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- TOPIC-

Performance improvement of Gas Turbine using an Earth Air Heat Exchanger

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Dedications

Although more beautiful than being able to share the best moments of your life with the objects you love.

At the end of my studies, I am very happy to dedicate this humble work to:

My dear parents

My sisters

My brother

My friend Amira and her family

My partner in this work

BOUTERFIF Asma

Dedications

Not every letter can find the right words ... every word can't find gratitude, love, respect, recognition ... so, it's just that.

To my dear parents, for all their sacrifices, their love, their tenderness, their support and their prayers throughout my studies. Thank you for all the noble values.

To my dear brothers and my sister for their moral support, this work is a testimony of my attachment and my love.

To all my family for their support throughout my university career

To my partner in this work for her sacrifices and encouragement.

Thanks for always being there for me.

LAKAS Anfal

Thanking

First of all, we thank **ALLAH,** the almighty, for giving us the courage and the will to accomplish this research work.

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Nomenclature

Symbol	Designation	Unit
T_{soil}	Soil temperature	$°C$
T_a	Air temperature inside the exchanger	$°C$
T_{ae}	Air temperature at the outlet of the exchanger	$°C$
T_{a1}	Air temperature at the outlet of the vertical exchanger 1	$°C$
T_{a2}	Air temperature at the outlet of the horizontal exchanger 2	$°C$
T_i	Initial temperature inside the soil	$°C$
S	Exchange surface	m^2
λ_{sol}	Thermal conductivity of the soil	$W/m.K$
h	Average convection air exchange coefficient	$W/m^2.K$
t	Time	s
δ	Depth of heat penetration into the ground	m
m	Air mass	kg
$\dot{m}$	Mass air flow	kg/s
C_p	Specific heat	$J/kg.K$
φ	Heat flow	$watt$
u	Axial air speed inside the exchanger	m/s
r_1	Inner shelf of buried tube	m
r_2	External radius of buried tube	m
r_3	Radius of the adiabatic layer of the soil	m
l	Longitudinal coordinate following the direction of the tube	m

x	Horizontal coordinate	m
α	Thermal diffusivity	m^2/s
z	Vertical cord	m
ρ	Density	$Kg/m3$
R	Thermal resistance	$m.K/W$
W	Power capacity	kW
$\dot{Q}$	Heat transfer rate	kW
γ	Specific heat ratio	Cp/Cv
ρ	Density	kg/m^3
v	Dynamic viscosity	$Pa\ s$
η_{THD}	Efficiency	%

Subscripts

a	Air
c	Compressor
f	Fuel
g	Flue gases
i	Tube inside
in	Input
o	Tube outside
p	Pipe
s	Soil
t	Turbine

th Thermal

Abbreviations

EAHE Earth air heat exchanger

GT Gas turbine

General Introduction

General introduction

Research context

Recently, the world's attention has shifted to renewable or alternative energies like geothermal energy. It is one of the oldest and cheapest alternative energies, its use has spread widely in cooling and heating greenhouses, as well as generating electricity using earth air heat exchangers EAHE.

Gas turbines play an essential role in generating electricity power, but any increase in temperature may negatively affect its functionality. It can affect material health, as well as extreme temperatures can cause the turbine blades to tear and bend.

Problematic:

The increase in the ambient air temperature during the summer, in southern Algeria, where there are large oil and gas production facilities (compression and pumping), directly weakens the efficiency of the gas turbine. The application of new operating cycles tends to become popular and to develop on auxiliary elements. Unfortunately, this had the disadvantage of greater energy (fuel) consumption; in fact, in order to support the execution of these applications, several modifications had to be made to these systems. These have thus seen their environmental conditions (temperature and humidity) and their performance change significantly. However, these two elements can be at the origin of a higher additional energy consumption.

The factor of the ambient temperature, although it is important for the correct functioning of this type of machine, is not taken into consideration during the process of developing their characteristics, despite the technological progress made in these types of use. The performance of the GT is still relatively limited.

- In literature, many researchers have modeled horizontal and vertical geothermal heat exchangers for any heat exchanger geometry.
- However, according to literature review, there are few studies interested in combining a gas turbine with a geothermal heat exchanger.
- The problems of heat accumulation in the adjacent soil around the buried tube arise in the continuous and long-term operation of the EAHE during the air-cooled period

- The relationship between earth air heat exchangers and the efficiency of gas turbine and preserving its parts from damage.

Objectives:

This study should achieve the following objectives:

- Provide a fairly complete study on the horizontal section of the EAHE
- Testing various factors that affect temperature drop, such as air velocity and air temperature .... etc.
- Presenting a study on the efficiency of the turbine and highlighting the relationship of between decrease of temperature in the inlet of compressor and the increase in efficiency and thus its relationship with the EAHE.

Organization of the thesis:

The first chapter of this study deals with general definitions of EAHE and gas turbine.

As for the second chapter, in a bibliographic study: We have made a summary of studies carried out by a group of researchers from different parts of the world on EAHE and gas turbines. This work allows us to predict the challenges that may face us in preparing this thesis.

The third chapter is devoted to mathematical modeling, in its first part, we do a mathematical modeling of the EAHE and the second part, we analyzed the gas turbine efficiency.

The fourth chapter is devoted to presenting and evaluating the various results obtained.

Chapter I:
General Definitions

1. Introduction

Recently, most industries have turned towards renewable energies, and if they do not work with them completely, they take a part of them, even if it is to improve production and productivity, that is, they work with them directly or indirectly, but this indicates the importance of renewable energies in the short or long term in all fields.

In this chapter, we will discuss general definitions about the heat exchanger and the gas turbine.

2. The Heat Exchanger air soil:

2.1. Definition:

Earth-to-air heat exchangers, also called underground tube heat exchangers, are an interesting technique for reducing energy consumption in a building. They can cool or heat the ventilation air, using the cold or heat accumulated in the ground. Several articles have been published in which a design method is described. Most of them are based on a discretization of the one-dimensional heat transfer problem in the tube. There are also complex three-dimensional models, solving the conduction and transport of moisture in the soil [1].

2.2. Principle:

The principle of the EAHE is that a pipe or several pipes buried in the ground. One end of the pipe system (the inlet) acts as the entrance for outdoor ambient air, whilst the other end of the pipe system (the outlet) releases air to the interior of a building. Ambient air is drawn into the pipe inlet, the air travelling through the pipe exchanging heat with the pipe walls which are in contact with the surrounding underground environment. In this way, heat is transferred to or from the surrounding soil by conduction through the pipe wall and convection with the tunnel air, tempering the air as it flows through the pipe Fig.1 illustrates this concept [2].

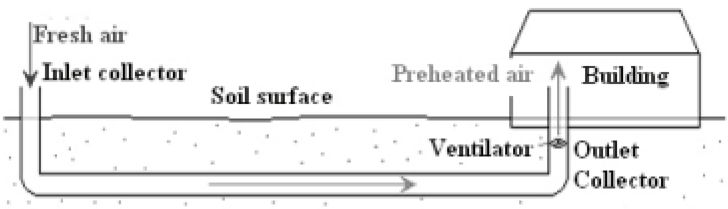


Figure 1.1. Simplified diagram of earth to air heat exchanger.

2.3.Different kinds of heat exchanger:

In addition to our heat exchanger there are another types of heat exchanger wich are:

a) Plate Heat Exchanger:

Heat exchangers are devices used to transfer energy between two fluids at different temperatures. They improve energy efficiency, because the energy already within the system can be transferred to another part of the process, instead of just being pumped out and wasted. In the new era of sustainability, the growing urgency to save energy and reduce overall environmental impacts has placed greater emphasis on the use of heat exchangers with better thermal efficiency. In this new scenario, the plate heat exchanger can play an important role [3].

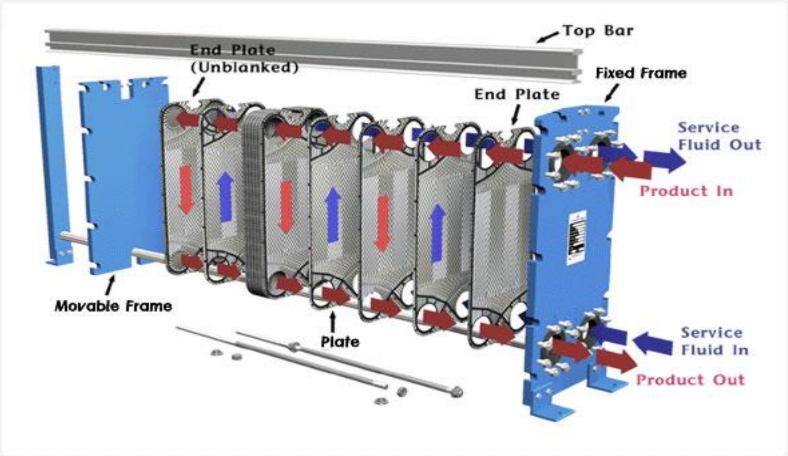


Figure 1.2. Plate Heat Exchanger [4].

b) Shell and Tube Oil Cooler:

Shell and tube heat exchangers are a common site throughout the engineering world and are one of the two most common types of heat exchanger; the other common type being the plate heat exchanger.

Shell and tube heat exchangers have a simple design, robust characteristics and relatively low purchase and maintenance costs. They also have a very high heat transfer rate although they require more space than a plate heat exchanger of similar thermal exchange capacity [3].

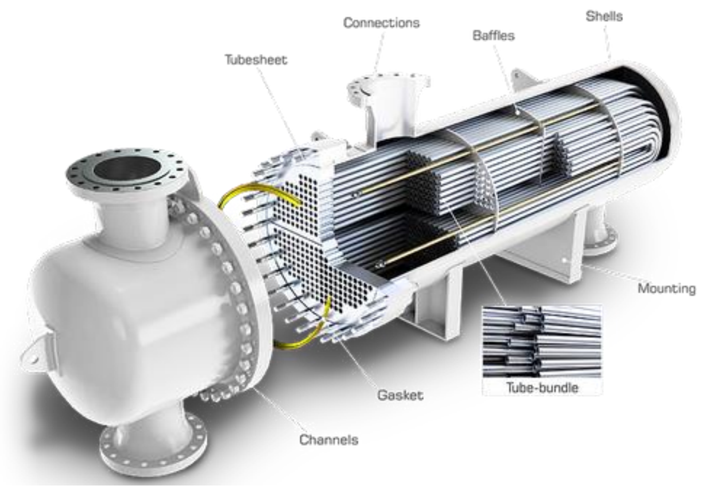


Figure 1.3. Shell and Tube Oil Cooler [5].

c) Intelligent Heat Exchanger Units:

Plate heat exchanger units have been widely used in industrial heat exchange filed, and its manufacturing technology become more and more mature. Intelligent heat exchange units Integrate with heat exchanger, circulating pump, water pump, meter, sensor, electric control and pipe valve [3].

Figure1.3. Intelligent Heat Exchanger Units[6]

d) Air-Oil Heat Exchanger :

Nowadays, more and more air cooling system were applied in oil cooling system because water cooling system cost is too high and same areas are not easy to secure water. Even more water treatment and discharge is also very difficult and expensive. Air-oil plate-fin heat exchanger has the characteristics of water-free, clean, compact, lightweight and very easy for piping [3].

Figure 1.4. Air-Oil Heat Exchanger [7].

Other Heat Exchanger with their capability:

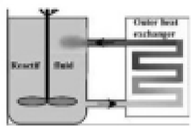
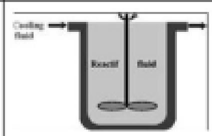
Process	Compact Multifunctionnal Heat Exchanger		Tubular exchanger reactor	Batch reactor with outer heat exchanger	Batch reactor with a double jacket
	Metallic foams [10] Re=1000	Offset Strip Fins [9] Re=2000			
Schematic diagram					
Specific Area, S/V ($m^2.m^{-3}$)	400	800	400	10	2.5
Global heat transfer coeff., U ($W.m^{-2}.K^{-1}$)	3500	5000	500	1000	400

Table 1.1. Heat exchange capability for different reactors [8].

