT exergy efficiency due to the fact that the GT turbine work output increases. As the load increases, this leads to a reduction in exergy destruction. Therefore, it can be concluded that TIT is the most important parameter in designing the gas turbine cycle due to the decrease in exergy destruction and increase in cycle exergy efficiency.

5.3 Conclusions

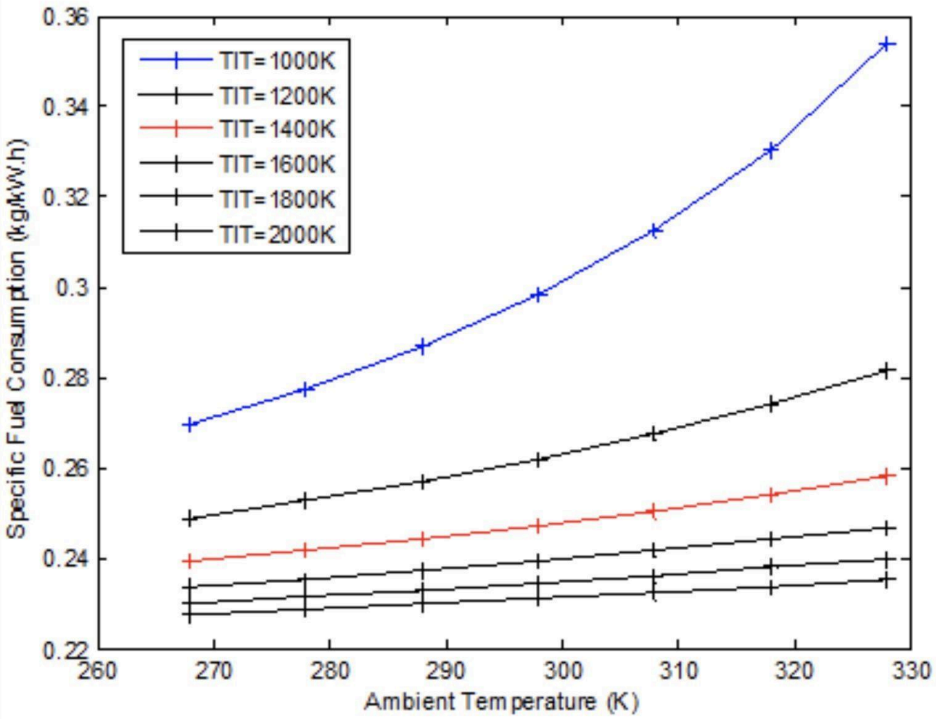


Fig 5.4

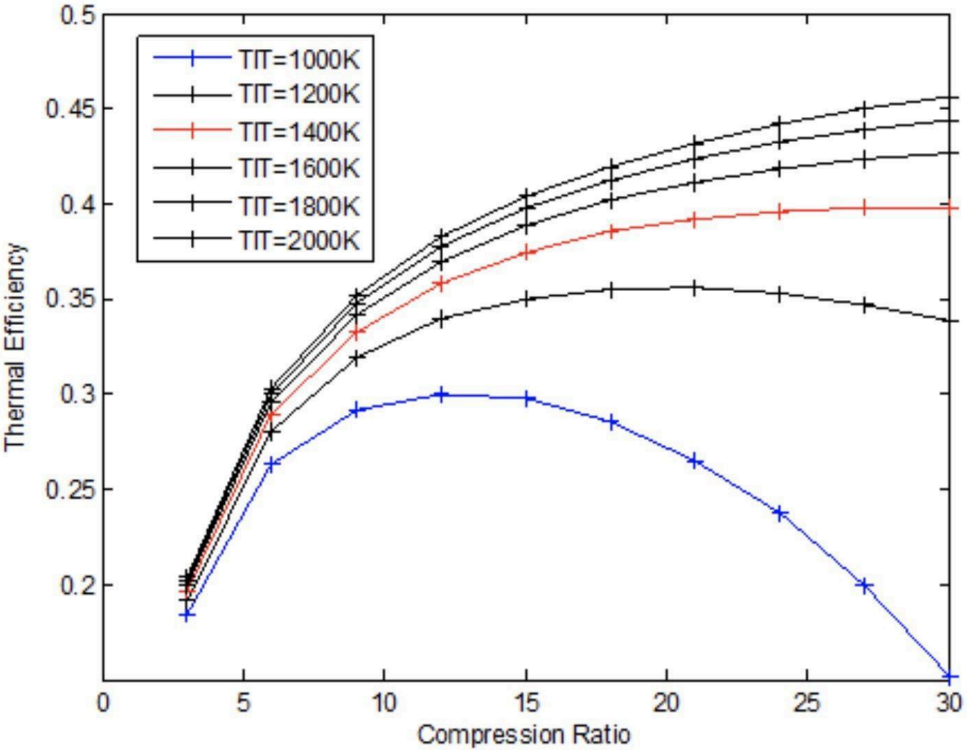


Fig 5.5

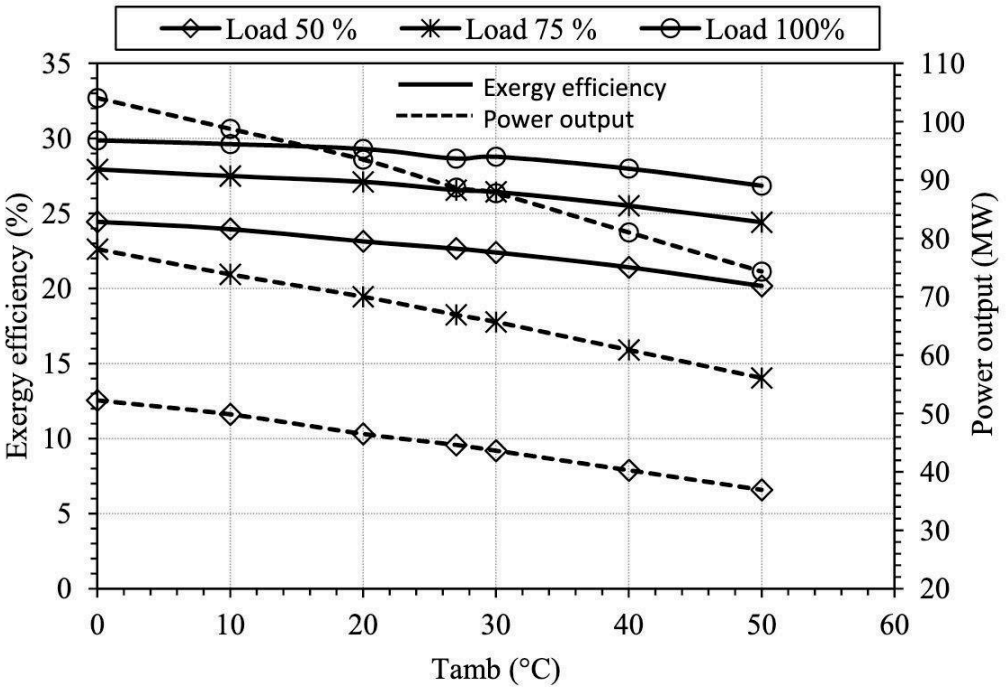

Load 50 %
Load 75 %
Load 100%
Exergy efficiency
Power output
Exergy efficiency (%)
Power output (MW)
Tamb (°C)
0
5
10
15
20
25
30
35
20
30
40
50
60
70
80
90
100
110
0
10
20
30
40
50
60

