UNIVERSITY OF GÄVLE

FACULTY OF ENGINEERING AND SUSTAINABLE DEVELOPMENT
Department of Building, Energy and Environmental Engineering

# GAS TURBINE THERMODYNAMIC AND PERFORMANCE ANALYSIS METHODS USING AVAILABLE CATALOG DATA

K.A.B.Pathirathna

2013

Student thesis, Master degree (two years), 30 HE
Energy Systems
Master Programme in Energy Systems

Supervisors: Dr. N. Senanayake / Hina Noor / Iresha Athanayake
Examiner: Prof. Torsten Fransson

# Abstract

The overall performance of an open gas turbine can be judged with the knowledge of the main design parameters, but this is quite challenging due to the limited data available from the manufacturer. The turbine manufacturers usually provide data about the turbine interface. However, the data (or information) required to analyze the thermodynamic quality of the gas turbine remains hidden. A theoretical model was developed to evaluate the thermodynamic performance of an open gas turbine (with one combustor) by using available catalog data. Realistic values (from experience) were assumed for the compressor polytropic efficiency, mechanical efficiency, electrical efficiency and pressure drops of the gas turbine. The Engineering Equation Solver (EES) tool has been used for implementing the theoretical model. The published catalog data from the manufacturers is kept as main inputs for the modeled program. The model calculates (or predicts) values for the unknowns i.e. temperature, isentropic and polytropic efficiencies of the individual components (inside the gas turbine), power output and some emission related parameters. The developed program is aimed to be used in the learning lab at the KTH energy department while being a part of the CompEdu learning platform. This program should provide students a possibility to fully analyze the performance of the open gas turbine cycle while judging consistency of gas turbine cycle data sets and completing incomplete gas turbine datasets.

# Acknowledgments

I would like to express my sincere gratitude to my supervisor at KTH Hina Noor for her supervision and the guidance given in completing the task. Then I would like to extend my thanks to my local supervisors Dr. N. Senanayake and Iresha Athanayake for their constant advices and supervision.

Next I would like to extend my thanks to Dr. Hans E. Wettstein for sharing his knowledge with my and his constant advices.

I would like to express my thanks to our DSEE program coordinator Ruchira Abeyweera for his support and guidance to completing this MSc.

Finally I would like to extend my sincere gratitude to Sunimal, Sasindu and Pradeep for their encouragement and various supports given to me.

# Nomenclature

## List of Abbreviations

| | |
|---|---|
| *AFR* | Air to Fuel Ratio |
| *C* | Velocity |
| *CK* | Kelvin Constant |
| *Cp* | Constant Pressure Heat Capacity |
| *DLE* | Dry Low Emission |
| *EES* | Engineering Equation Solver |
| *GUI* | Graphical User Interface |
| $h$ | Specific enthalpy |
| $h_0$ | Stagnation enthalpy |
| $h_1$ | Enthalpy at compressor inlet |
| $h_2$ | Enthalpy at compressor outlet |
| $h_3$ | Enthalpy at turbine inlet |
| $h_4$ | Enthalpy at turbine outlet |
| $h_{0X}$ | Stagnation enthalpy at x ( x = 1, 2, 3, 4 ) |
| *HRSG* | Heat Recovery Steam Generator |
| *ISO* | International Organization for Standardization |
| *KTH* | Kungliga Tekniska Hogskoolan (Royal Institute of Technology) |
| *LHV* | Lower Heating Value |
| $P$ | Pressure |
| $P_1$ | Pressure at compressor inlet |
| $P_2$ | Pressure at compressor outlet |
| $P_3$ | Pressure at turbine inlet |
| $P_4$ | Pressure at turbine outlet |
| $Q$ | Heat |
| $Q_{in}$ | Heat supply to gas turbine |
| *RH* | Relative Humidity |
| $r$ | Pressure ratio/Compression ratio |
| $T$ | Temperature |
| $T_0$ | Stagnation temperature |
| $T_1$ | Temperature at compressor inlet |
| $T_2$ | Temperature at compressor outlet |
| $T_3$ | Temperature at turbine inlet |
| $T_4$ | Temperature at turbine outlet |

| | |
|---|---|
| $T_{0X}$ | Stagnation temperature at x ( x = 1, 2, 3, 4 ) |
| *TA* | Turbine exit temperature |
| *TK* | Compressor exit temperature |
| *TIT* | Turbine Inlet Temperature |
| *TS* | Temperature-Entropy |
| *S* | Entropy |
| *W* | Work |
| $W_{12}$ | Compressor work |
| $W_{34}$ | Turbine work |
| $W_{net}$ | Gas turbine net work |

## Units

| | |
|---|---|
| *bar* | Bar |
| $^0C$ | Degree Celsius |
| $^0F$ | Degree Fahrenheit |
| *HP* | House Power |
| *K* | Kelvin |
| *psi* | Pounds per square inch |

## Greek symbols

| | |
|---|---|
| $\gamma$ | Heat Capacity Ratio |
| $\eta$ | Efficiency |

# Table of Contents

# List of figures

# List of tables

# 1 Introduction

Gas turbines are common power generators which are used mainly in power generation systems and propulsion systems. Most of the gas turbines are internal combustion machines while the rest are fired externally. The sizes of gas turbines can vary from 500kW to 250MW according to their applications (Energy and Environmental Analysis (an ICF International Company), 2008). Especially for high power applications the gas turbines are widely used compared to conventional reciprocating engines due to its high power density. Absence of the reciprocating and rubbing members inside the gas turbine is also a positive point compared to reciprocating engine and that will enhance gas turbines' reliability.

A typical gas turbine mainly consists of three components namely compressor, combustion chamber and turbine. The three main components of a gas turbine are illustrated in figure 1. Atmospheric air is compressed by the compressor then heated inside the combustion chamber and finally expanded inside the turbine. As a result of that the turbine produces work to the surrounding. Some fraction of that work is used by the compressor while the balance work can be considered as the net work.

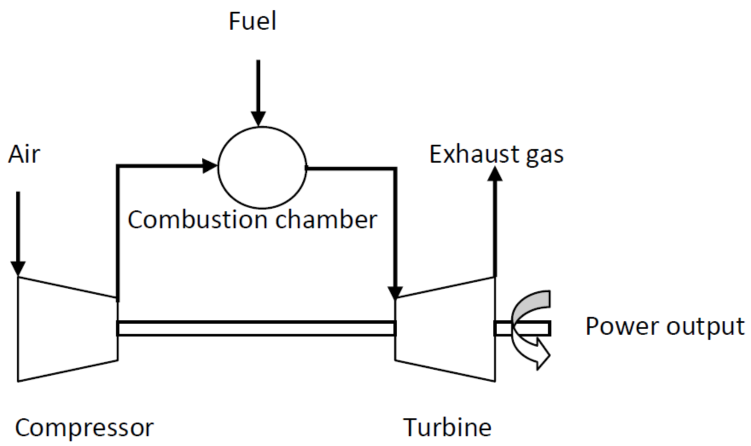



**Figure 1: Typical open gas turbine**

## 1.1 Gas turbine theory

The Brayton or the Joule cycle is commonly used to analyze the gas turbine systems and the figure 2 shows a Temperature-Entropy (TS) diagram representation of an ideal Brayton cycle. In figure 2, from point 1 to point 2 the air is isentropically compressed and the heat is supplied at constant pressure from point 2 to point 3. Finally the air is isentropically expanded from point 3 to point 4. In practice, the compression process and the expansion process always increase their entropy along the flow path due to the various losses inside the machines. Practically, the process from point 2 to point 3 also experiences the pressure drop along the flow path due to losses. Hence, the overall performance of the gas turbine highly deviates from the ideal cycle.

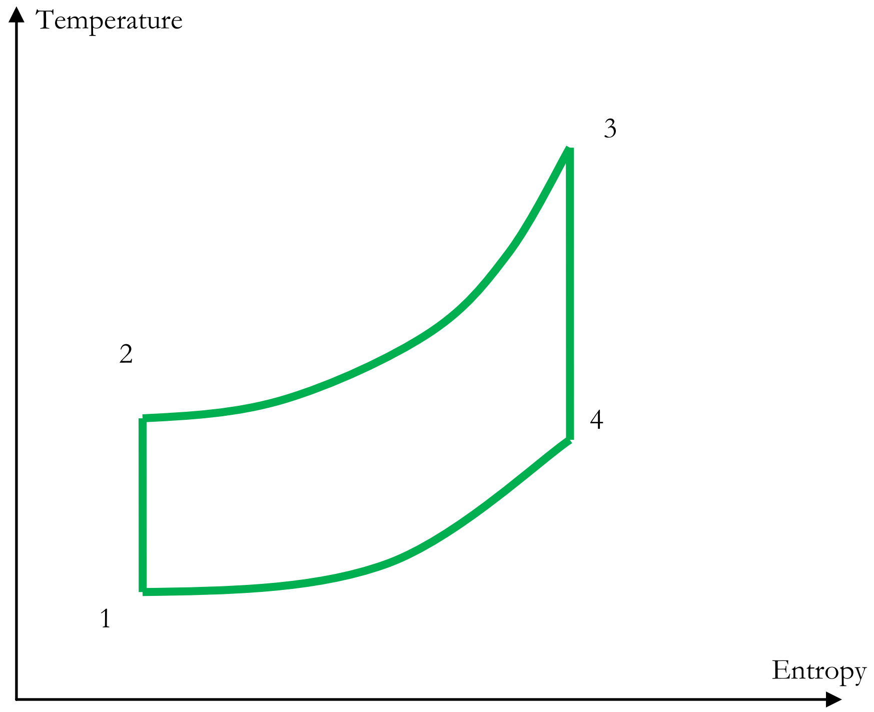



**Figure 2: T-S diagram for closed gas turbine cycle**

## 1.2 Background

When analyzing the overall performance of gas turbines, the importance of thermodynamic properties comes into play. Those thermodynamic properties lie along with the processes from points 1 to 4 in figure 2. Temperature, pressure, air to fuel ratio and the Relative Humidity (RH) are some of the important parameters which are needed for the performance analysis. The properties of the moist air need to be considered in the process 1 to 2 in the figure 2. During the process starting from point 2 to point 4 in the figure 2, the properties of the fuel-air mixture which is supplied to the gas turbine for combustion has also to be considered. Once all the parameters are known, the thermodynamic analysis can be carried out for the gas turbine.

## 1.3 Problem statement

During the design stage of any gas turbine, the thermodynamic properties influencing the gas turbine performances are optimized. The main objective is to manufacture highly efficient, more reliable machine for the energy market. However, when delivering the gas turbines to the customers the manufacturers do not expose thermodynamic property information or any other intermediate thermodynamic properties. They keep that data hidden and deliver only the interface data of the gas turbine which only gives a slight idea about the gas turbine design and performance. The turbine manufacturers usually supply five main parameters in their catalogues that are sufficient for installation requirements. Those are pressure ratio, gas turbine electrical output, overall efficiency, gas turbine exhaust temperature and exhaust mass flow rate. The given data is adequate from the operational point of view but, without knowing the hidden data it is not possible to fully analyze the specific gas turbine thermodynamically to gain complete knowledge of it for possible alterations depending upon circumstances. Consequently the importance of the revelation of hidden data of the commercially available gas turbines has arisen.

## 1.4 Objective

The main objective of this research is to determine a suitable method to reveal the hidden data related to the thermodynamic performance of commercial gas turbines from the catalog data.

# 2 Literature review on gas turbine performance

There is a considerable amount of literature under the topic of gas turbine performance and in most of the cases the authors tried to vary certain thermodynamic parameters and analyze the performances of the gas turbine. These thermodynamic parameters include compression ratio, ambient temperature, ambient pressure, humidity, heat rate, turbine inlet temperature, specific fuel consumption, air to fuel ratio, component efficiency. Therefore, it is considered important to discuss about the specified thermodynamic parameters.

## 2.1 Site dependent parameters

According to the ISO standards 3977-2 (Gas Turbines - Procurement - Part 2: Standard Reference Conditions and Ratings) the ISO ambient conditions for the industrial gas turbine are described as follows (Johnzactruba, 2009).

| | |
|---|---|
| Ambient temperature | 15 $^0$C/59 $^0$F |
| Relative humidity | 60 % |
| Ambient pressure | 1.013 bar/14.7 psi |

The above mentioned parameters are directly related to the density of air. Therefore, the deviation of ambient conditions from the above ISO ambient conditions results in change in the air density. As a result of that the amount of air mass enters the gas turbine changes. Since gas turbines are fixed displacement machines (Petchers, 2002) consequently, the performance of the gas turbine will change. Therefore, the change in the ambient conditions directly influences gas turbine performance.

### 2.1.1 Ambient temperature

Ambient temperature can be simply defined as the temperature of the surrounding or the temperature of the environment. Both the natural and manmade air breathing systems uses ambient air to keep it functioning properly. Internal combustion engines, gas turbines and compressors can be considered as manmade air breathing machines. In the following paragraph gas turbine performance and ambient temperature relationship is explained.

Increases in the ambient temperature can highly affect the gas turbine performance. When the inlet air is hot the net power of the gas turbine reduces. For every 1 $^0$C increment in the ambient temperature the amount of the reduction in power output is nearly 0.9% (Petchers, 2002). Figure 3 indicates how the ambient temperature affects the gas turbine power output. The Y-axis of the figure 3 represents the ratio between power output at any temperature and power output at reference temperature. That ratio is defined as power output correction factor. The unit is defined as $HorsePower_{any}/HorsePower_{reference}$ (HP/HP). The reference temperature for the curve in the figure 3 is 15 $^0$C (59 $^0$F) and for that reference temperature the gas turbine power output correction factor can be taken as 1 HP/HP (Y-axis). It can be seen that after the reference temperature (59 $^0$F) the output power correction factor reduces and vice versa. With the increase of the ambient temperature the density of the air decreases. Consequently the air mass flow rate into the turbine decreases. As a result of that the gas turbine power output reduces.

The thermal efficiency of the gas turbine also changes with the ambient temperature. For an increment of the ambient temperature by one Kelvin above the ISO condition, the reduction of the gas turbine thermal efficiency is nearly 0.1% (Sa, et al., 2011). The figure 4 represents the relationship between ambient temperature and thermal efficiency of the gas turbine. According to the figure 4 the thermal efficiency reduces with the increasing ambient temperature.

When the ambient temperature decreases, the density of the air tends to increase. Therefore the inlet air mass flow rate of the compressor increases. As a result, the fuel mass flow rate will increase, to keep air to fuel ratio constant, consequently the specific fuel consumption increases. With the decrease of the ambient temperature, both the air mass flow rate and the fuel mass flow rate increase (Rahman, et al., 2011).

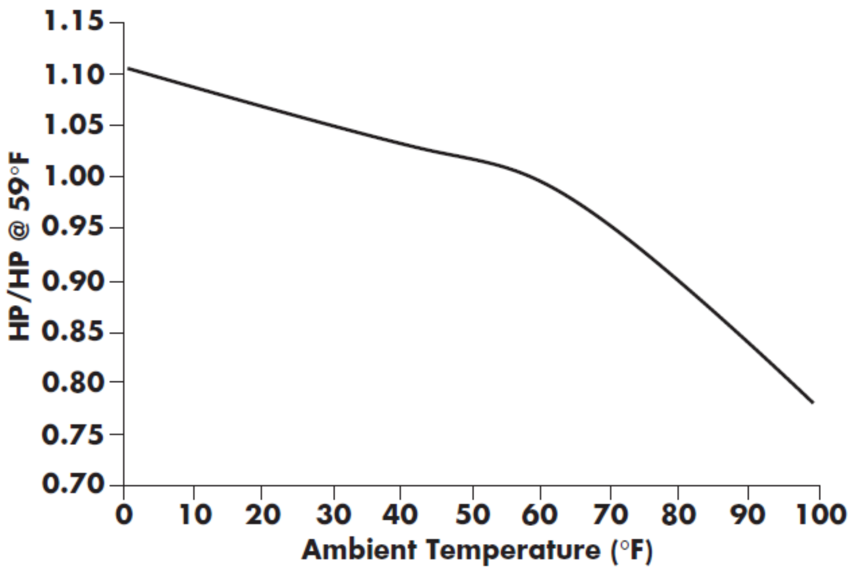



**Figure 3: Ambient Temperature Power Correction Factor Curve (Petchers, 2002)**

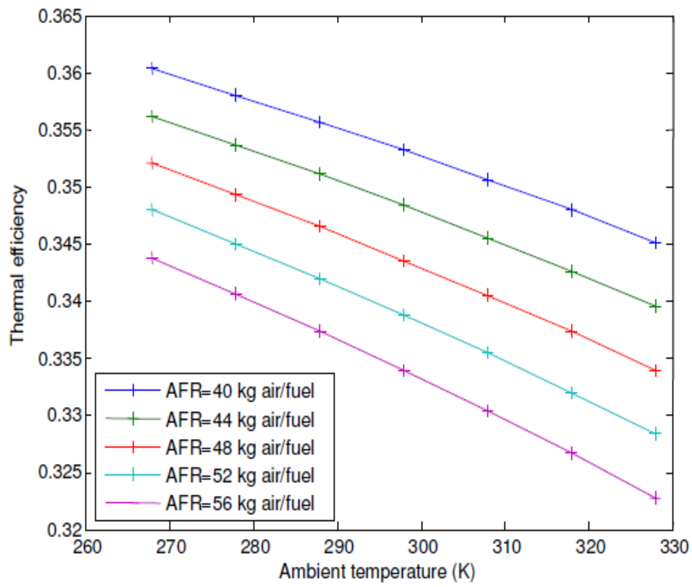



**Figure 4: Effect of ambient temperature and air to fuel ratio on thermal efficiency (Rahman, et al., 2011)**

### 2.1.2 Ambient Pressure

Ambient pressure is a site dependent parameter and it changes with the elevation. With the increases of the elevation, the density of the air reduces, thus ambient pressure reduces. As results of that mass flow rate, fuel rate and the power output of the gas turbine reduce nearly by 3.5% for each 1000 feet (305m) of elevation above the sea level (Petchers, 2002). Figure 5 describes how the power output of the gas turbine changes with the elevation. The Y-axis of the figure 5 represents the ratio between power output at any elevation and power output at reference elevation. This ratio is defined as power output correction factor. The unit is defined as $HorsePower_{any}/HorsePower_{reference}$ (HP/HP).The reference elevation of the curve in the figure 5 is 0 feet (sea level). According to the curve, the output power correction factor for the 0 feet is 1 HP/HP and it reduces with the elevation inclination.

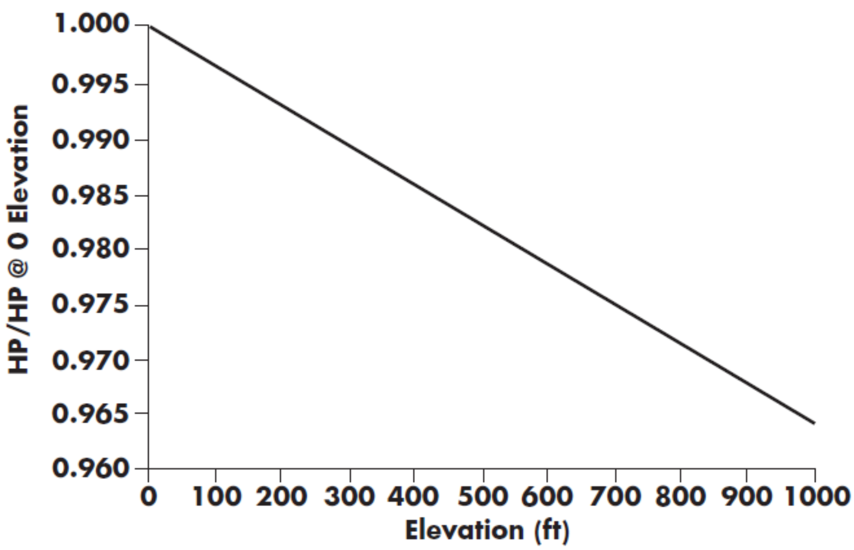



**Figure 5: Representative Altitude Power Correction Factor Curve**

### 2.1.3 Humidity

The atomic mass of the $H_2O$ is less than $N_2$ and $O_2$. Due to that reason mass of the humid air is less than the mass of the dry air (same volume). Therefore the humid air has less density than the dry air. As a result of low density air, the amount of dry air mass entering into the gas turbine reduces. Thus the performance of the gas turbine reduces.

Humid air exists in gas turbines by several means; ambient air is a one mode to does that. Normally the ambient air contains certain amount of moisture. Therefore the humid air directly goes through the gas turbine by means of the ambient air. Water or steam injection to reduce $NO_x$ is another way to increase the humidity level in the gas turbine (Brooks, 2005).

## 2.2 Pressure ratio

The amount of compressibility of the compressor can be considered as the pressure ratio. In similar manner it can be defined as the ratio between the exit pressure and the inlet pressure of the compressor. In an ideal open gas turbine cycle both the compressor inlet pressure and the turbine exit pressure is equal and it is same as ambient pressure. Both the points are in the same isobar line with different enthalpy and entropy values (1 and 4 points in the figure 6). The compressor exit pressure and the turbine inlet pressure are equal and those points are also in the same isobar line with different enthalpy and entropy values (2 and 3 points in the figure 6). However in practically the compressor exit pressure and the turbine inlet pressure are different due to pressure loss in the combustion chamber.

Consider an ideal open gas turbine cycle and the relative TS diagram. Suppose W and Q are the work and heat transfer per unit mass.

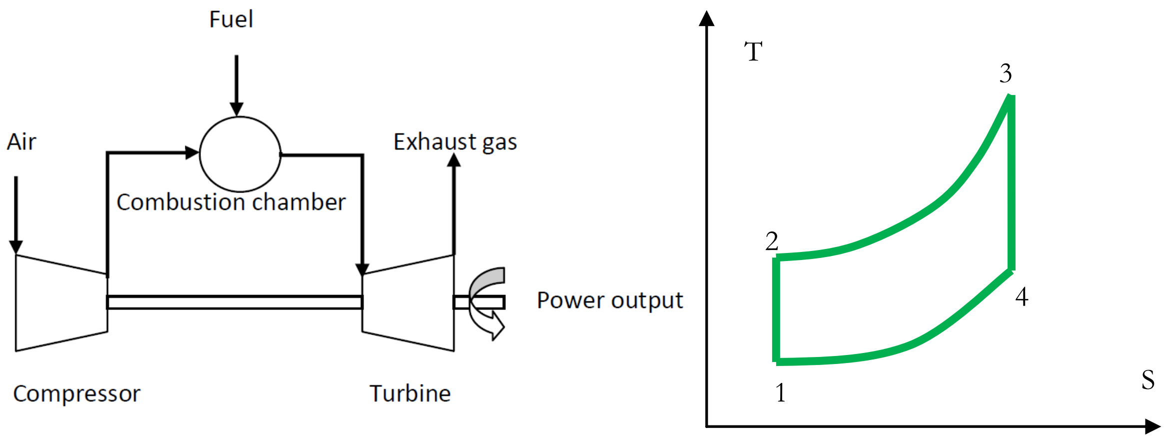



**Figure 6: Open gas turbine and T-S diagram**

By applying the energy balance to the ideal Brayton cycle, following equations can be derived; assume that the $C_p$ is constant for the process.

During the state 1 to 2 the air under goes isentropic compression process in the compressor and work input is given by,

$$W_{12} = -(h_2 - h_1) = -C_p(T_2 - T_1)$$

**Equation 1**

From the state 2 to 3 the compressed air under goes isobaric heat addition in the combustion chamber. This heat addition comes from the fuel.

$$Q_{23} = (h_3 - h_2) = C_p(T_3 - T_2)$$

**Equation 2**

From the state 3 to 4 the combust air goes through isentropic expansion inside the turbine. The work output of the turbine is given by,

$$W_{34} = (h_3 - h_4) = C_p(T_3 - T_4)$$

**Equation 3**

For the complete cycle the net-work output is given by;

$$W_{net} = C_p(T_3 - T_4) - C_p(T_2 - T_1)$$

**Equation 4**

Heat supplied for the cycle is given by;

$$Q_{in} = C_p(T_3 - T_2)$$

**Equation 5**

Hence the efficiency of the open ideal gas turbine cycle can be derived as;

$$\eta = \frac{Net\ work\ output}{Heat\ supplied} = \frac{C_p\ (T_3 - T_4)\ -\ C_p\ (T_2 - T_1)}{C_p\ (T_3 - T_2)}$$

**Equation 6**

From the isentropic temperature and pressure relation;

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}}$$

**Equation 7**

$\gamma$ is the heat capacity ratio.

For the ideal cycle the pressure ratio can be defined as follows;

$$\frac{P_2}{P_1} = \frac{P_3}{P_4} = r$$

**Equation 8**

Thus;

$$\frac{T_2}{T_1} = (r)^{\frac{(\gamma-1)}{\gamma}} = \frac{T_3}{T_4}$$

**Equation 9**

Finally the efficiency can be shown in term of pressure ratio;

$$\eta = 1 - \left(\frac{1}{r}\right)^{\frac{(\gamma-1)}{\gamma}}$$

**Equation 10**

According to the above relation the efficiency of the gas turbine depends only on the pressure ratio and gamma ($\gamma$). The efficiency of the gas turbine increases with the pressure ratio. For the same pressure ratio the efficiency increases with the increase of gamma.