2.4.Efficiency of the Heat Exchanger:

The main advantages of EAHE system are its simplicity, high cooling and pre-heating potential, low operational and maintenance costs, saving of fossil fuels and related emissions. Pre-heated fresh air supports a heat recovery system and reduces the space heating demand in winter. In summer, in combination with a good thermal design of the building, the EAHE can eliminate the need for active mechanical cooling and air-conditioning units in buildings, which will result in a major reduction in electricity consumption of a building if the EAHE is designed well. EAHEs are hence a passive cooling option in moderate climates [9].

The energy performance of EAHEs is described by the thermal interaction of heat conduction in the soil taking moisture in consideration, heat transport by flowing and ground water, heat transmission from the pipe to the air and changes in the air temperature and humidity. Different parametric and numerical models for EAHEs have been published in the last two decades.

Simulation models can be classified as models with an analytical or a numerical solution of the ground temperature field, and mixed models. Furthermore, mathematical or numerical optimization simply applies to one specified structure of the system whereas, often, structural modifications would be able to improve the cost effectiveness of the plant. Nevertheless, is not always possible or practical to develop a mathematical model for every promising design configuration of a system [9].

3.What is a Gas Turbine?

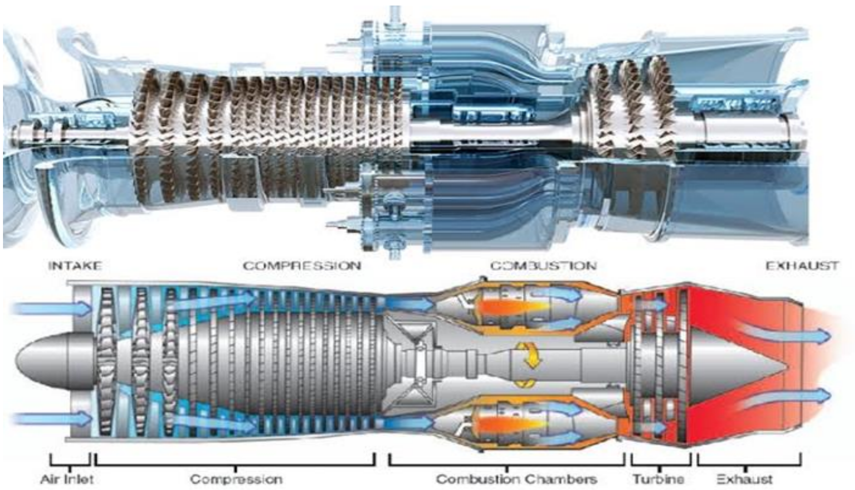


Figure 1.5. Gas Turbine [3].

3.1. Definition :

A Gas turbine (combustion turbine), is a thermodynamic rotating machine belonging to the family of internal combustion engines. whose role is to produce mechanical energy (rotation of a shaft) from the energy contained in a hydrocarbon (fuel, gas).

The turbojet is a particular gas turbine that uses the principle of reaction to propel certain types of fast planes. Each GT installation case must be personalized with specific parameters defined such as:

1. Type of fuel.
2. Operating time per year.
3. Extreme outdoor temperatures.
4. Editing, nuisances, etc [10].

4. History of Gas Turbines :

The development of gas turbine took place in several countries. Several different schools of thought and contributory designs led up to Frank Whittle's 1941 gas turbine flight. Despite the fact that NASA's development budget now trickles down to feed the improvement of flight, land based and marine engines, the world's first jet engine owed much to early private aircraft engine pioneers and some lower profile land-based developments [10].

The development of the gas turbine is a source of great pride to many engineer's worldwide and, in some cases takes on either industry sector fervor (for instance the aviation versus land based groups) or claims that are tinged with pride with one's national roots. People from these various sectors and subsectors can therefore get selective in their reporting [10].

So for understanding the history of the gas turbine, one would have to read several different papers and select material written by personnel from the aviation, and land-based sectors. At that point, one can "fill in the gaps" [10].

What follows therefore are two different accounts of the gas turbine's development. Neither of them is wrong. The first of these presents an aircraft engine development perspective [10].

Attempts to develop gas turbines were first undertaken in the early 1900's, with pioneering work done in Germany. The most successful early gas turbines were built by Holzwarth, who developed a series of models between 1908 and 1933. The first industrial application of a gas turbine was installed in a steel works in Hamborn, Germany, in 1933. In 1939 a gas turbine was installed in a power plant in Neuchâtel [10].

In 1931, U.S. army awards GE a turbine-powered turbosupercharger development contract.

In 1935 U.S. Army, Northrop, TWA, and GE combine to test fly a Northrop Gamma at 37,000 feet from Kansas City to Dayton. This led to a production contract for GE to build 230 units of the "Type B" supercharger and led to establishment of the GE Supercharger Department in Lynn, Massachusetts (later the site of the I-A development based on the Whittle engine) [10].

In 1938 Wright Aeronautical Corporation designs its own vaned superchargers for its own engines, although the superchargers were manufactured for Wright by GE [10].

In 1940 NACA joins with Wright, Allison and P&W to standardize turbo supercharger testing techniques. 1.1 Simple and Combined Cycles [10].

In 1925 R.E. Lasley of Allis-Chalmers receives the first of several patents on gas turbines. Around 1930 he forms the Lasley Turbine Motor Company in Waukegan, IL. with the goal of producing a gas turbine for aircraft propulsion [10].

In 1934 U.S. Army personnel from Wright Field visit Lasley's shop and inspected his hardware and the engine which he had filmed in operation earlier that year. However, neither the Army nor Navy would fund Lasley [10].

In 1939 GE studies gas turbine aircraft propulsion options and concludes the turbojet is preferable to the turboprop. Note, however, that two years later they changed their minds and proposed a turboprop to the Durand Committee [10].

In 1941 GE Turbo Supercharger Division (Lynn, Massachusetts) receives the Whittle W.1.X engine and drawings for the W.2.B improved version. A top secret effort begins to build an improved version, known as the I-A, for flight test in the Bell P-59 [10].

In 1941 Durand Committee also awards Navy contracts to Allis-Chalmers and Westinghouse. The Westinghouse W19, a small booster turbojet, resulted from this but Allis-Chalmers dropped out of the "gas turbine race" in 1943 [10].

In 1942 In April, the GE I-A runs for the first time in a Lynn test cell. In October, it powers the Bell P-59 on its first flight at Muroc Dry Lake, CA [10].

In 1929 Haynes Stellite develops Hastelloy alloy for turbine buckets, allowing operation up to gas temperatures of over 1800 F. This superior alloy was later crucial to the successful operation of the I-A and it gave U.S. turbine manufacturers the ability to use uncooled designs rather than include the complexity of blade cooling [11].

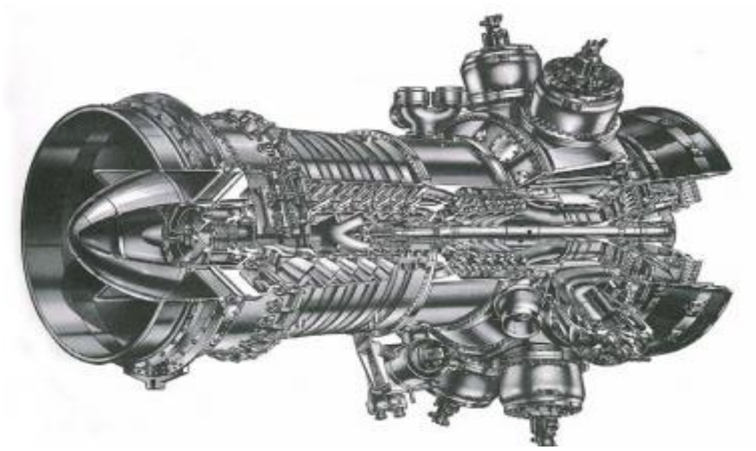


Figure 1.5. Rolls Royce RB211 Dry Low Emissions Gas Generator [7].

5. Main use of Gas Turbine :

5.1. Electricity production :

This application is extremely common: the turbine shaft drives a simplifier whose shaft at low speed drives an alternator. The mechanical system is simple and can be compared to a steam turbo-generator set for the production of electricity.

5.2. Combined heat-force production:

We also call cogeneration when we produce useful energies, electricity and heat, with primary energy sources, such as oil, gas or coal. This production is generally carried out within a thermal power station or, depending on its size, a combined cycle power station for domestic heating.

5.3. Pumping and compression:

In all types of GT, it is possible to replace the alternator driven by a pump, compressor or blower. The choice between a single-shaft or dual-shaft turbine depends on the type of machine coupled to the turbine and the intended operating style [10].

6. Classification of Gas Turbines:

Gas Turbines can be classified into two categories according to their mechanical design and technology:

- Industrial turbines can be one (1) shaft or two (2) shafts.
- Two (2) shaft aviation type turbines.

6.1. Industrial turbines:

Industrial turbines are heavy and fairly rustic machines where the main objective is long operating life. The latter leads to the application of mild operating parameters:

- The compression ratio between 6 and 8.
- The TET should be above 950 ° C (to overcome corrosion problems), then the average yields will be between 25-30%.

6.2. Single shaft turbine:

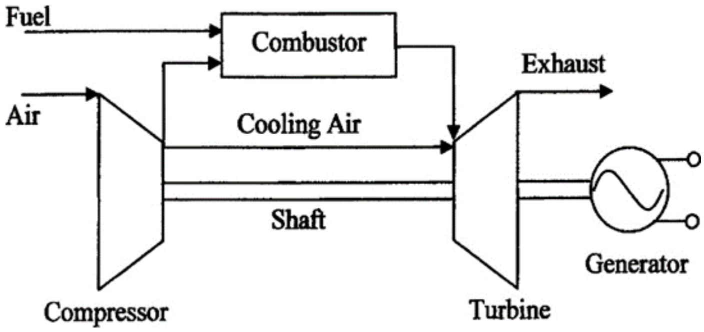


Figure 1.6. Schematic of a single shaft GT

In this class of GT, the air compressor, the expansion turbine and the driven device are on the same shaft. This installation has a positive side, due to its great simplicity, but also has a negative side, since it leads to a narrow operating range of speed, with low partial efficiency than a two-shaft turbine. In addition, the starter motor must drive the entire mobile crew. The second category of the turbine is generally the most used for the alternator drive, so the speed of rotation is constant [10].

6.2.1. Double (or twin) shaft turbine :

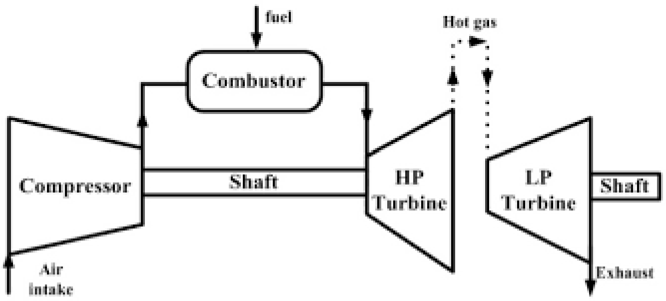


Figure 1.7. Schematic of a double shaft GT.

The expansion turbine is subdivided into two parts:

- The HP turbine drives the air compressor and it alone, via a shaft.
- The LP turbine guarantees the energy supply to the driven machin.

The first shaft line makes up the GG, the second shaft line for the production of mechanical energy.

From a mechanical point of view, the double shaft turbine is more complex than the single-shaft turbine; but, it makes it possible to obtain a better efficiency at partial load, it adapts particularly well to the drive of a compressor whose support is carried out during the increase in speed and makes it possible to work over a wide range rotational speed. Also note the reduced power of the starter motor which only drives the first shaft of the gas generator [10].

