Energy Analysis of a Topping Steam Turbine Driven Cogeneration System for a Prototype Hospital Building

BY

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THESIS

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I find myself in my bedroom in Chicago and I am thinking about the long journey that brought me here and all the people that I met along this journey that helped me get where I wanted to be. It seems yesterday that I began the university in Torino but almost six years have passed by and I would not have beeen able to achieve what I did without the help of some wonderful people.

First of all, I would not be here right now without the help of my grandfather, Clemente. Thank you for everything, you are able to help me, influence me and support me even now, I will always bring you with me in my heart.

Thanks to my family and especially to my mother, she has always been and I know that she will forever be my number one supporter, no matter what choices I make or where I end up. You are the strongest person on this Planet, even if you don't really know it, and I know how many sacrifices you made for me, thank you from the bottom of my hear.

I would like to thank my advisors as well. Thanks to prof. Ryan for helping me during this past year with this work and through all the difficulties that came up on the way. Thanks to prof. Masoero for the availability that he showed and his assistance.

Thanks to Eugenia for being by my side along this journey, from the very first day. It was never easy but we were able to be strong for one another every time.

I would also like to thank Edoardo, for being a true friend. I always carry with me your advice and I will never stop looking for my happiness, non matter where it brings me.

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LIST OF ABBREVIATIONS

CHP	Combined Heat and Power
eQUEST®	The QUick Energy Simulation Tool
NREL	National Renewable Energy Laboratory
DOE	Department of Energy
ASHRAE	American Society of Heating Refrigerating and Air-conditioning Engineers
MOB	Medical Office Building
HVAC	Heating, Ventilation and Air-Conditioning
VAV	Variable Air Volume
PT	Patient Tower
SHP	Separate Heat and Power
EPA	Environmental Protection Agency
GHG	Greenhouse gas
HEX	Heat Exchanger
CCHP	Combined Cooling, Heat and Power

SUMMARY

The goal of this thesis is to analyze and improve the energy needs of an ASHRAE Standard 90.1-2004 prototype health care facility located in Chicago, IL, USA. In order to do that, a simulation software has been used, eQUEST®– the Quick Energy Simulation Tool. The latter is a free open source software provided and developed by the US Department of Energy (DOE). Thanks to the software, the electric and heat demands of the building for the whole year have been analyzed and the electric and fuel consumption have been evaluated.

With the aim of improving the building baseline model a cogeneration system (CHP – combined heat and power) was introduced, using an electric generator and a heat recovery loop. In particular, a steam turbine was selected as generator, being this specific component never been used for this type of simulations on eQUEST®.

Cogeneration is an innovative way of producing energy which can lead to energy and cost savings as well as emissions' reduction. Over the last decade several plants have been installed in health care facilities, which are well suited for this purpose. Hospitals, in fact, usually run 24 hours a day every day of the year therefore they need a constant electric and thermal production that should also be very reliable. The benefits of the turbine introduction were evaluated, in terms of electricity generated and thermal heat recovered in order to satisfy the building loads. More in detail, considering that the electric efficiency of steam turbines does not reach very high values, an important control decision was made. The electric generator was set to follow the thermal load, which means that the main goal of the turbine operation is to satisfy the thermal load, while at the same time producing electricity that will partially cover the electric load. Electricity becomes, then, a byproduct.

Different sizes of the turbine from 500 kW to 2000 kW and steam conditions (in terms of inlet and outlet pressure) have been proposed and the optimal conditions have been found using as discriminant the overall efficiency of the system as well as the percentage reduction in source energy consumption. However, CHP efficiency is probably not the best parameter to analyze the performance of a system like this. In fact, when considering the reduction in source energy consumption, the best results where actually obtained with a 1000 kW turbine and steam expaning from 350 psi to 125 psi. The latter can be considered the best and most realistic situation that was simulated in this work, which can be used as a starting point for future works, performing for example an economic analysis.

CHAPTER 1

THE HOSPITAL FACILITY

This first chapter describes the health care facility that will be evaluated and studied in this thesis. Starting from the baseline model of the hospital, all the improvements and new conditions will be compared to the baseline itself, with the aim of understanding how the needs of the building are affected by the changes and if the latter were able to improve the facility loads, energy consumption and costs.

1.1 The Prototype Hospital

The model of the hospital was realized with the implementation on eQUEST®of the design addressed by the National Renewable Energy Laboratory (NREL) [1] [7]. Thanks to previous works, the model has already been validated and has undergone further improvements concerning some irregularities found through the analysis of the output records [8] [9].

The Commercial Buildings Group at NREL developed the hospital model and all its technical support documentation with the supervision of the U.S Department of Energy Building Technologies program. The guidelines for the definition of the prototype are the result of US Department of Energy (DOE) support to the development of energy codes and standards for commercial buildings. On the NREL's website the codes are available, in order to promote the transparency of the NREL's actions in its support, and also promote the validation of its work thanks to public review and application of the developed products. The main goal of the study was to enable the building to achieve energy savings up to 50 % or over the ASHRAE Standard 90.1-2004.

ASHRAE Standard 90.1 (Energy Standard for Building Except Low-Rise Residential Buildings) [7] is a US standard that provides the minimum accepted requirements for energy efficient buildings' design. The first release of ASHRAE Standard 90 was in 1975. Since then different upgrades have been released, in 2004, 2007, 2010, 2013 and 2016.

Starting from the ASHRAE 90.1 and NREL data, the prototype hospital was implemented on eQUEST®. In order to simulate the prototype facility, some steps were followed to describe the main features and characteristics of the building. The information can be summarized in the following categories:

- Building internal layout description;
- Building shell description;
- Internal loads;
- Occupation and equipment schedules;
- Lighting scheludes;
- Heating, ventilating and air conditioning information;
- Service water heating.

Considering that all the improvements to the facility were implemented starting from the prototype model of the hospital and that each one of them was compared to the basleine model itself, from now on, the prototype hospital will be referred as the base model.

1.2 Hospital layout and features

In Figure 1 there is a 3D representation of the entire hospital.

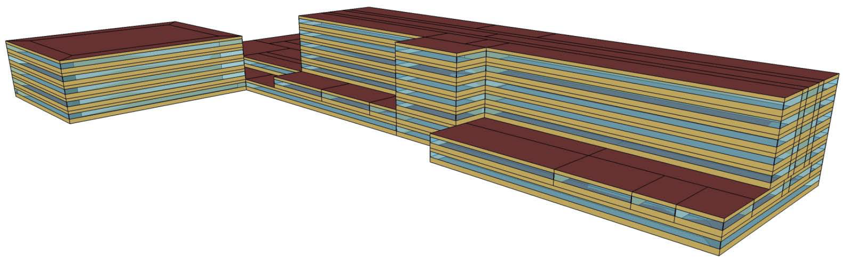


Figure 1: 3D Rendering of the health care facility [1]

The total surface of the base model is $527\,000\,\mathrm{ft}^2$ ($49\,000\,\mathrm{m}^2$) which is split in two main buildings: the hospital space of $427\,000\,\mathrm{ft}^2$, divided into seven stories, and the medical office building (MOB) of $10\,000\,\mathrm{ft}^2$, divided into five stories.

In Appendix A it is possible to see the specific maps of each floor. Table I summarizes the main features of the building.

TABLE I: **MAIN LAYOUT FEATURES OF THE PROTOTYPE BUILDING**

Hospital Parameter	**Value**
Total floor area	$527\,000\,\mathrm{ft}^2$ ($49\,000\,\mathrm{m}^2$)
Hospital floor area	$427\,000\,\mathrm{ft}^2$ ($39\,670\,\mathrm{m}^2$)
MOB floor area	$10\,000\,\mathrm{ft}^2$
Number of floors - hospital	7
Number of floors	5
Floor-to-floor height	10 ft (3.05 m)

The building is located in Chicago, Illinois, which corresponds to the climate zone that has been defined as cool-humid, 5A by ASHRAE 169-2006 [10].

Considering that no fixed prescription was available for the fenestration data, a 40 % fraction of fenestration to gross wall area had been adopted as an assumption. The structure of the building is characterized by a steel frame construction and an arrangement of the roof with insulation above the deck.

The equipment of the HVAC – heating, ventilation, and air conditioning system consist of central air handling units, boiler, chillers, chilled and hot water air handling unit coils and terminal units with hot water reheat coils. This description is valid only for the base model and a representation of this system on eQUEST®is shown in Figure 2.

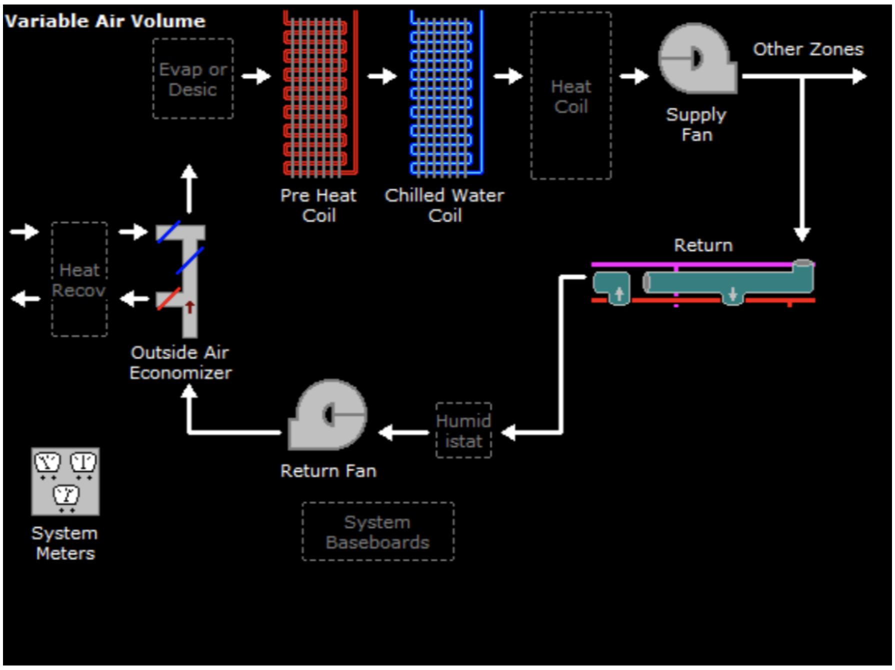


Figure 2: Layout of the HVAC system - Air Side taken from eQUEST®

Each room and space of the building has its specific occupancy and all the spaces of the facility are considered to operate with the same schedule:

- Hospital space 24/7;
- Medical office building (MOB) from 7:00 am to 5:00 pm;
- Extended hours from 5:00 am to midnight.