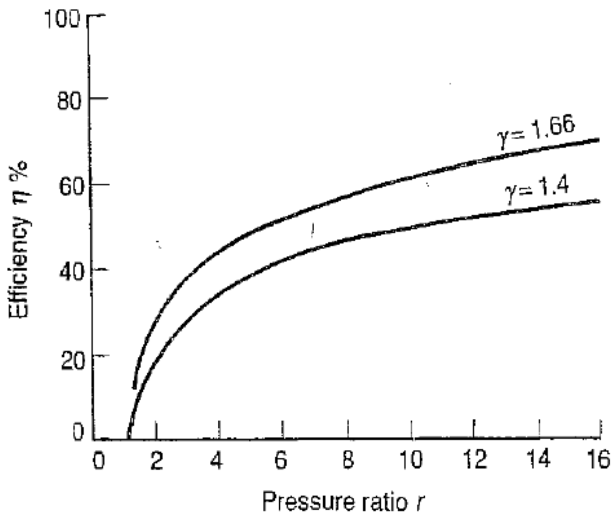



**Figure 7: Relation between pressure ratio and efficiency. (Cohen, et al., 1996)**

According to the figure 7 the efficiency increases with the pressure ratio of the gas turbine but there is a limitation for this explanation. According to the Rahman, et al., 2011, the thermal efficiency increases with the pressure ratio, but after certain value of the pressure ratio the efficiency decreases. In the figure 8 the curve of 1000K turbine inlet temperature starts to decrease around pressure ratio 12 and it reaches "zero" efficiency at pressure ratio 30 and the curve of 1200K turbine inlet temperature starts to decrease around pressure ratio 20. From the same research work it was described that the thermal efficiency decreases, with the increased ambient temperature for the same pressure ratio. For the high ambient temperature, the amount of compressor work is higher that the low ambient temperature due to density change in the air. Therefore the efficiency of the machine reduces in high ambient temperature. Figure 9 shows the variation of efficiency with the pressure ratio for several ambient temperatures.

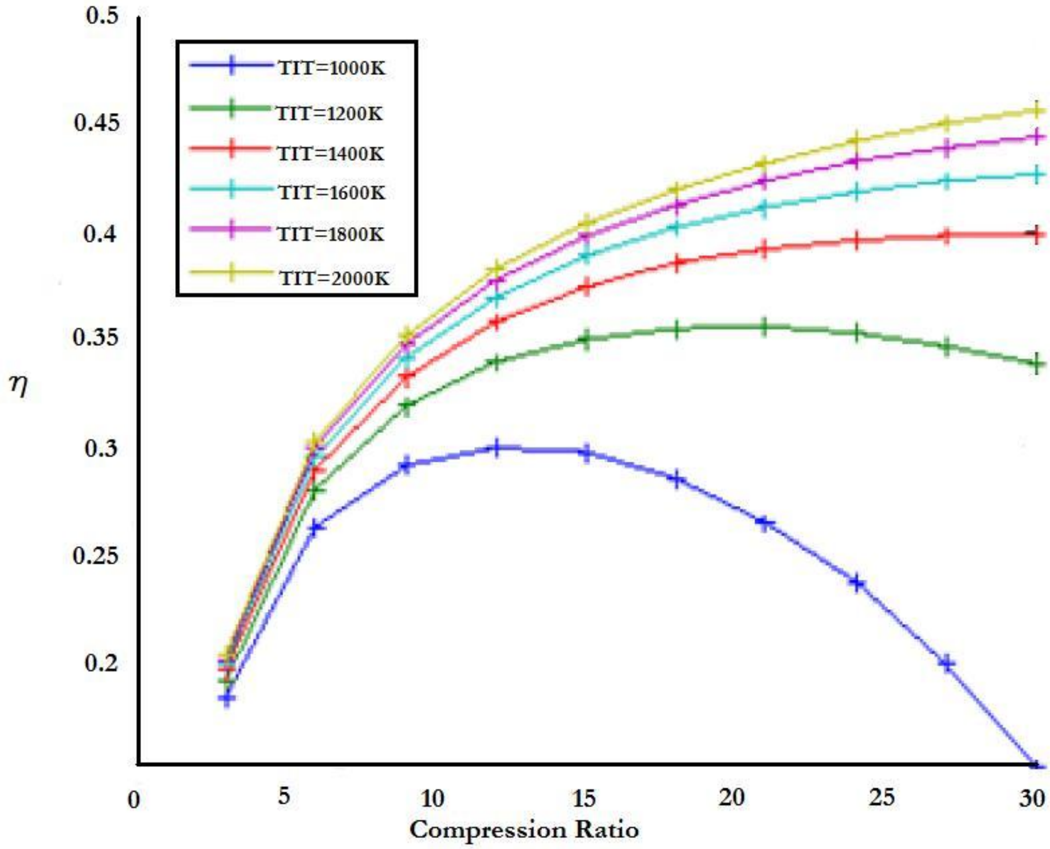



Figure 8: Variation of compression ratio and turbine inlet temperature on thermal efficiency (Rahman, et al., 2011)

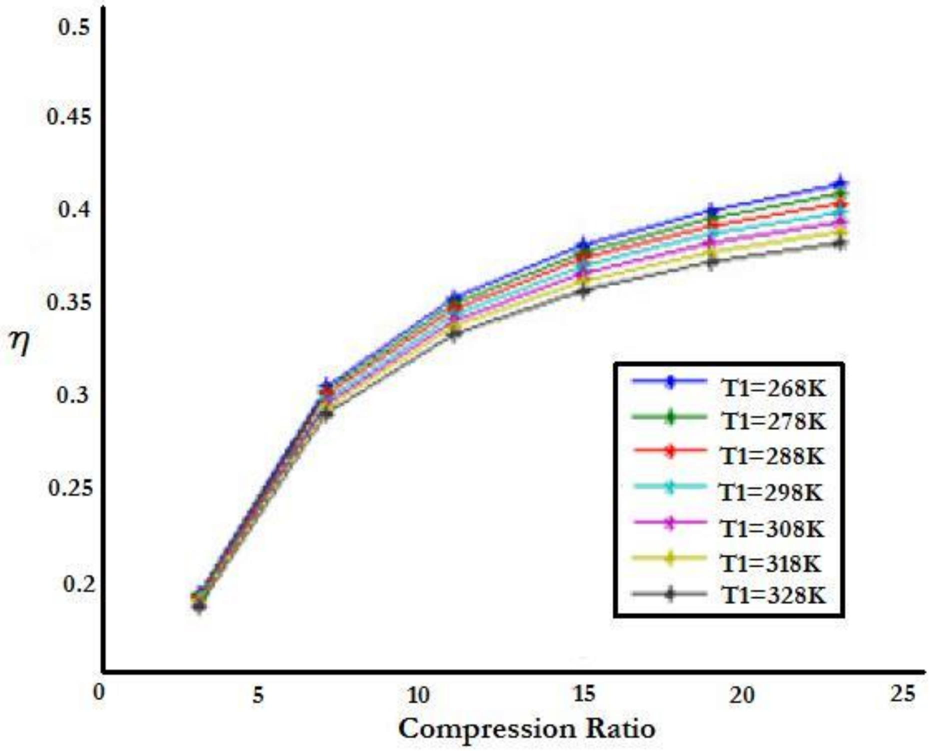



Figure 9: Variation of compression ratio and ambient temperature on thermal efficiency (Rahman, et al., 2011)

## 2.3 Turbine inlet temperature

Turbine inlet temperature (TIT) can be defined as the temperature of the air gas mixture at the inlet of the gas turbine and it is one of the most critical parameter which influences the gas turbine performance. In the turbine work output equation (equation 3) the $T_3$ is denoted as turbine inlet temperature. It is obvious that changes in the $T_3$ influences the turbine work output and consequently it affects to the net-work output. According to the equation 3, the higher TIT produces higher turbine work output and hence high net work output can be obtained from the gas turbine (equation 4). Thus for the better gas turbine performance it is desirable to have higher turbine inlet temperature. Both the power output and the thermal efficiency can be improved by increasing the TIT. Figure 10 shows that when the turbine inlet temperature increases the thermal efficiency of the gas turbine increases. Consider the 56 kg air/fuel curve in the figure 10. The thermal efficiency is around 0.18 at the TIT of 1040K of the pressure ratio 3 (pr=3) line and the thermal efficiency increases with increase of TIT. At the end point of the curve the thermal efficiency reaches around 0.19 at around 1300K of TIT in the same pressure ratio 3 line.

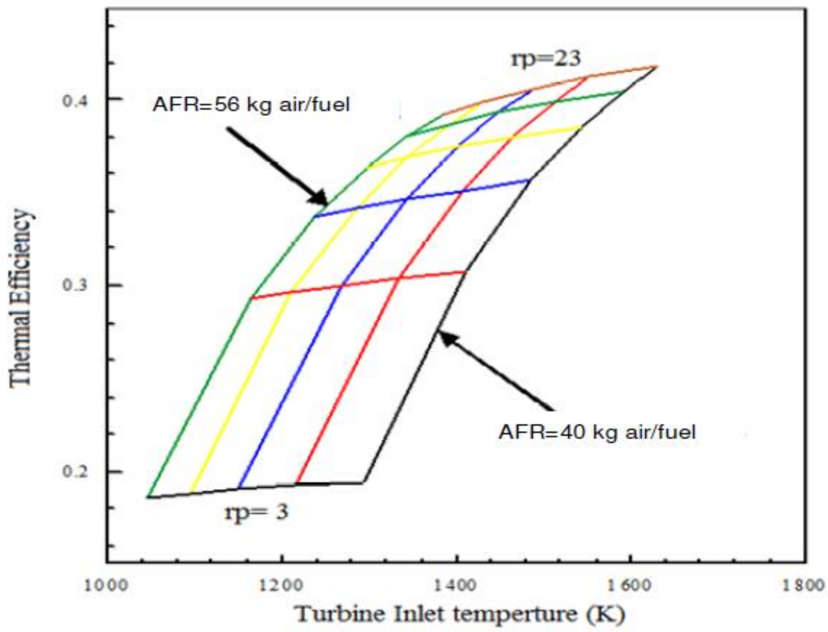



**Figure 10: Variation of turbine inlet temperatures with thermal efficiency for several compression ratio and air to fuel ratio (Rahman, et al., 2011)**

Although increased turbine inlet temperature gives a higher thermal efficiency and a higher power output, there are some practical difficulties for increasing the TIT. The main issue is the material property limitation. The turbine blade elements, casing, hub and combustor elements cannot withstand higher temperature above some thresholds. Therefore the gas path components undergo thermal and mechanical stresses at above threshold temperature, (Petek, et al., 2005). With the development of the gas turbine technology there are some methods to overcome this limitation. Currently available material failure mitigation methods for gas turbine are air, steam or water injection, use of special material such as high performance alloys, use of single-crystal material or use of the thermal barrier coating (Petek, et al., 2005). The other problem that limits the TIT of the gas turbine is the environment regulations. Basically that is due to $NO_x$ emission control. Normally the thermal $NO_x$ is generated in the high temperature environment. With the introduction of water and steam injection, lean premixed combustion, Dry Low Emission (DLE) and catalytic combustion the $NO_x$ emission is controlled successfully. Presently in most modern gas turbines the $NO_x$ emission reduces to single digit ppm value using these methods (Strand, 2005).

## 2.4 Components efficiency

Individual components inside the turbo machine play a major role for power output and the efficiency. Compressor, turbine, combustion chamber, casing and blades can be considered as individual components inside the turbo machine and among them compressor and turbine are the most critical components. By reducing the internal component losses of individual component, the overall performance of the gas turbine can be uplifted (Petek, et al., 2005).

Both the compressor and the turbine individual isentropic efficiencies are proportional to the overall gas turbine thermal efficiency (Rahman, et al., 2011)and figure 11 and figure 12 show the relationship between individual isentropic efficiencies and the overall thermal efficiency of the gas turbine. In both cases the thermal efficiency of the gas turbine increases with the individual component efficiencies.

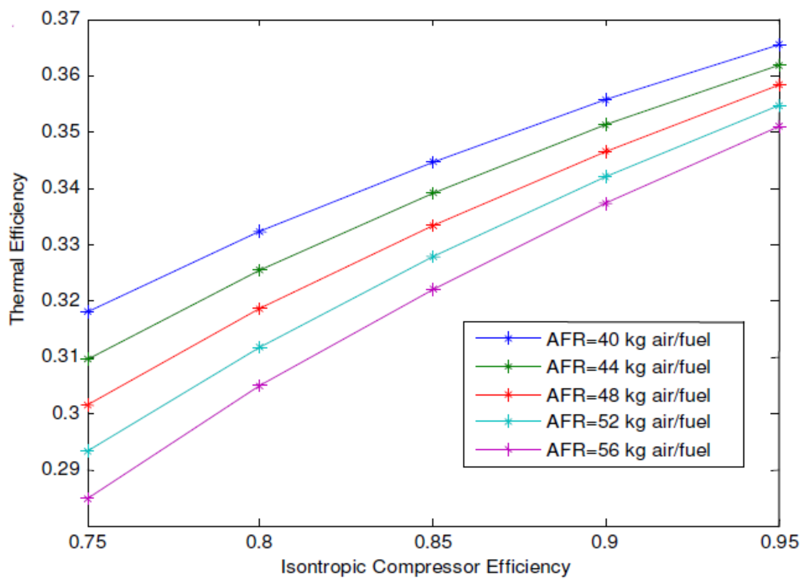



**Figure 11: Effect of isentropic compressor efficiency air to fuel ratio on thermal efficiency. (Rahman, et al., 2011)**

The difference between the turbine work and compressor work can be defined as net work output of the gas turbine. Thus the net work output has direct influence from the compressor and turbine work and their efficiency. Therefore the individual components efficiencies directly influence the gas turbine efficiency.

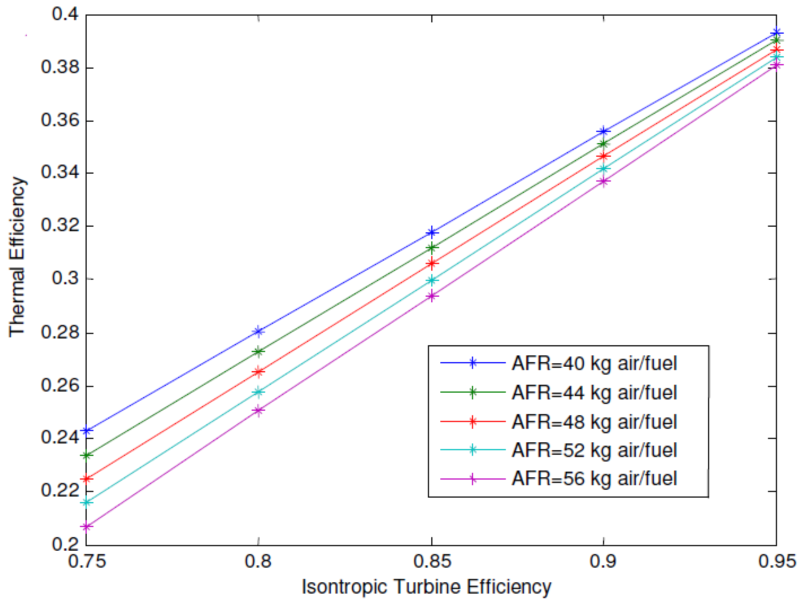



**Figure 12: Effect of isentropic turbine efficiency air to fuel ratio on thermal efficiency.** **(Rahman, et al., 2011)**

According to literature there are different types of efficiencies for the individual components such as isentropic efficiency total to total, isentropic efficiency total to static, and the polytropic efficiency. By means of the fundamental gas turbine equations, the different types of individual component efficiencies and their influence on the gas turbine performances can be measured. Before going to discuss about types of efficiencies it is beneficial to get familiarized with some fundamental thermodynamic principles related to compressible fluids.

### 2.4.1 Stagnation or total properties

Consider a gas flow with velocity 'C', temperature 'T' and enthalpy 'h'. The gas flow has brought to rest adiabatically without any work transfer, and the new enthalpy '$h_0$' is called stagnation (total) enthalpy and the new temperature '$T_0$' is called stagnation (total) temperature. By applying the steady state flow energy equation for the two states, the equation 11 can be obtained.

$$h + \frac{1}{2}C^2 = h_0 + 0$$

**Equation 11**

$$h_0 = h + \frac{1}{2}C^2$$

**Equation 12**

For the perfect gas,

$$h = C_p\,T$$

**Equation 13**

Substitute equation 13 to equation 12 the following relationship can be obtained.

$$T_0 = T + \frac{C^2}{2C_p}$$

**Equation 14**

## 2.4.2 Isentropic efficiency - Total to total

An isentropic process is a constant entropy process and it can be proved that any adiabatic and reversible process is an isentropic process. An adiabatic process is one in which no heat is transferred to or from the fluid during the process while in the reversible process, both the fluid and its surroundings can always be restored to their original state (Eastop, et al., 2002), but in reality it is hard to find any reversible process due to losses in the fluid flow path.

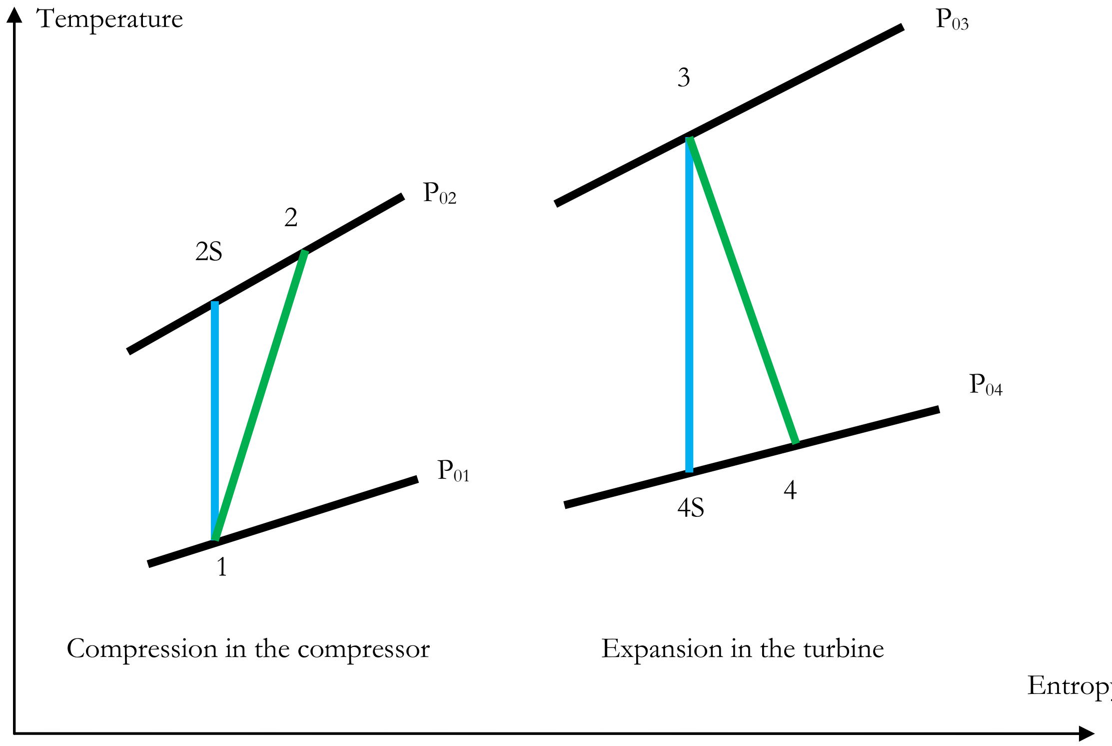



**Figure 13: T-S diagram for compression and expansion**

Figure 13 represents T-S diagram for the compression and the expansion of the gas turbine. The process 1 to 2S in figure 13 represents isentropic compression while process 1 to 2 represents an actual compression. From the enthalpy difference of the two cases, it is clearly indicated that the amount of energy needed to move a fluid particle from one pressure to another pressure is higher than that of ideal case. (i.e. isentropic case). For the expansion case in figure 13 the process 3 to 4S indicates the isentropic expansion while the process 3 to 4 represents the actual expansion. In the expansion process the amount of work that can be recovered during the expansion is less than the available amount of work (from the enthalpy difference). In order to get the better understanding of the performance of the individual components and their behavior inside the gas turbine the equation 15 is defined as following for compressor.

$$\boldsymbol{Isentropic\ efficiency_{compressor}} = \frac{\boldsymbol{Ideal\ change\ in\ energy}}{\boldsymbol{Actual\ change\ in\ energy}}$$

**Equation 15**

$$\eta_{is,c} = \frac{h_{02S} - h_{01}}{h_{02} - h_{01}}$$

**Equation 16**

On assuming constant mean $C_p$ throughout the flow, equation 16 can take following form by substituting the equation 13:

$$\eta_{is,c} = \frac{T_{02S} - T_{01}}{T_{02} - T_{01}}$$

**Equation 17**

$$T_{02} - T_{01} = \frac{1}{\eta_{is,c}}(T_{02S} - T_{01})$$

**Equation 18**

$$T_{02} - T_{01} = \frac{T_{01}}{\eta_{is,c}}\left(\frac{T_{02S}}{T_{01}} - 1\right)$$

**Equation 19**

$$T_{02} - T_{01} = \frac{T_{01}}{\eta_{is,c}}\left[\left(\frac{P_{02}}{P_{01}}\right)^{\frac{(\gamma-1)}{\gamma}} - 1\right]$$

**Equation 20**

$$\eta_{is,c} = \frac{T_{01}}{(T_{02} - T_{01})}\left[\left(\frac{P_{02}}{P_{01}}\right)^{\frac{(\gamma-1)}{\gamma}} - 1\right]$$

**Equation 21**

Similarly,

$$Isentropic\ efficiency_{turbine} = \frac{Actual\ change\ in\ energy}{Ideal\ change\ in\ energy}$$

**Equation 22**

$$\eta_{is,t} = \frac{h_{03} - h_{04S}}{h_{03} - h_{04}}$$

**Equation 23**

$$T_{03} - T_{04} = \eta_{is,t}T_{03}\left[1 - \left(\frac{1}{P_{03}/P_{04}}\right)^{\frac{(\gamma-1)}{\gamma}}\right]$$

**Equation 24**

$$\eta_{is,t} = \frac{T_{03} - T_{04}}{T_{03}\left[1 - \left(\frac{1}{P_{03}/P_{04}}\right)^{\frac{(\gamma-1)}{\gamma}}\right]}$$

**Equation 25**

### 2.4.3 Polytropic or small stage efficiency

According to the equation 21 and equation 25, the isentropic compressor efficiency and the isentropic turbine efficiency change with the pressure ratio. It is found that isentropic compressor efficiency tends to decrease and isentropic turbine efficiency tends to increase as the pressure ratio increases (Cohen, et al., 1996). Therefore it is not reasonable to define the efficiency by only considering the highest and the lowest pressures in the gas turbine. As a result of that another reasonable definition is needed when describing the efficiencies of the turbomachinary components in the gas turbine.

Apart from the two main pressure levels in the working of turbomachinary it consists of infinite numbers of small pressure levels. The pressure ratios in between those small pressure levels differ from each other. Consider the compression process in between constant pressures $P_{01}$ and $P_{02}$ in the figure 14. In between

$P_{01}$ and $P_{02}$ there are infinite numbers of constant pressure levels and they can be called as $P_{0x1}$, $P_{0x2}$, $P_{0x3}$… $P_{0xn}$. By adding small stage isentropic work outputs in between the $P_{0x1}$, $P_{0x2}$, $P_{0x3}$… $P_{0xn}$, the total flow work can be obtained;

$$\boldsymbol{Total\ isentropic\ flow\ work} = (h_{02Sx1} - h_{01}) + (h_{02Sx2} - h_{02x1}) + \cdots + (h_{02S} - h_{02xn})$$

**Equation 26**

Finally the polytropic efficiency for the compression is defined as follows;

$$\eta_{P,c} = \frac{\boldsymbol{Total\ isentropic\ flow\ work}}{\boldsymbol{Actual\ change\ in\ energy}}$$

**Equation 27**

$$\eta_{P,c} = \frac{(h_{02Sx1} - h_{01}) + (h_{02Sx2} - h_{02x1}) + \cdots + (h_{02S} - h_{02xn})}{h_{02} - h_{01}}$$

**Equation 28**

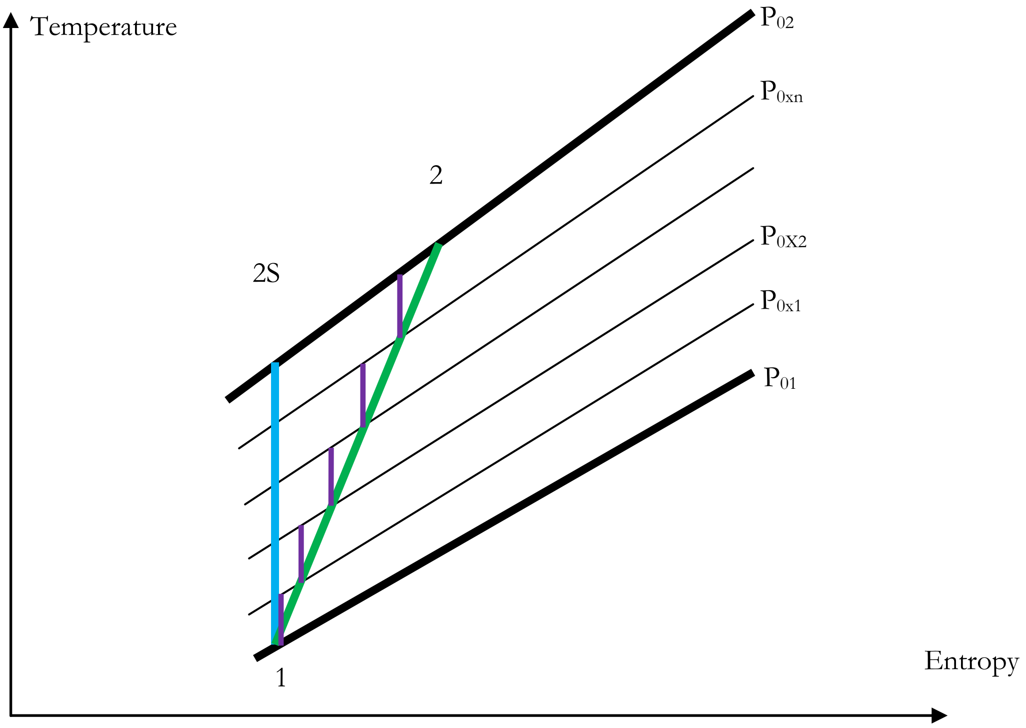



**Figure 14: T-S diagram for compression process**

From the fundamental thermodynamic relation for the entropy, equation 29 can be obtained;

$$Tds = dh - vdp$$

**Equation 29**

From partial differentiation of the equation 29 (for constant pressure) the equation 30 can be obtained.

$$(\partial h/\partial s)_p = T$$

**Equation 30**

According to the above relation $(\partial h/\partial s)_p$ represents the slope of the isobar line in h-S diagram and this slope is proportional to the temperature (Vogt, 2007). Therefore the slope increases with the temperature rise. Thus the equation 31 can be obtained.

$$(h_{02Sx1} - h_{01}) + (h_{02Sx2} - h_{02x1}) + \cdots + (h_{02S} - h_{02xn}) > h_{02S} - h_{01}$$

Equation 31

$$\frac{(h_{02Sx1} - h_{01}) + (h_{02Sx2} - h_{02x1}) + \cdots + (h_{02S} - h_{02xn})}{h_{02} - h_{01}} > \frac{h_{02S} - h_{01}}{h_{02} - h_{01}}$$

Equation 32

$$\eta_{P,c} > \eta_{is,c}$$

Equation 33

Similarly for the polytropic efficiency for the turbine,

$$\eta_{P,t} < \eta_{is,t}$$

Equation 34

## 2.5 Conclusion for the literature review

All the parameters discussed under the literature review directly influence the gas turbine performances. In order to carry out a thermodynamic performance analysis for a gas turbine the parameters discussed above should be taken into account.