Fig 5.6

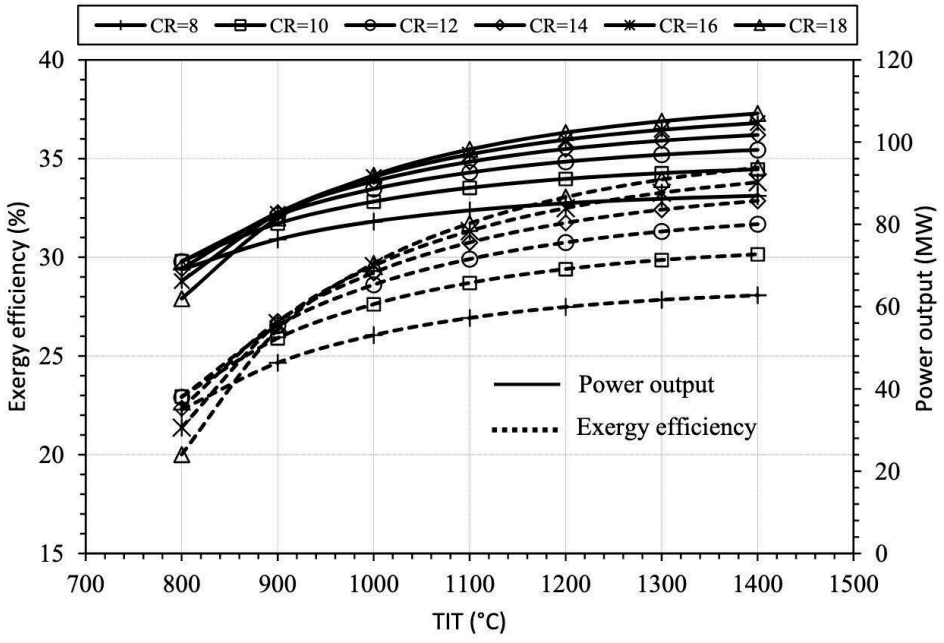

CR=8
CR=10
CR=12
CR=14
CR=16
CR=18
Power output
Exergy efficiency
Exergy efficiency (%)
Power output (MW)
TIT (°C)
15
20
25
30
35
40
0
20
40
60
80
100
120
700
800
900
1000
1100
1200
1300
1400
1500

Fig 5.7

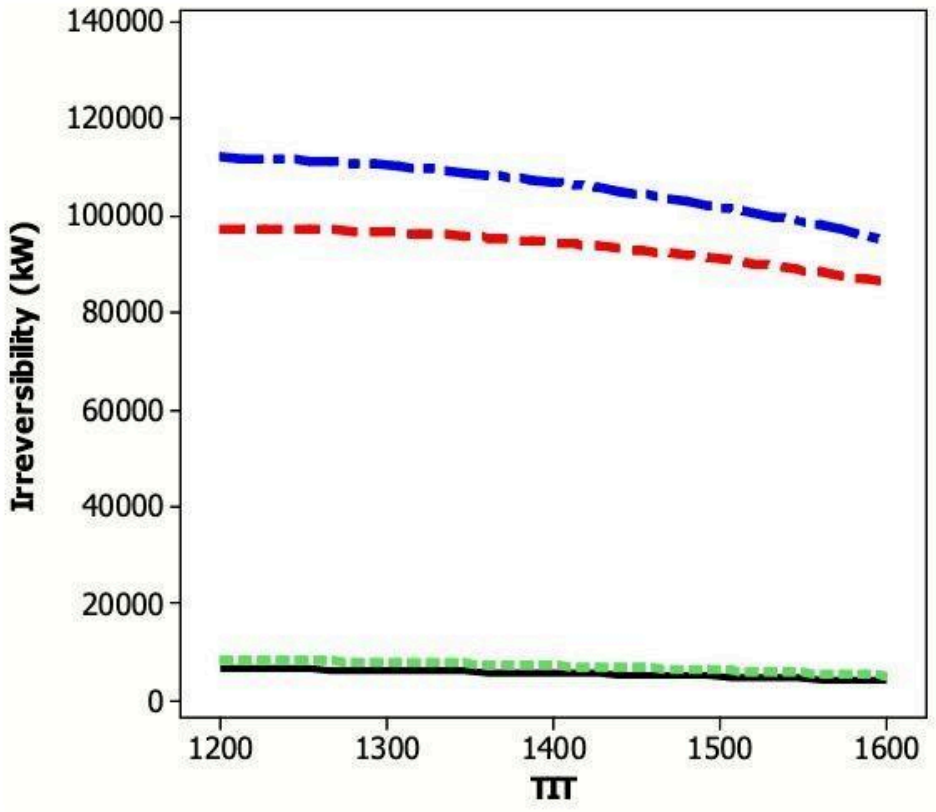


Fig 5.8

5.4 Methods of Improving Its Thermal Efficiency

Gas turbines work on a constant-volume cycle, so power output depends on air mass flow. Warm air is less dense than cold air, so power output is reduced. Also, warm air needs more energy to compress it than cold air, removing more work from the compressor, and increasing internal losses as it does so. Therefore, gas turbine efficiency can be improved by cooling the inlet air. This is especially effective in hot, dry climates. Gas turbine air inlet cooling can be achieved in a number of ways, including the use of mechanical and absorption chillers (with or without thermal energy storage), or the evaporative type (wetted media, fogging, and wet compression/overspray).