6.3.“Aviation” type turbines (Aero-derivative):

Aviation type turbines, also called “jet” turbines, consist of a GG reactor followed by an expansion turbine (in place of the nozzle) to produce mechanical energy. It is therefore a twin-shaft turbine using widely used aeronautical techniques. The design is obviously very different from industrial turbines since the criteria of weight and efficiency are essential.

The GG can no longer be repaired on site, it must be replaced with a new or refurbished generator. The repair is done by the manufacturer, in a specialized workshop taking into account safety [10].

7.Brayton cycle:

The way a gas turbine converts fuel energy into mechanical energy through the application of a thermodynamic process is known as the Brayton cycle. (shown in figure 06). The Brayton cycle taking place in the turbine is a regular and continuous process.

a. **Compression:** Atmospheric air is compressed by the axial compressor from 1 to 7 bar.
b. **Combustion:** Fuel is mixed with compressed air and the mixture is ignited, causing the hot gases to expand rapidly.
c. Expansion: Hot gases expand in the turbine section of the turbomachine, causing rotating torque in the process.
d. **Exhaust:** The gases used are evacuated into the atmosphere after having transmitted almost all of their energy to the turbine section [12].

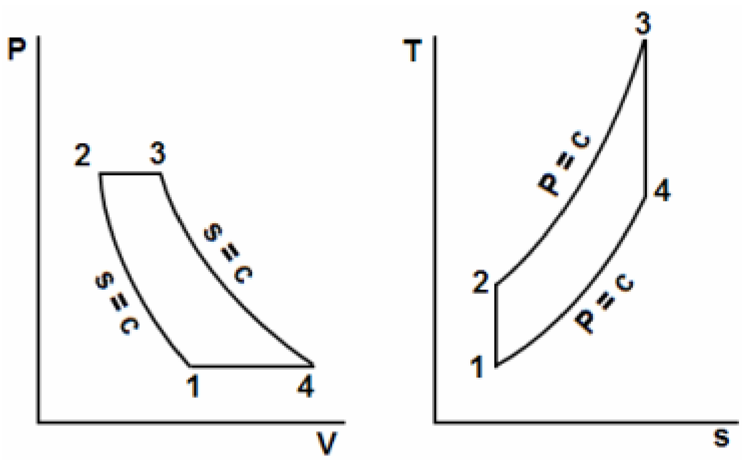


Figure 1.8. Brayton Cycle (The basic gas turbine cycle).

8. The efficiency of a gas turbine:

The efficiency of a gas turbine is changed depends on the type of turbine (turbine with a single blade or two blades), and in our study we will even improve this efficiency but with an earth-air exchanger and we will the explanation on the last 2 chapters.

9. Conclusion

Because of the pollution resulting from turbines and industries, due to their lack of production and efficiency, we decided to confront them or even reduce them by setting up mechanisms that allow us to maintain returns without increasing pollution, and that is what we will seek in the coming chapters.

By maintaining the turbine's efficiency, it may be possible to prevent an increase in pollution, thus preserving the environment.

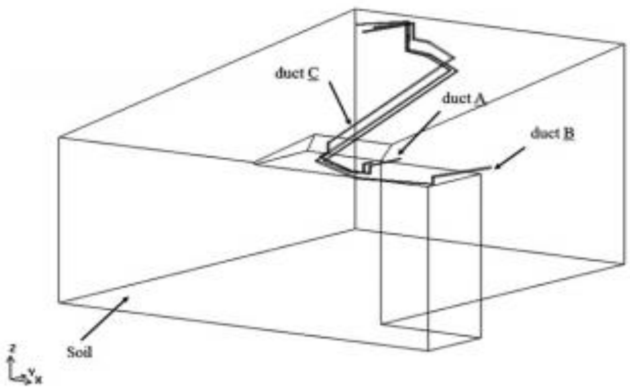
Chapter II:
Bibliographic Studies

1. Introduction:

In this chapter we are going to study earth-air heat exchangers in order to improve the efficiency of gas turbine. The goal is to conduct a bibliographic review summarizing the theoretical, analytical, and preparatory the experiences of some researchers working in this exchange.

2. Bibliographic studies:

J.Vaz et al. [13] analysed a numerical model of earth-air heat exchanger using a commercial code (FLUENT) which is based on the Finite Volume Method, they released the experimental setup in southern Brazil in the city of Viamão. the variation of air temperature inside the ducts, to an annual cycle was investigated. The transient temperature fields predicted numerically were compared with the experimental ones, the highest difference found was lower than 15% and they concluded that the validity and effectiveness of the employed computational model, enabling its use for future researches and projects developments about the earth–air heat exchangers.



Figur 2.1. Computational domain of the earth–air heat exchangers [14].

H.Al-Ansary et al. [14] did an Experimental Study of a Sand–Air Heat Exchanger for Use With a High-Temperature Solar Gas Turbine System. That system was being developed by King Saud University and the Georgia Institute of Technology to demonstrate the feasibility of using sand as the heat transfer and energy storage medium in central receiver systems. Experiments was conducted on silica sand and olivine sand, both of which was attractive options due to their wide availability. The apparatus included a tube bank consisting of eight electrically heated tubes arranged in three rows in a staggered formation. Heat transfer coefficient. the results were reported for bare and finned tubes for sand feed velocities of 1–3 mm/s. They founded in the range of 80–160 W/m^{2}K.

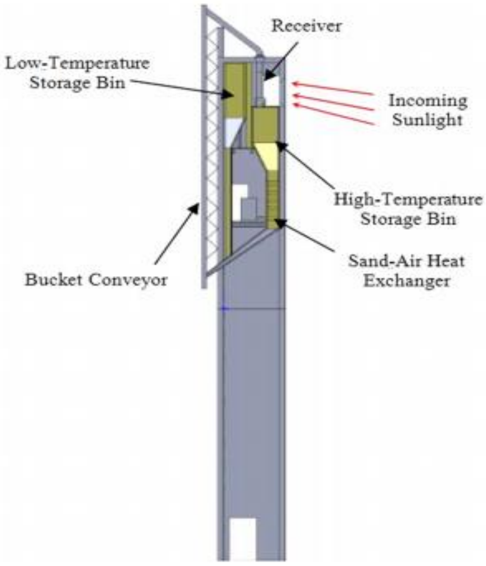


Figure 2.2. Section of the tower concept [15].

R.Vidhi et al. [15] did a parametric study of supercritical Rankine Cycle and EAHE for low temperature Power Generation. They considered temperature range of 125-200°C for the heat source, while ambient air cooled by EAHE, was used as a sink. Also, they studied the effect of various parameters (operating pressure, outlet temperature of the heat source and geometry of the EAHE) on the efficiency of the thermodynamic cycle. An optimum operating pressure was obtained for all the fluids studied; the results showed that the soil temperature increased only for a short distance around the pipe while the bulk temperature remained unaffected.

Figure 2.3. Schematic of an EAHE [16].

Y.BELLOUFI. [16] did a theoretical and experimental study about the use of geothermal energy in the heating or cooling of a heat transfer fluid used for the thermal comfort of home. He proposed an analytical model in a steady state to assess, the air temperature along a buried tube, also he developed a digital model in transient regime by applying the implicit finite difference method. The results showed that, when there are heat accumulation problems in the soil around the buried tube, an evaporator is integrated just at the end of the EAHE allowing the flowing air to reduce its temperature more and more by direct contact with it.

Figure 2.4. General view of the earth air heat exchanger [17].

R. S Brum et al. [17] studied a new computational modeling to predict the behavior EAHE. They found a new numerical model has the advantage of needing a lower computational effort, allowing the study about the influence of operational and constructive parameters, as well as, they were applicated geometric optimization methods in EAHE. A case study was developed where the influence of the installation depth in the thermal potential of an EAHE was investigated. They concluded that results were in agreement with those found in the literature; however, they were obtained with a reduction in the processing time of almost 45%.

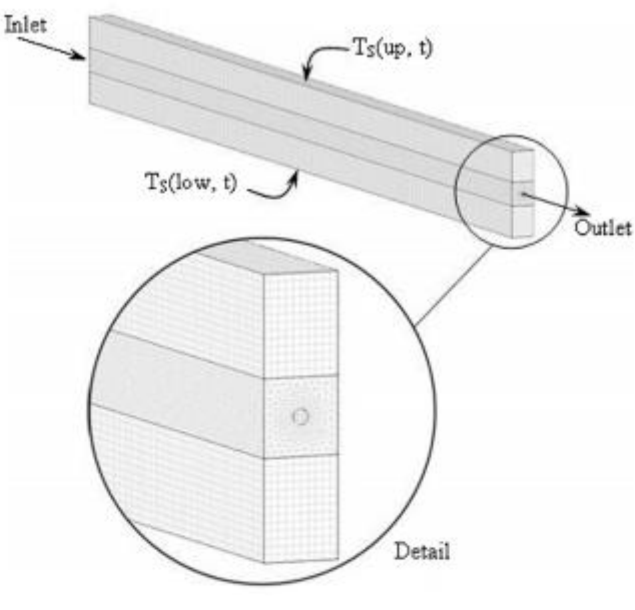


Figure 2.5. Computational domain developed for the Reduced model [19].

M.C Lekhal et al. [18] studied the Effect of geo-climatic conditions that effects on pipe materia the heating performance of an EAHE. They analyze the investigation on the heating performance and operating feasibility of two EAHEs in three different geo-climatic regions in Algeria. First, the performances of the EAHEs. The experiment allowed us to analyze the effect of the pipe material on the performance of the two EAHEs that are made of different materials (PVC and Zinc). The results revealed that the Zinc EAHE is more efficient in a temperate climate with a COP of 9.5 than in an arid or steppe climate with a COP of 8.2 or 8.1, respectively. However, the PVC EAHE is much better in an arid climate with a COP of 9.4 than in a temperate or steppe climate with a COP of 7.6 or 8.4, respectively. Otherwise, both EAHEs exhibited similar behavior in a steppe climate. They concluded that the thermal performance of EAHE mainly depends on geo-climatic conditions and the type of pipe material.

Figure 2.6. EAHEs under construction, installation and coupling with the test cell [19].

B.Singh et al. [19] studied the Evaluation of the Cooling Potential of Earth-Air Heat Exchanger Using Concrete Pipes in their study, they analyze the accomplishment of a low-cost cooling system of outdoor air for the hot-dry & hot-humid climate. The effect of velocity, length, and depth on the cooling potential of the system is studied at the inlet and outlet of the pipe. The novelty of this research is that enormous enhancement of the cooling potential has been observed in hot-humid climate than hot-dry climate, which is not available in previous studies. The results show that the maximum cooling potential in hot-dry climate is found 5643 kWh, 7375 kWh, 8939 kWh for the EAHE length of 15 m, 30 m, and 45 m, respectively, corresponding to the velocity 2.5 m s−1 and depth 1.5 m. Whereas in a hot-humid climate, the maximum cooling potential is achieved 13,373 kWh, 20,134 kWh, and 24,080 kWh with a length of 15 m, 30 m, and 45 m, respectively, for the given velocity and depth.

A. Minaei and H. Safikhani. [20] studied A new transient analytical model for heat transfer of EAHE they found that the Governing of equations for both pipe flow and soil surrounding the pipe was derived for the transient state and are solved using Laplacian transform. To validate the analytical model, the 3D numerical model of the EAHE is also simulated. Results of the analytical model and numerical simulations are compared with already reported experimental data. In the heating period at the velocity of 2 m/s, the discrepancy between the analytical model and reported experimental results is 0.87% and 1.31% for steel and PVC pipes, respectively. For velocity of 5 m/s this was as low as 0.4% for both pipe materials.