It is hard to find literature on gas turbine performance analysis using the manufacturers' published data. The only litrerature found on the subject is by Wettstein, 2007 and the CompEdu chapter (Wettstein, 2006). This available work is previousely used at KTH for teaching purpose. The gas turbine performance in the available work is evaluated using a Mathcad model. This Mathcad model was used in class room lectures and the feedback from the students are also available. The major findings from the students feedback (Wettstein, 2007) are given below.

- Students found it difficult to understand the notations of the program
- Need additional explanation about the program

During this MSc reaserch above mentioned available work is taken as the reference work and expected to analyze it. Subsequently it is expected to find the drawbacks of the available work. Finally new suggestions which are needed to improve it are to be given for this existing work. When proposing the new suggestions, the majar findings from the student feedbacks of available work are also to be considered.

# 3 Methodology

As mentioned in the conclusion for the literature review, there is an available work which was previously used by the KTH for teaching purpose, but students found it difficult to understand the notations of the program and they needed additional explanations for it. In order to cater the demands of the students, the available work should be analyzed first and the drawbacks of it should be identified. Therefore as an initial stage for this research, the available work is analyzed. During the analysis the drawbacks of the available work are to be found. Subsequently new suggestions for the available work are to be given. By means of the new suggestions it is expected to optimize and improve the available work. Finally the software model is to be improved by considering the suggestions and the results of the new work are to be validated.

## 3.1 Analysis the available work

There are two parts in the available work; those are software model and the CompEdu chapter. The software was modeled by using Mathcad. Under the section 3.1.1 the available Mathcad model is to be explained. The available CompEdu chapter is a user guide for the Mathcad model and it is expected to adjust soon after the modification of the software model.

### 3.1.1 The Mathcad program

The main objective of this program is to analyze the thermodynamic properties of a gas turbine by using commercially available catalog data. The program itself has several sub sections and each sub section has dedicated function for the main task. In order to understand the program sequence and the algorithm each sub sections is studied deeply and detailed descriptions for all sub sections are given below.

#### *3.1.1.1 Preparation of units and the gas data functions*

In the initial stage of this section some constants and units were defined in the Mathcad program. Those defined values are shown in the table 1.

| Unit 01 | Unit 02 |
|---|---|
| 1kJ | 1000 joule |
| 1MJ | 1000 kJ |
| 1MW | 1000 kW |
| 1bar | $10^5$ Pa |
| s | sec |
| 1MWh | 3600000kJ |
| kWh | kWhr |
| CK | 273.15 |
| 1 J | 1 joule |

**Table 1: Basic unit definition**

After defining the units and constant, program begins with determining the water ($H_2O$) content of the inlet air. In order to find the water content of the inlet air the program makes 3 assumptions. Those 3 assumptions are valid for the entire calculation.

- Temperature of the inlet air=15$^0$C
- Relative humidity of the inlet air=60%
- Pressure of the inlet air=1.013bar

After stating the above assumptions a function from the LibHuAir library uses for finding the water content of the incoming air for the gas turbine. The Mathcad itself does not have functions related to humid air properties calculations; therefore LibHuAir (an external library for the Mathcad software) is used. The LibHuAir consists of functions which are related to humid air. Such external libraries are added in the Mathcad to execute the calculation. Inside the LibHuAir library there is a function for the relative humidity and from that function the water content of the incoming air can be calculated. For the above assumed ambient conditions the calculated water content value is 0.00635083 g/kg$_{Air}$. Therefore during the entire calculation the water content of the incoming air is 0.00635083 g/kg$_{Air}$, because the calculation always uses fixed ambient condition. For the whole calculation the pressure, temperature and relative humidity are kept at 1.013bar, 15$^0$ C and 60% respectively. After calculating the water content of the inlet air, the Mathcad program calculates the gas mixture properties. For those gas properties calculation the program uses another external library for Mathcad. The name of the library is LibHuGas. Inside the LibHuGas library there are functions related to the humid gas mixtures. There are two types of functions inside the LibHuGas and they are chemical composition of air based on mole fractions functions and chemical composition of air based on mass fraction functions. Inside those functions the chemical composition of the air is defined in 8 most common chemical components and those chemical components are Argon, Neon, Nitrogen, Oxygen, Carbon monoxide, Carbon dioxide, Water and Sulfur dioxide. Those compositions should be entered into the program. The program uses functions from the LibHuGas for calculating the constant pressure heat capacity of the gas mixture, gas constant of the gas mixture and specific enthalpy of the gas mixture.

After finding the gas mixture properties, the properties related to pure methane and water are calculated using the program. Pure methane is the only fuel used in this program. Those property calculations use some general equations extracted from Wylen, et al., 1985. The constant pressure heat capacity of methane and water/steam are calculated from equation 36 and equation 37 where $\theta$ defined as

$\theta = T(Kelvin)/100$ (Wylen, et al., 1985)

$$C_{p(CH_4)} = \frac{-672.87+439.74\theta^{0.25} -24.875\theta^{0.75} +323.88\theta^{-0.5}}{16.04}$$

**Equation 35**

$$C_{p(H_2O)} = \frac{143.05-183.54\theta^{0.25} +82.751\theta^{0.5} -3.6989\theta}{18.02}$$

**Equation 36**

### *3.1.1.2 Input section 01*

As a first step to run the program the user must enter input parameters for the program. Once the parameters are set they will remain constant throughout the program. Some brief descriptions about the input parameters are given below. The variable name used in the Mathcad program for each parameter is indicated in within brackets.

**Relative inlet pressure loss until blading (Dpi)**

Pressure loss responsible for the ambient to until the compressor blading can be considered as Dpi and typically this value is in between 500 Pa to 1500 Pa (0.0049-0.014 with relative to ambient pressure ) for compressors with standard filters (Wilcox, et al., 2012).

**Relative outlet pressure loss (Blading to ambient) (Dpo)**

This value is responsible for pressure loss from turbine blading to the ambient.

**Relative internal pressure loss (Dpbk)**

The relative pressure loss incorporated from the compressor outlet up to the turbine inlet can be considered as Dpbk.

**Compressor relative diffuser loss (Related to plenum pressure) (Dpcd)**

This is the relative diffuser pressure loss of the compressor.

**$H_2O$ content of air at $15^0$ C with 60% humidity related to dry air ($xH_20$)**

This value is calculated in the previous section and it is equal to 0.00635083 $g/kg_{Air}$. This value is constant throughout the program.

**Ambient pressure (pu)**

This is the ambient pressure for the gas turbine unit.

**Ambient temperature (Tu)**

This is the ambient temperature for the gas turbine unit.

**Lower heating value of the fuel (pure methane) (HU)**

For the Mathcad model the pure methane is considered as the fuel for the gas turbine. Pure methane is the only fuel used in this program. Therefore the lower heating values of the pure methane should enter for the program.

**Fuel inlet temperature for the gas turbine (TGv)**

This is the inlet temperature of the incoming fuel and most of the cases this value is equal to the ambient temperature.

**Electrical efficiency ($\eta_e$)**

As an input variable the electrical efficiency of the electricity generator uses in the calculation. The realistic value for the electrical efficiency can be entering in the input section.

**Mechanical efficiency ($\eta_m$)**

This is the mechanical efficiency of the coupling. If the electric generator is coupled with the gas turbine directly, then the $\eta_m$ is equal to 1, because it doesn't loss any energy in the coupling.

#### 3.1.1.3 Input section 02

This section is the main input area for the program. As discussed in the problem statement there are 5 main parameters which can be found in the commercially available gas turbine catalogue. Those can be considered as catalog data for the gas turbine and those 5 parameters are mentioned in below.

- Pressure ratio (Ambient to compressor exit plenum)
- Base load power at generator terminals
- Exhaust mass flow rate
- Efficiency of the gas turbine
- Turbine exit gas temperature

Apart from the above values the compressor polytropic blading efficiency is needed additionally, for the calculation. The user of the software has to assume the compressor polytropic blading efficiency value and put in to input section 02. The equation 37 shows the relationship between compressor polytropic blading efficiency, pressure ratio and the compressor exit temperature. In the calculation core section this equation 37 is used for calculating the compressor exit temperature.

#### 3.1.1.4 Thermodynamic functions

Eight thermodynamic formulae which are related to gas turbine technology are considered in this section. All the formulae are written in Mathcad notations and most of them are in integral format. In the section 3.3.1 detailed descriptions about all thermodynamic functions are available.

#### 3.1.1.5 Calculation core section

The thermodynamic function section, which was previously described in the section 3.1.1.4 is only used for defining the thermodynamic functions, by the program. In other words it only stored functions which are needed for the gas turbine calculation. The purpose of introducing the calculation core section for the program body is to use the previously defined thermodynamic functions, according to the appropriate program steps when necessary. In the Mathcad program, using the previously defined functions is not a difficult task. After calling the appropriate functions in the program the set of output values can be generated from the models.

### 3.1.1.6 Results section

As an example the gas turbine 'SGT5-4000F (V94.3A)' was taken for testing the Mathcad model. The catalog data for the selected gas turbine are given below (Biasi, 2006).

| | | |
|---|---|---|
| 1. | Power at terminals (MW) | 286.60 |
| 2. | GT efficiency terminals | 39.50% |
| 3. | Outlet mass flow (kg/s) | 703.974 |
| 4. | Exhaust temperature (°C) | 577.2 |
| 5. | Pressure Ratio | 17.90 |

Apart from the above catalog data the compressor polytropic blading efficiency, the realistic pressure drop values, the mechanical efficiency and the electrical efficiency of the gas turbine used as inputs.

| | | |
|---|---|---|
| 1. | Polytropic compressor blading efficiency | 91.80% |
| 2. | Relative inlet pressure loss | 0.40% |
| 3. | Relative outlet pressure loss | 1.30% |
| 4. | Pressure loss between compressor and turbine blading | 6.00% |
| 5. | Mechanical efficiency | 100.0% |
| 6. | Electrical efficiency | 98.5% |

After executing the calculation command from the Mathcad software the calculation starts. It spends several seconds to complete the calculation. After completion of the calculation, results are transferred to the excel table. The excel table is placed at the bottom of the Mathcad program body. The table 2 shows the typical excel answer table of the software.

In the table 2 the row numbers 02 to 06 represent the used catalog data of the gas turbine calculation and those can be considered as input values for the Mathcad model. The row number 10 represents the assumed compressor polytropic blading efficiency. The row numbers 20 to 24 represent the realistic pressure loss valves and the efficiency values related to the gas turbine. Apart from the input rows in the table 2, other rows (7 – 9, 11 – 19, 25, 26) are used for the output values.

| | Data Source | GTW 2006 Handbook |
|---|---|---|
| 01 | Gas turbine Type | SGT5-4000F (V94.3A), simple cycle, reference |
| 02 | Power at terminals (MW) | 286.60 |
| 03 | GT efficiency terminals | 39.50% |
| 04 | Outlet mass flow (kg/s) | 703.974 |
| 05 | Exhaust temperature (°C) | 577.2 |
| 06 | Pressure Ratio | 17.90 |
| 07 | Compressor blading exit total temperature (°C) | 422.0 |
| 08 | Mixed combustor temperature (°C) "MCT" | 1235.2 |
| 09 | Mixed turbine inlet temp. (°C) "MTT" | 1235.2 |
| 10 | Polytropic compressor blading efficiency | 91.80% |
| 11 | Polytropic turbine blading efficiency. | 86.75% |
| 12 | Relative heat extraction | 0.00% |
| 13 | Fuel mass flow rate (kg/s) | 14.511 |
| 14 | Compressor inlet humid air mass flow rate (kg/s) | 689.462 |
| 15 | Fuel heat flow (MW) | 725.57 |
| 16 | Dry air inlet mass flow (kg/s) | 685.11 |
| 17 | Mixed isentropic compressor blading efficiency | 88.14% |
| 18 | Mixed isentropic turbine blading efficiency | 90.07% |
| 19 | Equivalent heat extraction temperature drop (°C) | 0.01 |
| 20 | Relative inlet pressure loss | 0.40% |
| 21 | Relative outlet pressure loss | 1.30% |
| 22 | Pressure loss between compressor and turbine blading | 6.00% |
| 23 | Mechanical efficiency "etam" | 100.0% |
| 24 | Electric efficiency "etae" | 98.5% |
| 25 | Total heat extraction (MW) | 0.01 |
| 26 | Relative heat balance mismatch | 8.1E-07 |

**Table 2: Results table of the Mathcad program (Wettstein, 2006)**

## *3.2* Optimization Improvement and Suggestion

After completion of this research, the computer model and the CompEdu chapter are expected to be used in technical and pedagogical tasks for a turbomachinery course at KTH. Therefore both the computer model and the CompEdu chapter should be improved for better understanding for students who follow the turbomachinary course. It is very much important to get a better understanding of the existing model in order to improve it. Due to that reason a study of the Mathcad software model was carried out. The major findings of the study are given below.

**To understand that the existing program is not much easy**

Although the existing program gives the thermodynamic analysis of the set of catalog data for a commercial gas turbine, understanding the core calculation is not that easy. By changing the input parameters of the program the user can get the set of output data related to the particular gas turbine. For that process the core calculations and the thermodynamic functions remain as a black box to the new user. If the user does not have any knowledge on the Mathcad programming, the situation goes bad to worse.

Especially in the air and gas data preparation section the program has introduced some functions from the external libraries LibHuAir and LibHuGas. The humid air and humid gas mixture properties can be easily calculated by using the mentioned libraries. The functions related to the LibHuAir and LibHuGas are directly used in the program. Those functions also confuse the user to some extent. In order to understand the Mathcad functions and the external library functions Mathcad help, Mathcad user's guide (Corporation, 2010), LibHuAir/LibHuGas help (Kretzschmar, et al.) need to be studied.

The Mathcad model has second option for non- LibHuAir/LibHuGas users also. Typically the program extracts constant pressure heat capacity (Cp) for the dry air from the LibHuAir/LibHuGas libraries. But from the non-LibHuAir/LibHuGas option the user can do the calculation without the libraries. Some previously extracted Cp values for the temperature range 200K to 2150K fed into the program via an interpolated function. As a result of that, both the LibHuAir/LibHuGas option and the non-LibHuAir/LibHuGas option available in the same program body. Therefore the person who studies the programming code gets confused because of this LibHuAir/LibHuGas and non- LibHuAir/LibHuGas options.

**Comments in the programming body are not adequate**

During the study of the existing program it was found that the comments in the program body are not adequate in order to understand the program. Especially for the gas data preparation section comments are essentially needed. For example the air composition vector used in the gas data preparation section need more comments in order to understand it. Without any indications or comments the air composition vector is placed inside the program code. The air composition vector is not a Mathcad function, and it is a function from LibHuAir/LibHuGas external libraries. Therefore inside the Mathcad help also it is impossible to find about this air composition vector. The LibHuAir/LibHuGas help files have to be used in order to get the real understanding of this air composition vector. If there are comments in the program body regarding the air composition vector, the understanding about the vector is much easier.

**The software model is not well organized**

Although the software model have all the necessary steps inside the body those steps are not clear to the new user. Some of programming steps in the gas data preparation section functions just hang around here and there. Due to that reason the software model programming steps are not neat inside the programming body. Then the user has to spend more and more time to understand the program.

**Suggestions**

**Suggestions in order to overcome the shortcomings are mentioned as follows**

1. Re arrange the programming body in an understandable manner. Neatness of the program should also be improved.
2. Put more comments to the programming body.
3. Introduce a graphical user interface to make the program more user-friendly.

## 3.3 New works

In order to get the above three requirements, two numbers of options have been identified. The first option is to edit the available model. Second option is to rewrite a new model. The first option is somewhat difficult due to some reasons.

1. Editing someone's code is a difficult task.
2. Mathcad does not have any Graphical User Interface (GUI) facilities.

Therefore the second option had been selected and the new model has been written by using the Engineering Equation Solver (EES). EES is an equation solving program which can numerically solve thousands of equations (fchart, 2012). EES also has the high accuracy thermodynamic and transport properties for hundreds of substances inside its data base (fchart, 2012). This is a major advantage over Mathcad. The previous program was used in some external libraries (LibHuAir and LibHuGas) and extracts the thermodynamic properties of dry air. Those external libraries are not freely available and the users have to buy those library files for Mathcad.

By means of the EES software the gas turbine was modeled with two limitations. Both the limitations are also in the Mathcad model and those are,

1. The modeled gas turbine is an open gas turbine with a one combustor
2. The model only limited to Methane as the fuel. (Interested user can change the fuel type, by changing the EES program carefully)

### 3.3.1 The Engineering Equation Solver (EES) program

The open gas turbine calculation in the model was done by using fundamental governing equations. The program sequence and the thermodynamic functions are same as Mathcad model.

**Calculation of water content in the incoming air**

Normally for open gas turbine systems the compressor sucks atmospheric air which is not 100% dry. In other words, the incoming air consists of two components, which can be defined as dry air and water. Therefore in the initial stage of the calculation the water content of the incoming air was calculated. Normally the water content of the air depends on main three parameters and they are air temperature, air pressure and relative humidity. By knowing the mentioned three parameter one can easily calculate the water content of the incoming air by simply calling the appropriate function in the EES.

During the calculation the user can use any ambient conditions for the calculation. The changing facility for the ambient condition is available in the GUI of the model.

**Calculation of the compressor exit temperature**

The exit temperature of the compressor is very important parameter. In order to calculate this parameter the polytropic efficiency of the compressor should be assumed by the user. The same approach is available in the Mathcad model also. The equation 37 gives the polytropic efficiency of the compressor (Wettstein, 2007).

$$\eta_C = \frac{ln(P_{ratio})}{\int_{T_1}^{T_2}\left[\frac{C_{p(Air)}.T+X_{H2O}.C_{p(H2O)}.T}{(R_{Air}+X_{H2O}.R_{H2O}).T}\right]dT}$$

**Equation 37**

The variables in the equation 37 can be defined as follows,

- $\eta_c$ = Polytropic efficiency of the compressor
- $T_1$ = Inlet air temperature (K)
- $T_2$ = Exit air temperature (K)
- $P_{ratio}$ = Pressure ratio between compressor blading region
- $X_{H2O}$ = Water content of the inlet air ($kg_{H2O}/kg_{Air}$)
- $C_{p(Air)}$ = Constant pressure specific heat capacity of dry air (kJ/kg)
- $C_{p(H20)}$ = Constant pressure specific heat capacity of water (kJ/kg)
- $R_{H2O}$ = Gas constant of water (0.4615 kJ/kg.K)
- $R_{Air}$ = Gas constant of dry air (0.2881 kJ/kg.K)
- T = Temperature (Variable for the integration)

By solving the equation 37 the unknown $T_2$ can be found.

**Compressor specific power**

After calculating the compressor exit temperature the specific power needed to run the compressor can be calculated from the equation 38. This is an extracted version of the equation 13 and from the equation 38 the specific enthalpy difference between the compressor inlet to outlet can be calculated for the unit mass flow rate of humid air. This specific enthalpy difference is same as the compressor specific power.

$$Compressor\ specific\ power = \int_{T_1}^{T_2}\left(C_{p(Air)}.T + X_{H2O}.C_{p(H2O)}.T\right)dT$$

**Equation 38**

**Pressure ratio correction**

The open gas turbine cycle experiences several types of pressure losses inside it. They are,

- Inlet pressure loss
- Outlet pressure loss
- Internal pressure loss
- Compressor diffuser pressure loss

The pressure drop values are fed to the model before the calculation is started. Most of the time commercially available catalog indicates the pressure ratio of the ambient to compressor plenum. So the commercial pressure ratio values should be corrected as per the equations used in the software. Basically there are two equations which incorporate pressure ratio. Those equations are compressor polytropic blading efficiency (Equation 37) and turbine polytropic blading efficiency (Equation 42). Inside the mentioned equations (37 & 42) the pressure ratio is defined only for the blading region of that turbomachine. Therefore the commercial pressure ratio is subjected to correction inside the software program. The pressure ratio correction calculation is given below.

$$\boldsymbol{P_{CB1} = P_1 . (1 - P_{inlet\ pressure\ loss})}$$

**Equation 39**

$$\boldsymbol{P_{CB2} = P_1 . P_{ratio\ catalog} . \left(1 + P_{diffuser\ pressure\ loss}\right)}$$

**Equation 40**

$$\boldsymbol{P_{ratio\ model} = \frac{P_{CB2}}{P_{CB1}}}$$

**Equation 41**

In the equation 39 the $P_{CB1}$ denotes the pressure at the compressor blading entrance, while $P_{CB2}$ denotes the pressure at the compressor blading exit in the equation 40. $P_1$ denotes the ambient pressure and $P_{CB1}$ is calculated by using the inlet pressure loss of the compressor in the equation 39. In equation 40 the $P_{ratio\ catalog}$ is the pressure ratio adopted from the catalog data and from the equation 40 the $P_{CB2}$ calculated by considering the diffuser pressure loss of the compressor. After calculating the $P_{CB1}$ and $P_{CB2}$ corrected pressure ratio can be calculated from the equation 41 and the corrected pressure ratio is denoted as $P_{ratiomodel}$. Figure 15 represents the pressure losses and pressure ratios inside the compressor.

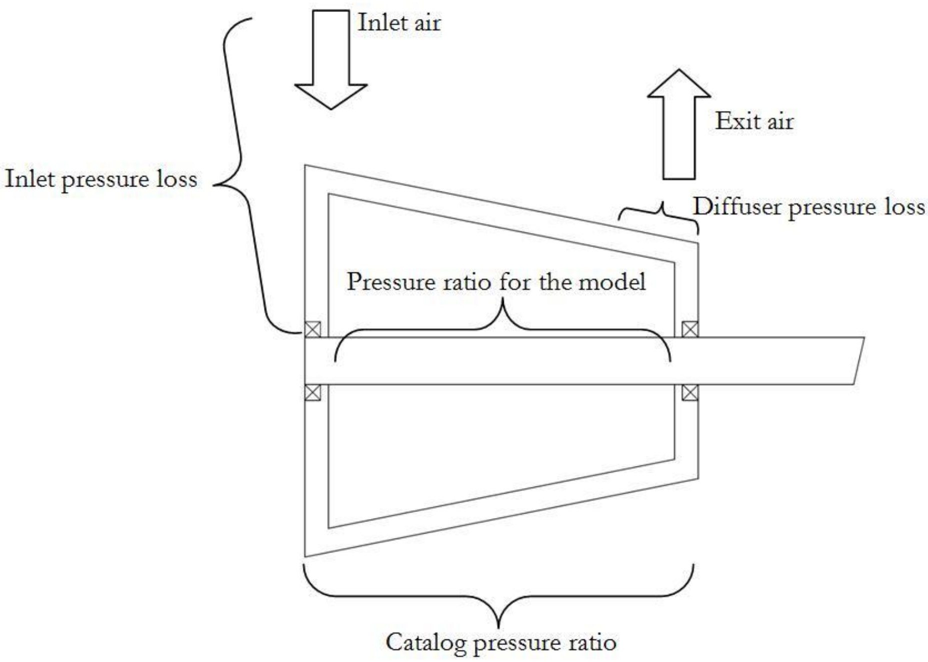



**Figure 15: Pressure losses and pressure ratios in the compressor**

**Calculation of the turbine inlet temperature, polytropic turbine efficiency, fuel to air ratio and turbine power**

By solving the below equations (42, 43, 44, 45 and 46) the inlet temperature of the turbine, polytropic blading turbine efficiency, fuel to air ratio and turbine power can be calculated. For the equation 42 the corrected pressure ratio for the turbine is used.