As already mentioned, thanks to previous works [9], some improvements were introduced to the base model in order to fix some discrepancies between the data provided by the thermostats in each space and their actual design temperature. Some spaces, for a considerable number of hours throughout the year, were at a temperature lower than their design one. The problem was due to the interaction of the thermal load with some default parameters in the eQUEST®-HVAC system properties. The improvements include:

- The introduction of pre-heat coils in each of the variable air volume (VAV) HVAC system;
- The increase of the maximum increase in temperature for supply air passing through the reheat coils, which in the software is called *Reheat Delta T*;
- The increment of the sizing ratio.

Another important thing to point out is the definition of the different spaces inside the building. A space is represented by one or more rooms that are characterized by the same desired thermal condition. Each space is managed by the same VAV system. In order to simulate the building performance and needs on the software it is necessary to divide the whole building in different spaces. The number of spaces is decided based on the complexity of the thermal control strategy of the building. The hospital has been divided in nine main spaces:

- MOB - Medical Office Building, five stories;
- PT - Patient Tower, five stories (from third to seventh floor);
- Building 3, two stories;
- Building 4, two stories;
- Building 5, one story;
- Building 6, two stories;

- Building 7, three stories;
- Building 8, one story;
- Building 9, two stories, located below the PT.

The zones are shown in Figure 3.

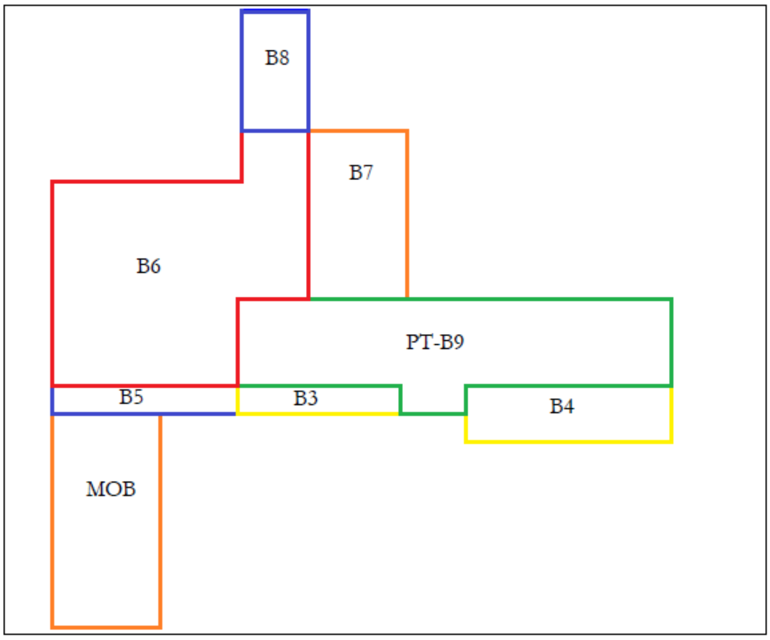


Figure 3: Layout of the different spaces in the building [1]

All the features and assumptions described in this section have been used to develop the base model of the prototype building. The model has been implemented as already mentioned, in eQUEST®and in the next section the energy needs of the facility will be shown and evaluated, being the starting point of this study.

1.3 Building Loads and Consumption Profiles

Thanks to the simulation tool eQUEST®, the hospital energy loads and consumption profiles have been determined. The software allows the user to see this profile throughout the whole year hour-by-hour. The electric consumption profile and fuel consumption profile have been extrapolated from the hourly reports provided by the software itself and post processing on those data has been done thanks to other softwares such as Excel and MATLAB.

The electric demand expressed in MW is shown in Figure 4 and it represents the electric load throughout all the 8760 hours of the year.

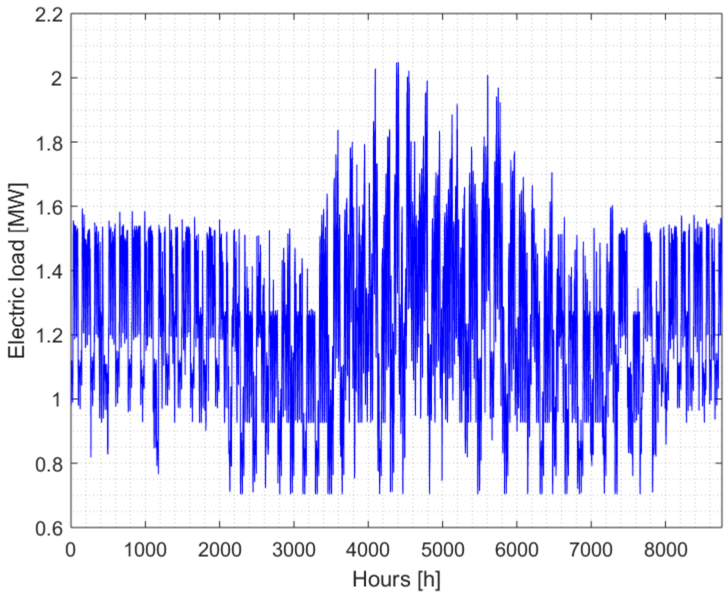


Figure 4: Annual electric load for the baseline model

By looking at the previous figure we can understand the trend of the electric load over the entire year. It is noticeable that the load never decreases to zero, because the hospital needs electricity over the entire year due to its constant and continuous operations. Of course, there are some fluctuations due to the time of the day and the seasons. The peak demand occurs during summer when, due to the cooling load, the air conditioning system is on, and it requires a large amount of electricity to run the electric chillers.

Thanks to eQUEST®it is also possible to show the monthly electric demand, month by month, expressed in GWh and it is shown in Table II.

TABLE II: **MONTHLY ELECTRIC DEMAND**

Month	**Electric Demand (GWh)**
January	0.77
February	0.69
March	0.77
April	0.76
May	0.83
June	0.9
July	1.01
August	0.98
September	0.84
October	0.8
November	0.73
December	0.77
Total	9.84

In addition, it is possible to check what are the main categories that consume electricity and how much they influence the electric load and the consequent electric consumption. Table III and Figure 5 show them.

The largest contribution to the electric laod and consequent consumption is due to lighting, as it could have been expected, considering the coninuous operation of the hospital throughout the entire year. The second largest contribution is due to *Miscellaneous Equipment*, while *Space Cooling* is not a high percentage because it mainly occurs during summer, even though it is responsible for the peak demand in terms of power.

TABLE III: **ANNUAL ELECTRIC CONSUMPTION BY CATEGORY**

Category	**Annual Electric Consumption (GWh)**
Space Cooling	1.03
Heat Rejection	0.12
Ventilation Fans	0.9
Pumps and Auxiliary	1.18
Miscellaneous Equipment	3.03
Area Lights	3.57

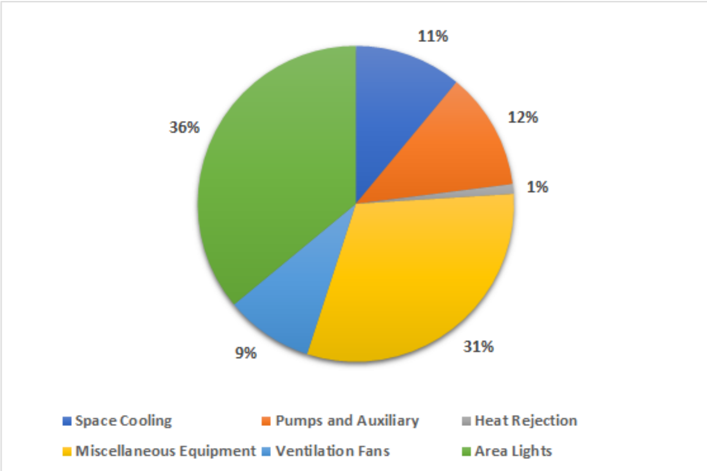


Figure 5: Annual distribution of the electric consumption

Another useful tool to understand the behavior of the electric demand and consequent consumption, is the duration curve. This curve represents the electric load over the entire year sorted in descendent order, starting from the highest value (peak demand) to the lowest one. It is represented in Figure 6

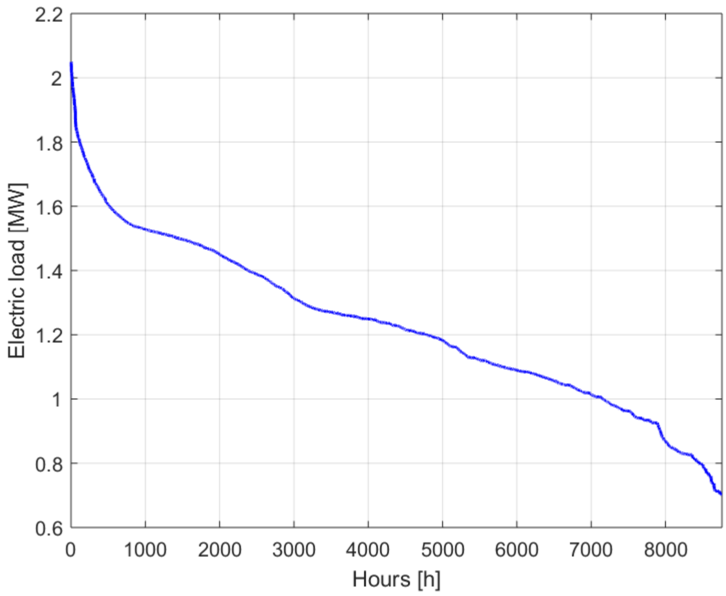


Figure 6: Duration curve - electric load for the baseline model

The same procedure was followed for the fuel cosumption. Figure 7 shows the fuel consmuption over the entire year, while Figure 8 represents the duration curve.

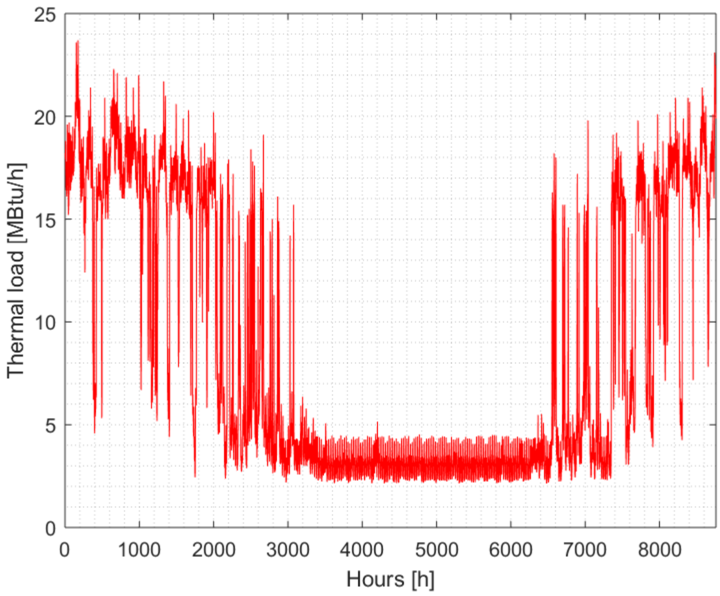


Figure 7: Annual fuel load for the baseline model

As expected, the fuel consumption is higher during the winter season due to space heating, while during summer is lower and pretty much constant because of the only presence of the hot water demand. The peak is reached in January and it is around 24 MBtu/h, while the summer average is around 4 MBtu/h.