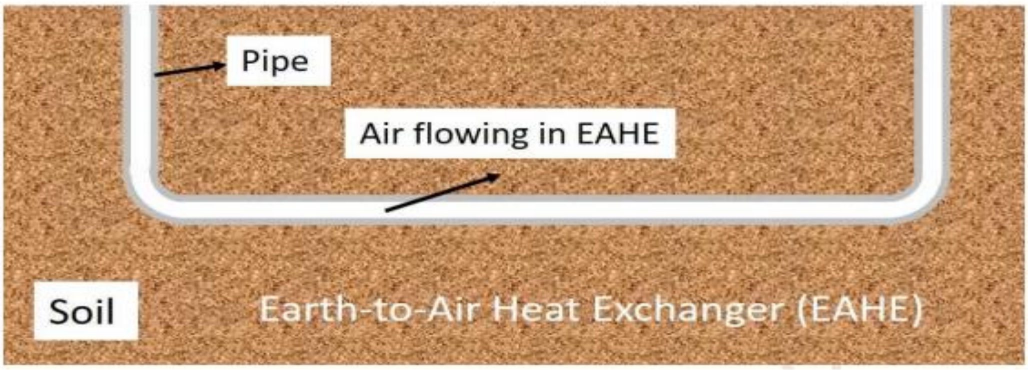


Figure 2.7. Schematic of ground to air exchanger [21].

S.F.Ahmed et al. [21] studied Physical and hybrid modeling techniques for EAHE in reducing building energy consumption: Performance, applications, progress, and challenges and they found that hybrid modeling was more effective than physical models for accurate prediction. On the contrary, the hybrid models suffer from high complexity if EAHE operating conditions and all key parameters are considered during the model development. The outcome of this study also provides valuable information regarding the physical and hybrid EAHE modeling techniques to the scientists, researchers, and so on in adopting the most appropriate EAHE modeling technique for their climates.

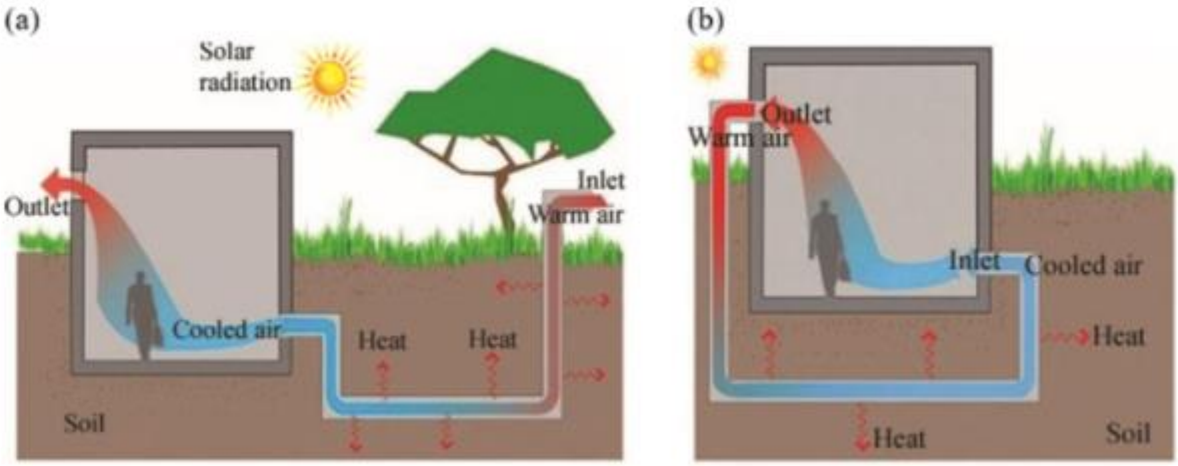
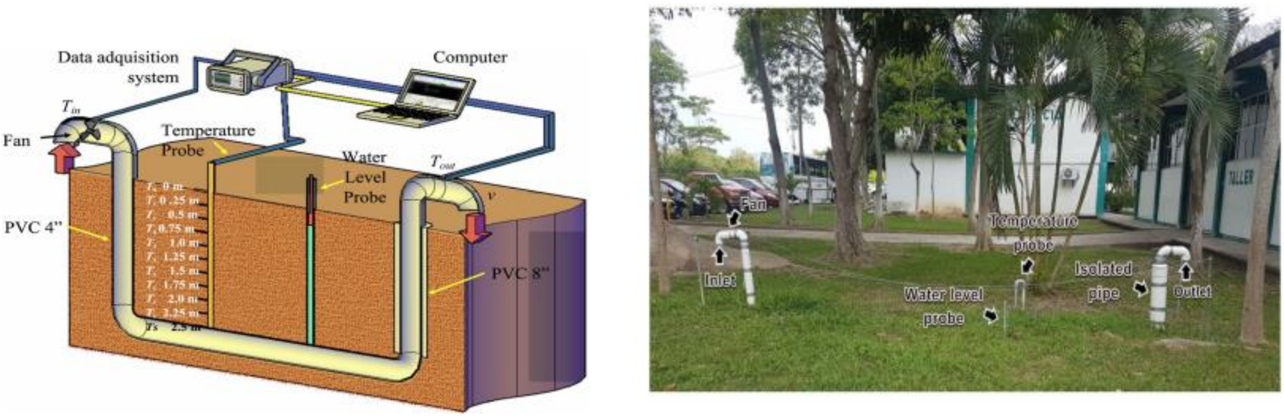


Figure 2.8. Earth-air heat exchanger types during summer (a) Open-loop system (b) Closed-loop system reprinted [22].

H. P. Díaz-Hernández et al. [22] did an Experimental study of (EAHE) for warm humid climatic conditions. They found that The EAHE has a horizontal configuration with three sections of 101.6 mm diameter PVC pipe, a horizontal pipe of 6 m long and two vertical sections of 3 m long for the inlet and outlet sections. It was buried at 2.5 m and to improve the performance of the EAHE, the outlet section was thermally insulated. they monitored for 6 months the soil temperature from 0 to 2.5 m in intervals of 0.25 m, and the air temperature at the inlet and the outlet. Specific heat was determined for five samples of soil analyzed using thermogravimetry. The average temperature of the soil at 2.5 m depth remained between 27–28C, and the obtained specific heat of the soil ranged between 0.726 and 0.910 Jg.K showing a homogeneous behavior. The EAHE behaved as a cooler during the day and as a heater at night, except in the winter season that worked most of the time as a heater. The results showed that with the EAHE it is possible to cool an airflow under humid-warm weather conditions.



a). shematic diagram

b). Photograph

Figure 2.9. EAHE experimental set up [23].

Alexey Kolychev et al. [11] studied the effect of vane thermal emission cooling on the efficiency of the Gas Turbine power plant they found that to improve the efficiency of a gas turbine is to increase the gas temperature in front of the turbine. And Cooling of gas turbine elements is difficult. One of the solutions to the problem may be the method of thermal emission cooling. The purpose of that work is to estimate the potential effect of thermal emission cooling of turbine blades on efficiency. And to achieve that mentioned aim they were analyzed the main factors influencing the efficiency of the power gas turbine unit. The results showed that the blade temperature of the turbine with thermal emission cooling can reach the value of about 1000 K at the electron work function 1 eV and at the gas temperature in front of the turbine 2700 K (by 7-8 % higher than modern power gas turbines).

Figure 2.10. Alstom's GT-8C2, 50/60Hz gas turbine with basic specification [12].

GuangyaZhu et al. [23] did a comparative study on humidified gas turbine cycles with different air saturator designs, they used to the performance comparison between two types of air saturator designs of humidified gas turbine cycles. Type 1 air saturator was a hybrid design that combined an indirect evaporative cooler with a Maisotsenko cycle while Type 2 was a conventional indirect evaporative cooler. Through system simulations, it was found that all the humidified gas turbine cycle systems offered higher system efficiencies than a simple gas turbine system with recuperator. Besides, parametric studies were conducted which highlighted the effects of system inlet air temperature, turbine inlet temperature, water injection rate, and part-load ratio on the performances of the different humidified gas turbine cycle designs. The results showed that The employment of Type 1 air saturator offered 9.34% and 23.55% enhancement in the system efficiencies as compared to those using Type 2 air saturator under the design and 50% part-load ratio conditions respectively. This reinforced the benefit of applying Maisotsenko cycle to the air saturator design of humidified gas turbine cycle for the enhancement of system efficiency.

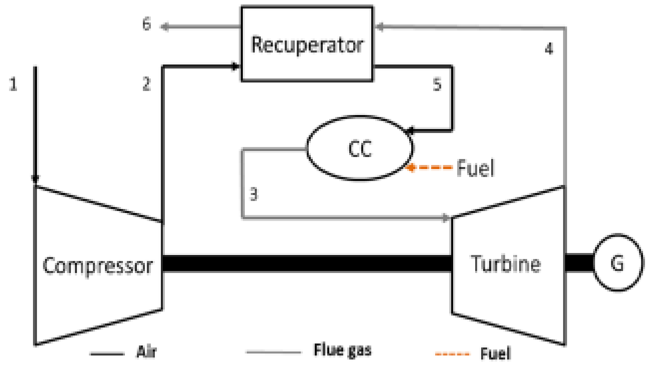


Figure 2.11. Simple gas turbine cycle with recuperator [24].

ZhenXu et al. [24] did an Experimental study on the off-design performances of a micro humid air turbine cycle: Thermodynamics, emissions, and heat exchange. they did a detailed experimental study was presented on a mHAT converted from a recuperated microturbine by introducing a humidifier, an aftercooler, and an economizer the steady-state thermodynamic performances, and the combustion and emission characteristics are evaluated at rated and part load condition. The off-design performances of the heat exchangers are also evaluated the results showed that the specific output power and the electrical efficiency are relatively increased by 45% and 18.6% respectively compared with the recuperated cycle at the rated load, and show a greater relative increment at part load, such as 58.1% and 21% respectively at 62.5% load. Introducing water into the air reduces the risk of NOx formation, but increases dramatically the risk of CO formation and incomplete combustion in the combustion chamber, thus the combustion efficiency deteriorates up to 92.5% at 50% of full load.

KamalAbudu et al. [25] studied the Impact of gas turbine flexibility improvements on combined cycle gas turbine performance they found that There are different approaches to improving gas turbine flexibility, and they had performance implications for the bottoming cycle in the combined cycle gas turbine (CCGT) operation. The CCGT configuration was favorable in generating more power output, due to the higher thermal efficiency that was key to the economic viability of electric utility companies. However, the flexibility benefits obtained in the gas turbine were often not translated to the overall CCGT operation. the results showed that The MEL extension on the gas turbine that bringed about a reduction in the engine power output results in a higher steam turbine power output due to the higher exhaust gas temperature of the former.

M N Khan et al. [26] studied the Improvement of the efficiency of gas turbine-air bottoming combined cycle by heat exchangers and bypass control valves, they did a comparative energy and exergy analysis of a Proposed gas turbine cycle (PGTC) with a simple gas turbine cycle (SGTC), maintaining the same rate of fuel supply in both cycles. In the PGTC, the air bottoming cycle is operated by exhaust gases from the topping gas turbine cycle by exchanging heat in the heat exchangers (H.E.s) by controlling the path of exhaust gases and compressed air through the bypass control valves. The bypass valve in the topping, as well as the bottoming cycle, directs the combustible product from the combustion chamber to the H.E., in such a way that it optimizes the performance of the proposed combined cycle as compared to SGTC. The results showed that the maximum increase in Work net output and thermal efficiency in PGTC compared to SGTC is 65.7% at rp = 4 and turbine inlet temperature (TIT) = 1500 K, whereas the exergy loss by the exhaust gases in the PGTC was much less than the exergy loss by the exhaust gases in the SGTC. The maximum difference in the exergy loss by the exhaust gases in PGTC and SGTC is also observed at rp = 4 and TIT = 1500 K.