The equation 42 is used for calculating the turbine polytropic blading efficiency.

$$\boldsymbol{\eta_T = \frac{\int_{T_4}^{T_3}\left[\frac{C_{p(Air)}.T + X_{H2O}.C_{p(H2O)}.T + X.C_{p(CH4)}.T}{(R_{Air} + X_{H2O}.R_{H2O} + X.R_{CH4}).T}\right]dT}{ln[P_{ratio}]}}$$

**Equation 42**

The mass balance is applied to calculate the dry air mass flow rate to the gas turbine. For the mass balance it is assumed that all the inlet air and the pure methane are removed from the gas turbine as an exhaust gas. The equation 43 represents the mass balance of the gas turbine.

$$\boldsymbol{m_{dry\ air} = \frac{m_{exhaust}}{(1 + X_{H2O} + X)}}$$

**Equation 43**

$X_{H2O}$ = Water content of the inlet air

X = Pure methane to air ratio

The specific power of the turbine can be calculated by using the equation 44. The specific enthalpy difference between the turbine inlet to outlet can be calculated from the equation 44.

$$\boldsymbol{P_{turbine} = \int_{T_4}^{T_3}\left(C_{p(Air)}.T + X_{H2O}.C_{p(H2O)}.T + X.C_{p(CH4)}.T\right)dT}$$

**Equation 44**

By subtracting the specific compressor power (equation 38) from the specific turbine power (equation 44) the specific gas turbine power can be calculated.

$$\boldsymbol{P_{GT} = P_{turbine} - P_{compressor}}$$

**Equation 45**

Electrical output of the gas turbine can be calculated by using the equation 46.

$$\boldsymbol{P_{el} = \eta_{el}.\eta_{mech}.m_{dry\ air}.P_{GT}}$$

**Equation 46**

**Calculation of the mixed combustor temperature**

In order to calculate the mixed combustor temperature the equation 47 (Wettstein, 2007)can be used. This is the same equation which was used in the Mathcad model.

$$\boldsymbol{X = \frac{\int_{T_2}^{T_5}(C_{p(Air)}.T + X_{H2O}.C_{p(H2O)}.T)dt}{LHV - \int_{T_1}^{T_5}(C_{p(CH4)}.T)dt}}$$

**Equation 47**

Variables used in the equation 47 can be defined as follows,

| | | |
|---|---|---|
| x | =Fuel to air ratio |
| $T_5$ | =Mixed combustor temperature |
| LHV | =Lower heating value for the fuel |

### 3.3.2 Results from the EES model

After inserting the catalog data and all other input data to the EES model (via interface) the calculation can be carried out. The calculation process needs time to generate set of output. This time duration depends on the processor power of the computer. High speed computers need less time, while old computers need more time for the calculation process. Intel® Core™2 Duo CPU 2.00GHz processor with 3.00 GB memory (RAM) computer can solve a typical calculation in around 18 seconds, but for the Mathcad calculation the same computer consumes only 6 seconds. After the calculation all the results can be seen in the output section of the GUI. The figure 16 represents the GUI of the EES model and the table 3 displays all the output variables which can be generated from the model.

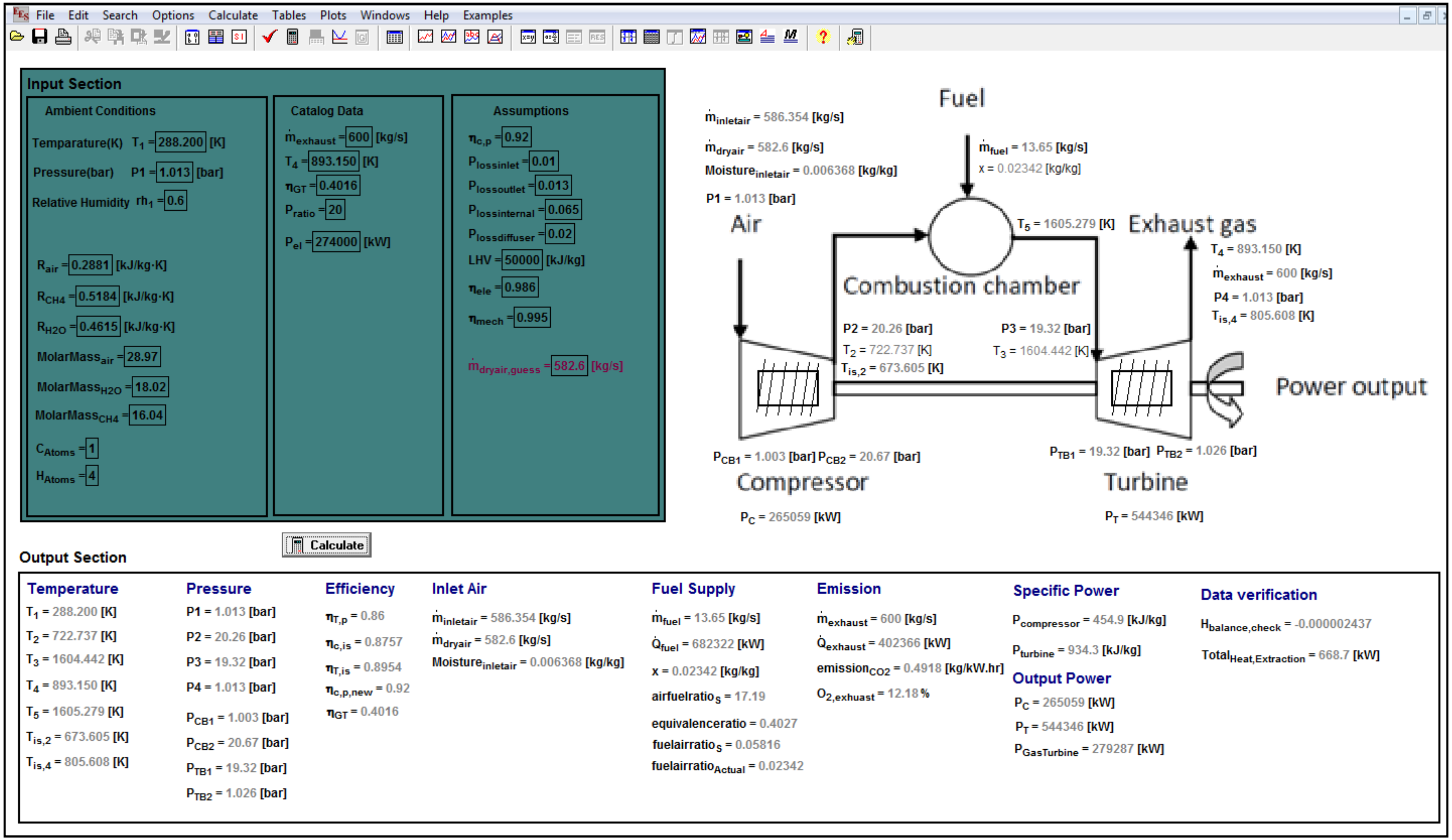



**Figure 16: GUI of the EES model**

| | Symbol | Description |
|---|---|---|
| 01 | $Moisture_{inletair}$ | Water content of the inlet air |
| 02 | $T_2$ | Compressor exit temperature (K) |
| 03 | $T_3$ | Turbine inlet temperature (K) |
| 04 | $T_5$ | Mixed combustor temperature (K) |
| 05 | x | Fuel to air ratio |
| 06 | $m_{fuel}$ | Fuel mass flow rate (kg/s) |
| 07 | $m_{dryair}$ | Dry air mass flow rate (kg/s) |
| 08 | $m_{inletair}$ | Inlet air mass flow rate (kg/s) |
| 09 | $Q_{exhaust}$ | Available exhaust heat (kW) |
| 10 | $Q_{fuel}$ | Heat from the fuel (kW) |
| 11 | $P_C$ | Power needed to run the compressor (kW) |
| 12 | $P_T$ | Power generated from the turbine (kW) |
| 13 | $p_c$ | Specific compressor power (kJ/kg) |
| 14 | $p_t$ | Specific turbine power (kJ/kg) |
| 15 | $h_{T,p}$ | Polytropic efficiency of the turbine |
| 16 | $h_{T,IS}$ | Isentropic efficiency of the turbine |
| 17 | $h_{C,IS}$ | Isentropic efficiency of the compressor |
| 18 | $H_{extract}$ | Extracted heat from the combustion chamber (kW) |
| 19 | $T_{IS,2}$ | Isentropic compressor exit temperature (K) |
| 20 | $T_{IS,4}$ | Isentropic turbine exit temperature (K) |
| 21 | $P_2$ | Pressure after the compressor (bar) |
| 22 | $P_3$ | Pressure after the combustion chamber (bar) |
| 23 | $P_{CB1}$ | Pressure at the compressor blading inlet (bar) |
| 24 | $P_{CB2}$ | Pressure at the compressor blading exit (bar) |
| 25 | $P_{TB1}$ | Pressure at the turbine blading inlet (bar) |
| 26 | $P_{TB2}$ | Pressure at the turbine blading exit (bar) |
| 27 | $H_{balance}$ | Heat balance check |

**Table 3: Results output from the EES model**

By means of the ambient conditions the water content of the inlet air to the compressor can be calculated. After that the compressor exit air temperature ($T_2$), combustor exit temperature ($T_5$) and the turbine inlet temperature ($T_4$) are calculated. The isentropic compressor exit temperature ($T_{IS,2}$ ) and the isentropic turbine exit temperature ($T_{IS,4}$ ) also can be calculated from the model. The temperature values are the most important values in the gas turbine cycle calculation. Because most of the calculation steps use the mathematical integration, temperature is the integration variable and integration limits are also fixed at some temperature values. Not only that but also the temperature values are used for calculating the isentropic compressor and turbine efficiency.

Specific power of the compressor, specific power of the turbine and specific net electric power of the gas turbine can be calculated from the ESS model. The mathematical integration is used in those calculations.

By multiplying specific power values from dry air mass flow rate the power consumed by the compressor and the generated power of the turbine can be calculated.

By assuming the compressor polytropic efficiency the turbine polytropic efficiency can be calculated. The isentropic efficiency of the turbine as well as compressor can also be calculated from this EES model. Finally the compressor polytropic blading efficiency is calculated from the model in order to verify the results.

Inside the open gas turbine cycle there are several pressure stages available. Those pressure values can be used to understand the gas turbine cycle and can be used to plot the Temperature-Entropy (TS) diagram of the specific gas turbine. The most important main pressure values inside the open gas turbine cycle can be generated from the modeled EES software. The EES model starts its calculation with limited known pressure values. The ambient pressure, the pressure ratio, the inlet pressure loss, the internal pressure loss, the outlet pressure loss and the compressor diffuser pressure loss are the limited pressure values for the calculation. After the EES model calculation the main unknown pressure values of the open gas turbine cycle can be generated. Pressure at the compressor blading inlet ($P_{CB1}$), Pressure at the compressor blading exit ($P_{CB2}$), Pressure at the turbine blading inlet ($P_{TB1}$), Pressure at the turbine blading exit ($P_{TB2}$), Pressure after the compressor ($P_2$), Pressure after the combustion chamber ($P_3$) are the main unknown pressure values for the open gas turbine. The figure 16 and 17 indicate the main pressure values incorporated in the compressor and the turbine.

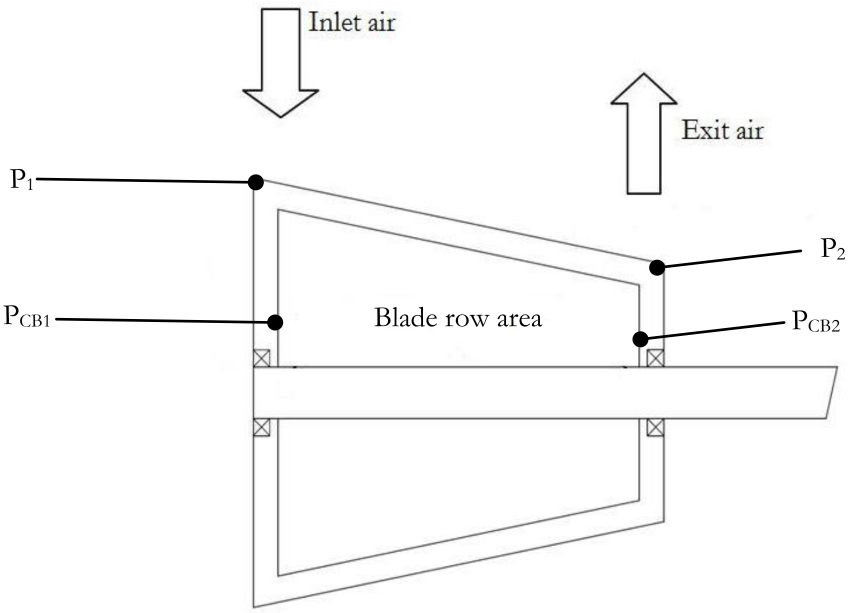



**Figure 17: The pressure values in the compressor**

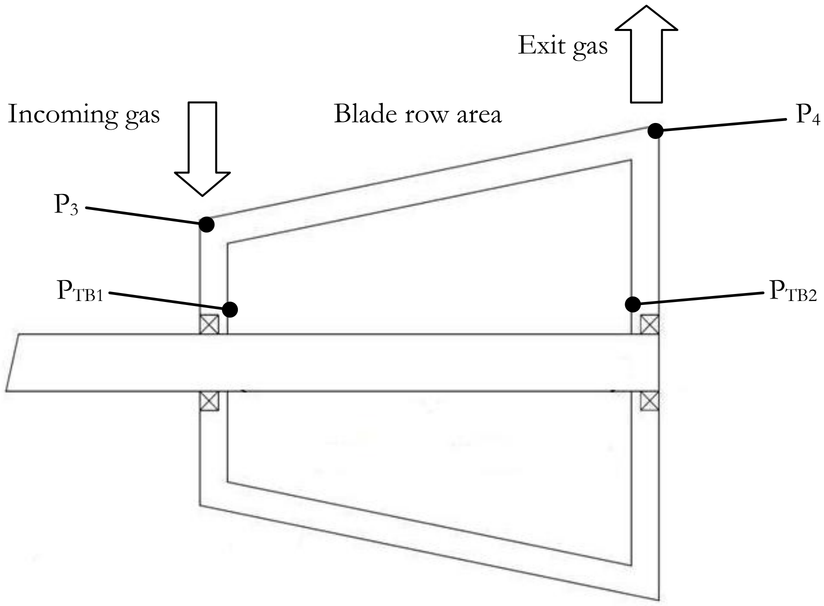



**Figure 18: The pressure values in the turbine**

### 3.3.3 Results validation

In order to validate the results from the EES model, the Mathcad model was used. Results from the Mathcad model were used as reference values for the validation process. The Siemens SGT6-8000H gas turbine was chosen and the pure methane was taken as the fuel for the machine. The typical catalog data for the Siemens SGT6-8000H are as follows.

| | |
|---|---|
| Power at terminals | 274 MW |
| GT efficiency terminals | 40.16% |
| Outlet mass flow | 600 kg/s |
| Exhaust temperature | 620 °C |
| Pressure Ratio | 20 |

The following assumptions were used for both the models.

| | |
|---|---|
| • Polytropic blading efficiency of the compressor | 92% |
| • Inlet pressure loss | 1% |
| • Outlet pressure loss | 1.3% |
| • Internal pressure loss | 6.5% |
| • Compressor diffuser pressure loss | 2% |

The results from the both models are given below.

| | Parameters | MathCAD Model | EES Model | Difference | Percentage error |
|---|---|---|---|---|---|
| 01 | Gas turbine Type | SGT6-8000H | SGT6-8000H | | |
| 02 | Power at terminals (MW) | 274 | 274 | 0 | 0.00% |
| 03 | GT efficiency terminals | 40.16% | 40.16% | 0 | 0.00% |
| 04 | Outlet mass flow (kg/s) | 600 | 600 | 0 | 0.00% |
| 05 | Exhaust temperature (°C) | 620 | 620 | 0 | 0.00% |
| 06 | Pressure Ratio (compressor exit plenum to ambient) | 20 | 20 | 0 | 0.00% |
| 07 | Compressor blading exit total temperature (°C) | 448.8 | 449.587 | 0.787 | 0.18% |
| 08 | Mixed combustor temperature (°C) "MCT" | 1329.2 | 1332.071 | 2.871 | 0.22% |
| 09 | Mixed turbine inlet temp. (°C) "MTT" | 1328.5 | 1331.297 | 2.797 | 0.21% |
| 10 | Polytropic compressor blading efficiency | 92.00% | 92.00% | 0 | 0.00% |
| 11 | Polytropic turbine blading efficiency. | 85.99% | 86.00% | 0.0001 | 0.01% |
| 12 | Relative heat extraction | 0.08% | 0.09091% | 0.0001299 | 16.24% |
| 13 | Fuel mass flow rate (kg/s) | 13.645 | 13.645 | 0 | 0.00% |
| 14 | Compressor inlet humid air mass flow rate (kg/s) | 586.355 | 586.355 | 0 | 0.00% |
| 15 | Fuel heat flow (MW) | 682.27 | 682.27 | 0 | 0.00% |
| 16 | Dry air inlet mass flow (kg/s) | 582.645 | 582.645 | 0 | 0.00% |
| 17 | Mixed isentropic compressor blading efficiency | 88.26% | 87.57% | -0.0069 | -0.78% |
| 18 | Mixed isentropic turbine blading efficiency | 89.53% | 89.54% | 0.0001 | 0.01% |
| 19 | Equivalent heat extraction temperature drop (°C) | 0.64 | 0.837 | 0.36 | 56.25% |
| 20 | Relative inlet pressure loss | 1.00% | 1.00% | 0 | 0.00% |
| 21 | Relative outlet pressure loss | 1.30% | 1.30% | 0 | 0.00% |
| 22 | Pressure loss between compressor and turbine blading | 6.50% | 6.50% | 0 | 0.00% |
| 23 | Mechanical efficiency | 99.50% | 99.50% | 0 | 0.00% |
| 24 | Electric efficiency | 98.60% | 98.60% | 0 | 0.00% |
| 25 | Total heat extraction (MW) | 0.52 | 0.6203 | 0.1003 | 19.29% |
| 26 | Relative heat balance mismatch | 9.10E-07 | 2.435E-06 | 1.53E-06 | 168.13% |

**Table 4: Results comparison Mathcad model and EES model**

In the first 5 rows (power at terminals, GT efficiency terminals, outlet mass flow, exhaust temperature, pressure Ratio) in the table 4 have same values for both Mathcad and EES model, because those values are input values for the model. Those five values are the well known catalog data for the gas turbine. The row numbers 10, 20, 21, 22, 23 and 24 (polytropic compressor blading efficiency, relative inlet pressure loss, relative outlet pressure loss, pressure loss between compressor and turbine blading, mechanical efficiency, electric efficiency) also have the same values for both the models. Those values are also input values for both the models. The row 07 represents the air exit temperature of the compressor. There is 0.18% difference in between the two values obtained. There are 0.22% and 0.21% differences in the mixed combustor temperature and the turbine inlet temperature of the two models respectively. The turbine polytropic blading efficiency value and the turbine isentropic efficiency value of the two models have the similar values. Differences in those values are small and can be neglected. The compressor isentropic efficiency values in the two models have minor difference and the difference is 0.78%. The fuel mass flow rate and the fuel heat flow of the two models have same valves. There is 0% difference in the dry air mass flow values, but in the humid air mass flow rate to the compressor have same values.

Therefore according to the table 4, some of the results generated from the MathCAD model and the EES model are having minor differences. In both models the gas properties of dry air, water and methane come from some data bases and functions. For the MathCAD model, the external libraries LibHuAir and LibHuGas and some formulas were used. For the EES model the inbuilt internal libraries were used. , but two models give two different values for the $C_p$ for the same temperature value. Therefore the different results are due to different $C_p$ valves for the dry air, water and methane used in the two models.

The table 5 shows the $C_p$ values for dry air used in the Mathcad model and the EES model and their respective differences. The values in the table 5 plotted in the figure 20. It is obvious that there are differences in between the two $C_p$ values. The temperature from 200K to 300K the difference is 0.001kJ/kg. From 300K to 1000K the difference maintains at low values. After 1000K the difference increases rapidly. The main reason for difference in the $C_p$ values of two models is due to the dissociation of air in high temperature (Wettstein, 2006). In Mathcad model the air data is selected by considering dissociation at high temperature and hence it gives higher $C_p$ values for higher temperature. The deviation of the Cp value stars from 800 $^0$C (Wettstein, 2006). But in the EES model the air data is selected by not considering the dissociation at high temperature.

| | Temp(K) | Cp(MathCAD)kJ/kg | Cp(EES)kJ/kg | Difference kJ/kg |
|---|---|---|---|---|
| 01 | 200 | 1.007 | 1.002 | 0.005 |
| 02 | 250 | 1.006 | 1.003 | 0.003 |
| 03 | 300 | 1.007 | 1.005 | 0.002 |
| 04 | 350 | 1.009 | 1.008 | 0.001 |
| 05 | 400 | 1.014 | 1.013 | 0.001 |
| 06 | 450 | 1.021 | 1.020 | 0.001 |
| 07 | 500 | 1.030 | 1.029 | 0.001 |
| 08 | 550 | 1.040 | 1.040 | 0.000 |
| 09 | 600 | 1.051 | 1.051 | 0.000 |
| 10 | 650 | 1.063 | 1.063 | 0.000 |
| 11 | 700 | 1.075 | 1.075 | 0.000 |
| 12 | 750 | 1.087 | 1.087 | 0.000 |
| 13 | 800 | 1.099 | 1.098 | 0.001 |
| 14 | 850 | 1.111 | 1.110 | 0.001 |
| 15 | 900 | 1.122 | 1.121 | 0.001 |
| 16 | 950 | 1.132 | 1.131 | 0.001 |
| 17 | 1000 | 1.142 | 1.141 | 0.001 |
| 18 | 1050 | 1.152 | 1.150 | 0.002 |
| 19 | 1100 | 1.162 | 1.159 | 0.003 |
| 20 | 1150 | 1.171 | 1.167 | 0.004 |
| 21 | 1200 | 1.180 | 1.174 | 0.006 |
| 22 | 1250 | 1.188 | 1.181 | 0.007 |
| 23 | 1300 | 1.197 | 1.188 | 0.009 |
| 24 | 1350 | 1.205 | 1.194 | 0.011 |
| 25 | 1400 | 1.214 | 1.200 | 0.014 |
| 26 | 1450 | 1.222 | 1.206 | 0.016 |
| 27 | 1500 | 1.230 | 1.211 | 0.019 |
| 28 | 1550 | 1.239 | 1.216 | 0.023 |
| 29 | 1600 | 1.248 | 1.220 | 0.028 |
| 30 | 1650 | 1.257 | 1.225 | 0.032 |
| 31 | 1700 | 1.266 | 1.229 | 0.037 |
| 32 | 1750 | 1.276 | 1.233 | 0.043 |
| 33 | 1800 | 1.286 | 1.237 | 0.049 |
| 34 | 1850 | 1.297 | 1.240 | 0.057 |
| 35 | 1900 | 1.309 | 1.244 | 0.065 |
| 36 | 1950 | 1.322 | 1.247 | 0.075 |
| 37 | 2000 | 1.336 | 1.250 | 0.086 |
| 38 | 2050 | 1.353 | 1.252 | 0.101 |
| 39 | 2100 | 1.372 | 1.255 | 0.117 |
| 40 | 2150 | 1.393 | 1.258 | 0.135 |
| 41 | 2200 | 1.418 | 1.260 | 0.158 |

**Table 5: Difference between the Mathcad $C_p$ values and EES $C_p$ values for dry air**

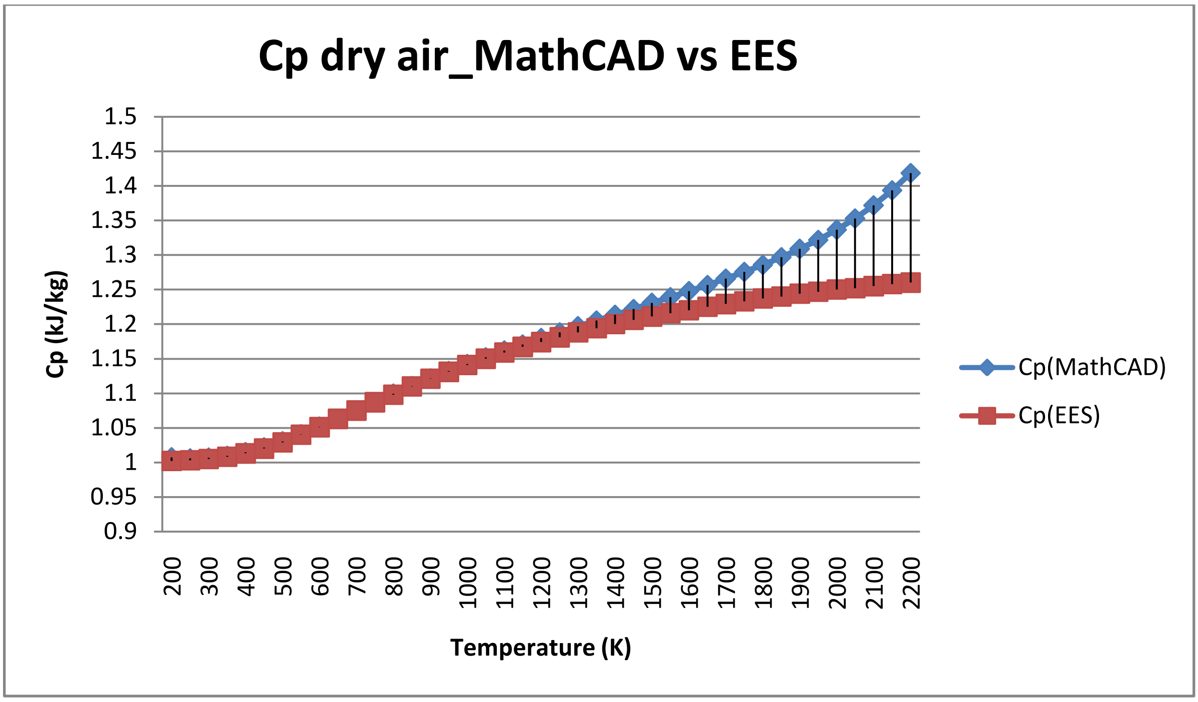



**Figure 19: Difference between the Cp (Mathcad) and Cp(EES) for dry air (200K-2150K)**

In the Figure 19 the light blue color curve represents the $C_p$ values used in the Mathcad model and the brown color curve represents the $C_p$ values used in the EES model. After the 1000K point of the graph the difference is rapidly increasing.