We again ran the analysis with a slightly low inlet temperature with a change of 40 kelvin with all other parameters remaining the same and found a significant increase in the final

efficiency obtained. The parameters we used for the second run of thermodynamic analysis are:

efficiency	0.3553
eta_c	0.83
eta_t	0.87
h2_id	244.3
h4_id	313.4
h[1]	114.8
h[2]	270.8
h[3]	645.9
h[4]	356.6
m_dot	37519
PressureRatio	14
P[1]	14
P[2]	196
P[3]	196
P[4]	14
q_23	375.1
s2_id	1.337
s4_id	1.578
s[1]	1.337
s[3]	1.578
T[1]	480
T[2]	1118
T[3]	2500
T[4]	1452
V_dot	7943
w_c	-156
W_dot	5.0E+06
w_net	133.3
w_t	289.3

Fig 5.9

On this new set of values we were able to achieve efficiency of ***35.53%*** from our Gas Turbine Power Plant. An increase of about **1.4%** from the last reading.

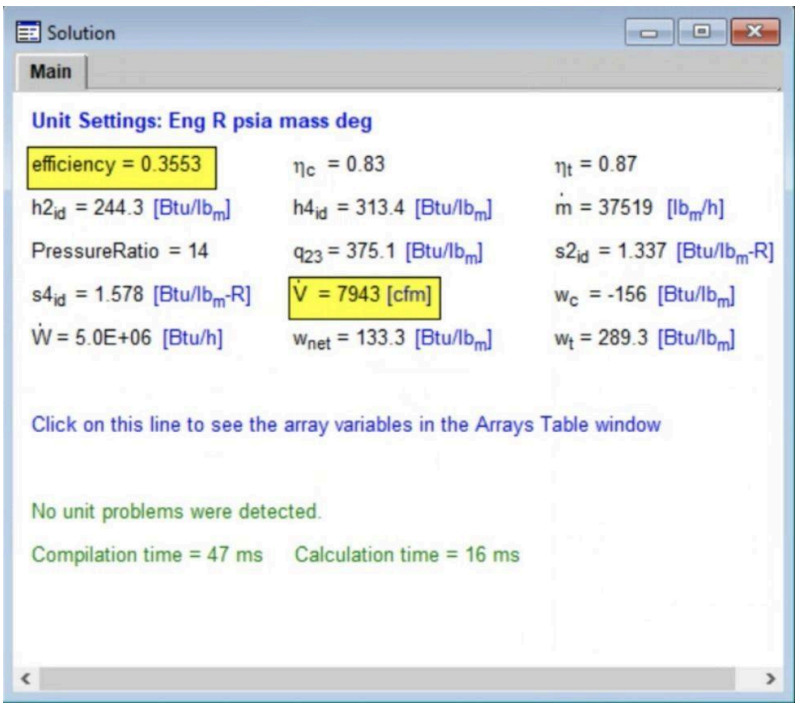


Fig 5.10

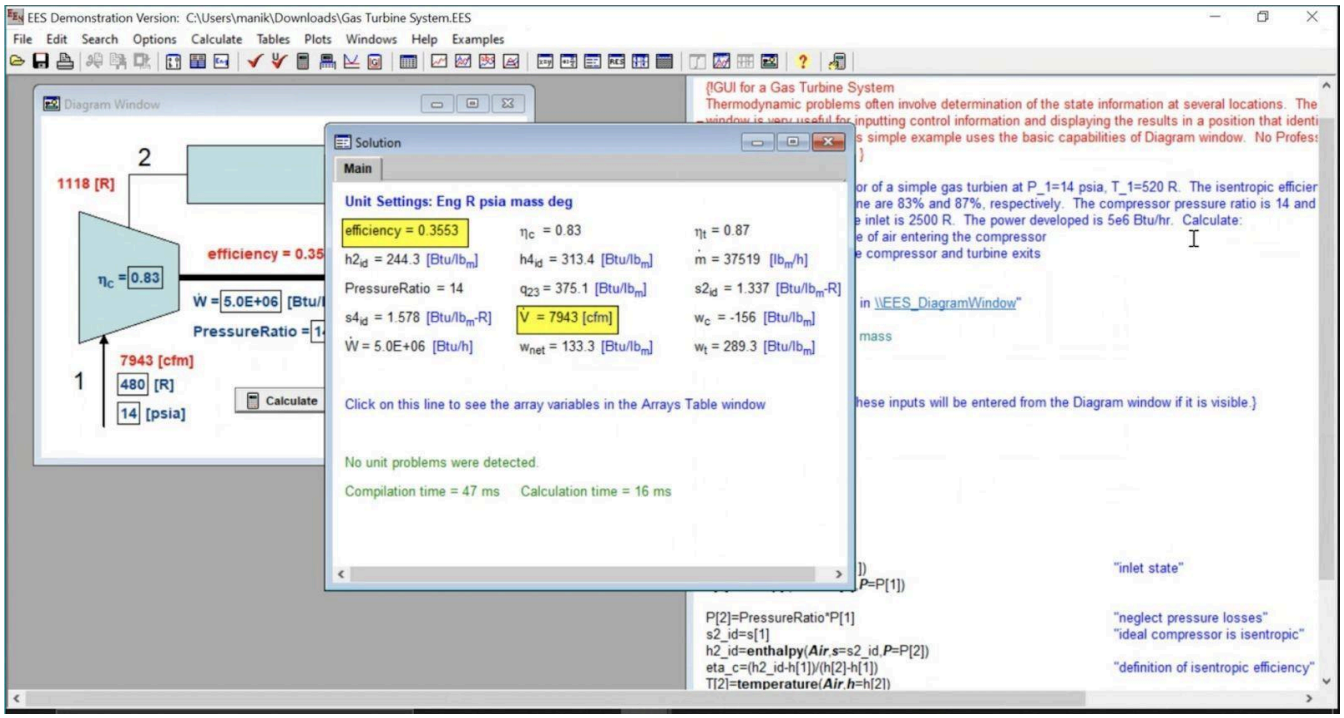


Fig 5.11

Chapter 6

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Appendix (Code used)

```
h[1]=enthalpy(Air,T=T[1])                                   "inlet state"
s[1]=entropy(Air,T=T[1],P=P[1])

P[2]=PressureRatio*P[1]                                     "neglect pressure losses"