S. Barakat et al. [27] studied the Enhancement of gas turbine power output using (EAHE) cooling system where ion of (EAHE) as an inlet air cooling system on gas turbine performance has been investigated. Transient, one-dimensional model was developed for predicting the thermal performance of EAHE. Gas turbine output power, efficiency and specific fuel consumption were assessed with application of EAHE. MATLAB program is developed for solving the discrete numerical equations. Damietta power plant is selected as case study. The results showed that the output power and thermal efficiency of gas turbine increases by 9% and 4.8%; respectively. In addition, the annual revenue will increase by 1.655 ⁄ 106 $ with payback period of 1.2 year.

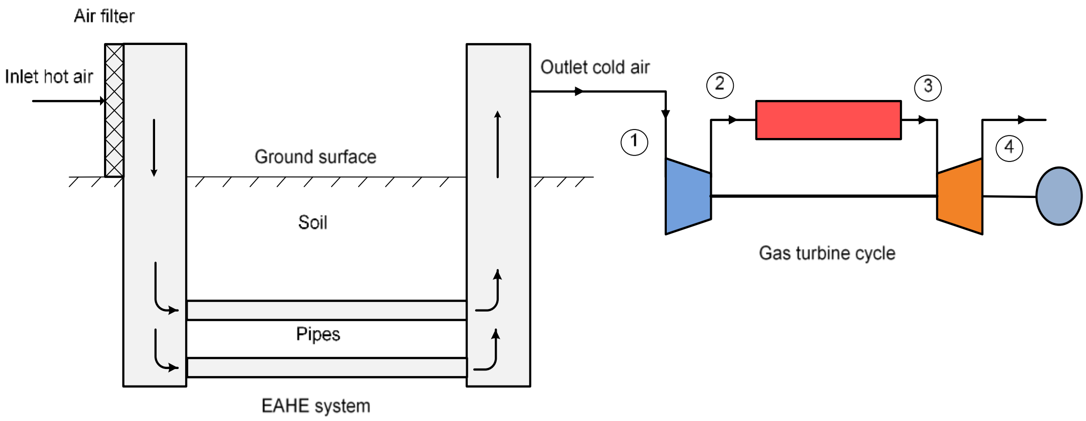


Figure 2.12. Schematic diagram of EAHE system installed in gas turbine power plant [27].

3. Conclusion:

The higher firing temperature reflects the high efficiency of the gas turbine. Whereas, the use of higher temperatures is limited because it may cause the turbines to shear, warp, or bend. Hence, developing an efficient internal gas turbine blade cooling system is essential. At the same time, it is necessary to ensure the lowest possible penalty on the thermodynamic performance cycle and this is what we want to study in Chapter 3.

Chapter III: Mathematical Model

1. Introduction:

In this chapter, we will conduct a numerical and analytical study using Matlab program for model EAHE and gas turbine. An analytical model will be developed using the general conduction equation to study the various influences that can contribute in one way or another to lowering the temperature in order to improve the efficiency of the turbine.

2. Soil temperature modeling:

It is important to estimate the factors affecting the lowering of the air temperature inside the ground-to-air heat exchanger we will study the horizontal part of the heat exchanger

2.1. Steady station of soil temperature:

2.1.1. Variable ground surface temperature:

make the following unidirectional conduction equation:

$$\frac{\partial^2 T}{\partial z^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \tag{1}$$

Suppose that:

T: Ground temperature in [degrees].

t: Time in [s].

z: Vertical coordinate in [m].

α: Thermal diffusivity of the soil [α = λ/(ρ.cp)] (m2/s).

λ: Thermal conductivity of the soil in (w/(m. degree)).

ρ: Density of the soil in (kg/m3)

Cp: The specific heat of the soil in (j/kg.degree)

Change the variables:

$$\theta(z,t) = T(z,t) - T_i \tag{2}$$

Equation (1) becomes:

$$\frac{\partial \theta}{\partial t} = \alpha \frac{\partial^2 \theta}{\partial z^2} \tag{3}$$

$$\alpha = \frac{\lambda}{\rho Cp} \tag{4}$$

The boundary conditions for solving equation are explicitly given by equation (5) :

$$\begin{cases} \theta(z = 0, t) = A.cos(wt) \\ \theta(z \to \infty, t) = 0 \\ \theta(z, t = 0) = 0 \end{cases} \tag{5}$$

Where:

T_i : Annual average temperature at the ground surface, it also represents the invariant temperature of the subsoil.

w : Angular frequency [w = 2π/365] (rad/j).

A : Amplitude of temperature variation

Using separating variables method to solve equation:

$$\theta(z, t) = Z\,(z).\tau(t) \tag{6}$$

Replace equation (6) in equation (3), we will have:

$$Z(z).\tau'(t) = \alpha.Z''(z).\tau(t) \tag{7}$$

Divide the two sides of the equation by Z (z). τ (t) we get:

$$a\frac{Z''(z)}{Z(z)} = \frac{\tau'(t)}{\tau(t)} = \beta = cste \tag{8}$$

Such that: β = iw (Constant complex number)

From equation (8), we get:

$$z''(y) - \frac{\beta}{a}\,.z(z) = 0 \tag{9}$$

$$\tau'(t) - \beta.\tau(t) = 0 \tag{10}$$

The solution of equations (9) and (10) will be presented by equations (11) and (12) namely:

$$Z(z) = a.exp\left(-\sqrt{\frac{\beta}{a}}z\right) + b.exp\left(\sqrt{\frac{\beta}{a}}z\right) \tag{11}$$

$$\tau(t) = c.exp(\beta t) \tag{12}$$

The general solution of equation (3) takes the following form:

$$\theta(z,t) = \left[C_1.exp\left(-\sqrt{\frac{\beta}{a}}z\right) + C_2.exp\left(\sqrt{\frac{\beta}{a}}z\right)\right].exp(\beta t) \qquad (13)$$

C_1 and C_2 are two integration constants, we apply the boundary conditions mentioned previously, to determine them:

- So that the solution remains finite when z ⟶ ∞; constant C_2 must be zero.
- On the wall z = 0, we must have:

$$\theta(0,t) = real[C_1.exp(\beta t)] = real[C_1.(cos(wt) + i.sin(wt))] = A.cos[wt] \qquad (14)$$

Noting that: $\sqrt{i.w} = \sqrt{\frac{w}{2}}\,.(1+i)$

The complex solution of equation (13) can be written as:

$$\theta(z,t) = A.exp\left(-\sqrt{\frac{\omega}{2a}}(i+1).z\right).exp(iwt) \qquad (15)$$

which turns into:

$$\theta(y,t) = A.exp\left(-\sqrt{\frac{\omega}{2a}}z\right).\left[exp(i\omega.t).exp\left(-i\sqrt{\frac{\omega}{2a}}z\right)\right] \qquad (16)$$

Let's introduce the trigonometric writing then:

$$\theta(y,t) = A.exp\left(-\sqrt{\frac{\omega}{2a}}z\right).\left[\left(cos\,\omega t + i\,sin\,\omega t\right).\left(cos(-\sqrt{\frac{\omega}{2a}}z)\right)\right] \qquad (17)$$

After arrangement, we will have:

$$\theta(y,t) = A.exp\left(-\sqrt{\frac{\omega}{2a}}z\right).\left[\left(cos(\omega t).cos\left(-\sqrt{\frac{\omega}{2\alpha}}z\right) + i\,cos(t - t_0).sin\left(-\sqrt{\frac{\omega}{2a}}z\right) + i\,sin(\omega t)\,cos\left(-\sqrt{\frac{\omega}{2a}}z\right)\right) + i^2\,sin(\omega t).sin\left(-\sqrt{\frac{\omega}{2a}}z\right)\right] \qquad (18)$$

$$\theta(z,t) = A.exp\left(-\sqrt{\frac{\omega}{2a}}z\right).\left[cos(\omega t).cos\left(-\sqrt{\frac{\omega}{2\alpha}}z\right) - sin(\omega t).sin\left(-\sqrt{\frac{\omega}{2a}}z\right) + i\left(cos(\omega t).sin\left(-\sqrt{\frac{\omega}{2a}}z\right) + sin(\omega t).cos\left(-\sqrt{\frac{\omega}{2a}}z\right)\right)\right] \qquad (19)$$

The desired fluctuation in temperature is the real part of the complex solution found, that is so say:

$$\theta(z,t) = A.exp\left(-\sqrt{\frac{\omega}{2a}}z\right).\left[cos\,\omega(t)\,.\,cos\left(\sqrt{\frac{\omega}{2\alpha}}z\right) + sin\,\omega t\,.\,sin\left(\sqrt{\frac{\omega}{2a}}z\right)\right] \qquad (20)$$

Finally, the expression T(z,t) will take the following form:

$$T(z,t) = T_i + A.exp\left(-\sqrt{\frac{\omega}{2a}}z\right).cos\left(\omega t - \sqrt{\frac{\omega}{2\alpha}}z\right) \qquad (21)$$

3.Thermal model of the earth air heat exchanger

3.1. Description of the system

It is a PVC tube buried at a specific depth so that the ground neighbor of the tube remains less invariant over the year. The air flow to the interior of the tube is provided by a variable flow fan. Heat exchange between the ground and the air flowing inside the tube (fig. 1) depends on the thermal and geometric characteristics of the soil and the tube, of the temperature air inlet and soil temperature near the tube

3.2. Simplifying assumptions

We admit here the following simplifying assumptions:

i) Air is incompressible and its thermal properties are stable.

ii) The ground far from the exchanger keeps a constant temperature during earth air heat exchanges.

iii) The convective exchange coefficient between the air and the tube is constant along the earth air heat exchanger.

iv) The air flow is unidirectional with radial speed assumed constant.

v) Longitudinal conduction and humidity are negligible.

vi) The EAHE is buried at a depth δ deep enough for soil temperature (Tsoil) remains constant equal to (Ti) fig.3.1.

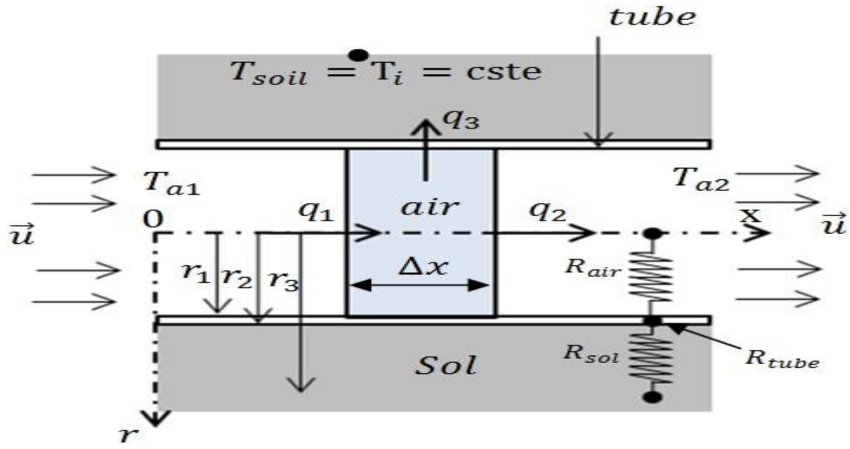


Figure 3.1. Descriptive diagram of EAHE for a cooling cycle.