The 7$^{th}$ row of the table 4 represents the compressor exit air temperature and it is 722.737K (449.587 $^0$C). The corresponding difference of the value is 0.787K. In the row 8$^{th}$ and 9$^{th}$ of the table 4 represent the mixed combustor temperature and the turbine inlet temperature. The corresponding differences in those values are 2.871K and 2.797K respectively. Therefore it is obvious that in high temperature values, the differences are higher than the low temperature values. The main reason for the higher difference in the high temperature values is the higher deviation of the $C_p$ values in the high temperature region.

The figure 20 shows the $C_p$ values for methane used in both the Mathcad and EES models. The light blue color curve represents the $C_p$ values used in the Mathcad model. The $C_p$ values for Mathcad model were extracted from the equation 35. The brown color curve of the figure 20 represents the $C_p$ values used in the EES model. Those values were extracted from the EES internal library. Between the temperature ranges 200K to 600K the difference in the two curves is not significant, but after 600K the diffidence is significant. It is obvious that the considerable difference in the two $C_p$ curves between the temperature range 600K to 2200K.

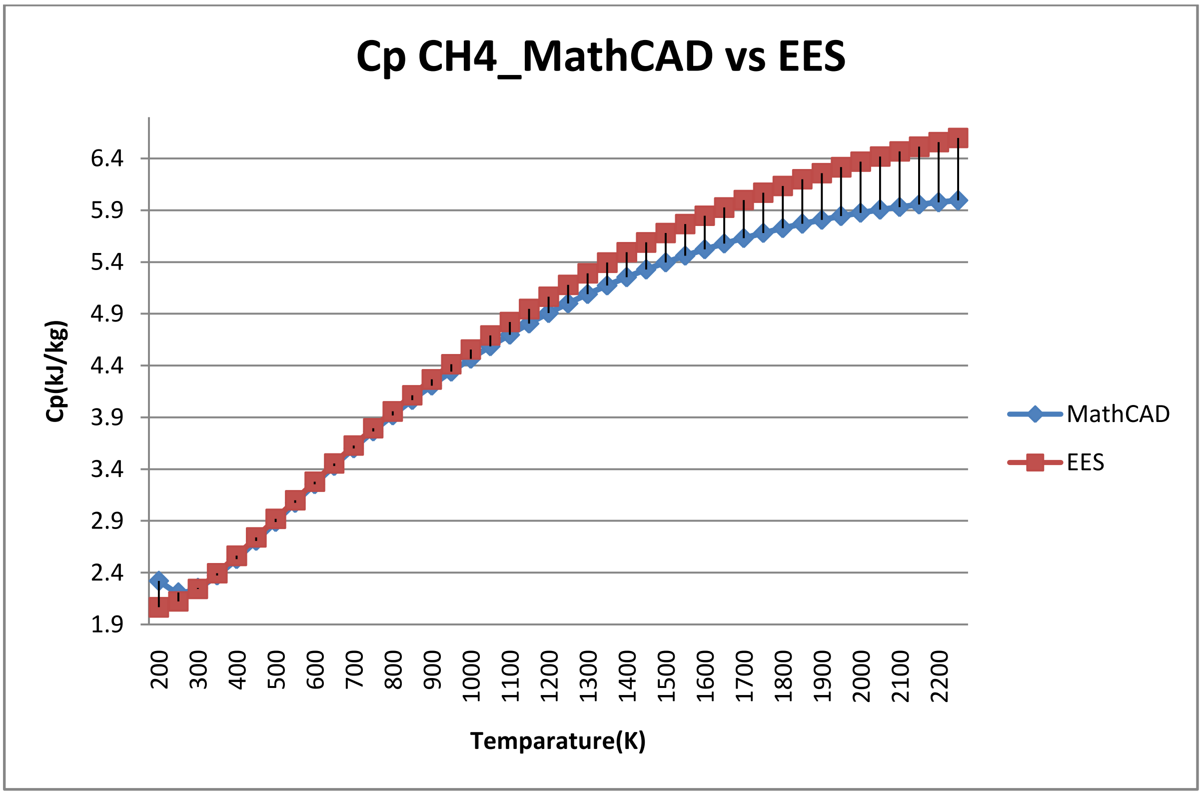



Figure 20: Difference between the Cp (Mathcad) and Cp(EES) for Methane (200K-2250K)

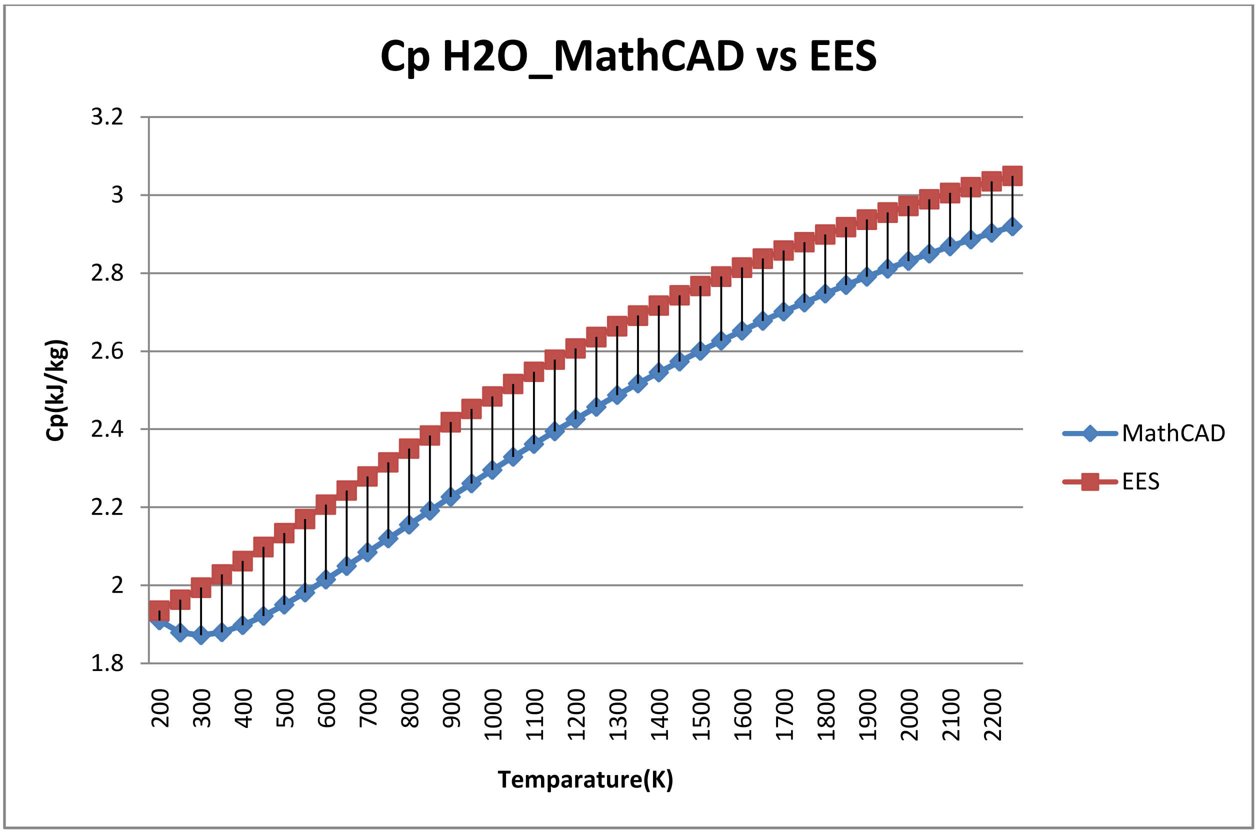



Figure 21: Difference between the Cp(Mathcad) and Cp(EES) for water (200K-2250K)

The Figure 21 represents the $C_p$ values for water used in the Mathcad and ESS models. The light blue curve represents the $C_p$ values used in the Mathcad model. The values for the curve were extracted from the equation 36. The brown curve represents the $C_p$ values used in the EES model and those values were extracted from the EES internal library. In figure 21, it is clearly indicates the different between both $C_p$ values used in the models.

Apart from the $C_p$ values, the calculation of the water content of the inlet air also uses the library functions. The Mathcad uses LibHuAir library for this calculation, while the EES uses internal library. The value of the water content in the Mathcad model is 0.00635083 $g/kg_{air}$ (page 25) and 0.006368 $g/kg_{air}$ for the EES model. During the calculation of the gas turbine, both the Mathcad and EES model use above values. Due to the difference in the water content, the result of both models can vary as water content value is constant throughout the calculation.

In order to validate the results obtained from the EES model, the available MathCAD model was considered as a reference model. According to the table 4, some results from the both model have different values. The differences in the $C_p$ values and the water content value of the incoming air in the two models are the main reasons for different results. If both the models use the same libraries and functions for the calculation, then the results should be the same. Therefore the $C_p$ function for the dry air, water, methane and the water content of the incoming air extracted from the EES libraries were used them inside the MathCAD model. Then the new MathCAD results and EES results compared.

| | **Parameters** | **MathCAD Model with EES database** | **EES Model** | **New difference** | **Percentage error** |
|---|---|---|---|---|---|
| 01 | Gas turbine Type | SGT6-8000H | SGT6-8000H | | |
| 02 | Power at terminals (MW) | 274 | 274 | 0 | 0.00% |
| 03 | GT efficiency terminals | 40.16% | 40.16% | 0 | 0.00% |
| 04 | Outlet mass flow (kg/s) | 600 | 600 | 0 | 0.00% |
| 05 | Exhaust temperature (°C) | 620 | 620 | 0 | 0.00% |
| 06 | Pressure Ratio (compressor exit plenum to ambient) | 20 | 20 | 0 | 0.00% |
| 07 | Compressor blading exit total temperature (°C) | 448.7 | 449.587 | 0.887 | 0.20% |
| 08 | Mixed combustor temperature (°C) "MCT" | 1330.7 | 1332.071 | 1.371 | 0.10% |
| 09 | Mixed turbine inlet temp. (°C) "MTT" | 1330.5 | 1331.297 | 0.797 | 0.06% |
| 10 | Polytropic compressor blading efficiency | 92.00% | 92.00% | 0 | 0.00% |
| 11 | Polytropic turbine blading efficiency. | 86.00% | 86.00% | 0 | 0.00% |
| 12 | Relative heat extraction% | 0.02% | 0.09091% | 0.000709 | 354.50% |
| 13 | Fuel mass flow rate (kg/s) | 13.645 | 13.645 | 0 | 0.00% |
| 14 | Compressor inlet humid air mass flow rate (kg/s) | 586.355 | 586.355 | 0 | 0.00% |
| 15 | Fuel heat flow (MW) | 682.27 | 682.27 | 0 | 0.00% |

| | | | | | |
|---|---|---|---|---|---|
| 16 | Dry air inlet mass flow (kg/s) | 582.654 | 582.645 | -0.009 | 0.00% |
| 17 | Mixed isentropic compressor blading efficiency | 88.26% | 87.57% | -0.0069 | -0.78% |
| 18 | Mixed isentropic turbine blading efficiency | 89.55% | 89.54% | -0.0001 | -0.01% |
| 19 | Equivalent heat extraction temperature drop (°C) | 0.18 | 0.837 | 0.657 | 365.00% |
| 20 | Relative inlet pressure loss | 1.00% | 1.00% | 0 | 0.00% |
| 21 | Relative outlet pressure loss | 1.30% | 1.30% | 0 | 0.00% |
| 22 | Pressure loss between compressor and turbine blading | 6.50% | 6.50% | 0 | 0.00% |
| 23 | Mechanical efficiency | 99.50% | 99.50% | 0 | 0.00% |
| 24 | Electric efficiency | 98.60% | 98.60% | 0 | 0.00% |
| 25 | Total heat extraction (MW) | 0.15 | 0.6203 | 0.4703 | 313.53% |
| 26 | Relative heat balance mismatch | 4.40E-06 | 2.435E-06 | -2E-06 | -45.45% |

**Table 6: New results comparison Mathcad model and EES model**

After using the EES library data to the Mathcad model the new results obtained and those results are shown in the table 6. The compressor exit air temperature value has 0.2% difference in between two models. The mixed combustor temperature and the turbine inlet temperature have 0.1% and 0.06% difference in the two models respectively and the new differences are lower than the initial stage. The turbine polytropic blading efficiency is equal in both the models. The turbine isentropic efficiency has 0.01% difference in the two models and this difference can be negligible. The compressor isentropic efficiency has 0.78% difference in the two models and this value is same as the previous case. Fuel mass flow rate and the fuel heat flow of both models have same values. The dry air mass flow rate has 0.0015% difference in the two models.

Although the Mathcad model used the EES library value for the calculation the differences in the two models remain, but the differences are less than the previous case. Therefore the reasons for differences in the two models are partially depend on the library data. During the calculation process both the models undergo some mathematical integration and iteration processes. Both the models use different intervals for the integration and different step sizes for the iteration. Therefore the results from the two models depend on these mathematical operations also. Different intervals and different step size affect the final solution of the mathematical operation. Therefore the reasons for difference results in two models can be considered as different library data and different mathematical operation in the two models.

### 3.3.4 CompEdu Chapter

CompEdu is an internet base learning platform, which is used by the division of Heat and Power Technology of KTH for their teaching purpose. The CompEdu is very popular among students of KTH and especially among the distance based students. It is a convenient way to transfer the knowledge among students.

As mentioned in the previous sections of this report there is an existing CompEdu chapter for the gas turbine calculation and it can be considered as the user manual to the Mathcad model. During this project a new software model for the gas turbine calculation was introduced. After the completion of the new software model the existing CompEdu chapter was changed according to the new model.

### 3.3.5 The User Manual

Apart from the CompEdu chapter, a separate user manual is developed for the new EES model. The user manual consists of description about all the input and output parameters, the governing equations and the programming sequence of the EES model. Later part of the manual consists of a brief introduction about the EES software, which can be beneficial for the users, who do not have any previous experience of the EES. Please refer Appendix B for the User Manual.

## 3.4 Discussion

During the validation process the previous Mathcad model and the new EES model results were compared. There were similarities in some results of both models and there were minor differences in some results due to the different $C_p$ values and different mathematical operations incorporated in the calculations. As a whole the results from the EES model can be acceptable. After the validation process the results from the EES model can be summarized as follows.

The main three temperature values incorporate in the gas turbine calculation are the compressor exit air temperature, mixed combustor temperature and turbine inlet temperature. All the temperature values obtained from the EES model have minor differences with the Mathcad model. For the compressor exit air temperature and the turbine inlet temperature, the difference is less than 1K. The mixed combustor temperature has a slightly higher difference and it is less than 1.5K. There are four types of efficiency values in the gas turbine calculation and those are compressor polytropic blading efficiency, turbine polytropic blading efficiency, compressor isentropic efficiency and turbine isentropic efficiency. As discussed in the methodology chapter the compressor polytropic efficiency is used as an input value. In other words it is assumed in initial stage of the calculation. But later, inside the EES model the same value is recalculated for the verification. The results obtained for the turbine polytropic efficiency and the turbine isentropic efficiency have similar values in both the Mathcad and EES models. The value for the compressor isentropic efficiency has minor difference in both models and the difference is 0.0069. The dry air mass flow rate, the compressor humid air mass flow rate, the fuel mass flow rate are calculated from the EES model. The zero differences were obtained for the humid air mass flow rate and fuel mass flow rate. A minor difference obtained for the dry air mass flow rate.

The main purpose for implementing this EES model is to use for technical tasks of the turbomachinery course at KTH. Therefore the model should able to be easily handled by the students and it should be more user friendly and the programming code of the EES model should be easily understood by the students. The implemented EES model has all the requirements discussed above and the EES software is appropriate for this application. Furthermore the EES is more popular among students and they know how to use the EES platform. That is also an advantage of new EES model. Consequently the newly implemented EES model can be used for the technical and teaching tasks of the turbomachinery course without any difficulties.

The availability of a Graphical User Interface (GUI) is one advantage for the EES model. From the GUI the student is able to add input parameters for the EES model, and they can able to run the model by clicking the ‘calculate’ button in the GUI. After the calculation process all the output results can be seen in the GUI. By introducing the GUI for the model, both the user friendliness and the attraction can be improved. The typical GUI of the EES model is shown in the figure 16. The detailed description about the GUI is available in ”Gas Turbine Thermodynamic and Performance Analysis Methods Using Available Catalog Data-User Manual & Exercises”.

The other advantage of the EES model, it can accept any ambient condition for the model. In the Mathcad model there is only one ambient condition which can be used in the model.

# 4 Conclusion

The gas turbine manufacturers only publish some interface parameters for the commercial gas turbines, which only give little information about that specific machine. Most of the important parameters of the gas turbine remain as hidden for the user. By knowing the hidden parameter, one can do the thermodynamic analysis in order to evaluate the performance of that gas turbine. Therefore the main objective of this thesis is to find the suitable way to extract hidden thermodynamic properties of gas turbines from the published catalog data.

A software model developed, in order to extract hidden data from the catalog data for a gas turbine. The Engineering Equation Solver (EES) is used to model the software. As discussed in the previous part of this report the modeled software will be used for technical and pedagogical tasks for a turbomachinery course at KTH. However at the time of beginning this thesis work the KTH has one software model which can produce set of hidden data from catalog data. Therefore the primary target for this thesis work is to improve the existing software model. As discussed in the initial part of the methodology chapter the existing model studied and gave some suggestions to improve it. Then according to the listed suggestions the new software for the thesis was modeled. The results from the EES model has been validated in the later part of this thesis work.

The software code in the EES program is understandable. Any student has knowledge on basic thermodynamic and turbomachinery theory can easily grab the program. The program body consists with appropriate comments and the programming code divides in to sub sections in order to increase the user friendliness. By using the 'formatted equation window' of the EES interface the user can see the programming code in easy to read mathematical notations. The programming code can be seen in the appendix A in this report. A separate user manual for the EES program is also introduced to increase the reliability of the model. The user of the model can refer the user manual in order to get the basic understanding of it. Some basic introduction about the EES software is also included for the user manual. As a final conclusion it can say that the EES model has all the necessary requirement in order to use as technical and pedagogical purposes of turbomachinery course at KTH.

## 4.1 Future works

For the EES model there are some future works remaining. Any interested person can attend the further development of the program. Further development suggestion is given below.

**Extend the fuel use in the EES model.**

The EES model only accepts pure methane as a fuel, but the model can be extended to any composition of the natural gas. There is an EES model which can be used to any natural gas compositions. For that model the results are not validated due to unavailability of the reference results. Any interested person can carry out the further works.

# References


1. Akbari, P., & Müller, N. 2003. *Performance Investigation of Small Gas Turbine Engines. 39th AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit. Alabama*: American Institute of Aeronautics and Astronautics.
2. Brooks, F. J. 2005. *GE Gas Turbine Performance Characteristics*. NY: GE Power Systems.
3. Cohen, H., Rogers, G., & Saravanamuttoo, H. 1996. *Gas Turbine Theory-4th edition*. Essex: Longman Group Limited.
4. Corporation, P. T. 2010. Mathcad help.
5. Corporation, P. T. 2010. User's Guide Mathcad 15.
6. Eastop, T. D., & McConkey, A. 2002. *Applied Thermodynamics for Engineering Technologists*. Dilhi: Pearson Education (Singapore) Pte. Ltd.
7. Energy and Environmental Analysis (an ICF International Company). 2008. *Technology Characterization: Gas Turbines*. Virginia.
8. fchart. 2012. *EES Engineering Equation Solver*. (F-Chart) Retrieved 08 26, 2012, from F-Chart Software: http://www.fchart.com/ees/
9. johnzactruba. 2009, 11 15. *What is ISO rating of Gas Turbines ?* Retrieved 02 24, 2012, from bright hub: http://www.brighthub.com/engineering/mechanical/articles/25425.aspx
10. Kretzschmar, H. J., Stoecker, I., Kunick, M., Knobloch, K., Hermann, S., & Salomo, B. (n.d.). Property Software for Humid Air Calculated as Ideal Mixture of Real Fluids.
11. Kretzschmar, H. J., Stoecker, I., Kunick, M., Knobloch, K., Kleemann, L., & Seibt, D. (n.d.). Property Library for Humid Gas Mixtures.
12. OLSON, R. G. GAS TURBINES.
13. Petchers, N. 2002. *Combined Heating, Cooling & Power Handbook: Technologies & Applications*. Georgia: The Fairmont Press, Inc.
14. Petek, J., & Hamilton, P. P. 2005. Performance Monitoring for Gas turbines. In ORBIT (pp. 65-74). GE Energy.
15. Rahman, M. M., Ibrahim, T. K., & Abdalla, A. N. 2011. Thermodynamic performance analysis of gas-turbine power-plant. *Vol. 6* (14), 3539-3550.
16. Sa, A. D., & Zubaidy, S. A. 2011. Gas turbine performance at varying ambient temperature. *Applied Thermal Engineering*, 2735-2739.
17. Strand, T. 2005, 01 30. Combustion theory lecture note. *On the development of a Dry Low NOx combustion system for SGT800*.
18. Van Wylen, G. J., & Sanntag, R. E. 1985. *Fundamentals of classical thermodynamic* (Vol. 3rd Edition SI Version). New York: John Wiley & Sons.
19. Vogt, D. 2007, 09 01. Turbomachinery Lecture Notes. *Efficiencies*.
20. Vogt, D. 2007, 09 11. Turbomachinery Lecture Notes. *Turbine and Compressor Lines in h-s Diagrams*.
21. Wettstein, H. E. 2006. *CompEdu Chapter*. KTH.

22. Wettstein, H. E. 2007. Lecture On Gas Turbine Cycle Analysis Aimed At Application To Catalog Data. *ASME Turbo Expo 2007*. Montreal: ASME.
23. Wettstein, H. E. 2011. Natural Gas Fired Gas Turbine Combined Cycles For The Energy Supply In An Economy With Growing Renewable Energy Sources. *World Engineers' Convention*. Geneva.
24. Wilcox, M., Kurz R., & Brun K. Technology Review of Modern Gas Turbine Inlet. *International Journal of Rotating Machinery*. Hindawi Publishing Corporation, 2012. *Vol. 2012*.