s2_id=s[1]                                                  "ideal compressor is isentropic"
h2_id=enthalpy(Air,s=s2_id,P=P[2])
eta_c=(h2_id-h[1])/(h[2]-h[1])                              "definition of isentropic efficiency"
T[2]=temperature(Air,h=h[2])
w_c=h[1]-h[2]

q_23=h[3]-h[2]                                              "heat input per unit mass of air"
h[3]=enthalpy(Air,T=T[3])
s[3]=entropy(Air,T=T[3],P=P[3])
P[3]=P[2]                                                   "neglect pressure losses"

s4_id=s[3]                                                  "ideal turbine is isentropic"
P[4]=P[1]
h4_id=enthalpy(Air,s=s4_id,P=P[4])
eta_t=(h[3]-h[4])/(h[3]-h4_id)                              "definition of isentropic efficiency"
T[4]=temperature(Air,h=h[4])
w_t=h[3]-h[4]

w_net=w_t+w_c                                               "net work is the sum of the turbine and
compressor work"
efficiency=w_net/q_23                                       "cycle thermal efficiency"
m_dot=W_dot/w_net                                           "mass flow rate of air through cycle"
V_dot=m_dot*volume(Air,T=T[1],P=P[1])*convert(ft^3/h,cfm)   "volumetric flow of inlet air"
```