Just like it was done for the electric demand, the monthly fuel consumption has been tabulated and the contribution of each component (*Space Heating* and *Domestic Hot Water*) to the demand over the entire year has been plotted.

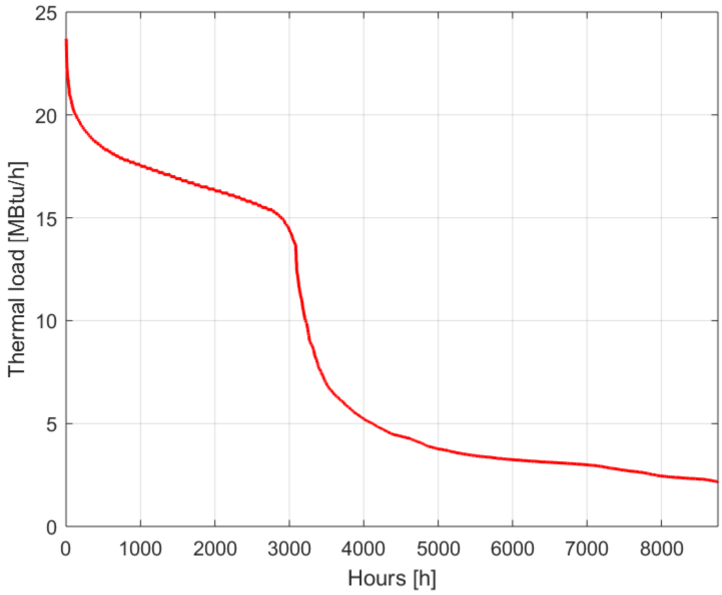


Figure 8: Duration curve - fuel load for the baseline model

TABLE IV: **MONTHLY FUEL DEMAND**

Month	**Fuel Demand (GBtu)**
January	7.91
February	6.41
March	5.95
April	3.85
May	2.73
June	2.23
July	2.28
August	2.28
September	2.31
October	3.36
November	5.07
December	7.27
Total	51.64

TABLE V: **ANNUAL FUEL CONSUMPTION BY CATEGORY**

Category	**Annual Fuel Consumption (GBtu)**
Space Heating	28.27
Domestic Hot Water	23.37

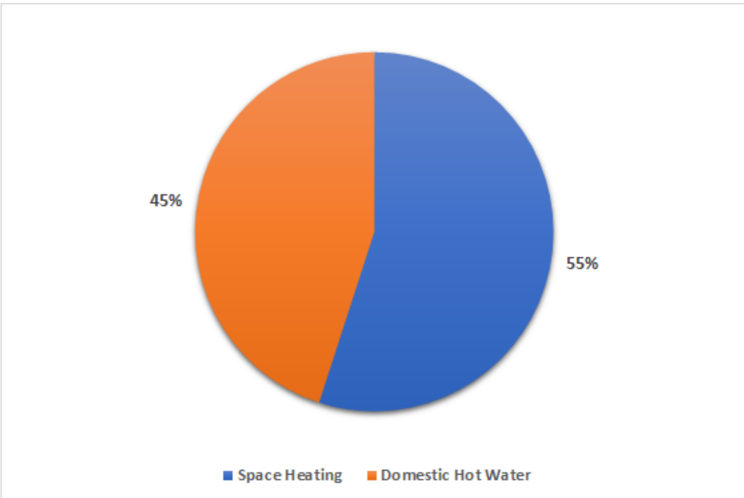


Figure 9: Annual distribution of the fuel consumption

CHAPTER 2

CHP - COMBINED HEAT AND POWER

This chapter focuses on the description of the technology that has been used to improve the needs of the hospital, cogeneration, also known as CHP - Combined Heat and Power. This section represents an intermediate step, before analyzing the specific components added to the system and the final results, needed in order to better understand what is the key idea of this study. The concepts behind CHP applications must be addressed and a knowledge on how they work should be established, knowing also what benefits they bring.

2.1 Overview of the technology

The term Combined Heat and Power (CHP) refers to technologies that efficiently produce electricity and at the same time capture or recover heat that would otherwise be wasted, with the aim of providing useful thermal energy, which can be delivered in the form of steam or hot water, for space heating or domestic hot water use [4]. Thus, in CHP systems we have the simultaneous generation of two forms of energy starting from a single fuel or energy source, electricity and heat.

CHP represents a more efficient and productive way of using a fuel, because it re-uses the heat that would be wasted during the production of electricity. In opposition to traditional separate heat and power (SHP) facilities, cogeneration systems are much more efficient. In fact, in traditional power generation plant, almost two third of the energy used is wasted in the form of heatr ejected to the atmosphere. In addition to that, during the transmission and distribution of electricity to the end-users, energy is wasted too.

According to the US Department of Energy (DOE), there are more than 4,400 sites around all the United States and its territories, reaching a capacity of 81.3 GW installed [2]. The data were updated last August (2018) and includes data through December 31, 2017, including newer CHP systems in Puerto Rico and the U.S. Virgin Islands. This new update, also states that these CHP systems save 1.8 quads of fuel each year, thus avoiding 241 million metric tons of CO2 emissions per year [2].

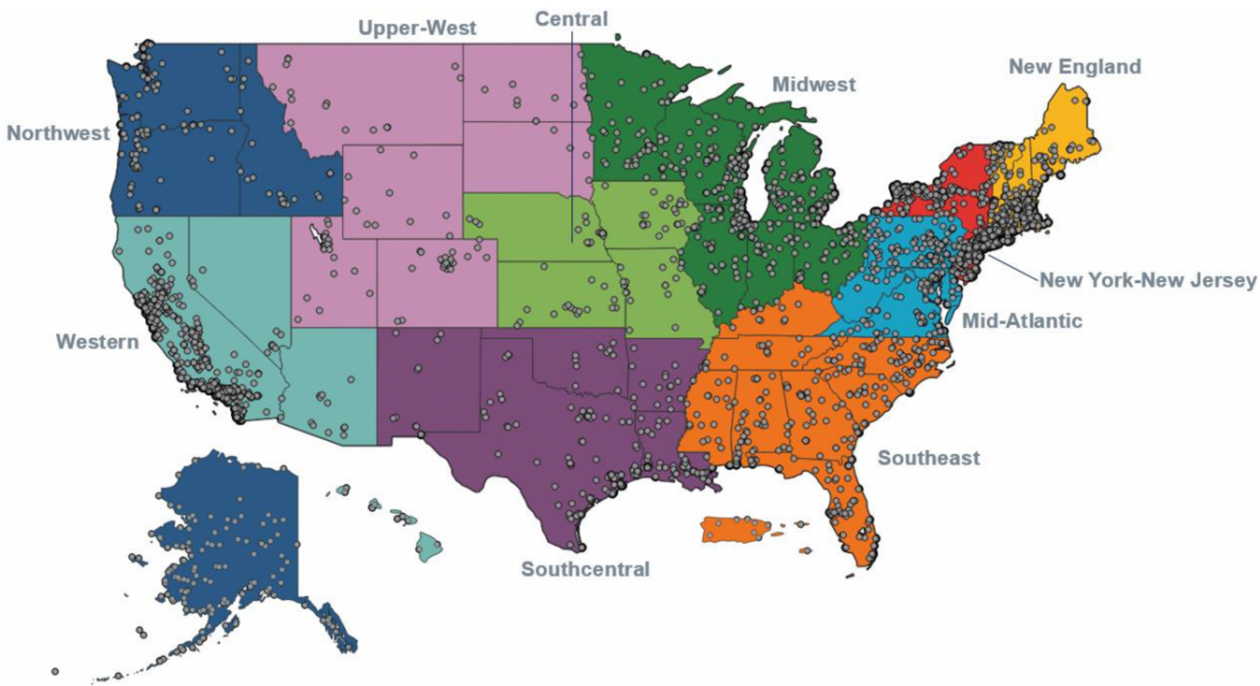


Figure 10: Current CHP systems installed in the U.S. [2]

The main applications of CHP systems are:

- **Commercial buildings.** Office buildings, hotels, health clubs;
- **Residential.** Condos, apartments;
- **Institutional.** Universities and colleges, hospitals, prisons;
- **Municipial.** District energy systems, wastewater treatment facilities;
- **Manufacturers.** Food processing, pulp and paper, refining, chemical, glass manufacturing.

2.2 CHP Benefits

As already mentioned, CHP systems bring a lot of benefits compared to traditional separate heat and power plants. The main benefits can be summurized into four main categories: efficiency benefits, environmental benefits, economic benefits and reliability benefits [3].

- **Efficiency benefits**

  The average efficiency of traditional fossil fuel-powered plants ranges from 30 % to 45 % [11]. What this means is that when producing electricity in power plants, around two third of the energy used is wasted. The most common form in which energy is wasted is heat discharged in the atmosphere. Subsection 2.3 shows the way the efficiency of CHP systems is calculated and correlated to traditional power plants. Typically the efficiencies reache with CHP plants range from 60 % to 80 % [3].

- **Environmental benefits**

  By recovering potentially wasted heat, cogeneration systems require less fuel compared to traditional plants, in which electricity is purchased and thermal energy is produced on-site. This implies a reduction in greenhouse gas (GHG) emissions that can reach 30 % [3].

- **Economic benefits**

  Due to the higher efficiency and the fact that the fuel needed to run the system is lower, CHP can make the plant save money. More in detail, there are several economic benefits:

  - Reduced energy cost;
  - Avoided capital cost;
  - Less exposure to electricity rate increase;
  - Protection of revenue streams.

- **Reliability benefits**

  Considering that cogeneration systems are small and widely distributed and that they are located at the point of use, they are are independent in terms of fuel and operation. Therefore, they are less vulnerable to problems and issues related to the transimission lines and they are able to provide power in case of an emergency. This allows CHP systems to be more reliable than traditional plants.

2.3 CHP Efficiency

As mentioned in section 2.2, one of the benefits of CHP systems is the increased efficiency. The efficiency of CHP systems is higher because every cogeneration system recovers heat that would be otherwise wasted.