3.3. Steady state modeling of the air temperature along the tube:

$$m.C_{pair}.\frac{DT_a}{Dt}=q_1 - q_2-q_3 \quad (22)$$

The equation 1 represent the energy balances for cooling where:

$$q_3 = \frac{T_{air}-T_{soil}}{R_{total}} \quad (23)$$

Therefore, we observe from equation (1) that the balance energy for the operating cycles is summarized in an energy balance which is written as follows:

$$m.C_{pair}.\left(\frac{\partial T_a}{\partial t} + u\frac{\partial T_a}{\partial x}\right) = -\lambda.S.\frac{\partial T_a}{\partial x}\bigg|_x + \lambda.S.\frac{\partial T_a}{\partial x}\bigg|_{x+dx} + \frac{(T_{soil}-T_a)}{R_{total}} \quad (24)$$

Where (u) is the average air flow velocity inside the EAHE. The total thermal resistance (R_{total}) is made up on the one hand of the resistance to conduction of the ground (R_{soil}) and that of the tube (R_{tube}), and on the other hand of the resistance air convective (R_{air}).

$$R_{total} = R_{tube} + R_{soil} + R_{cv} \quad (25)$$

Where the resistance of the tube is written as follows:

$$R_{tube} = \frac{1}{\lambda_{tube}.2\pi.\Delta x} ln\left(\frac{r_2}{r_1}\right) \quad (26)$$

The resistance of the soil is given by the relation (6)

$$R_{soil} = \frac{1}{\lambda_{soil}.2\pi.\Delta x} ln\left(\frac{r_3}{r_2}\right) \quad (27)$$

$when\ r_3 = 2r_1$

On the other hand, that of air is written in the form below:

$$R_{air} = \frac{1}{h_{cv}.2\pi.r.\Delta x} \tag{28}$$

Dividing equation 3 by the unit of the differential element Δx, we get:

$$m.C_{pair}.\left(\frac{\partial T_a}{\partial t} + u\frac{\partial T_a}{\partial x}\right) = \frac{-\lambda.S.\frac{\partial T_a}{\partial x}\Big|_x + \lambda.S.\frac{\partial T_a}{\partial x}\Big|_{x+dx}}{\Delta x} + \frac{(T_{soil} - T_a)}{R_{itotal}} \tag{29}$$

Let s denote by s, the internal section of the tube, $s = \pi.\ r_1^2$ and h = (Nu.k) /2 r_1, the average coefficient of heat transfer by convection.

Where the Nusselt number, $Nu\ =\ 0.023\ Re^{0.8}\ Pr^{0.3}$

and the number Reynolds, Re = ρv /μ [16].

R_{itotal} the total thermal resistance through unit length.

$$m.C_{pair}.\left(\frac{\partial T_a}{\partial t} + u\frac{\partial T_a}{\partial x}\right) = \lambda.S\ \frac{\partial^2 T_a}{\partial x^2}\ + \frac{(T_{soil} - T_a)}{R_{itotal}} \tag{30}$$

$$\rho.\pi.r_1^2.C_{pair}.u\frac{dT_a}{dx} = \frac{T_{soil} - T_a}{R_{itotal}} \tag{31}$$

and

$$ln(T_a - T_{soil}) = -\left(\frac{1}{\rho.\pi.r_1^2.C_{pair}.u.R_{itotal}}\right)x + C \tag{32}$$

With $T_a(x = 0) = T_{a1}$

Where $Ta1$ is the inlet air temperature.

Therefore, the temperature profile along the EAHE is written in the following form:

$$T_a(x) = T_{soil} + (T_{a1} - T_1)exp\left(\frac{-x}{\rho.\pi.r_1^2.C_{pair}.u.R_{itotal}}\right) \tag{33}$$

4.Gas turbine model:

the gas turbine works according to the cycle invented by George Bryton (1870), the Bryton cycle is mainly used for the production of electricity. This cycle consists of four thermodynamic evolutions:

1 → 2 Isentropic compressors: the air drawn from the atmosphere is isentropic compressed in the compressor (generally of the axial type).

2 → 3 Isobaric combustions: the air leaving the compressor is heated (at P = cst) in the combustion chamber where fuel is injected.

3 → 4 Isentropic expansions: the burnt gases leaving the combustion chamber at high pressure and temperature expand isentropically in the turbine to be actuated and therefore develop work.

4 → 1 Isobaric exhaust: Discharge of combustion gases to the atmosphere at constant pressure (exhaust and intake gases in the open cycle).

A simplified analytical model for gas turbine cycle will be considered, based on following assumptions [27]:

1. Air and combustion products are ideal gases.

2. The fuel supplied to the system is assumed to be natural gas.

3. Temperature at turbine inlet is constant.

If air enters the compressor at T1, the outlet temperature T2 can be calculated as:

$$T_2 = T_1 \left\{ 1 + \left[\frac{1}{\eta_c} \left(r_c^{\frac{\gamma_a -}{\gamma_a}} - 1 \right) \right] \right\} \tag{34}$$

where η_c is the isentropic efficiency of the compressor and rc is the compression ratio.

The compressor power consumption can be estimated for an adiabatic compressor as follows:

$$W_c = \dot{m}_a Cp_{a.avg} T_1 \left\{ \left[\frac{1}{\eta_c} \left(r_c^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right) \right] \right\} \tag{35}$$

where $\dot{m}_a$ is the inlet air mass flow rate, $Cp_{a.avg}$is the specific heat of dry air at average temperature across the compressor.

By knowing the temperature of combustion gas at turbine inlet T_3, the heat delivered by the combustor is estimated from the energy balance as follows:

$$\dot{Q}_{in} = \dot{m}_a Cp_{a.avg} (T_3 - T_2) \tag{36}$$

where $Cp_{a.avg}$is the specific heat of flue gases at average temperature across the combustor.

The mass flow rate of fuel used is defined as:

$$\dot{m}_f = \frac{\dot{Q}_{in}/LHV}{\eta_{comb}} \tag{37}$$

where η_{comb} is the combustion efficiency.

The temperature of the flue gases leaving the turbine T_4 can be expressed as:

$$T_4 = T_3\left\{1 - \left[\eta_t\left(1 - r_e^{\frac{1-y_g}{y_g}}\right)\right]\right\} \tag{38}$$

Turbine power is evaluated as:

$$W_t = (\dot{m}_a + \dot{m}_f)Cp_{a.avg}T_3\left\{\eta_t(1 - r_e^{\frac{1-y_g}{y_g}}\right\} \tag{39}$$

Hence, the net power output is:

$$W_{net} = W_t - W_c \tag{40}$$

4.1. Thermodynamic efficiency of a gas turbine:

$$(Q_c - Q_f) + (W_{comp} - W_{turb}) = h_s - h_e \tag{41}$$

Where:

$$Q_c = h_3 - h_2 = C_p(T_3 - T_2) \tag{42}$$

And

$$Q_f = h_1 - h_4 = C_p(T_1 - T_4) \tag{43}$$

useful work will be:

$$W_u = W_{turb} + W_{comp} = C_p\big((T_4 - T_3) + (T_2 - T_1)\big) \tag{44}$$

where:

$$\eta_{THD} = 1 + \frac{Q_f}{Q_c} \tag{45}$$

so:

$$\eta_{THD} = 1 - \frac{C_P(T_4 - T_1)}{C_p(T_3 - T_2)} = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} \tag{46}$$

we notice from the isentropic evolution (1 >> 2) and (3 >> 4) that:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4} \tag{47}$$

Denote by $\tau = \frac{P_2}{P_1}$ the compressor rate

So:

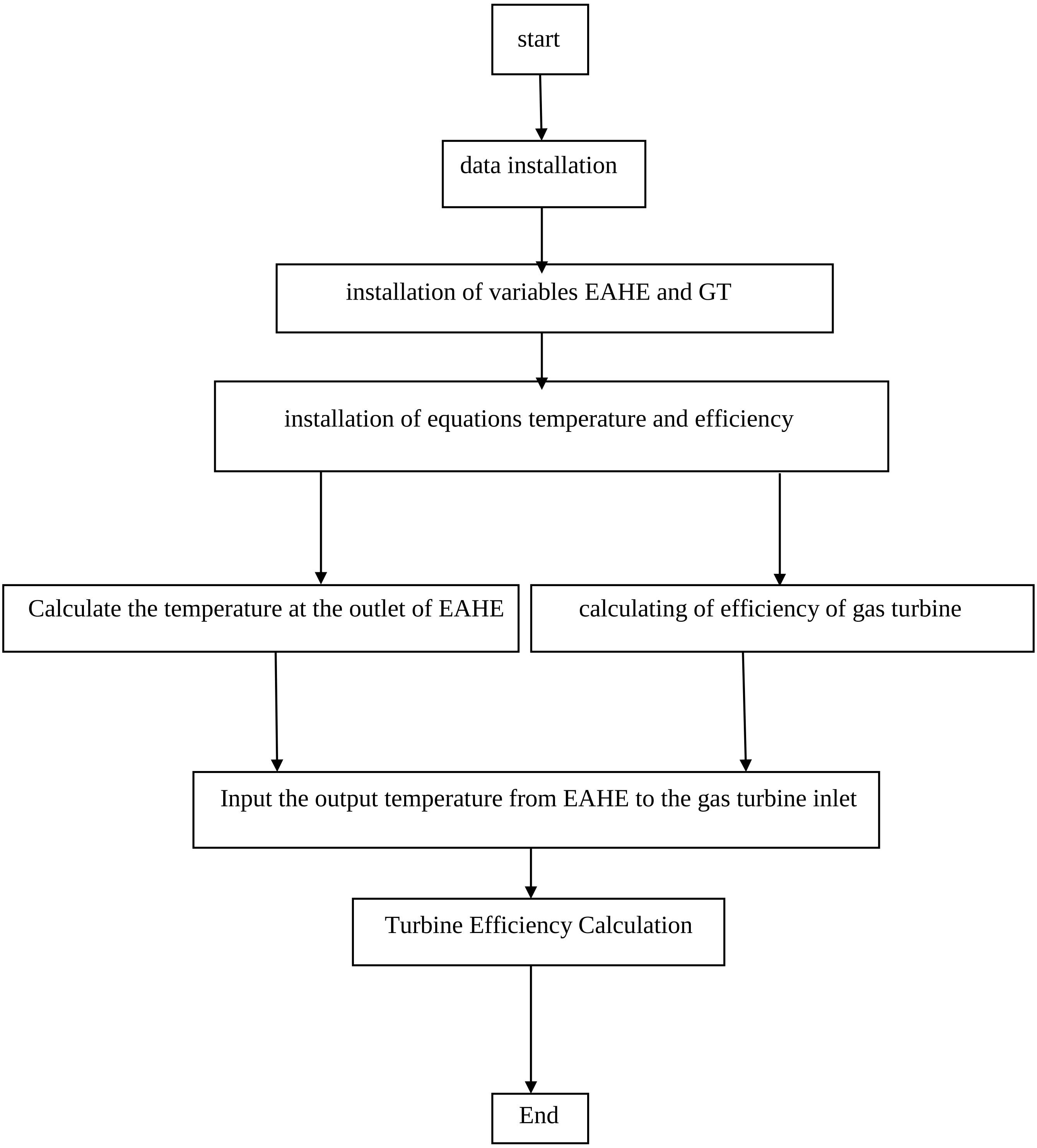
$$\eta_{THD} = 1 - \frac{\left(\tau^{\frac{\gamma-1}{\gamma}} - 1\right)}{\tau^{\frac{\gamma-1}{\gamma}}(\tau^{\frac{\gamma-1}{\gamma}} - 1)} \tag{48}$$

$$\eta_{THD} = 1 - \frac{1}{\tau^{\frac{\gamma-1}{\gamma}}} \tag{49}$$

5. Conclusion:

In this chapter, we have presented thermal mathematical models based on the general equation of conduction and the principle of energy balances, the purpose of this study which is to predict the evolution of temperature in the soil and that inside the buried tube. also we studied the evolution of efficiency of gas turbine using thermodynamic models, a MATLAB language program was developed to determine the essential parameters which allowed us to specify the different ones that carry out the improvement of the efficiency of gas turbine.

6.Calculation flowchart:



Chapter IV :

Results and discussions

1. Introduction :

This chapter, will provide the inlet air cooling system for gas turbines using EAHE cooling system. To implement this system, the soil behavior must first be examined by presetting the soil temperature underground at different depths throughout the year and determining the optimum depth for high performance. Next, the effects must be studied are: air velocity passing through the buried tube, thermal conductivity of the soil surrounding it and its diameter with the temperature drop and the improvement of efficiency.

2. Comparative study:

Results of model validation against both the experimental and theoretical data of us and BELLOUFI [16].