## Appendix A: EES programming code

```
"====================================Compressor Exit Temperature Calculation=======================================================" 

SubProgram Compressor(R_air,R_H2O,rh_1,T_1,P1,P_ratiocompressor,a:eta_c_p,Moisture_inletair)         "Define the Sub program called Compressor"

   Moisture_inletair=HumRat(AirH2O,T=T_1,r=rh_1,P=P1)                                                "Calculation of the water content in the 
inlet air"
   L=LN(P_ratiocompressor)
   M=(integral(((Cp(Air,T=T)+Moisture_inletair*Cp(H2O,T=T))/(T*(R_air+Moisture_inletair*R_H2O))), T, T_1, a))
   eta_c_p=L/M

end

"====================================General calculation of Turbine================================================================"
SubProgram Turbine(eta_mech,eta_ele,m_dot_exhaust,T_4,P_compressor,x,Moisture_inletair,b:P_turbine,P_el,m_dot_dryair)      "Define the Sub 
program called Turbine"

   m_dot_dryair=m_dot_exhaust/(1+x+Moisture_inletair)
   P_turbine=integral((Cp(Air,T=T)+x*Cp(CH4,T=T)+Moisture_inletair*Cp(H2O,T=T)),T,T_4,b)              "Turbine specific power calculation"
   P_GT=P_turbine-P_compressor                                                                        "Gas Turbine specific power"
   P_el=m_dot_dryair*P_GT*eta_mech*eta_ele

end
"============="
SubProgram Temp(T_1,AAA,Moisture_inletair,I1:x)

   II=(integral((Cp(Air,T=T)+Moisture_inletair*(Cp(H2O,T=T))),T,T_1,AAA)-I1)
   HH=integral((Cp(CH4,T=T)),T,T_1,AAA)
   x=II/(50000-HH)

end
"====================================Module for calculate compressor specific power=====================================================" 

MODULE Compressor1(A,B,Moisture_inletair:I)
   I=integral((Cp(Air,T=T)+Moisture_inletair*(Cp(H2O,T=T))),T,A,B)
END

"====================================Module for calculate turbine specific power=======================================================" 

MODULE Turbine1(C,D,x,Moisture_inletair:I)
   I=integral((Cp(Air,T=T)+x*Cp(CH4,T=T)+Moisture_inletair*Cp(H2O,T=T)),T,C,D)
END

"====================================Module for calculate new polytropic efficiency of the compresso
   r====================================================" 

MODULE P(T_1,T_2,R_air,Moisture_inletair,R_H2O,P_ratiocompressor:eta_PC1)
   LL=LN(P_ratiocompressor)
   MM=(integral(((Cp(Air,T=T)+Moisture_inletair*Cp(H2O,T=T))/(T*(R_air+Moisture_inletair*R_H2O))), T, T_1, T_2))
   eta_PC1=LL/MM
END
```

```
"====================================INPUT======================================================================================"

{rh_1=0.6}                                          "Relative Humidity of the inlet air"
{T_1=288.15[K]}                                     "Ambient Temparature"
{P1=1.013[bar]}                                     "Ambient Pressure"

{P_lossinlet=0.01}                                  "Inlet pressure loss for the gas turbine"
{P_lossoutlet=0.013}                                 "Outlet pressure loss for the gas turbine"
{P_lossinternal=0.065}                              "Internal pressure loss for the gas turbine"
{P_lossdiffuser=0.02}                                "Diffuser loss"
{P_ratio=20}                                         "Pressure ratio, this is a catalog data"


"=============================calculation of Intermediate pressures and pressure ratios=========================================================="

P_CB1=P1*(1-P_lossinlet)                            "Calculation of compressor inlet blading pressure, corrected for inlet pressure loss"
P2=P1*P_ratio                                       "Pressure after the compressor"
P_CB2=(P_ratio*(1+P_lossdiffuser))*P1               "Calculation of compressor outlet blading pressure, corrected for diffuser loss"
P_ratiocompressor=P_CB2/P_CB1                       "pressure ratio between compressor blading region"
P4=P1                                               "Gas turbine exit pressure, this sholud be equel to the ambinet pressure"

P_TB2=P4/(1-P_lossoutlet)                           "Calculation of turbine outlet blading pressure, corrected for outlet pressure loss"
P3=P_ratio*P1*(1-P_lossinternal)*(1+P_lossdiffuser) "Pressure after the combustion chamber,corrected for internal pressure loss"
P_TB1=P3                                            "Turbine blading inlet pressure, this is equel to P3"
P_ratioturbine=P_TB1/P_TB2                          "pressure ratio between turbine blading region"

"==================================INPUT=========================================================================================="

{eta_c_p=0.92}                                      "Polytropic efficiency of the compressor, this is assumed value"

{R_air=0.288}                                       "Gas constant, AIR"
{R_H2O=0.4615}                                      "Gas constant WATER"
{R_CH4=0.5184}                                      "Gas constant CH4"




"=================================Calling sub program to calculate the compressor exit temparature=========================================="

Call Compressor(R_air,R_H2O,rh_1,T_1,P1,P_ratiocompressor,a:eta_c_p,Moisture_inletair)

T_2=a                                               "Compressor exit temparature"

"=================================Calling module to calculate the compressor specific power==========================================="

CALL Compressor1(T_1,T_2,Moisture_inletair:I1)
P_compressor=I1
```

```
"=================================Calling module to calculate the new polytropic compressor
efficiency===========================================" 

CALL P(T_1,T_2,R_air,Moisture_inletair,R_H2O,P_ratiocompressor:eta_c_p_new)


"==================================INPUT=============================================================================="
{eta_mech=0.995}                                    "Mechanical efficiency of the gas turbine"
{eta_ele=0.986}                                     "Electrical efficiency of the generator"
{eta_GT=0.4016}                                     "Gas turbine efficiency, this is a catalog data"
{LHV=50000}                                         "Lower heating value of the fuel"
{P_el=274000}                                       "Gas turbine electrical output, this is a catalog data"


{m_dot_dryair_guess}                                "Guess value for the dry air mass flow rate"
eta_GT=P_el/(m_dot_dryair_guess*x*LHV)              "Calculation of the dry air mass flow rate"

"==================================INPUT=============================================================================="

{T_4=893.15[K]}                                     "Gas turbine exhaust temparature,this is a catalog data"
{m_dot_exhaust=600[kg/s]}                           "Gas turbine exhaust mass flow rate,this is a catalog data"


"=================================Calling subprogram Turbine to calculate the turbine inlet
temparature=========================================================================" 


CALL Turbine(eta_mech,eta_ele,m_dot_exhaust,T_4,P_compressor,x,Moisture_inletair,b:P_turbine,P_el,m_dot_dryair)



T_3=b                                               "Turbine inlet temparature"
H=integral((Cp(Air,T=T)+x*Cp(CH4,T=T)+Moisture_inletair*Cp(H2O,T=T))/(T*(R_air+x*R_CH4+Moisture_inletair*R_H2O)),T,T_4,T_3)
G=LN(P_ratioturbine)
eta_T_p=H/G                                         "Polytropic efficiency of the turbine"



m_dot_fuel=x*m_dot_dryair                           "Calculation of the fuel rate"
P_T=m_dot_dryair*P_turbine                          "Power output of the turbine"
P_C=m_dot_dryair*P_compressor                       "Power needed to run the compressor"

CALL Temp(T_1,AAA,Moisture_inletair,I1:x)

T_5=AAA

CALL Turbine1(T_3,T_5,x,Moisture_inletair:I3)
```

```
H_extract=I3*m_dot_dryair

m_dot_inletair=m_dot_dryair*(Moisture_inletair+1)

"=====================================Calling MODULE Turbine1 to calculate the available exhaust
heat================================================================="
CALL Turbine1(T_1,T_4,x,Moisture_inletair:I2)

Q_dot_exhaust=m_dot_dryair*I2                         "Available exhaust heat"


"==================================INPUT=============================================================================="

{MolarMass_air=28.965}                                "Molar mass of the air"
{MolarMass_H2O=18.016}                                "Molar mass of the water"
{MolarMass_CH4=16.035}                                "Molar mass of the fuel"

{C_Atoms=1}                                           "Number of carbon atoms in the fuel"
{H_Atoms=4}                                           "Number of hydrogen atoms in the fuel"


"==================================Calculating the equivalence
ratio=============================================================================="

n=C_Atoms+(H_Atoms/4)

airfuelratio_S=4.76*n*(MolarMass_air/MolarMass_CH4)  "air to fuel ratio"

equivalenceratio=x*airfuelratio_S                     "Equivalence ratio"

"=========================================Emission
calculations=================================================================================="

Oxygen_inexhaust=0.21*(1-equivalenceratio)/(1+x+Moisture_inletair)                                 "Available oxgen in the exhaust flow"


Molfraction_water=1/(1+((MolarMass_H2O*1)/(MolarMass_air*Moisture_inletair)))                      "Mol fraction of water"

Molfraction_fuel=1/(1+((MolarMass_CH4*1)/(MolarMass_air*x)))                                       "Mole fraction of the fuel"

m_CO2exhaust=m_dot_dryair*x*44/(MolarMass_CH4)                                                     "Carbon dyoxide flow rate"

emission_CO2=m_CO2exhaust*3600/P_el                                                                "carbon dyoxide emmision per kWh"

Q_dot_fuel=m_dot_dryair*x*LHV                                                                      "Heat from the incoming fuel"
```

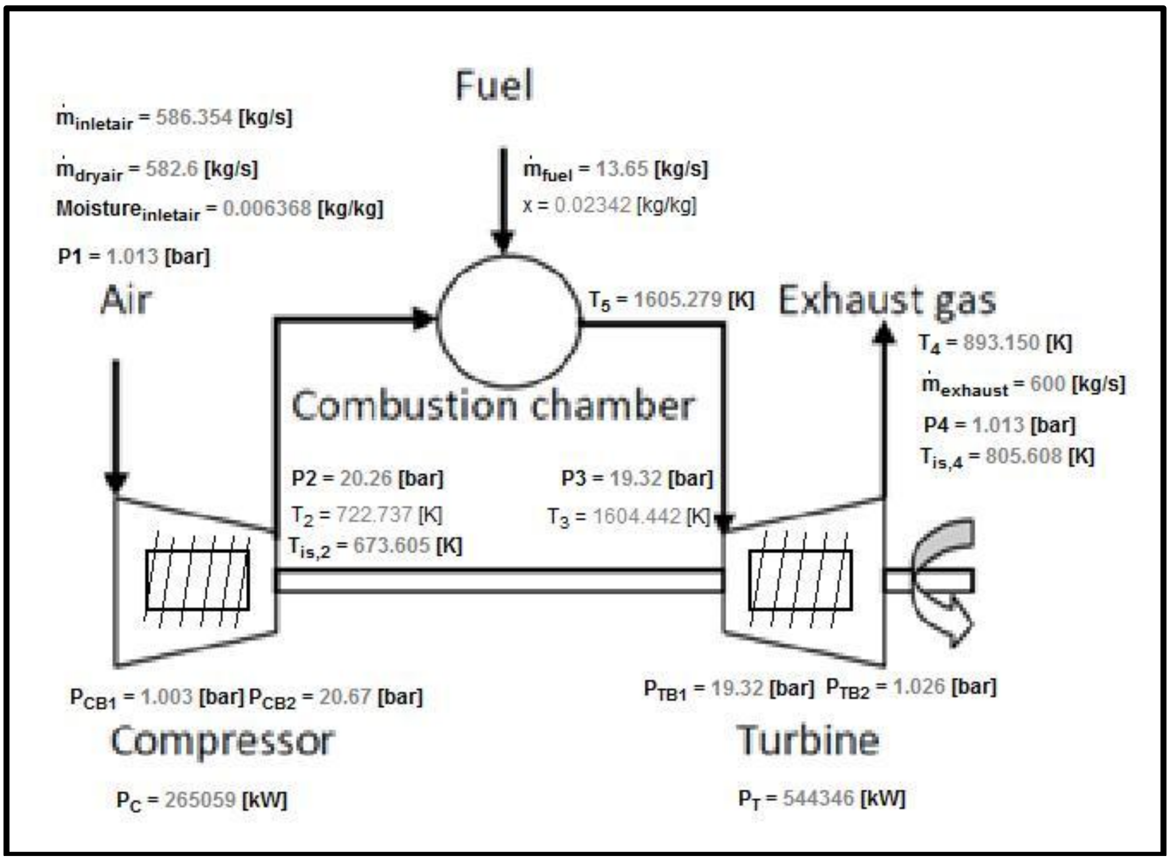
```
Relative_Heat_Extraction=(H_extract/Q_dot_fuel)*100

H_balance=(Q_dot_exhaust+(P_el/(eta_mech*eta_ele))+H_extract)/(Q_dot_fuel)
Total_Heat_Extraction=(Q_dot_fuel-(P_T-P_C)-Q_dot_exhaust)

"==========================================Calculation of the isentropic efficiency of the compressor=============================================================="

cp[1]=Cp(Air,T=T_1)
cp[2]=Cp(H2O,T=T_1)
cp[3]=Cp(Air,T=T_2)
cp[4]=Cp(H2O,T=T_2)
cp[6]=cp[1]+Moisture_inletair*cp[2]
cp[7]=cp[3]+Moisture_inletair*cp[4]
h[1]=cp[6]*T_1                                        "Actual enthalphy of the compressor inlet"
h[2]=cp[7]*T_2                                        "Actual enthalphy of the compressor outlet"

s[1]=Entropy(Air,T=T_1,P=P1)
s[2]=Entropy(H2O,T=T_1,P=P1)
s[3]=s[1]+Moisture_inletair*s[2]                      "entropy of the compressor inlet"

s[3]=Entropy(Air,T=T_is_2,P=P2)+Moisture_inletair*Entropy(H2O,T=T_is_2,P=P2)                          "calculation of the isentropic compressor outlet temparature"
cp[8]=Cp(Air,T=T_is_2)
cp[9]=Cp(H2O,T=T_is_2)
cp[10]=cp[8]+Moisture_inletair*cp[9]
h[4]=cp[10]*T_is_2                                    "Isentropic enthalphy of the compressor outlet"

eta_c_is=(h[4]-h[1])/(h[2]-h[1])                      "Isentropic compressor efficiency"


"==========================================Calculation of the isentropic efficiency of the turbine=============================================================="

cp[11]=Cp(Air,T=T_3)
cp[12]=Cp(H2O,T=T_3)
cp[13]=Cp(CH4,T=T_3)
cp[14]=cp[11]+Moisture_inletair*cp[12]+x*cp[13]
h[5]=cp[14]*T_3

cp[15]=Cp(Air,T=T_4)
cp[16]=Cp(H2O,T=T_4)
cp[17]=Cp(CH4,T=T_4)
cp[18]=cp[15]+Moisture_inletair*cp[16]+x*cp[17]
h[6]=cp[18]*T_4

s[4]=Entropy(Air,T=T_3,P=P3)
s[5]=Entropy(H2O,T=T_3,P=P3)
s[6]=Entropy(CH4,T=T_3,P=P3)
```

```
s[7]=s[4]+Moisture_inletair*s[5]+x*s[6]

s[7]=Entropy(Air,T=T_is_4,P=P4)+Moisture_inletair*Entropy(H2O,T=T_is_4,P=P4)+x*Entropy(CH4,T=T_is_4,P=P4)

cp[19]=Cp(Air,T=T_is_4)
cp[20]=Cp(H2O,T=T_is_4)
cp[21]=Cp(CH4,T=T_is_4)

cp[22]=cp[19]+Moisture_inletair*cp[20]+x*cp[21]

h[7]=cp[22]*T_is_4

eta_T_is=(h[5]-h[6])/(h[5]-h[7]
```

## Appendix B: Gas Turbine Thermodynamic and Performance Analysis Methods Using Available Catalog Data-User Manual & Exercises

KTH Industrial Engineering
and Management

# Gas Turbine Thermodynamic and Performance Analysis Methods Using Available Catalog Data
# User Manual & Exercises




**K.A.B. Pathirathna**

**Master of Science Thesis**

KTH School of Industrial Engineering and Management
Energy Technology EGI-2013-147MSC EKV993
SE-100 44 STOCKHOLM

# 1 Introduction

A program has been developed to determine the gas turbine performance and the thermodynamic performance of the gas turbine by using the available catalog data. The program is developed using the Engineering Equation Solver (EES) software. This application includes fundamental governing equations of the gas turbine and the thermodynamic theories. The target group of this program is MSc students whose follow the Turbomachinary course for their studies.

Limitations of the program

- Only considered open gas turbine cycle and has only one combustor.
- The acceptable fuel for the gas turbine is Methane.
- Initial temperature of the fuel is same as ambient temperature.

Pre-requites

- Theory on thermodynamics and turbomachinery
- Basic knowledge on open gas turbine
- Basic knowledge on Engineering Equation Solver (EES)

Software model

- File name of the software model is “GT.EES”

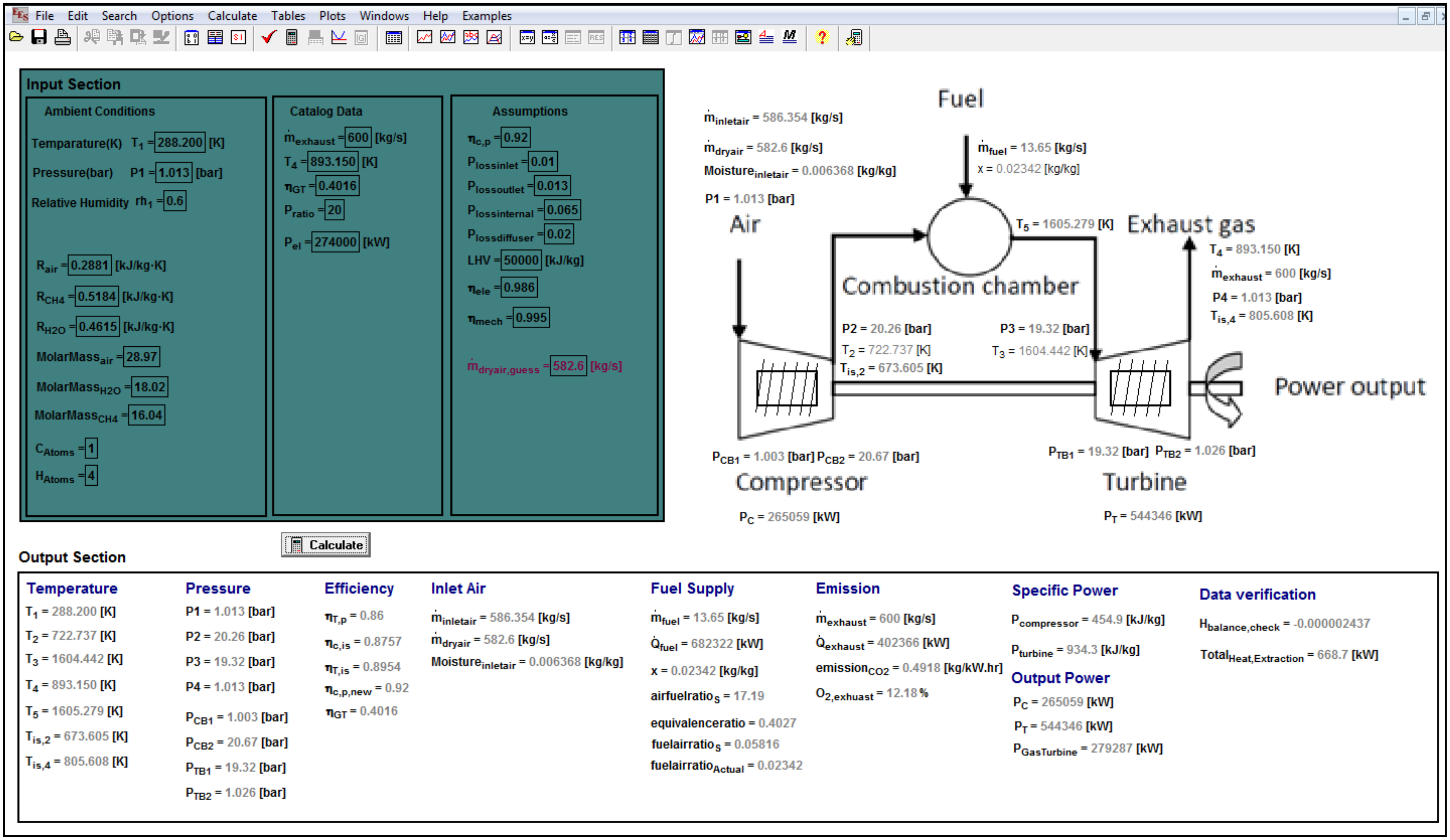



***Figure 22:*** *The Interface of the GT.EES software*

# 2 Software interface

There are 3 main sections in the software interface and they are;

- Input section
- Output section
- Gas turbine figure

## 2.1 Input section

Inside the input section there are three sub sections. The main section is catalog data input section. The user of the model has to enter the available gas turbine catalog data through this section. Those catalog data are published by the gas turbine manufactures. There are 05 main catalog data values for a gas turbine and those are as follows.

**Gas turbine electrical power output ($P_{el}$)**

The electrical power output at the electricity generator terminal can be considered as this parameter. The unit for this parameter is kW.

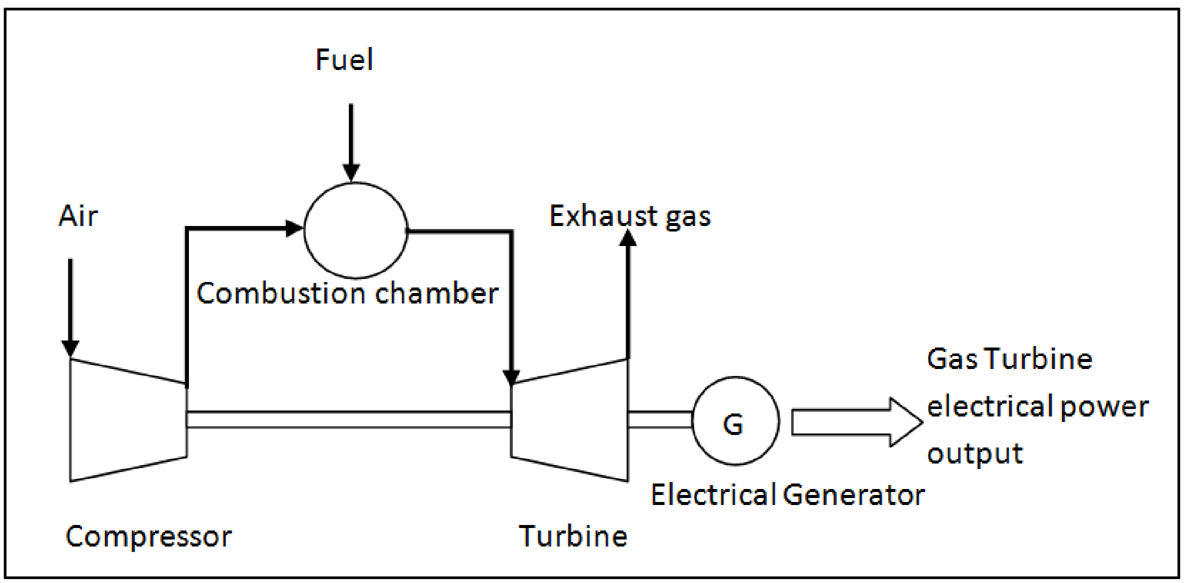



***Figure 23:*** *Typical open gas turbine*

**Overall gas turbine efficiency ($h_{GT}$)**

The overall gas turbine efficiency can be defined as follows.

$$Gas\ turbine\ efficiency = \frac{Gas\ turbine\ electrical\ power\ output}{Available\ energy\ in\ \ incoming\ fuel}$$ Eq. 1

Normally in the published data the gas turbine efficiency is available in percentage vise, but for this soft ware the user should not use the percentage value. User must use the decimal value of the polytropic efficiency. The value should be in between ‘0’ and ‘1’.

**Pressure ratio ($P_{ratio}$)**

Pressure ratio between compressor exit plenum and ambient can be consider as this value. This is a unit less parameter.

**Exhaust gas mass flow rate ($\dot{m}_{exhaust}$)**

This is the total exhaust mass flow rate of the gas turbine. This exhaust mass flow rate measured from the exit port of the gas turbine. This value is more important when the combined cycle point of view. The unit of this parameter is kg/s.

**Exhaust gas temperature ($T_4$)**

This is the temperature of the exhaust gas in the turbine exit. This temperature value is also important for the combined cycle operation. The amount of heat available in the exhaust gas is highly depended on this temperature. The user must use Kelvin temperature for the software.

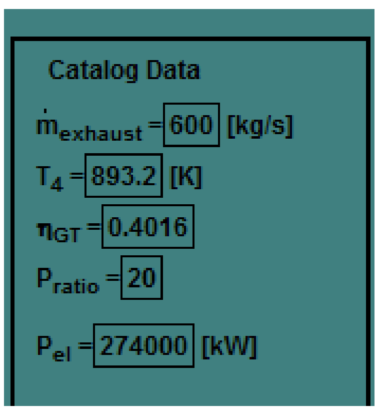



***Figure 24:*** *Catalog data input sub section*

The second input sub section is for the known parameters for the software model. Those are,

**Ambient temperature ($T_1$)**

This is the ambient temperature of the gas turbine. The calculation inside the model start from this parameter and all the calculation will depend on this variable. The user of the software must use the Kelvin temperature of the ambient temperature.

**Ambient pressure (P1)**

This is also an important parameter for the calculation. This is the pressure of the surrounding of the gas turbine. The unit for this parameter is bar.

**Relative humidity ($rh_1$)**

This is the relative humidity of the surrounding air of the gas turbine. During the calculation the water content of the incoming air, is calculated inside the software model. For that psychrometric calculation the relative humidity, ambient pressure and the ambient temperature are incorporated. This relative humidity is a unit less parameter.

**Gas constant of the air ($R_{air}$)**

The user should enter the gas constant of the dry air. Normally it is equal to 0.2881 kJ/kg.K.

**Gas constant of the Methane ($R_{CH4}$)**

The user should enter the gas constant of methane. Normally it is equal to 0.5184 kJ/kg.K.

**Gas constant of the water/steam ($R_{H2O}$)**

The user should enter the gas constant of the water. Normally it is equal to 0.4615 kJ/kg.K.

**Molar mass of the air ($MolarMass_{air}$)**

The user should enter the molar mass of dry air. Normally it is equal to 28.97 kg/kmol.

**Molar mass of the methane ($MolarMass_{CH4}$)**

The user should enter the molar mass of the methane. Normally it is equal to 18.02 kg/kmol.

**Molar mass of the water/steam ($MolarMass_{H2O}$)**

The user should enter the molar mass of the water. Normally it is equal to 16.04 kg/kmol.

**Number of carbon atoms in Methane ($C_{Atoms}$)**

The user should enter the number of carbon atoms in Methane. Normally the value is "1".

**Number of hydrogen atoms in Methane ($H_{Atoms}$)**

The user should enter the number of hydrogen atoms in Methane. Normally the value is "4"

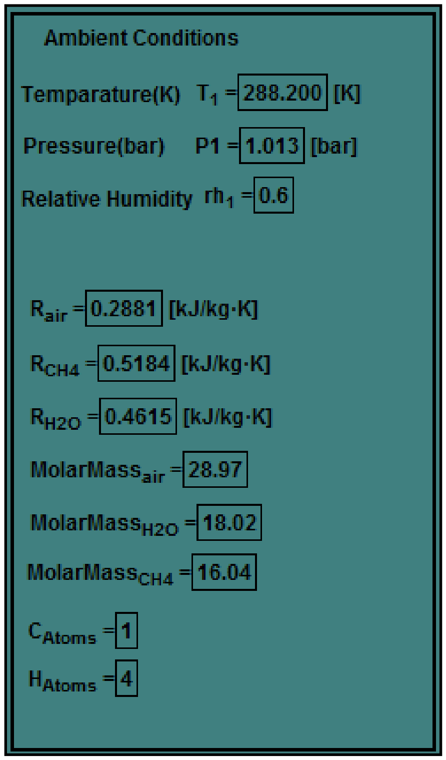



***Figure 25:*** *Known values input sub section*

The third input section is for the assumptions of the gas turbine calculation. Those are as follows.

**Polytropic efficiency of the compressor ($h_{C,P}$)**

This is the most important assumed value in this section, because without this assumed polytropic efficiency, the calculation is more complex. Normally for radial compressors the value is in between 80-88% and for the axial compressors the value is in between 89-92%. This parameter does not have any units. The user should not use the percentage value in here. User must use the decimal value of the polytropic efficiency. The value should be in between '0' and '1'.

**Inlet pressure loss ($P_{lossinlet}$)**

This value corresponds to the inlet pressure loss of the compressor with standard filters. This is a unit less parameter.

**Outlet pressure loss ($P_{lossoutlet}$)**

This value corresponds to the outlet pressure loss of the turbine. This is a unit less parameter.