Figure 11 shows how the efficiency of a cogeneration plant compared to conventional generation. In the example, the CHP system is driven by a 5 MW Natural Gas Combustion Turbine while the conventional generation is represented by a power plant for the production of electricity, having an efficiency of 33 %, and an on-site 80 % efficient boiler for the useful thermal energy procution.

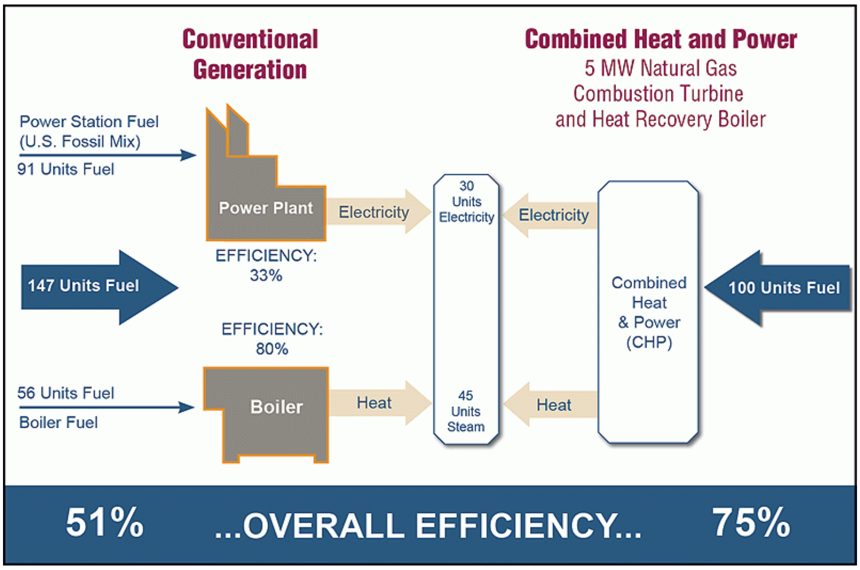


Figure 11: Overall efficiency of a CHP systems versus conventional generation [3]

According to EPA [3], there are two main definitions of efficiency when dealing with CHP systems.

- **Total system efficiency ($\mathbf{EFF_{CHP}}$)**

  It is often used to compare the efficiency of CHP plants to conventional systems that have the separate production of electricity (purchased from the grid) and theraml power (on-site boiler). The equation used to calculate the total system efficiency is represented in (Equation 2.1).

$$\mathrm{EFF_{CHP}} = \frac{\mathrm{E} + \mathrm{Q}}{\mathrm{F}} \tag{2.1}$$

  Where:

  - E = Net useful electric output of the electric generator;
  - Q = Net useful thermal output, as to say the useful heat that is recovered from the generator that would be otherwise wasted;
  - F = Total energy fuel input that is consumed in order to run the generator.

  This term represents the combined outputs of the cogeneration system based on the fuel consumption.

- **Effective Electric Efficiency (EE)**

  It represents the measure used when comparing the electricity produced by CHP systems to the electricity produced by conventional power plants. It can be calculated as represented in (Equation 2.2).

$$\mathrm{EE} = \frac{\mathrm{E}}{\mathrm{F} - \frac{\mathrm{Q}}{\alpha}} \tag{2.2}$$

Where:

- α = Efficiency of the conventional technology that would be used in order to produce thermal energy in the case there was no CHP system.

2.4 CHP Components and Management

The main components of a cogeneration plant are:

- **Prime Mover.** It is the core of the system, because it is the component that produces the needed mechanical energy, which is converted in the generator into electricity, and from which waste enegy is recovered into useful thermal energy. The most common prime movers are: Combustion Turbines, Reciprocating Engines, Steam Turbines, Microturbines, and Fuel Cells.
- **Generator.** It is connected to the prime mover with a shaft and converts mechanical power into useful electricity.
- **Heat Recovery Unit.** This component represents the main difference between traditional plants and CHP systems. It is able to recover the waste heat coming out from the prime mover and transfer it to a fluid (hot water or steam) which will cover the space heating and domestic hot water needs.
- **Boiler.** This component is only present when the prime mover is a steam turbine. It generates the needed amount of steam in order to run the steam turbine.

Considering the above-mentioned components of typical CHP systems, it is possible to identify two main system configurations for CHP plants. They are represented in Figure 12 and Figure 13. As it will be explained in the following chapters, the configuration that was used in this work is the second one, represented in Figure 13.

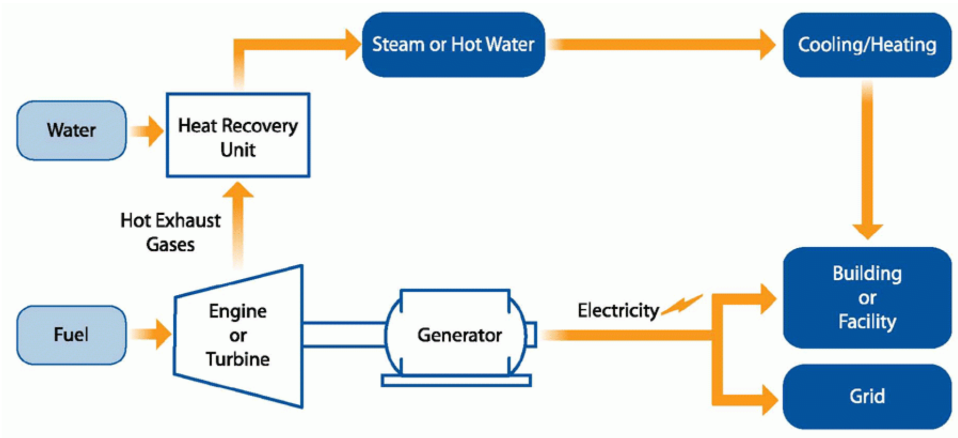


Figure 12: System configuration in the case of a Combustion Turbine, Mictroturbine or Reciprocating Engine as prime mover [4]

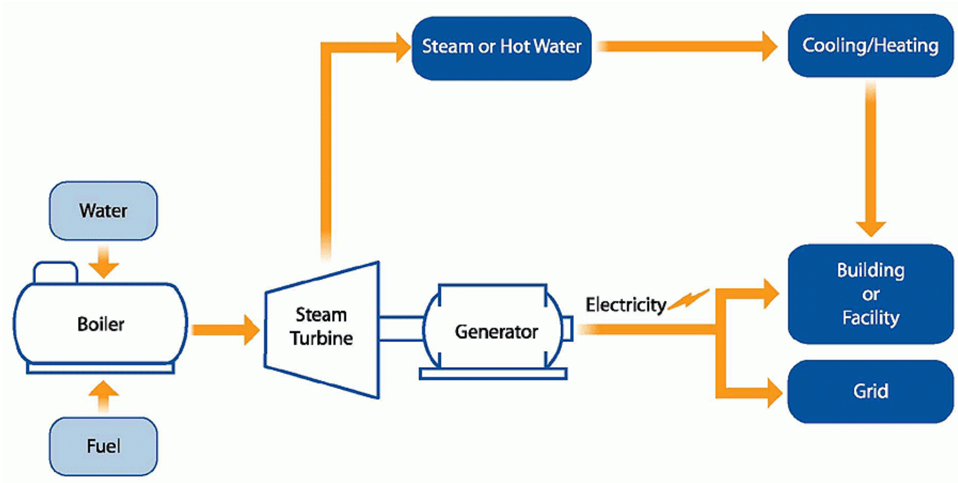


Figure 13: System configuration in the case of a steam turbine as prime mover [4]

In the first case, the fuel is burnt directly in the prime mover, being a Reciprocating Engine, a Combustion Turbine or a Microturbine. In the second case there is a boiler because the prime mover is a Steam Turbine and the fuel is fed to the boiler where there is the production of high pressure steam that will be directed to the turbine.

Based on the prime mover that is chosen to "run" the system the costs (initial and operation and maintenance) and the specific characteristics of the plant will change. In addition, every prime mover has different features and performances that will impact the overall system efficiency, savings and emissions.

In Table VI and Figure 14 and Figure 15 it is possible to see the number and corresponding percentage of sites and installed capacity of different CHP plants in the U.S. based on the prime mover. The data has been taken from [12] which evaluates the state of the art up to 2014.

TABLE VI: **U.S. CHP INSTALLED SITES AND CAPACITY BASED ON THE PRIME MOVER**

Prime Mover	**Sites**	**Sites share (%)**	**Installed Capacity (MW)**	**Capacity share (%)**
Reciprocating Engine	2,194	51.9	2,288	2.7
Gas Turbine	667	15.8	53,320	64
Steam Turbine	734	17.4	26,741	32.1
Microturbine	355	8.4	78	0.1
Fuel Cell	155	3.7	84	0.1
Other	121	2.8	806	1
Total	**4.395**	**100**	**82,598**	**100**

Concerning the system management strategies, there are two possibilities: following the electric load or following the thermal load. In other words, the generator can be ”set” to track either the electric demand or the thermal one. In the first scenario, the prime mover is run so that the electric load is satisfied and the waste heat is recovered as extra product to cover part of the thermal demand. In the second case, the system is designed to satisfy the thermal load and the electricity produced becomes the extra product that will be used to partially cover the electricity load instead of purchasing it from the grid.

Nowadays, systems usually track the electric load because of the economic benefits that come from choosing this solution. Electricity, in fact, is more expansive than natural gas, thus making this approach more economic efficient. However, as it will explained later on, the solution that was adopted in this thesis will be to follow the thermal load, producing as first output useful thermal energy in the form of steam, installing on top of the steam system a steam turbine which produces electricty.