Table 4.1. EAHE parameters

EAHE parameters: (EAHE with a pipe in PVC)	**Dimension**
r_1	0.055 m
Air velocity	1.5 m/s
Air temperature	48.87°C
Thermal conductivity of the buried tube	0.17 W/m.k
Thermal conductivity of the soil	1.25 W/m.k
Thermal conductivity of the air	0.024 W/m.k

Table 4.2. Validation of proposed model with experimental results of BELLOUFI.Y [1]: The Results obtained at u = 3.5 m / s of the day of 07 August 2013. at 15.30.

Length(m)	**T_exprimental results(°c)**	**T_theorical results (°c)**	**Error %**
0	48,87	48,87	0
3,63	44,94	43,39	3,572252
7,69	39,34	38,85	1,261261
11 ,73	35,87	35,6	0,758427
16,04	34,29	33,11	3,563878
20,07	32,95	31,44	4,802799
26,37	31,26	29,69	5,287976
29,07	30,82	29,19	5,584104
33,1	30,23	28,57	5,810291
37,01	29,5	28,15	4,795737
38,86	29,25	28	4,464286
40,82	29,03	27,85	4,236984
45,1	28,44	27,61	3,006157
48,8	28,28	27,45	3,023679

3. Thermal model of the soil

3.1. Soil depth temperature:

Figure 1 shows the average annual variation in soil temperature for different depths. We notice that the average temperature of the soil decreases when we penetrate more into the soil. The depth of the soil plays a very important role in cooling

Table 4.3. Thermo-physical properties of the soil.

Soil	Soil specific heat capacity (J/Kg .K)	Soil thermal conductivity(W/m.°c)	Soil density (kg/m3)
	1840	0.52	2050

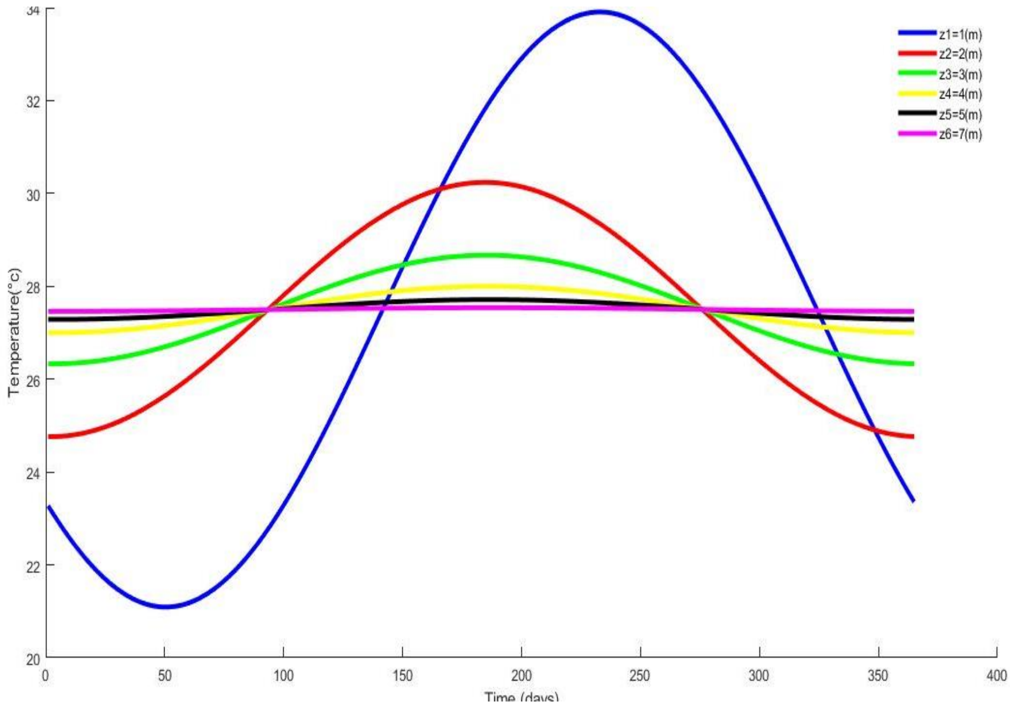


Figure 4.1. Annual evolution of soil temperature as a function of depth and time.

3.2. Effect of thermal conductivity of soil surrounding the tube:

Three different thermal conductivities (0.5 W/m.K, 1.5 W/m.K and 4 W/m.K) are taken into account to evaluate the thermal performance and determine the optimum conductivity of the soil. We noticed in fig.4.2. that the thermal performance is strongly affected by the soil having low thermal conductivity. On the other hand, the high thermal conductivity of the soil facilitates the evacuation of the accumulated heat towards the layers of the soil away from the EAHE.

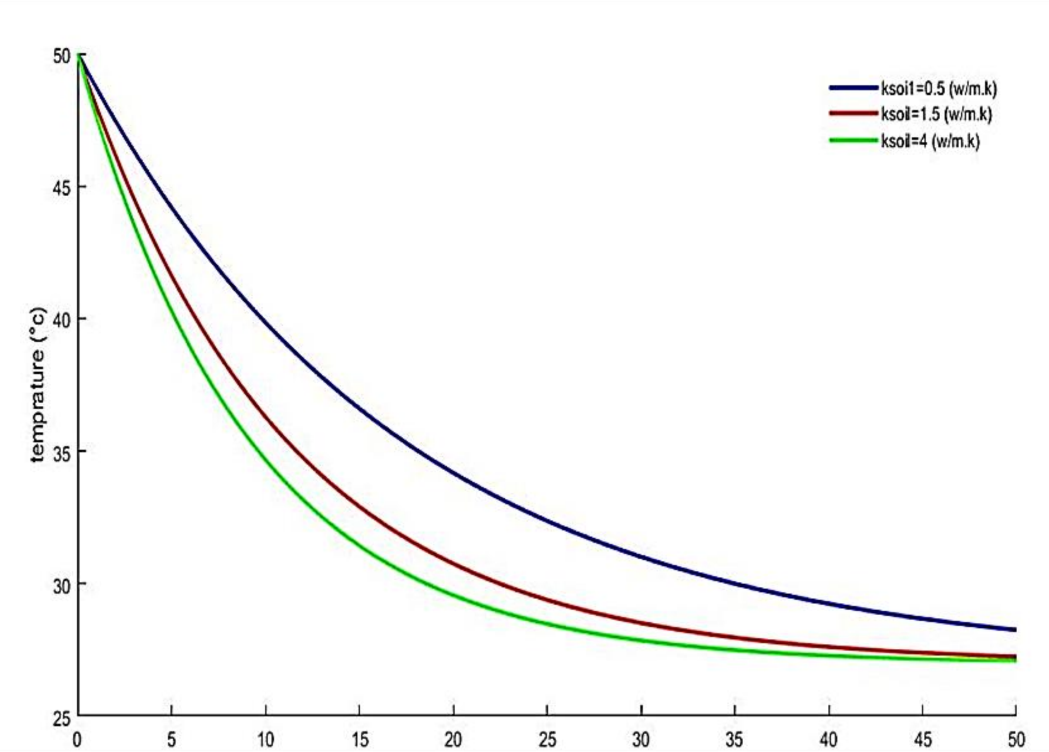


Figure 4.2. Air temperature variation for different thermal conductivity.

3.3. Effect velocity of air entering to the EAHE:

Three air flow velocities (1.5 m/s, 2 m/s and 4 m/s) are considered in the analysis of the thermal performance of the EAHE. The fig.4.3 represents the temperature drop curves for different air flow velocities. Air temperatures rises with increasing air flow velocity. That means, there is a direct proportion between air velocity and temperature. It’s clear that the decrease of air flow velocity deteriorates the thermal performance of EAHE, because of the fluid didn’t have enough time to exchange its heat with the soil surrounding the buried tube.

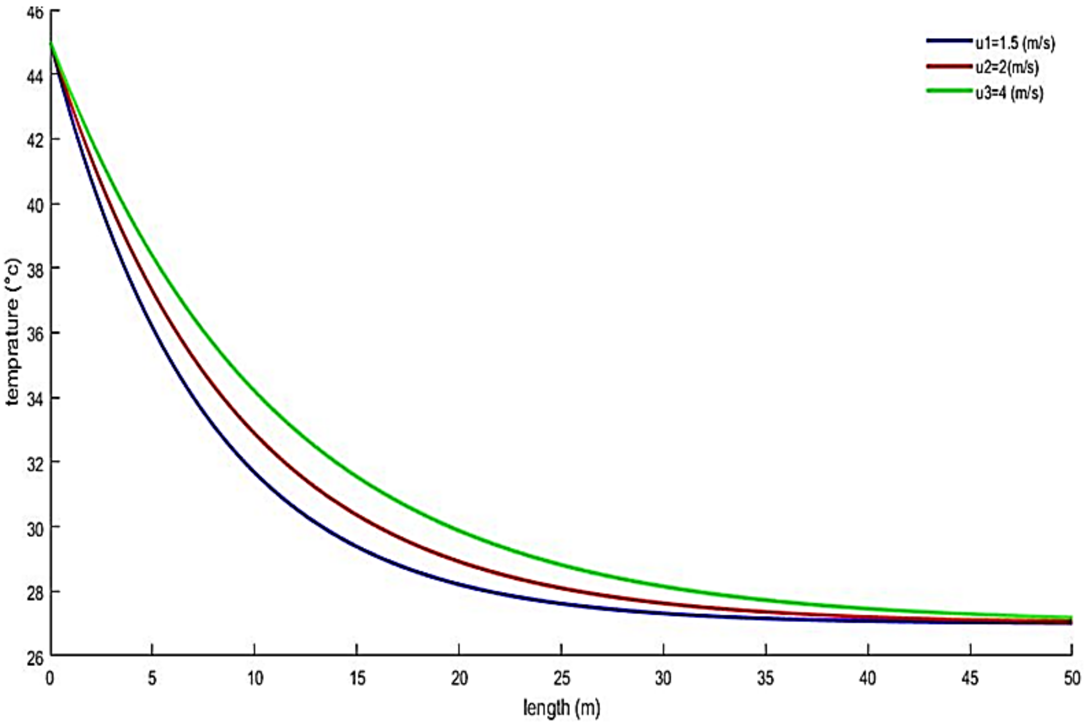


Figure 4.3. Effect of air velocity air entering the heat exchanger.

3.4. Effect of buried tube diameter on temperature drop:

Three radius (r11=0.055 m, r12=0.058 m, r13=0.061m) were taken in order to study its narrow and wide with the drop in temperature. Where we notice, as shown in ig.4.4, that the air temperature gradually increases as we expand the radius of the tube. However, it should be noted that the thermal performance deteriorates with the decrease in the radius of the tube, which is necessary for the success of the experiment.

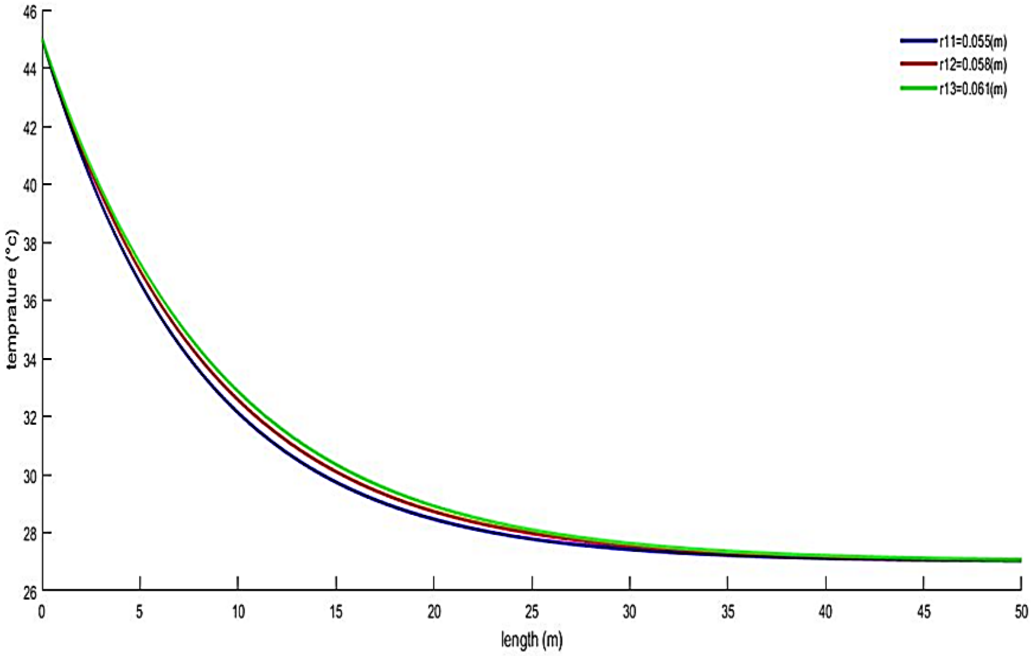


Figure 4.4. Effect of buried tube diameter on temperature drop.