**Internal pressure loss ($P_{lossinternal}$)**

This value corresponds to the internal pressure loss of the open gas turbine cycle. This is a unit less parameter.

**Diffuser pressure loss ($P_{lossdiffuser}$)**

This value corresponds to the diffuser pressure loss of the compressor. This is a unit less parameter.

**Lower Heating Value of the fuel (LHV)**

This is the LHV of the methane. The unit for this parameter is kJ/kg.

**Electrical efficiency ($h_{ele}$)**

This is the electrical efficiency of the generator. The user should not use the percentage value in here. User must use the decimal value of the polytropic efficiency. The value should be in between ‘0’ and ‘1’.

**Mechanical efficiency ($h_{mech}$)**

This is the mechanical efficiency of the gas turbine. The user should not use the percentage value in here. User must use the decimal value of the polytropic efficiency. The value should be in between ‘0’ and ‘1’. If the gas turbine directly coupled with the generator, the mechanical efficiency value should equal to ‘1’.

**Dry air mass flow rate ($\dot{m}_{dryair,guess}$)**

This is the guess value of the dry air mass flow rate. At the starting point of the calculation the user must enter this dry mass flow rate as a guess value. The user can enter some value which is slightly lower than the exhaust mass flow rate for this guess value. After entering that value the gas turbine calculation can be done by pressing the ‘calculate’ button. After finishing the calculation the results can be seen in the result section. Under that result section the calculated dry air mass flow rate can be seen. After getting the calculated dry air mass flow rate, the user must re-do the calculation by adding the calculated dry air mass flow rate as the guess dry air mass flow rate. After repeating the above step several times the correct dry air mass flow rate can be obtained. At that point the assumed dry air mass flow rate and the calculated dry air mass flow rate should be equal. The unit for this parameter is kg/s.

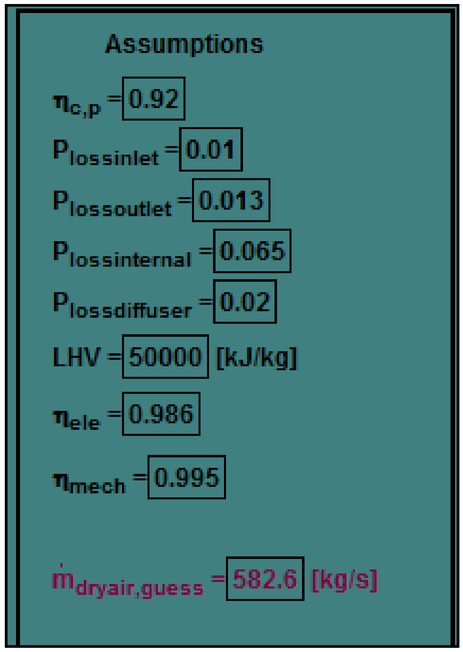



***Figure 26:*** *Assumptions input sub section*

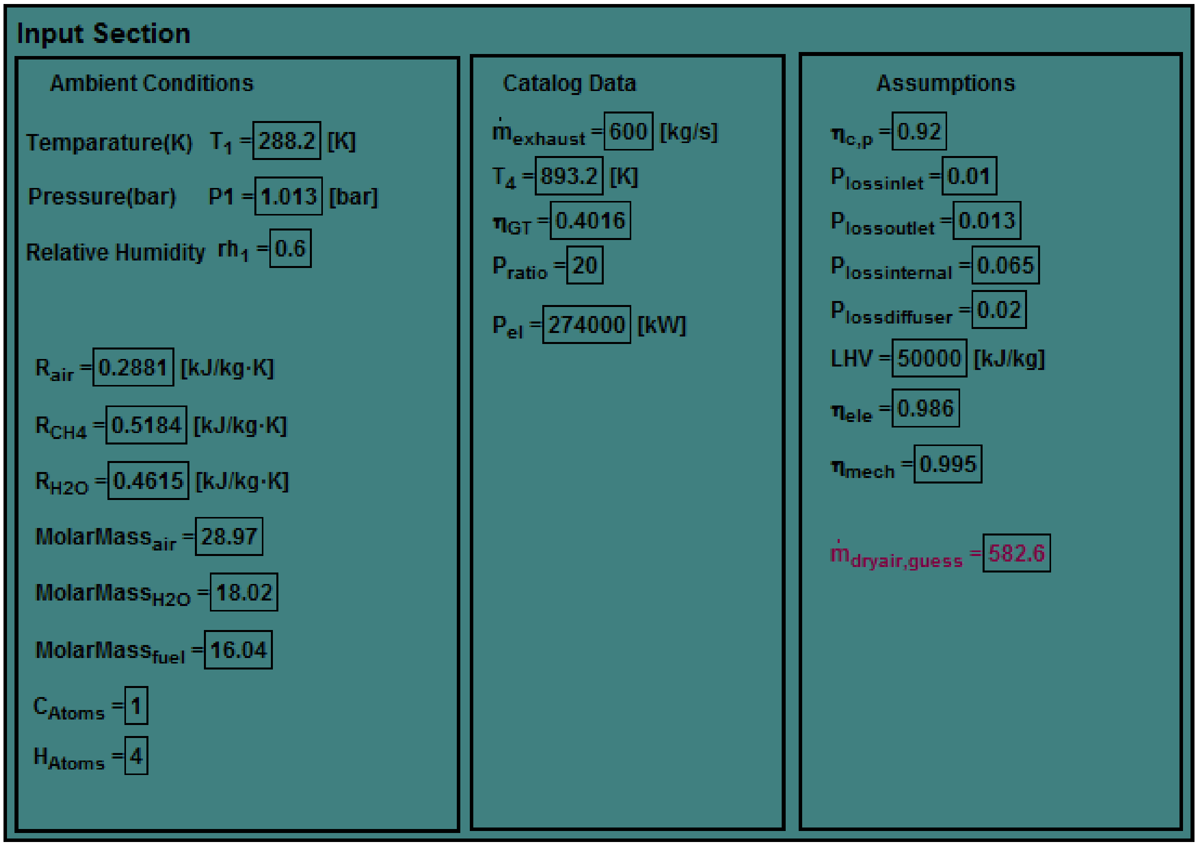



***Figure 27:*** *Input section*

## 2.2 Output section

Inside this section the user can see the results of the calculation.

### Compressor exit air temperature ($T_2$)

This is the outlet air temperature of the compressor. The unit for this parameter is Kelvin.

### Turbine inlet temperature ($T_3$)

This is the turbine inlet temperature. The unit of this parameter is Kelvin.

### Mixed combustor temperature ($T_5$)

The temperature after the combustion chamber can be considered as mixed combustor temperature. The unit of this parameter is Kelvin.

### Isentropic compressor exit temperature ($T_{is,2}$)

This is the isentropic exit temperature of the compressor. Unit for this parameter is Kelvin.

### Isentropic turbine exit temperature ($T_{is,4}$)

This is the isentropic exit temperature of the turbine. Unit for this parameter is Kelvin.

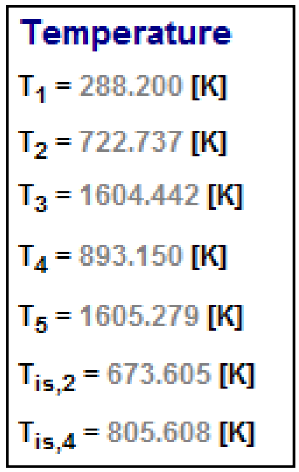



***Figure 28:*** *Typical temperature values in the output section*

### Pressure after the compressor (P2)

This is the pressure value after the compressor. Unit for this parameter is bar.

### Pressure after the combustion chamber (P3)

This is the pressure value after the combustion chamber. This same pressure is available at the turbine inlet. Unit for this parameter is bar.

### Pressure after the turbine (P4)

This is the pressure value after the turbine. This value is same as ambient pressure. Unit for this parameter is bar.

### Pressure at the compressor blading inlet ($P_{CB1}$)

This is the pressure value at the compressor blading inlet. Unit for this parameter is bar.

### Pressure at the compressor blading outlet ($P_{CB2}$)

This is the pressure value at the compressor blading outlet. Unit for this parameter is bar.

**Pressure at the turbine blading inlet ($P_{TB1}$)**

This is the pressure value at the turbine blading inlet. Unit for this parameter is bar.

**Pressure at the turbine blading outlet ($P_{TB2}$)**

This is the pressure value at the turbine blading outlet. Unit for this parameter is bar.

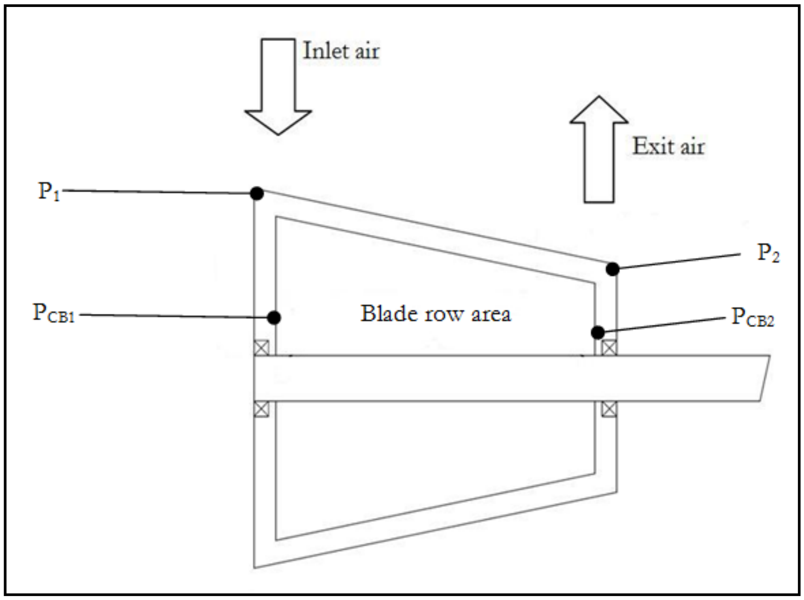



***Figure 29:*** *The pressure values in the compressor*

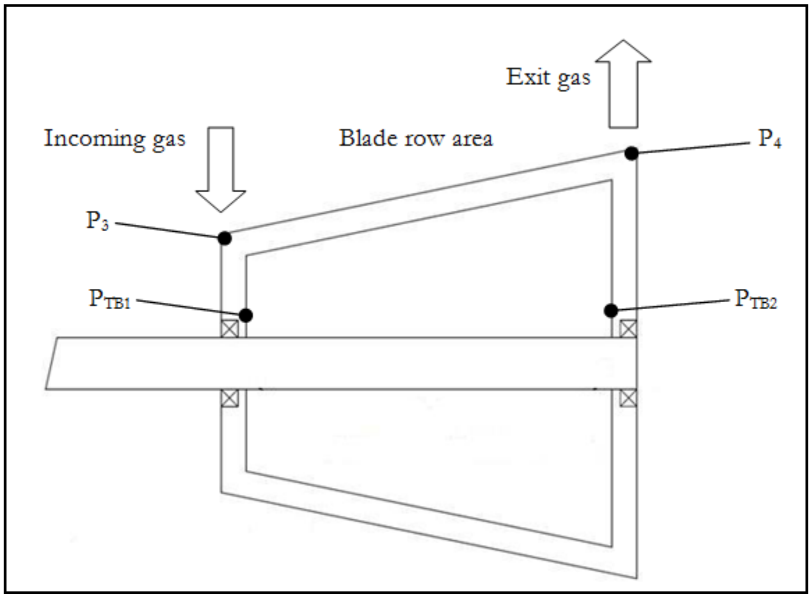



***Figure 30:*** *The pressure values in the turbine*

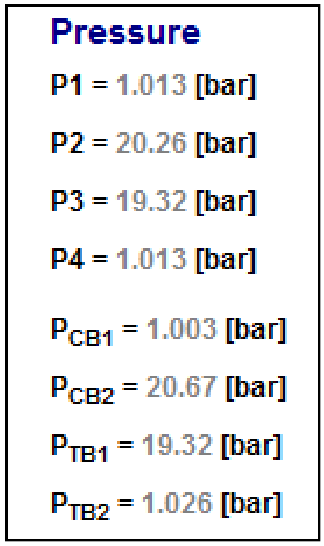



***Figure 31:*** *Typical pressure values in the output section*

**Calculated compressor polytropic blading efficiency ($h_{C,P,new}$)**

This is the new polytropic efficiency of the compressor. The main reason for adding this parameter is to re-calculate the same parameter, because the polytropic efficiency of the compressor was assumed in the initial stage. Therefore it is essential to re-calculate it. This parameter does not have any unit.

**Turbine polytropic blading efficiency ($h_{T,p}$)**

This is the polytropic efficiency of the turbine and this also does not have any unit.

**Compressor isentropic efficiency ($h_{c,is}$)**

This is the isentropic efficiency of the compressor. This parameter does not have any unit.

**Turbine isentropic efficiency ($h_{T,is}$)**

This is the isentropic efficiency of the turbine. This parameter does not have any unit.

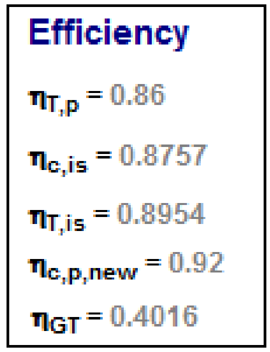



***Figure 32:*** *Typical efficiency values in the output section*

**Inlet air mass flow rate ($\dot{m}_{inletair}$)**

This is the inlet air mass flow rate of the gas turbine cycle. The unit of this parameter is kilogram of inlet air per second ($kg_{dryair}/s$).

**Dry air mass flow rate ($\dot{m}_{dryair}$)**

This is the dry air mass flow rate of the gas turbine cycle. This value is constant throughout of the open gas turbine cycle. The unit of this parameter is kilogram of dry air per second ($kg_{dryair}/s$).

**Water content of the incoming air ($Moisture_{inletair}$)**

This is the water content of the incoming air, to the compressor. The unit for this parameter is kilogram per kilogram of dry air ($kg/kg_{dryair}$).

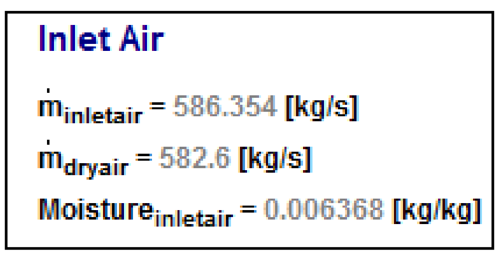



***Figure 33:*** *Typical inlet air values in the output section*

**Fuel rate ($\dot{m}_{fuel}$)**

This is the fuel rate of the gas turbine. The unit of this parameter is kilogram of fuel per second ($kg_{fuel}/s$).

**Fuel power ($Q_{fuel}$)**

This parameter gives the amount of power available in the fuel. The unit of this parameter is kW.

### Methane to dry air ratio (X)

This is the methane to dry air ratio for the gas turbine. The unit of this parameter is kilogram of fuel per kilogram of dry air ($kg_{fuel}/kg_{air}$).

### Stoichometric air to fuel ratio ($airfuelratio_S$)

This is the stoichometric air to fuel ratio. This is a unit less parameter.

### Equivalence ratio (equivalenceratio)

This is the equivalence ratio of the combustion process.

### Stoichometric fuel to air ratio ($fuelairratio_S$)

This is the stoichometric fuel to air ratio. This is a unit less parameter.

### Actual fuel to air ratio ($fuelairratio_{Actual}$)

This is the actual fuel to air ratio. This is a unit less parameter.

**Fuel Supply**

$\dot{m}_{fuel}$ = 13.65 [kg/s]

$\dot{Q}_{fuel}$ = 682322 [kW]

x = 0.02342 [kg/kg]

$airfuelratio_S$ = 17.19

equivalenceratio = 0.4027

$fuelairratio_S$ = 0.05816

$fuelairratio_{Actual}$ = 0.02342

***Figure 34:*** *Typical fuel supply values in the output section*

### Exhaust gas power ($Q_{exhaust}$)

This is the power available in the exhaust fuel gas. It is assumed that the entire exhaust gases cool down to the ambient temperature and during that process the released energy is calculated as this parameter. The unit for this parameter is kW.

### Carbon dioxide emission ($emission_{CO2}$)

This is an emission parameter and this indicates how much carbon dioxide release for the one kilowatt hour.

### Oxygen percentage in the exhaust gas ($O_{2,exhaust}$)

This is the available oxygen percentage in the exhaust gas.

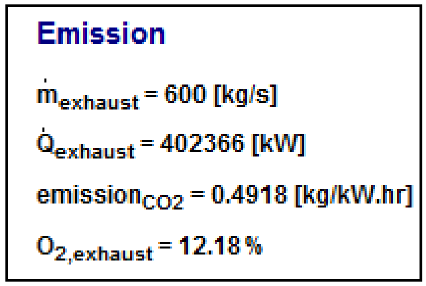



***Figure 35:*** *Typical emission values in the output section*

### Compressor specific power ($P_{compressor}$)

This is the specific power of the compressor. The unit of this parameter is kJ/kg.

### Turbine specific power ($P_{turbine}$)

This is the specific power of the turbine. The unit of this parameter is kJ/kg.

### Compressor power ($P_C$)

This is the power needed to run the compressor. The unit of this parameter is kW.

### Turbine power ($P_T$)

This is the power produce by the turbine. The unit of this parameter is kW.

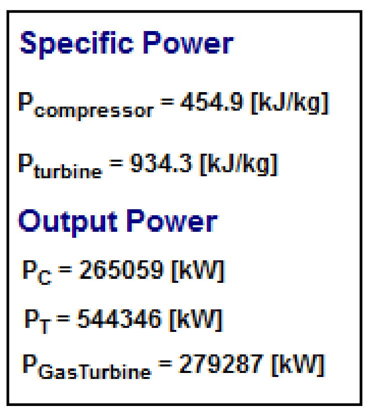



***Figure 36:*** *Typical power values in the output section*

### Heat balance check ($H_{balance,check}$)

This parameter uses for data verification purpose. This value should be nearly “0” for the successful set of results.

**Total heat extraction ($Total_{Heat,Extraction}$)**

This is the extracted heat inside the combustion chamber. This value should be positive value for the successful set of results.

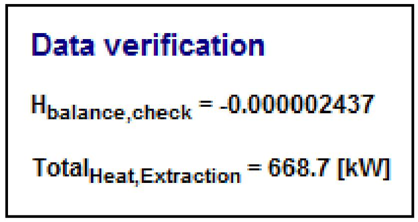



***Figure 37:*** *Data verification values in the output section*

## 3 Gas property data

It is essential to get the gas property data for the complete calculation of the open gas turbine. The Engineering Equation Solver (EES) itself has a collection of gas property data libraries. Those library data can use during the calculation. All the gases incorporated in the model were assumed as ideal gases. The three main gases in the gas turbine model are dry air, methane and water/steam.

**Air**

The thermodynamic properties of dry air in the EES internal library provide the values by assuming the dry air is an ideal gas. During the calculation the specific heat of the dry air calculates for the required temperatures. The values are available 60K to 3500K temperature range.

**Water/steam ($H_2O$)**

For the $H_2O$ also the ideal gas condition was assumed and the thermodynamic properties for gaseous water can be generate over the temperature range of 200K to 3500K from the EES internal libraries.

**Methane ($CH_4$)**

For the gas turbine calculation the fuel is assumed as Methane. The built in libraries of the EES provides the ideal thermodynamic properties of the Methane. The properties are valid for the temperature range of 250K to 3500K.

# 4 Program sequence

The open gas turbine software was modeled by using fundamental governing equations and the program sequences are as follows.

## 4.1 Water content of the incoming air

Typically the air in the atmosphere consist some amount of humid. Normally for the open gas turbine systems compressor sucks atmosphere air which is not dry. In other words, the incoming air consists of two components, which can define as dry air and water. Therefore, in the initial stage of the calculation the water content of the incoming air was calculated. Normally the water content of the air depends on the three main parameters and they are air temperature, air pressure and relative humidity of that air. By knowing the mentioned parameter, one can easily calculate the water content of the incoming air by simply using the appropriate function in the EES.

## 4.2 Compressor exit temperature

The exit temperature of the compressor ($T_2$) is a very important parameter. The equation 2 is used in the model in order to calculate the $T_2$.

$$\boldsymbol{\eta_C = \frac{\ln(P_{ratio})}{\int_{T_1}^{T_2}\left[\frac{C_{p(Air)}.T+X_{H2O}.C_{p(H2O)}.T}{(R_{Air}+X_{H2O}.R_{H2O}).T}\right]dT}}$$

**Eq. 2**

$T_1$= Inlet air temperature

$T_2$= Exit air temperature

$P_{ratio}$= Pressure ratio between compressor blading region

$X_{H2O}$= Water content of the inlet air to the compressor ($kg_{H2O}/kg_{Air}$)

$C_{p(Air)}$= Specific heat capacity of dry air

$C_{p(H20)}$= Specific heat capacity of water/steam

$R_{Air}$= Gas constant of dry air

$R_{H2O}$= Gas constant of water

T= Temperature (Variable for the integration)

The only unknown variable in the equation is $T_2$. By solving the equation 2, $T_2$ can be found.

## 4.3 Compressor specific power

After calculating the compressor exit temperature the specific power needed to run the compressor can be calculated. The constant pressure heat capacity of dry air and water/steam integrate from the compressor inlet temperature to the compressor exit temperature.

$$P_c = \int_{T_1}^{T_2} \left( C_{p(Air)}.T + X_{H2O}.C_{p(H2O)}.T \right) dT$$

Eq. 3

## 4.4 Pressure ratio correction

The open gas turbine cycle experiences several types of pressure losses inside it. They can be stated as follows,

- Inlet pressure loss
- Outlet pressure loss
- Internal pressure loss
- Diffuser loss

During the calculation of the model the realistic pressure drop values are utilized accordingly. Most of the time commercially available catalog indicates the pressure ratio of the ambient to compressor exit plenum. So the commercial pressure ratio should be corrected as per the equations used in the software. Basically there are two equations which incorporate with the pressure ratio. Those equations are compressor polytropic blading efficiency and turbine polytropic blading efficiency. Inside the mentioned equations the pressure ratio defines only for the blading region of that particular turbomachine. Therefore, the commercial pressure ratio subjected to corrected inside the software program. See the “calculation of Intermediate pressures and pressure ratios” section in the software code.

## 4.5 Turbine inlet temperature, Polytropic turbine efficiency, Methane to dry air ratio

In order to find the above parameters several numbers of equations are used in the software model. Those equations are as follows,

The equation 4 uses for calculate the turbine polytropic blading efficiency.

$$\eta_T = \frac{\int_{T_4}^{T_3} \left[\frac{C_{p(Air)}.T + X_{H2O}.C_{p(H2O)}.T + X.C_{p(CH4)}.T}{(R_{Air} + X_{H2O}.R_{H2O} + X.R_{CH4}).T}\right] dT}{ln[P_{ratio}]}$$

**Eq. 4**

$T_3$= Turbine inlet temperature

$T_4$= Turbine exit temperature

$P_{ratio}$= Pressure ratio between turbine blading region

$X_{H2O}$= Water content of the inlet air to the compressor ($kg_{H2O}/kg_{Air}$)

$C_{p(Air)}$= Specific heat capacity of dry air

$C_{p(H20)}$= Specific heat capacity of water/steam

$C_{p(CH4)}$= Specific heat capacity of methane

$R_{Air}$= Gas constant of dry air

$R_{H2O}$= Gas constant of water

$R_{CH4}$= Gas constant of methane

T= Temperature (Variable for the integration)

X= Methane to dry air ratio

The equation 5 uses for calculation of the dry air mass flow rate of the gas turbine.

$$m_{dry\ air} = \frac{m_{exhaust}}{(1 + X_{H2O} + X)}$$

**Eq. 5**

The specific power of the turbine can be calculated by using the equation 6. The constant pressure heat capacity of the dry air, water and methane integrate from turbine inlet temperature to turbine exit temperature.

$$\mathbf{P_{turbine}} = \int_{T_4}^{T_3} \left(C_{p(Air)}.T + X_{H2O}.C_{p(H2O)}.T + X.C_{p(CH4)}.T\right)dT$$

**Eq. 6**

By subtracting the specific compressor power (equation 3) from the specific turbine power (equation 6) the specific gas turbine power can be calculated. It is indicated in the equation 7.

$$P_{GT} = P_{turbine} - P_{compressor}$$

**Eq. 7**

Electrical output of the gas turbine can be calculated by using the equation 8.

$$P_{el} = \eta_{el}.\eta_{mech}.m_{dry\ air}.P_{GT}$$

**Eq. 8**

By using the equation 9 the methane to dry air ratio can be calculate.

$$X = \frac{\int_{T_2}^{T_5}(C_{p(Air)}.T + X_{H2O}.C_{p(H2O)}.T)dT}{LHV - \int_{T_1}^{T_5}(C_{p(CH4)}.T)dT}$$

**Eq. 9**

$T_5$= Mixed combustor temperature

LHV=Lower heating value

By solving the above equations during the calculation the polytropic efficiency of the turbine, dry air mass flow rate, the turbine inlet temperature and the methane to dry air ratio can be calculated. Those can be seen in the source code of the EES model and required comments are also available for better understanding.

# 5 Basic information about EES

### Equations Window

All the equations that are used in the program are entered in this window and it is very much similar to word processor. When the EES launched for the first time, the equation window opens automatically. All the programming arguments, functions could be entered and some editing functions such as cut, copy, paste, undo etc are also available inside equation window.

There is another method to access this equation window. By clicking the equation window button in the main menu user can open it.