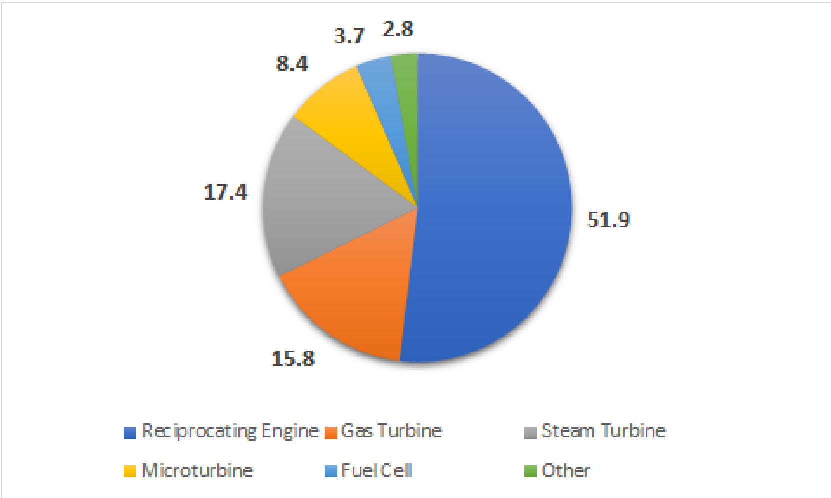


Figure 14: U.S. CHP Sites percentage by prime mover

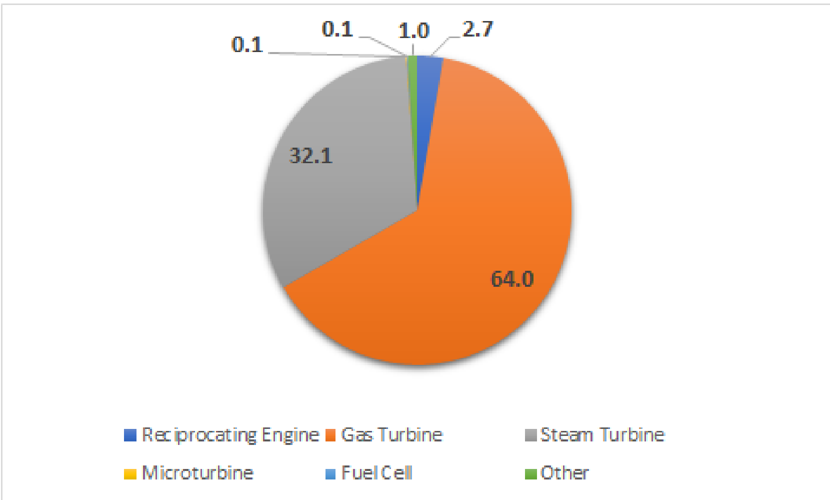


Figure 15: U.S. CHP Installed Capacity percentage by prime mover

2.5 CHP for Hospital Facilities

As already mentioned in the previous chapter, hospitals make ideal candidates for CHP applications and they can take advantage from the full suite of the CHP benefits. There are several reasons why hospitals are one of the best application of cogeneration systems and they can be summirized in: practical reasons, proven technology, economic benefits, reliability, clean energy [13].

- **Practical reasons**

  Hospitals represent on of the most energy intensive businesses of the commercial sector, they consume almost more than twice energy per square foot compared to the average commercial building. They operate continuosly, 24/7 and have large needs for electricity, heating and cooling, throughout the whole year.

- **Proven technology**

  According to [14] in the U.S. there are 219 CHP plants installated in hospital facilities with an installed capacity over 750 MW. This is a proven and well understood technology which can be maintained easily with a trained staff [15].

- **Economic benefits**

  By looking at the overall market of energy usage, it is possible to see that hospitals are accounted for around 10 % of the energy use in commercial buildings in the U.S. and the 8 % of all GHG emissions [15]. The fact that health care facilities have to run without interruption can lead to high energy costs. An efficient CHP plant can lead to attractive investments for these facilities to achieve interesting electric and thermal savings. This is even more true in regions where there are high electric rates and lower natural gas costs, because producing power on-site rather than purchasing it from the utility can be less expensive.

- **Reliability**

  As it was highlighted when describing CHP systems benefits, reliability is a key concept of cogeneration. CHP systems are able to maintain both power and heat output in case of extreme weather events. Considering the need for health care facilities to have a continuous electric and thermal production, this is one of the most important aspects when evaluating the possibility of implementing a CHP system in this type of buildings.

- **Clean energy**

  Cogeneration systems are considered much more "green" than traditional separated heat and power systems because the fuel required to satisfy the loads is lower. Being hospitals one of the most energy-consuming facilities, the impact of CHP concerning the emissions should not be underestimated. A hospital can reduce greenhouse gas emissions by almost 20 % by investing in a high efficiency CHP system [15].

2.6 CHP in eQUEST®

In Figure 16 the layout of the system in eQUEST®is represented. In particular, the *HVAC system - water side* as it is called in the software. The CHP system consists of the prime mover, a recovery heat exchanger that supplies the hot water demand of the hospital facility. In the layout of the system the prime mover is not represented.

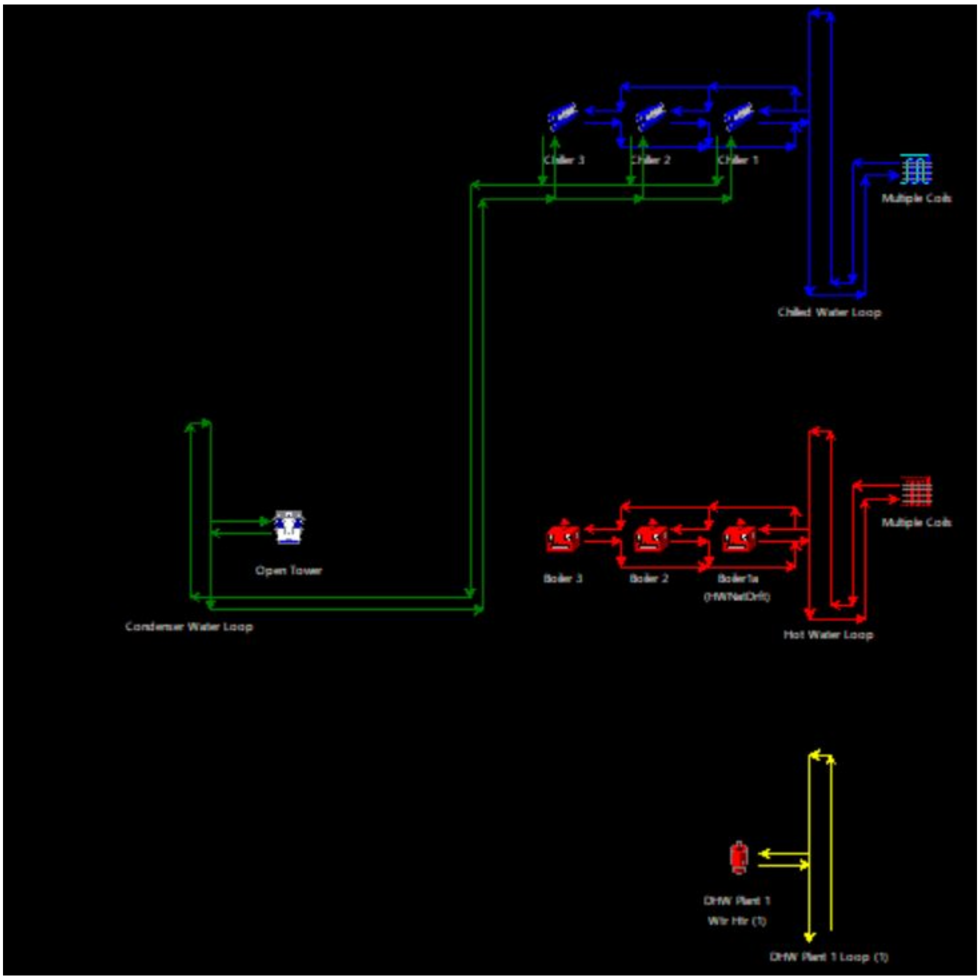


Figure 16: Layout of the CHP system in eQUEST®

Three main loops can be identified in the system: the *Hot Water Loop*, the *Domestic Hot Water Loop* and *the Chilled Water Loop*.

The *Hot Water Loop* is the one connected to the boilers that produces hot water for the space heating system. The heat recovered from the turbine will be connected to this loop in order to satisfy the thermal demand. When the heat demand is not covered by the waste heat from the turbine, the natural gas fueled boiler are operated.

The *Domestic Hot Water Loop* is the one that supplies the bulding with hot water, however, because of a limitation of the software its water heater capacity has been set to zero. In fact, eQUEST®is not able to attach multiple loops to the recovery heat exchanger, therefore, the domestic hot water load has been "assigned" to the hot water loop as a miscellaneous load. This approximation does not represent the reality but it is the only way the CHP system could work on the software. This is the reason why the capacity of the water heater of the DHW loop was set to zero. In addition, eQUEST®requires the presence of the domestic hot water loop, therefore it could not be deleted. By following this approximation, the total value of the thermal demand does not change. The following procedure was performed in order to incorporate the domestic hot water into the space heating load. Thanks to (Equation 2.3) the hot water process load was calculated

$$\mathrm{Q} = \mathrm{G} \cdot \rho \cdot \mathrm{c} \cdot \Delta T \tag{2.3}$$

Where:

- Q is the domestic hot water process load;
- G is the domestic hot water process flow;
- ρ is the water density;
- c is the specific heat;

- ΔT is the temperature difference between inlet and outlet temperature.

The process flow can be found in the software (the exact location can be found in Appendix B), while the temperature difference is 80 F, since the inlet temperature is $T_{\text{in}} = 45\,\text{F}$ and the outlet temperarure is $T_{\text{out}} = 125\,\text{F}$.

Finally, the *Chilled Water Loop* is the one responsible for the cooling demand of the hospital. Electric chillers are connected to this loop to satisfy the cooling load.

The detailed procedure for implementing and setting the CHP system in eQUEST®is explained in Appendix B.

CHAPTER 3

STEAM TURBINES

Chapter 3 describes the main component of the CHP system, the prime mover that has been selected to run the system. The objective is to analyze the performance of the prototype building. The decision of utilizing a steam turbine as prime mover for the system was derived from the fact that this type of generator has never been implemented in previous works. In fact, previous theses which analyzed the same prototype hospital, included the implementation as prime mover of: reciprocating engine, gas turbine, microturbines and fuel cells.

3.1 Technology Description

Steam turbines represent one of the oldest technologies of prime movers used to drive a generator. It has been descovered that the very first example of steam turbine was created by Hero of Alexandria in 150 BC. However, the first modern steam turbine appeared in 1884 and it was invented by Sir Charles Parsons. His model, in connection to a dynamo, could generate 7.5 kW (10 hp) of electricity [16]. After that, steam turbines rapidly started to replace reciprocating engine due to their lower costs and higher efficiencies [5]. Today, steam turbines are used in a variety of applications, including CHP systems, all around the world. Their size can range from 50 kW to around 250 MW.

The main difference between steam turbines and, for example, combustion turbines and reciprocating engines, is the fact that while for the latter heat is a byproduct of power generation, for steam turbines electricity becomes a byproduct of the production of steam (heat). In fact, if you feed the fuel into a reciprocating engine or a gas turbine, in the case of a steam turbine system, the fuel is fed to the boiler, where there is the production of high pressure steam that will be sent to the turbine for expansion. Having this specific configuration, with a physical separation from the production of steam and its expansion, allows steam turbine driven systems to use a variety of fuels, like natural gas, coal, wood or wood waste.