4.Gas turbine model:

we choose a gas turbine monoshaft to do that study and the table below shows the Technical specification of the selected gas turbine[27].

Table 4.4. Technical specification of the selected gas turbine.

Item	Adopted values
Gas turbine output, MW	124.269
Air inlet temperature (ISO), °C	15
Relative humidity, %	60
Ambient pressure, bar	1.009
Average air mass flow rate, kg/s	446.7431
Fuel gas mass flow rate, kg/s	6.640592
Inlet temperature to turbine, °C	1150
Exhaust gases temperature, °C	552.1111
Gas lower heating value, kJ/kg	48,741
Compression ratio	12.046
Combustion chamber pressure loss, bar	1.5
Isentropic efficiency of compressor, %	85
Isentropic efficient of turbine, %	88
Combustion efficiency,%	99
Natural Gas lower heating value, kJ/kg	47,040
Efficiency, %	37.2

4.1. Gas turbine performance :

The performance of selected gas turbine is evaluated due to using EAHE system The cooling performance curve of EAHE is presented in Fig.4.5. Operational results confirm the cooler ability to reduce the ambient temperature from 44 °C to 27.318° so the efficiency rises from 33.6 % to 34.34 % .

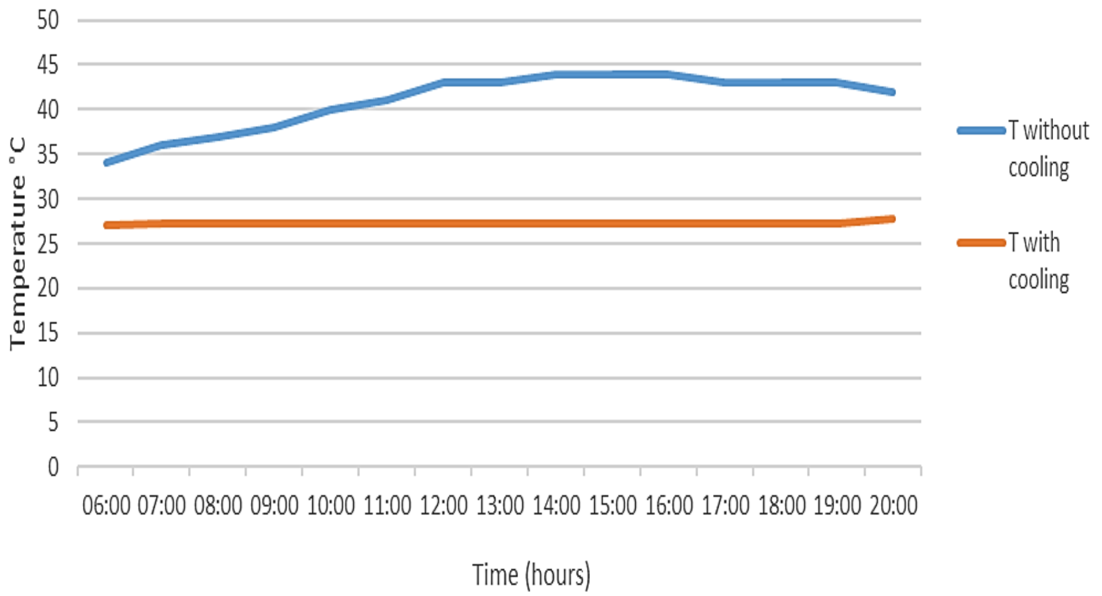


Figure 4.5. Variation of outlet air temperature from EAHE with time. The day of 08.06.2021 in Ouargla.

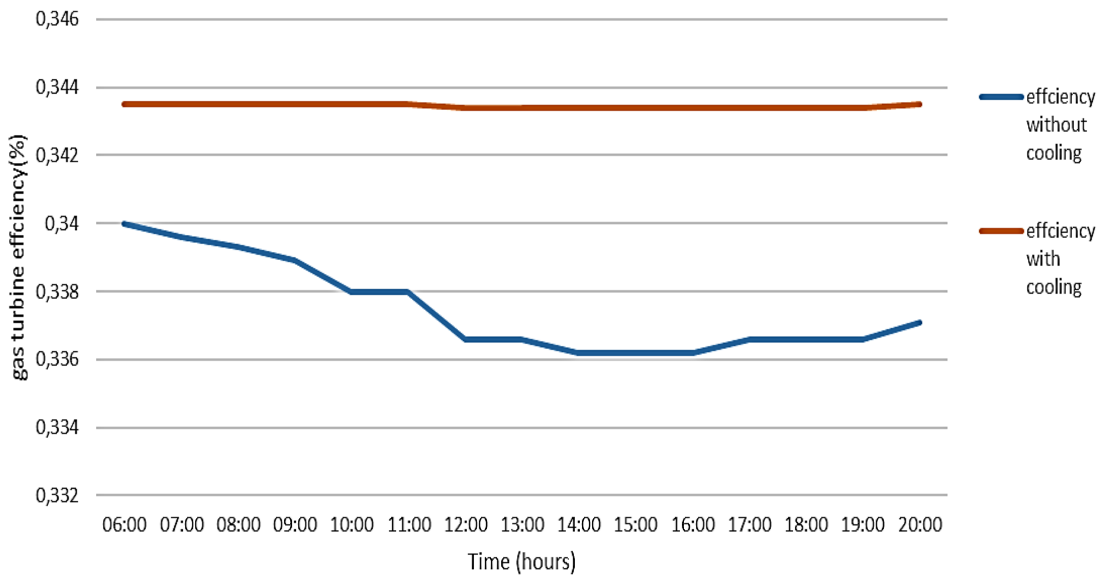


Figure 4.6. Variation of thermal efficiency with time.

Fig 4.6. shows the impact of utilizing the proposed cooling system on the thermal efficiency. As the ambient temperature decreases from 44°C to 27.318 °C, the efficiency of the system will increased from 33.6% to 34.34%. This because of the additional output power accomplished by the cooling system is higher than the extra fuel used.

5. Conclusion:

In the present chapter, EAHE system was suggested for gas turbine cycle inlet air-cooling. Transient one-dimensional model was developed for predicting the impact of main geometrical and dynamical parameters including tube length, inner tube diameter and inlet air velocity on the performance of the EAHE system. MATLAB program was developed for solving an analytic model based on the energy balance principal.

General conclusion

General conclusion

In the face of the growing use of gas turbines in the industrial field of petroleum, reducing them to replace them with renewable energies is unlikely. As we belong to this field and try to develop and adopt it, we decided to integrate it into the energy industry to contribute to raising the efficiency of GT through EAHE.

At the end of this work, we presented the study and development of several techniques for cooling the intake air temperature applied in a GT cycle, which increased the mass load. The study of the conditions on-site, precisely the ambient temperature, influences the effectiveness of the GT (more particularly in the Algerian desert). The method applied was evaporative cooling of the temperature of the intake air in the axial compressor.

According to bibliographic research, ambient air cooling is linked to climatic conditions; it is infrequent in the world because it is used in hot regions of the planet (the case of Algeria). This research found similar work dealing with ambient air cooling by a method of use in previous studies. These studies were our reference points for validating our results.

A computer code (in the MATLAB program) was used to apply a methodology analyzing the potential of EAHE and its role in ambient air cooling as a first step, and the parameters were changed for each result.

In the second step, by the GT calculation code, we see the efficiency once it is without cooling and again with the cooling, which is our goal.

The numerical simulation results allowed us to come out with several observations on the effects of the conditions on the efficiency of the GT. We summarize in the following the main results obtained:

1. As we go deeper into the earth, the ambient temperature decreases more, which is limited.
2. The performance of the GT is highly dependent on climatic conditions.
3. Cooling the ambient air temperature increases air density, which directly affects the mass load of the circulating fluid, increasing power.
4. The temperature in the outlet of EAHE decreases from 44°C to 27°C.
5. The efficiency of GT is improving from 33.6% to 34.34%.

Finally, we hope that our modest research based on a theoretical study can be a source of work for the students and be considered in the future, more precisely in practice. Thus, this work was carried out to continue future research work to contribute to renewable energy, energy efficiency, and pollution control. Atmospheric when operating the Gas Turbines.

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Abstract:

The use of earth air heat exchangers doesn't limit only in cooling of buildings also can be used to improvement of efficiency of gas turbine to get the high performance. One-dimensional model was developed for predicting the thermal performance of EAHE, also gas turbine efficiency assessed with application of EAHE. MATLAB program is developed for solving an analytic model based on the energy balance principal. A validation study was devoted between our proposed model and experimental results. It shows that the mean error was 3.58% also the ambient temperature in outlet of EAHE decreases from 44°C to 27.4°C so that resulting to improve the efficiency of gas turbine from 33.6% to 34.34%.

Keywords: Gas turbine, Earth air heat exchanger (EAHE), Cooling.

Résumé:

L'utilisation d'échangeurs de chaleur air sol ne se limite pas seulement au refroidissement des bâtiments, elle peut également être utilisée pour améliorer l'efficacité de la turbine à gaz afin d'obtenir des performances élevées. Un modèle unidimensionnel a été développé pour prédire les performances thermiques de l'EAHE, ainsi que l'efficacité des turbines à gaz évaluée avec l'application de l'EAHE. Le programme MATLAB est développé pour résoudre un modèle analytique basé sur le principe du bilan énergétique. Une étude de validation a été consacrée entre notre modèle proposé et les résultats expérimentaux. Il montre que l'erreur moyenne était de 3,58% et que la température ambiante en sortie d'EAHE diminue de 44°C à 27,4°C, ce qui entraîne une amélioration du rendement de la turbine à gaz de 33,6% à 34,34%.

Mots clés: Turbine à gaz, Echangeur de chaleur air-sol (EAHE), Refroidissement.

ملخص:

لا يقتصر استخدام المبادلات الحرارية للهواء الأرضي على تبريد المباني فقط ، كما يمكن استخدامه لتحسين كفاءة التوربينات الغازية للحصول على أداء عالٍ. تم تطوير نموذج أحادي البعد للتنبؤ بالأداء الحراري لـ EAHE، وكذلك كفاءة التوربينات الغازية التي تم تقييمها مع تطبيق EAHE. تم تطوير برنامج MATLAB لحل نموذج تحليلي يعتمد على مبدأ توازن الطاقة. تم تخصيص دراسة التحقق بين نموذجنا المقترح والنتائج التجريبية. يوضح أن متوسط الخطأ كان 3.58٪ كما انخفضت درجة الحرارة المحيطة بمخرج EAHE من 44 درجة مئوية إلى 27.4 درجة مئوية مما أدى إلى تحسين كفاءة التوربينات الغازية من 33.6٪ إلى 34.34٪.

الكلمات المفتاحية: التوربينات الغازية، المبادل الحراري للهواء الأرضي (EAHE)، تبريد.