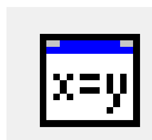



***Figure 38:*** *Equation window button*

### Formatted Equation Window

The equations entered in the equation window are in the word processor format and they are in a single line. Therefore the user of the software cannot see the mathematical notations in clear manner. To see the mathematical notation of the equations, the formatted equation window can be used.

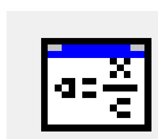


***Figure 39:*** *Formatted equation window button*

### Check command

After entering the equation in the equation window the user can check the equations for errors. The number of equations, the number of variables and the syntax errors are checked during the checking period. After finishing the check, the results are displayed in a separate window.

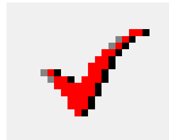


***Figure 40:*** *Check command button*

**Solve command**

After executing the solve command, the EES software check the syntax errors in the equations. If there are not any syntax errors and the number of variables and number of equations are equal, then the compiling of equations will be started. Then the solution algorithm is started and the information dialog box opens to indicate the progress of the calculation. After the calculations are completed the information dialog box holds the final status of the calculation. After clicking the OK button in the information box, the user can access the solutions of the calculation.

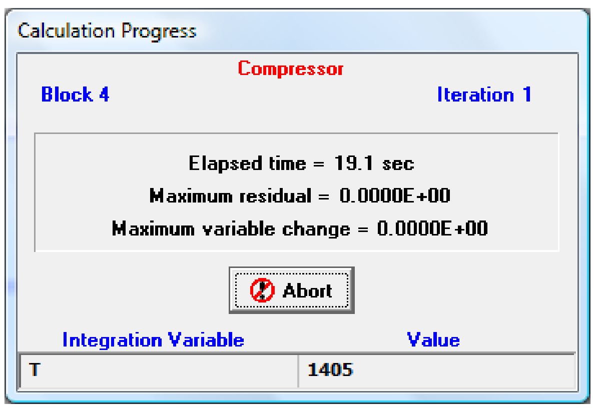



***Figure 41:*** *Status of the information box, during the calculation*

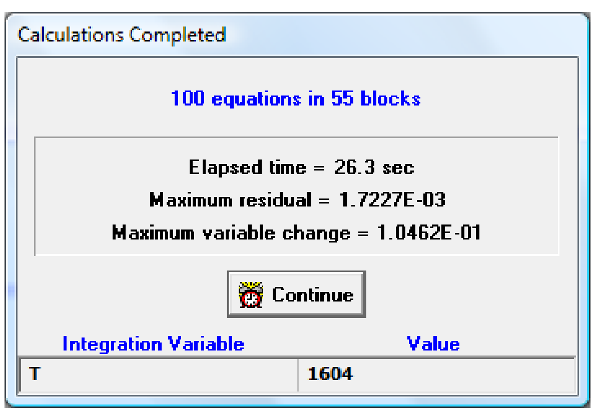



***Figure 42:*** *Status of the information box, after the calculation*

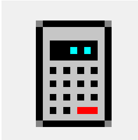


***Figure 43:*** *Solve button*

**Solution Window**

After the compiling process completed all the results are shown in the solution window. This will appear in front of all the windows. This solution window can access only after the compiling of the program. If the EES cannot solve the program up to the end point the compiling process terminates and the last iteration results will be displayed in the solution window. The errors in the unit system are also displayed in the solution window.

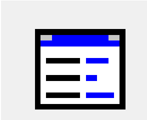


***Figure 44:*** *Solution window button*

**Diagram Window**

Diagram window can be used to display some diagrams of the program. This also can be used as a graphical user interface to give input to the program and display the output of the program.

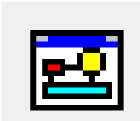


***Figure 45:*** *Diagram window button*

**Thermodynamic property functions used in the EES**

Other than the mathematical functions in the EES there are thermophysical functions available in the internal libraries. EES provides list of thermophysical property data for most of the available fluids. The fluids inside the EES are group in to three categories and those are real fluid, ideal fluid and brines. For the gas turbine calculation ideal fluids were taken into account.

**Instructions:**

Before going to do the exercise 01 read the "Gas Turbine Thermodynamic and Performance Analysis Methods Using Available Catalog Data – User Manual & Exercises"

**Exercise 01:**

The following table gives five main catalog data for well-known gas turbine.

| Model | Power MW | Efficiency | Pressure Ratio | Flow kg/sec | Exhaust Temp |
|---|---|---|---|---|---|
| **Siemens SGT6-8000H** | 274 | 40.16% | 20 | 600 | $620^0$C |

Use the "GT.EES" program and calculate the unknown pressure values and temperature values inside the gas turbine cycle. Hence plot the TS diagram for the gas turbine.

Calculate the compressor power and turbine power by using TS diagram and compare the results with the EES results.

Hints:

| | |
|---|---|
| Ambient temperature | $15^0$C |
| Ambient pressure | 1.013bar |
| Relative humidity | 0.6 |
| Compressor polytropic blading efficiency | 0.92 |
| Inlet pressure loss | 0.01 |
| Outlet pressure loss | 0.013 |
| Internal pressure loss | 0.065 |
| Diffuser pressure loss | 0.02 |
| Electrical efficiency | 0.986 |
| Mechanical efficiency | 0.995 |

**Exercise 02**

The following table gives some catalog data sets for the commercially available gas turbines. The data sets were extracted from the GTW 2006 catalog section "Simple cycle power plants", page 70 & 75

| Model | Year | ISO Base Rating kW | Heat Rate Btu/kWh | Efficiency | Pressure Ratio | Flow lb/sec | Turbine Speed rpm | Exhaust Temp |
|---|---|---|---|---|---|---|---|---|
| **Rolls Royce Trent 60 DLE** | 1996 | 51504 | 8104 | 42.1% | 33.0 | 334.0 | 3000 | $832^0$F |
| **Rolls Royce Trent 60 DLE** | 1996 | 51685 | 8138 | 41.9% | 34.0 | 341.0 | 3600 | $825^0$F |
| **General Electric LM6000PC** | 1992 | 43471 | 8112 | 42.0% | 26.1 | 282.1 | 3600 | $824^0$F |
| **General Electric LM6000PC Sprint** | 1997 | 50080 | 8434 | 40.5% | 31.3 | 299.1 | 3600 | $819^0$F |

Hints

- These kinds of aero-derivative engines may have a power turbine rotating at grid frequency. Therefore no gear loss is to be considered.
- Both engines have modern axial compressor blading.

Use the given program "GT.EES" to calculate the below parameters.

1. Mixed turbine inlet temp (°C)
2. Polytropic turbine blading efficiency
3. Draw the TS diagram with calculated pressure values, temperature values

# Appendix C: Answers for the Exercises

**Exercise 01:**

The following table gives five main catalog data for well-known gas turbine.

| Model | Power MW | Efficiency | Pressure Ratio | Flow kg/sec | Exhaust Temp |
|---|---|---|---|---|---|
| **Siemens SGT6-8000H** | 274 | 40.16% | 20 | 600 | $620^0$C |

Use the "GT.EES" program and calculate the unknown pressure values and temperature values inside the gas turbine cycle. Hence plot the TS diagram for the gas turbine.

Calculate the compressor power and turbine power by using TS diagram and compare the results with the EES results.

Hints:

| | |
|---|---|
| Ambient temperature | $15^0$C |
| Ambient pressure | 1.013bar |
| Relative humidity | 0.6 |
| Compressor polytropic blading efficiency | 0.92 |
| Inlet pressure loss | 0.01 |
| Outlet pressure loss | 0.013 |
| Internal pressure loss | 0.065 |
| Diffuser pressure loss | 0.02 |
| Electrical efficiency | 0.986 |
| Mechanical efficiency | 0.995 |

**Answer:**

| Temperature (K) | | Pressure (bar) | | Efficiency | | Fuel Supply | | Emission | |
|---|---|---|---|---|---|---|---|---|---|
| $T_1$ | 288.200 | $P_1$ | 1.013 | $h_{T,p}$ | 0.86 | $\dot{m}_{fuel}$ | 13.65 kg/s | $\dot{m}_{exhaust}$ | 600 kg/s |
| $T_2$ | 722.737 | $P_2$ | 20.260 | $h_{C,is}$ | 0.8757 | $Q_{fuel}$ | 682322 kW | $Q_{exhaust}$ | 402366 kW |
| $T_3$ | 1604.442 | $P_3$ | 19.320 | $h_{T,is}$ | 0.8954 | X | 0.02342 kg/kg | $emission_{CO2}$ | 0.4918 kg/kW.hr |
| $T_4$ | 893.150 | $P_4$ | 1.013 | $h_{C,p,new}$ | 0.92 | $airfuelratio_s$ | 17.19 | $O_{2,exhaust}$ | 12.18% |
| $T_5$ | 1605.279 | $P_{CB1}$ | 1.003 | $h_{GT}$ | 0.4016 | equivalenceratio | 0.4027 | | |
| $T_{is,2}$ | 673.605 | $P_{CB2}$ | 20.670 | | | $fuelairratio_s$ | 0.05816 | | |
| $T_{is,4}$ | 805.608 | $P_{TB1}$ | 19.320 | | | $fuelairratio_{Actual}$ | 0.02342 | | |
| | | $P_{TB2}$ | 1.026 | | | | | | |

| Inlet Air | | Specific Power | | Output Power | | Data Verification | |
|---|---|---|---|---|---|---|---|
| $\dot{m}_{inletair}$ | 586.354 kg/s | $P_{compressor}$ | 454.9 kJ/kg | $P_C$ | 265059 kW | $H_{balance,check}$ | -0.000002437 |
| $\dot{m}_{dryair}$ | 582.6 kg/s | $P_{turbine}$ | 934.3 kJ/kg | $P_T$ | 544346 kW | $Total_{Heat,Extraction}$ | 668.7 kW |
| $Moisture_{inletair}$ | 0.006368 kg/kg | | | $P_{GasTurbine}$ | 279287 kW | | |

**Exercise 02**

The following table gives some catalog data sets for the commercially available gas turbines. The data sets were extracted from the GTW 2006 catalog section "Simple cycle power plants", page 70 & 75

| Model | Year | ISO Base Rating kW | Heat Rate Btu/kWh | Efficiency | Pressure Ratio | Flow lb/sec | Turbine Speed rpm | Exhaust Temp |
|---|---|---|---|---|---|---|---|---|
| **Rolls Royce Trent 60 DLE** | 1996 | 51504 | 8104 | 42.1% | 33.0 | 334.0 | 3000 | $832^0$F |
| **Rolls Royce Trent 60 DLE** | 1996 | 51685 | 8138 | 41.9% | 34.0 | 341.0 | 3600 | $825^0$F |
| **General Electric LM6000PC** | 1992 | 43471 | 8112 | 42.0% | 26.1 | 282.1 | 3600 | $824^0$F |
| **General Electric LM6000PC Sprint** | 1997 | 50080 | 8434 | 40.5% | 31.3 | 299.1 | 3600 | $819^0$F |

Hints

- These kinds of aero-derivative engines may have a power turbine rotating at grid frequency. Therefore no gear loss is to be considered.
- Both engines have modern axial compressor blading.

Use the given program "GT.EES" to calculate the below parameters.

4. Mixed turbine inlet temp (°C)
5. Polytropic turbine blading efficiency
6. Draw the TS diagram with calculated pressure values, temperature values

**Answer:**

**Rolls Royce Trent 60 DLE (3000 rpm)**

**Input Section**

**Ambient Conditions**

Temparature(K) $T_1$ = 288.200 [K]

Pressure(bar) P1 = 1.013 [bar]

Relative Humidity $rh_1$ = 0.6

$R_{air}$ = 0.2881 [kJ/kg·K]

$R_{CH4}$ = 0.5184 [kJ/kg·K]

$R_{H2O}$ = 0.4615 [kJ/kg·K]

$MolarMass_{air}$ = 28.97

$MolarMass_{H2O}$ = 18.02

$MolarMass_{CH4}$ = 16.04

$C_{Atoms}$ = 1

$H_{Atoms}$ = 4

**Catalog Data**

$\dot{m}_{exhaust}$ = 151.499 [kg/s]

$T_4$ = 717.594 [K]

$\eta_{GT}$ = 0.421

$P_{ratio}$ = 33

$P_{el}$ = 51504 [kW]

**Assumptions**

$\eta_{c,p}$ = 0.92

$P_{lossinlet}$ = 0.004

$P_{lossoutlet}$ = 0.013

$P_{lossinternal}$ = 0.065

$P_{lossdiffuser}$ = 0

LHV = 50000 [kJ/kg]

$\eta_{ele}$ = 0.986

$\eta_{mech}$ = 1

$\dot{m}_{dryair,guess}$ = 148.109 [kg/s]

| Temperature (K) | | Pressure (bar) | | Efficiency | | Fuel Supply | | Emission | |
|---|---|---|---|---|---|---|---|---|---|
| $T_1$ | 288.200 | $P_1$ | 1.013 | $h_{T,p}$ | 0.8571 | $\dot{m}_{fuel}$ | 2.447 kg/s | $\dot{m}_{exhaust}$ | 151.499 kg/s |
| $T_2$ | 827.903 | $P_2$ | 33.43 | $h_{C,is}$ | 0.8822 | $Q_{fuel}$ | 122337 kW | $Q_{exhaust}$ | 69630 kW |
| $T_3$ | 1459.542 | $P_3$ | 31.26 | $h_{T,is}$ | 0.9037 | X | 0.01652 kg/kg | $emission_{CO2}$ | 0.4691 kg/kW.hr |
| $T_4$ | 717.594 | $P_4$ | 1.013 | $h_{C,p,new}$ | 0.92 | $airfuelratio_s$ | 17.19 | $O_{2,exhaust}$ | 14.7% |
| $T_5$ | 1461.966 | $P_{CB1}$ | 1.009 | $h_{GT}$ | 0.421 | equivalenceratio | 0.284 | | |
| $T_{is,2}$ | 770.569 | $P_{CB2}$ | 33.43 | | | $fuelairratio_s$ | 0.05816 | | |
| $T_{is,4}$ | 630.874 | $P_{TB1}$ | 31.26 | | | $fuelairratio_{Actual}$ | 0.01652 | | |
| | | $P_{TB2}$ | 1.026 | | | | | | |

| Inlet Air | | Specific Power | | Output Power | | Data Verification | |
|---|---|---|---|---|---|---|---|
| $\dot{m}_{inletair}$ | 149.052 kg/s | $P_{compressor}$ | 571.3 kJ/kg | $P_C$ | 84609 kW | $H_{balance,check}$ | 0.000000148 |
| $\dot{m}_{dryair}$ | 148.109 kg/s | $P_{turbine}$ | 923.9 kJ/kg | $P_T$ | 136844 kW | $Total_{Heat,Extraction}$ | 472.6 kW |
| $Moisture_{inletair}$ | 0.006368 kg/kg | | | $P_{GasTurbine}$ | 52235 kW | | |

**Rolls Royce Trent 60 DLE (3600 rpm)**

**Input Section**

**Ambient Conditions**

Temparature(K) $T_1$ = 288.200 [K]

Pressure(bar) P1 = 1.013 [bar]

Relative Humidity $rh_1$ = 0.6

$R_{air}$ = 0.2881 [kJ/kg·K]

$R_{CH4}$ = 0.5184 [kJ/kg·K]

$R_{H2O}$ = 0.4615 [kJ/kg·K]

$MolarMass_{air}$ = 28.97

$MolarMass_{H2O}$ = 18.02

$MolarMass_{CH4}$ = 16.04

$C_{Atoms}$ = 1

$H_{Atoms}$ = 4

**Catalog Data**

$\dot{m}_{exhaust}$ = 154.674 [kg/s]

$T_4$ = 713.706 [K]

$\eta_{GT}$ = 0.419

$P_{ratio}$ = 34

$P_{el}$ = 51685 [kW]

**Assumptions**

$\eta_{c,p}$ = 0.92

$P_{lossinlet}$ = 0.004

$P_{lossoutlet}$ = 0.013

$P_{lossinternal}$ = 0.065

$P_{lossdiffuser}$ = 0

LHV = 50000 [kJ/kg]

$\eta_{ele}$ = 0.986

$\eta_{mech}$ = 1

$\dot{m}_{dryair,guess}$ = 151.244 [kg/s]

| Temperature (K) | | Pressure (bar) | | Efficiency | | Fuel Supply | | Emission | |
|---|---|---|---|---|---|---|---|---|---|
| $T_1$ | 288.200 | $P_1$ | 1.013 | $h_{T,p}$ | 0.8541 | $\dot{m}_{fuel}$ | 2.467 kg/s | $\dot{m}_{exhaust}$ | 154.674 kg/s |
| $T_2$ | 834.916 | $P_2$ | 34.44 | $h_{C,is}$ | 0.8819 | $Q_{fuel}$ | 123353 kW | $Q_{exhaust}$ | 703386 kW |
| $T_3$ | 1458.028 | $P_3$ | 32.2 | $h_{T,is}$ | 0.902 | X | 0.01631 kg/kg | $emission_{CO2}$ | 0.4714 kg/kW.hr |
| $T_4$ | 713.706 | $P_4$ | 1.013 | $h_{C,p,new}$ | 0.92 | $airfuelratio_s$ | 17.19 | $O_{2,exhaust}$ | 14.78% |
| $T_5$ | 1460.784 | $P_{CB1}$ | 1.009 | $h_{GT}$ | 0.419 | equivalenceratio | 0.2805 | | |
| $T_{is,2}$ | 776.705 | $P_{CB2}$ | 34.44 | | | $fuelairratio_s$ | 0.05816 | | |
| $T_{is,4}$ | 624.915 | $P_{TB1}$ | 32.2 | | | $fuelairratio_{Actual}$ | 0.01631 | | |
| | | $P_{TB2}$ | 1.026 | | | | | | |

| Inlet Air | | Specific Power | | Output Power | | Data Verification | |
|---|---|---|---|---|---|---|---|
| $\dot{m}_{inletair}$ | 152.207 kg/s | $P_{compressor}$ | 579.1 kJ/kg | $P_C$ | 87587 kW | $H_{balance,check}$ | 0.000000177 |
| $\dot{m}_{dryair}$ | 151.244 kg/s | $P_{turbine}$ | 925.7 kJ/kg | $P_T$ | 140006 kW | $Total_{Heat,Extraction}$ | 668.7 kW |
| $Moisture_{inletair}$ | 0.006368 kg/kg | | | $P_{GasTurbine}$ | 52419 kW | | |

**General Electric LM6000PC**

## Input Section

### Ambient Conditions

Temparature(K) $T_1$ = 288.200 [K]

Pressure(bar) P1 = 1.013 [bar]

Relative Humidity $rh_1$ = 0.6

$R_{air}$ = 0.2881 [kJ/kg·K]

$R_{CH4}$ = 0.5184 [kJ/kg·K]

$R_{H2O}$ = 0.4615 [kJ/kg·K]

$MolarMass_{air}$ = 28.97

$MolarMass_{H2O}$ = 18.02

$MolarMass_{CH4}$ = 16.04

$C_{Atoms}$ = 1

$H_{Atoms}$ = 4

### Catalog Data

$\dot{m}_{exhaust}$ = 130.000 [kg/s]

$T_4$ = 713.150 [K]

$\eta_{GT}$ = 0.42

$P_{ratio}$ = 26.1

$P_{el}$ = 43471 [kW]

### Assumptions

$\eta_{c,p}$ = 0.92

$P_{lossinlet}$ = 0.004

$P_{lossoutlet}$ = 0.013

$P_{lossinternal}$ = 0.065

$P_{lossdiffuser}$ = 0

LHV = 50000 [kJ/kg]

$\eta_{ele}$ = 0.986

$\eta_{mech}$ = 1

$\dot{m}_{dryair,guess}$ = 127.120 [kg/s]

Note: Used 2% increased exhaust mass flow rate in order to get the reasonable data

| Temperature (K) | | Pressure (bar) | | Efficiency | | Fuel Supply | | Emission | |
|---|---|---|---|---|---|---|---|---|---|
| $T_1$ | 288.200 | $P_1$ | 1.013 | $h_{T,p}$ | 0.8764 | $\dot{m}_{fuel}$ | 2.07 kg/s | $\dot{m}_{exhaust}$ | 130 kg/s |
| $T_2$ | 774.475 | $P_2$ | 26.44 | $h_{C,is}$ | 0.8847 | $Q_{fuel}$ | 103503 kW | $Q_{exhaust}$ | 59074 kW |
| $T_3$ | 1406.460 | $P_3$ | 24.72 | $h_{T,is}$ | 0.9149 | X | 0.01628 kg/kg | $emission_{CO2}$ | 0.4703kg/kW.hr |
| $T_4$ | 713.150 | $P_4$ | 1.013 | $h_{C,p,new}$ | 0.92 | $airfuelratio_s$ | 17.19 | $O_{2,exhaust}$ | 14.79% |
| $T_5$ | 1408.509 | $P_{CB1}$ | 1.009 | $h_{GT}$ | 0.42 | equivalenceratio | 0.28 | | |
| $T_{is,2}$ | 723.755 | $P_{CB2}$ | 26.44 | | | $fuelairratio_s$ | 0.05816 | | |
| $T_{is,4}$ | 642.732 | $P_{TB1}$ | 24.72 | | | $fuelairratio_{Actual}$ | 0.01628 | | |
| | | $P_{TB2}$ | 1.026 | | | | | | |

| Inlet Air | | Specific Power | | Output Power | | Data Verification | |
|---|---|---|---|---|---|---|---|
| $\dot{m}_{inletair}$ | 127.930 kg/s | $P_{compressor}$ | 511.8 kJ/kg | $P_C$ | 65064 kW | $H_{balance,check}$ | 0.000000359 |
| $\dot{m}_{dryair}$ | 127.120 kg/s | $P_{turbine}$ | 858.7 kJ/kg | $P_T$ | 109152 kW | $Total_{Heat,Extraction}$ | 340.6 kW |
| $Moisture_{inletair}$ | 0.006368 kg/kg | | | $P_{GasTurbine}$ | 44088 kW | | |

## General Electric LM6000PC Sprint

### Input Section

#### Ambient Conditions

Temparature(K) $T_1$ = 288.200 [K]

Pressure(bar) P1 = 1.013 [bar]

Relative Humidity $rh_1$ = 0.6

$R_{air}$ = 0.2881 [kJ/kg·K]

$R_{CH4}$ = 0.5184 [kJ/kg·K]

$R_{H2O}$ = 0.4615 [kJ/kg·K]

$MolarMass_{air}$ = 28.97

$MolarMass_{H2O}$ = 18.02

$MolarMass_{CH4}$ = 16.04

$C_{Atoms}$ = 1

$H_{Atoms}$ = 4

#### Catalog Data

$\dot{m}_{exhaust}$ = 135.669 [kg/s]

$T_4$ = 710.372 [K]

$\eta_{GT}$ = 0.405

$P_{ratio}$ = 31.3

$P_{el}$ = 50080 [kW]

#### Assumptions

$\eta_{c,p}$ = 0.92

$P_{lossinlet}$ = 0.004

$P_{lossoutlet}$ = 0.013

$P_{lossinternal}$ = 0.065

$P_{lossdiffuser}$ = 0

LHV = 50000 [kJ/kg]

$\eta_{ele}$ = 0.986

$\eta_{mech}$ = 1

$\dot{m}_{dryair,guess}$ = 132.353 [kg/s]

The GE LM6000 Sprint gas turbine uses inter-cooling design in order to increase the mass flow rate of the inlet air. The system injects atomized water at both the low pressure and high pressure compressor inlet plenums. Therefore the results from the GT.EES cannot consider as 100% correct due to the GT.EES is not modeled for compressor inter-cooling.

| Temperature (K) | | Pressure (bar) | | Efficiency | | Fuel Supply | | Emission | |
|---|---|---|---|---|---|---|---|---|---|
| $T_1$ | 288.200 | $P_1$ | 1.013 | $h_{T,p}$ | 0.8869 | $\dot{m}_{fuel}$ | 2.473 kg/s | $\dot{m}_{exhaust}$ | 135.669 kg/s |
| $T_2$ | 815.597 | $P_2$ | 31.71 | $h_{C,is}$ | 0.8828 | $Q_{fuel}$ | 123654 kW | $Q_{exhaust}$ | 61475kW |
| $T_3$ | 1460.540 | $P_3$ | 29.65 | $h_{T,is}$ | 0.9239 | X | 0.01869 kg/kg | $emission_{CO2}$ | 0.4877 kg/kW.hr |
| $T_4$ | 710.372 | $P_4$ | 1.013 | $h_{C,p,new}$ | 0.92 | $airfuelratio_s$ | 17.19 | $O_{2,exhaust}$ | 13.9% |
| $T_5$ | 1525.105 | $P_{CB1}$ | 1.009 | $h_{GT}$ | 0.405 | equivalenceratio | 0.3213 | | |
| $T_{is,2}$ | 759.799 | $P_{CB2}$ | 31.71 | | | $fuelairratio_s$ | 0.05816 | | |
| $T_{is,4}$ | 642.648 | $P_{TB1}$ | 29.65 | | | $fuelairratio_{Actual}$ | 0.01869 | | |
| | | $P_{TB2}$ | 1.026 | | | | | | |

| Inlet Air | | Specific Power | | Output Power | | Data Verification | |
|---|---|---|---|---|---|---|---|
| $\dot{m}_{inletair}$ | 133.196 kg/s | $P_{compressor}$ | 557.5 kJ/kg | $P_C$ | 73789 kW | $H_{balance,check}$ | -0.000000497 |
| $\dot{m}_{dryair}$ | 132.353 kg/s | $P_{turbine}$ | 941.3 kJ/kg | $P_T$ | 124580 kW | $Total_{Heat,Extraction}$ | 11389 kW |
| $Moisture_{inletair}$ | 0.006368 kg/kg | | | $P_{GasTurbine}$ | 50791 kW | | |