Given the fact that steam turbine size range is wide they are well suited for different type of applications:

- **Combined Heat and Power (CHP)** - Primarily used in industrial processes like paper mills, where there is solid or waste fuels available for being directed to and used in boilers.
- **Mechanical Drive** - The turbine can drive various equipments, from feedwater pumps, air compressors or chillers, instead of producing useful electric power.
- **District Heating and Cooling Systems** - The low pressure steam coming out of the turbine is used into the distribution system or to run absorption chillers or produce chilled water for air conditioning.
- **Combined Cycle Power Plants** - They are large power plants in which the waste gas coming from a combustion turbine is recovered and used to produce high pressure steam to run a steam turbine and produce additional electric power. These plants are not considered traditional CHP systems because they usually do not provide heat, just electricity.

Concerning CHP systems, the low pressure steam that is extracted from the turbine can be used directly in a process or for district heating, or it could be converted in other forms of thermal energy such as hot or chilled water [17].

3.1.1 Thermodynamics of Steam Turbines

The thermodynamic cycle that describes the behavior of steam turbines is the Rankine Cycle. In Figure 17 is shown an example of a Rankine Cycle with superheated steam.

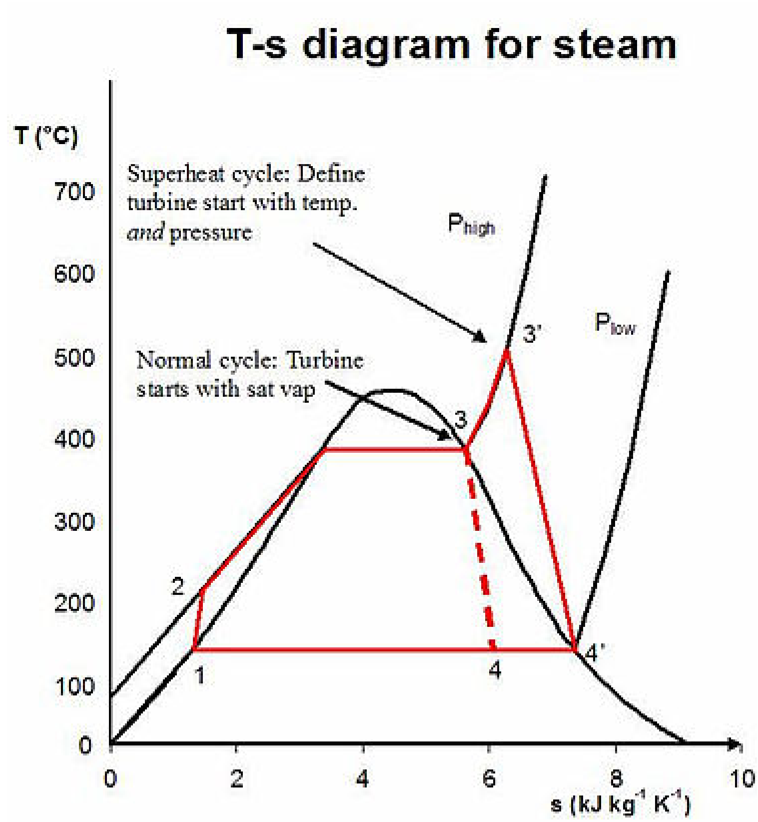


Figure 17: Example of Rankine Cycle with Superheated Steam

Looking at Figure 17, starting from point 1, a working fluid is pumped from a low pressure value to high pressure (point 2). The energy required by the pump is low due to the fact that the fluid is at liquid stage. The liquid, now at high pressure, enters the boiler and it is heated at constant pressure (Phigh), first it reaches its saturation point, then it reaches the vapor condition (saturated vapor at point 3) and it usually becomes superheated steam (point 3'). Then, in the turbine, the expansion process occurs and the steam pressure decreses to a lower value (Plow) generating mechanical power that will be converted into electricity inside the generator. Once the fluid has reached the condition of wet vapour, it is condensed at constant pressure and the condensate returns to the initial condition (point 1) in order to restart the cycle.

There are some variations from this basic cycle. For example it is possible to have a reheating cycle, where two turbines (the first one at high pressure and the second one at lower pressure) are connected in series and there are two expansion processes. The purpose is to eliminate and remove the possible moisture that could be carried by the stem at its final stages of the expansion process. In this case, after the first expansion, the steam at lower pressure goes back into the boiler to be reheated and then enters the second turbine at lower pressure. Another variation is the rigenerative rankine cycle. In this case, the condensate coming out of the condeser is heated up by the steam that comes out of the hot stages of the cycle. Rigenerative cycles can increase the efficiency of the system.

3.1.2 Components of Steam Turbine Systems

The schematic of a steam turbine system is represented in Figure 18.

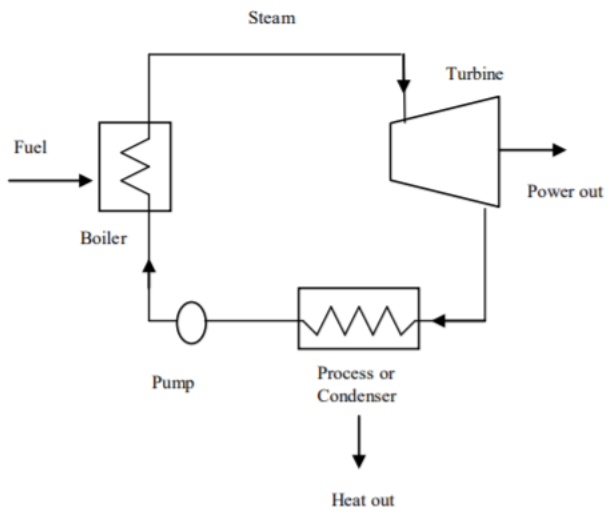


Figure 18: Schematic of a Steam Turbine Driven System [5]

The one shown in Figure 18 is a simple system in which there are four main components:

- The Feedwater Pump;
- The Condenser or a Heat Recovery System in a CHP system;
- The Steam Boiler;
- The Steam Turbine.

The feedwater pump is needed to increase the pressure of the working fluid to the desired value at which it will be sent to the boiler. Considering that the fluid is at liquid phase at this stage, the energy required by the pump is very low and ofen negligible.

The condensder is a simple heat exchanger (HEX) in which the low pressure steam coming out of the turbine is cooled to liquid phase thank to process water. The heat recovered in the condenser, in the case of a CHP plant, can become the useful thermal output of the system.

The boiler is a very important component because it differs these systems from other prime movers like reciprocating engine, internal combustion engines and gas turbines. In the case of the latter systems, the fuel is burned inside of the generator itself, while for steam turbine systems the fuel is fed and burned inside of the boiler. Therefore, the steam production and the power generation happen in two different components. The average efficiency of boilers is around 80 % [18].

The steam turbine is that component in which the high pressure steam is expanded to low pressure values with the aim of providing power to a shaft connected to a generator where mechanical power is converted into electricity. Steam turbines can be classified into different categories based on different parameters such as working pressures, size and their construction. However, in general, there are two main types of steam turbines depending on their blades: impulse turbines and reaction turbines. The main difference among them is the way steam passes through the turbine body and is expanded. In Figure 19 is represented the different blade configuration of these two types and the pressure drop from inlet to outlet.

As it is shown in Figure 19 impulse turbines are characterized by the alternation of moving blades and fixed nozzles. The real expansion happens in the fixed nozzles (stationary nozzles) while the pressure is constant when steam is passing over the blades. This means that the entire pressure drop takes place inside the stationary nozzles, which accelerates the steam to

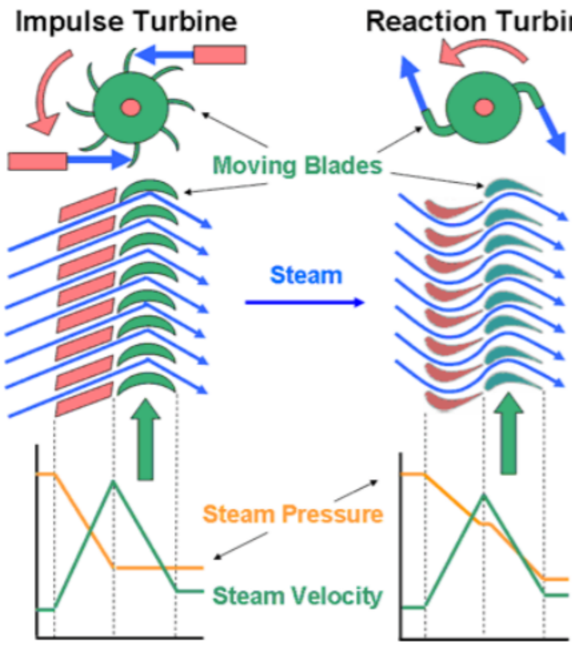


Figure 19: Diagram showing impulse and reaction turbine [5]

high velocity. In reality there is a little pressure drop across the moving blades too, but it considered negligible. The blades are responsible for the change in direction of the steam flow. A force is created on the blades after this direction change which manifests itself as torque applied to the shaft on which the moving blades are attached to. The most important feature of impulse turbines is the fact that the pressure drop in a single stage can be rather large, allowing bigger blades and a smaller number of stages. For bigger-size turbines, multiple stages in series are usually deployed, while for smaller turbines (like the ones used in the case study of this thesis) there is usually one single expansion stage. Examples of impulse steam turbine are the Curtis turbine, the Rateau turbine and the Brown-Curtis turbine.

Reaction turbines are composed of moving blades (nozzles) which are alternated with fixed nozzles. Unlike impulse turbine, in reaction turbines steam is continually expanding and the pressure drop happens in both the fixed and moving nozzles. In the moving blades there are both velocity and pressure loss. The steam is expanded through the fixed nozzles where potential energy is converted into kinetic energy and its velocity increases. Then, this high-velocity steam impacts the moving nozzles and is further expanded, while its velocity decreases after the impact with the moving nozzles. The impact generates a force on the moving blades, which manifests itself as torque on the shaft to which the blades are connected to. In this case, the pressure drop of a single stage is low, therefore the blades become smaller and the number of stages should be increased. As a consequence, usually reaction turbine are implemented with mutiple stages in series.

Reaction turbine are usually more efficient than impulse turbine and they would require almost twice as much blade rows in order to obtain the same output electricity with the same steam flow [19].

In eQUEST®, it is not possible to declare or decide if the implemented turbine is a reaction or impulse type. However,the model can be considered to be an impulse turbine, considering the fact that the size of the turbine is relatively small and works with only one expansion stage.

Another way of classifying steam turbines is based on the working pressure. We can identify: condensing turbines, non-condensing turbines or back-pressure turbines and extraction turbines. The main difference is related with the phase at which the working fluid exits the turbine, which is strictly correlated with its pressure. A schematic of these three technologies is represented in Figure 20 (from left to right: Non-condensing Turbine, Back-pressure Turbine, Extraction Turbine).

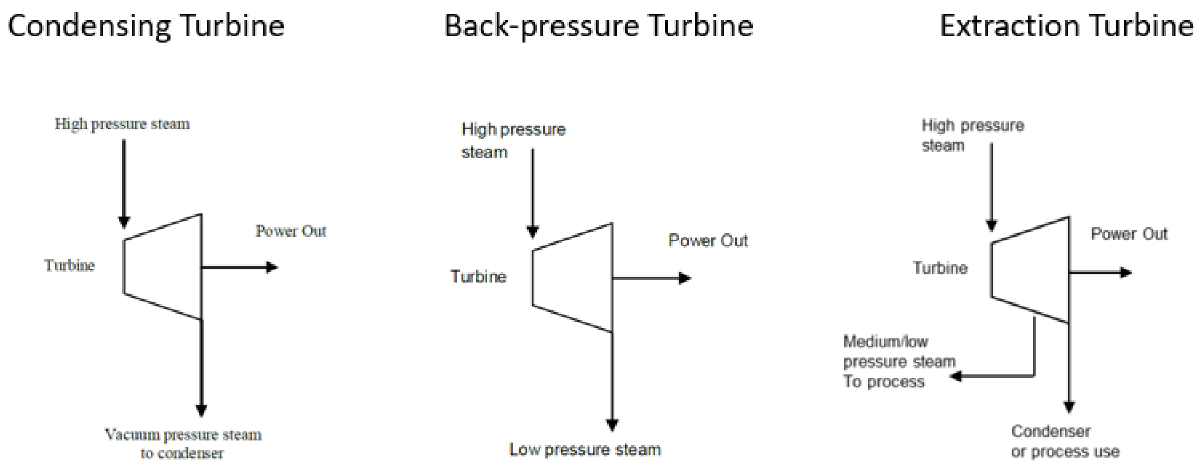


Figure 20: Schematics of three steam turbine types based on the working pressure [5]

Condensing turbines are typically used in larger power generation plants. These systems require only the prodution of electricty, therefore they wish to extract the maximum amount of energy from the steam. This is achieved by ”sending” the exhaust steam at vacuum conditions to a condenser which condenses the steam at constant low pressure values. Typical values of exhaust pressure of the steam are well below atmospheric pressure (14.7 psi). In CHP applications these turbines are not used because the goal of cogeneration is being able to recover useful thermal energy for a process, which means having steam that comes out of the turbine at higher pressure.

Non-condensing or back-pressure turbines differ from condensing turbines because they do not exploit the whole energy that is "carried" by the steam. The pressure of the exhaust steam is well above atmospheric pressure and it can therefore be used for different type of processes, like district heating or to produce hot or chilled water. The most typical pressure levels for steam distribution systems are 50, 150 and 250 psi. The lower value is usually used in district heating systems. The turbine that is used for the simulation scope of this thesis is a back-pressure turbine because the main goal of the CHP system is to recover useful thermal energy for space heating and domestic hot water.

Extraction turbines usually have one or more openings in their casing, used for extracting a portion of the steam that is expanding inside the turbine at intermediate conditions. The steam that is extracted may be used for for different processes' purposes in CHP plants or for feedwater heating. The remaining steam continues the expansion process and can be exahust into a condenser at vacuum conditions or be used in another low pressure appliction.

3.2 Performance characteristic

To mesaure the perfomance of steam turbines, it is possible to consider the isentropic efficiency (η_{is}), which can be defined as the ratio between the actual performance of the turbine to the ideal performance that would be achieve by an isentropic turbine with no internal and frictional losses. In Figure 21 there is a graphical representation of the isentropic efficiency on the Mollier diagram.

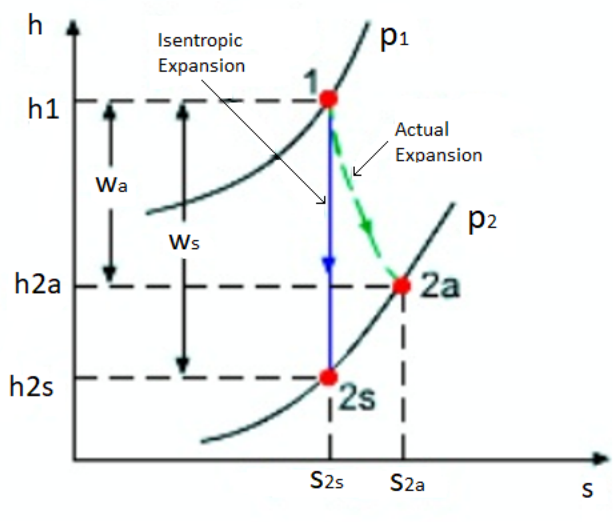


Figure 21: Graphical representation of the isentropic efficiency (η_{is}) on the Mollier Diagram [6]

The steam initial pressure is the same one in both ideal and actual case, the difference is the type of expansion that is performed and the outlet condition that is reached. The blue line in Figure 21 represents the isentropic expansion, from high pressure (P1) to low pressure (P2), at constant entropy (s), which means with no losses. The actual expansion is represented by the green dot line from point 1 to point 2a, taking into consideration the internal losses. In (Equation 3.1) there is the formula that can be used to calculate the isentropic efficiency of a steam turbine. It is possible to see that the efficiency can be calculated as ratio of the enthalpy difference between inlet condition and outlet conditions, for the actual case and the isentropic expansion. When calculating the efficiency, the heat that is lost to the environment is assumed to be negligible.

$$\eta_{\mathrm{is}} = \frac{\text{Actual turbine work}}{\text{Isentropic turbine work}} = \frac{\mathrm{W_a}}{\mathrm{W_s}} = \frac{h_1 - h_{2\mathrm{a}}}{h_1 - h_{2\mathrm{s}}} \tag{3.1}$$

It is important not to confuse the isentropic efficiency with the electrical efficiency of a turbine. The former, as shown above, is a parameter that allows to understand how a turbine is performing compared to the ideal and isentropic case, without internal losses. The electrical efficiency can be defined as the ratio of the net power generated by the turbine to the total fuel introduced into the system (fed to the boiler for producing the steam). In other words, the electrical efficiency measures how efficiently the turbine is able to extract power from the steam supplied to the turbine itself. The electrical efficiency can be calculated as shown in (Equation 3.2) and usually does not reach values higher than 10 %.

$$\eta_{\mathrm{electrical}} = \frac{\text{Net useful electricity output}}{\text{Total fuel input into the system}} \tag{3.2}$$

The main reason why the electrical efficiency of steam turbines is low is because in these systems, electricity is a byproduct of heat generation and their primary objective is the production of large amounts of steam. In fact, when considering the power to heat ratio of steam turbines, defined as the ratio of the net useful electricicty output to the useful heat output, the resulting values are very low, and the reason is the same, the production of heat is much larger than the net electricity produced.

If the electrical efficiency of steam turbines is pretty low, the effective electrical efficiency is not. The latter is calculated as it is shown in (Equation 3.3).

$$\eta_{\text{effective electrical}} = \frac{\text{Net useful electricity output}}{\text{Total fuel into boiler} + \frac{\text{Steam to process}}{\eta_{\text{boiler}}}} \tag{3.3}$$

The values reached by the effective electrical efficiency are very high, around 70 %, because almost all the difference in energy between the high pressure steam that exits the boiler and the lower presure steam that leaves the turbine is converted into electricty.

As a consequence, the overall CHP efficiency of the system when the prime mover is a steam turbine are high. Typal values range from 70 % to 85 % depending on the boiler type and fuel. In fact, the CHP efficiency usually reaches the boiler efficiency level, 80 %. The formula used to calculate the CHP efficiency is represented in (Equation 3.4).

$$\text{CHP}_{\text{Eff}} = \frac{\text{Net useful electricity output} + \text{Useful heat output}}{\text{Total fuel input into boiler}} \tag{3.4}$$

Table VII reports some typical performance values and working conditions for three different sizes of steam turbines that are present on the market. The table was taken from [5] and the turbine parameters are average values of turbines from Elliot Group [20] until 2014.

TABLE VII: **BACK-PRESSURE STEAM TURBINE PERFORMANCE CHARACTERISTICS**

Parameter	System 1	System 2	System 3
Turbine size (kW)	500	3,000	15,000
Isentropic Efficiency (%)	52.5	61.2	78.0
Generator Efficiency (%)	94	94	94
Steam flow (lb/h)	20,050	152,600	494,464
Inlet pressure (psig)	500	600	700
Inlet temperature (F)	550	575	650
Outlet pressure (psig)	50	150	150
Outlet temperature (F)	298	373	379.7
Boiler Efficiency (%)	80	80	80
Electric Efficiency (%)	6.27	4.92	7.31
Fuel Input (MMBtu/h)	27.2	208.3	700.1
Steam to Process (MMBtu/h)	19.9	155.7	148.484
Total CHP Efficiency (%)	79.60	79.68	79.70
Power/Heat Ratio	0.086	0.066	0.101
Effective Electrical Efficiency (%)	75.15	75.18	76.84

3.3 Steam Turbine Model in eQUEST®

In order to proceed with the simulations on the software (chapter 4) it is important to understand how to create a CHP system and the desired electric generator in eQUEST®. The detailed procedure to implement the CHP system and set the generator can be found in Appendix B. In addition, the model of the stem turbine should be evluated, understood and explained.

When creating an electric generator the first thing that the softwares requires is the type and after that, if the steam turbine is selected, the capacity of the generator in kW and the loop that will provide steam to the turbine, called the *Steam Loop*. Once the electric generator is created, it is possible to change the different parameters that characterize this specific component. In Figure 22 and Figure 23 the interface of the steam turbine in eQUEST®, where the user can modify the parameters is shown.

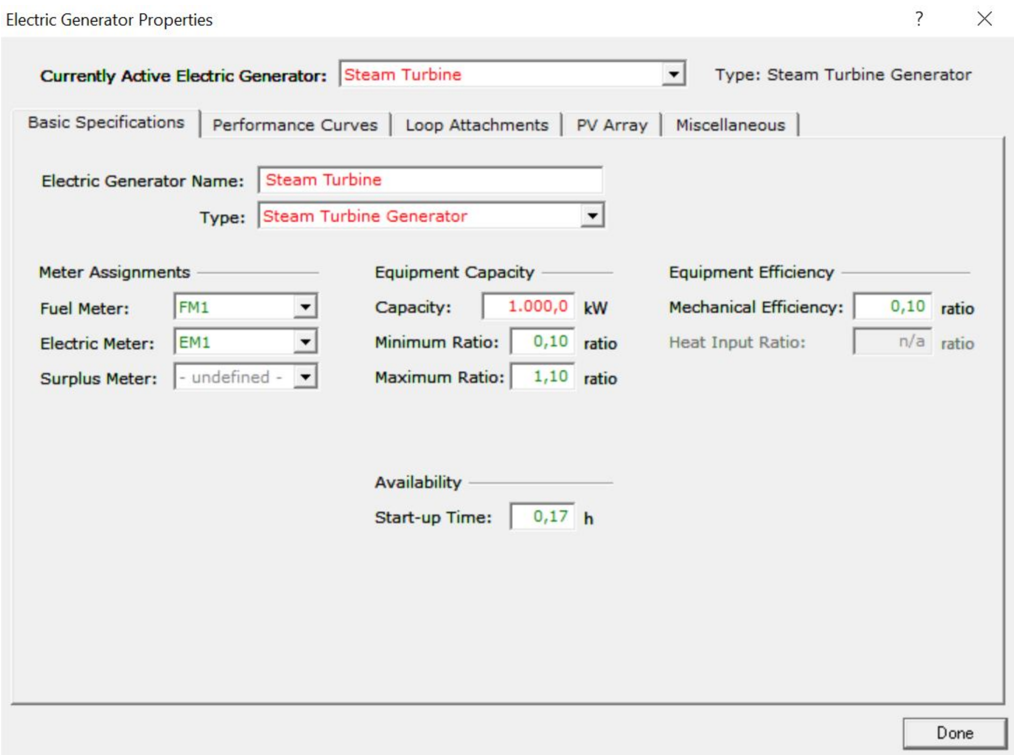


Figure 22: Steam turbine section in eQUEST®, tab *Basic Specifications*

As Figure 22 shows, most of the general parameters can be changed in the first tab called *Basic Specifications*, but not all of them. Starting from the left side of the figure, we can see the meters that are assigned to the turbine, the electric and the fuel meter. In the building model

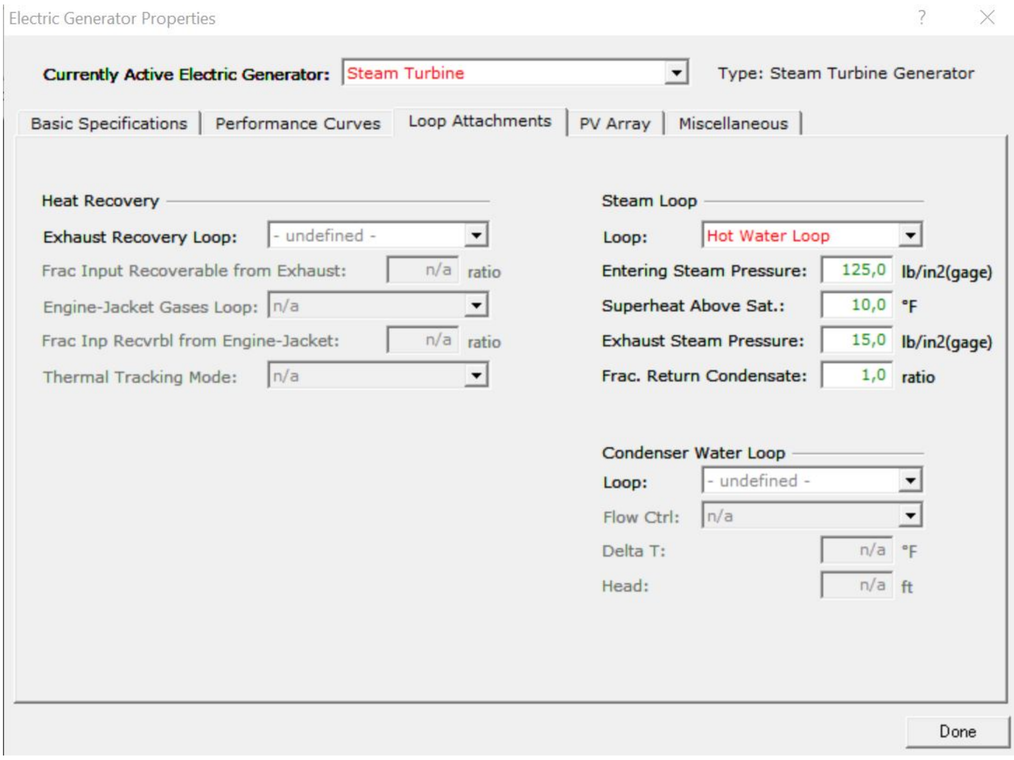


Figure 23: Steam turbine section in eQUEST®, tab *Loop Attachments*

there are only two meters, *EM1* - primary electric meter and *FM1* - primary fuel meter. The steam turbine is connected to both of them. Concerning the *Equipment Capacity*, the size of the turbine can be changed at any time and it accepts values in kW. In addition, it is possible to change the *Minumum Ratio* and *Maximum Ratio*, which represent the lower and upper limits at which the turbine can run expressed as unit values of the part load ratio of the turbine. The default values are 0.10 and 1.10. On the right-hand side of the screen there is the *Equipment Efficiency* part, in which the only parameter that can be changed is the *Mechanical Efficiency* whome default value is 0.10. The *Heat Input Ratio* is not applicable for steam turbines.

The tab *Loop Attachments* represented in Figure 23 allows the user to modify the loops that are connected to the turbine and the conditions of the steam. It is possible to select, on the left side, the *Exhaust Recovery Loop* from the existing loops, or by creating a new loop. This loop is the one to which heat will be supplied in order to recover useful thermal energy for space heating and domestic hot water. By default the turbine does not recover heat, therefore it is necessary, in order to have a CHP system, to select the *Hot Water Loop* as the one that will be supplied by the recovered heat. On the right side of the screen the steam conditions can be changed according to the need of the building. In particular the parameters that can be modified are:

- *Entering Steam Pressure* - The pressure value in (psig) of the steam entering the turbine. The default value is 125 psi.
- *Superheat Above Saturation* - The temperature difference above saturation condition of the steam entering the turbine. The default value is 0 psi.
- *Exhaust Steam Pressure* - The pressure value in (psig) of the steam leaving the turbine. The default value is 15 psi.
- *Fraction Return Condensate* - The fraction, from 0 to 1, of condensate that after having been directed to the process returns to the turbine.

It is possible to select if the condensation of the steam leaving the turbine is performed by a *Condenser Water Loop* too. By default the software does not do that.

The model of the steam turbine in eQUEST®is not very recent and has its limitations. However, it was not changed for this work, because of the complexity of the procedure and the fact that no performance and data sheets for turbines present on the market, similar to the one needed in this simulation, have been found. Most of the data availabe online refer to larger size steam turbines for industrial and power generation plant applications. The steam turbine implemented in this thesis is a small turbine with limited size, reaching in some simulations the maximum value of 2000 kW. The main limitation regarding the model is the fact that the software cannot simulate steam loops. The steam loop feeding the turbine is approximated by using a hot water loop having the right characteristics in terms of pressure and temperature and neglecting thermal losses typical of steam loops and auxuliary equipments that could be used in steam systems like de-aerators. In addition, the default parameters, as shown in previous figures, do not allow the turbine to reach very high values of total efficiency, therefore they should be changed accordingly to what are the needs of the user and the examined building.

The software uses the steam conditions that are inserted by the user in the *Loop Attachments* tab and consequently calculates the steam enthalpy for the inlet and outlet condition. For the inlet condition it is trivial having both pressure and temperature of the steam, for the outlet condition the program assumes saturated steam. With the enthalpy values it is possible to evaluate the maximum isentropic efficiency of the turbine η_{is} just like it has been described in section 3.2. The overall efficiency of the turbine is calculated as product of the mechanical efficiency inserted by the user and the maximum isentropic efficiency calculated by eQUEST®as shown in (Equation 3.5).

$$\eta_{\text{overall}} = \eta_{is} * \eta_{\text{mechanical}} \tag{3.5}$$

The model is further represented by two performance curves that can be found in the tab called *Performance Curves* of the steam turbine window. The two curves are called: *Steam Consumption Rate* and *Heat Input Ratio*.

- *Steam Consumption Rate* (*flow-fPSIG*).

  This curve calculates the theoretical steam consumption rate in (lbs/kWh) of the turbine as a function of both the *Entering Steam Pressure* and the *Exhaust Steam Pressure*. This theoretical rate is calculated assuming an isentropic expansion corresponding to the maximum possible efficiency that has been calculated with those specific steam conditions expressed by the user. The actual steam rate is adjusted by the *Mechanical Efficiency* as reported in (Equation 3.6). For part load performance the steam rate is further adjusted by the *HIR-fPLR* curve.

$$\text{Actual Steam Rate} = \frac{\text{Theoretical Steam Rate}}{\eta_{is}} \tag{3.6}$$

  In (Equation 3.7) is reported the equation of this curve which is a bi-quadratic curve, where Z represents the Steam Consumption Rate, X represents the steam pressure entering the turbine and Y represents the steam pressure leaving the turbine.

$$Z = \text{a} + \text{b}X^2 + \text{c}X^2 + \text{d}Y + \text{e}Y^2 + \text{f}XY \tag{3.7}$$

With:

- $a = 38.79236221$;
- $b = -0.21138562$;
- $c = 0.00052878$;
- $d = 1.02008748$;
- $e = 0.00091660$;
- $f = -0.00349944$.

Since we are dealing with a bi-quadratic curve the best way of representing it is by fixing the value of one of the variables and plot the remaining two in a two-axis plot. In Figure 24 there are several curves that show (Equation 3.7) in different pressure conditions. On the x-axis there is the inlet steam pressure and each curve represents a value of outlet steam pressure. The limitation concerning this curve is the fact that eQUEST®sets a minimum and maximum limit for the steam consumption rate: 15 lbs/kWh - 100 lbs/kWh. This is the reason why the examples shown in Figure 24 do not increase the inlet pressure over 350 psi, otherwise (Equation 3.7) would not be reliable. Concerning the outlet pressure, the values that have been considered go from 50 psi to 150 psi because the steam turbine for this system is a back-pressure turbine, thus having an exhaust pressure above the atmospheric one. In addition, the steam needed for supplying a hospital facility in terms of thermal demand should be around an average value of 125 psi.

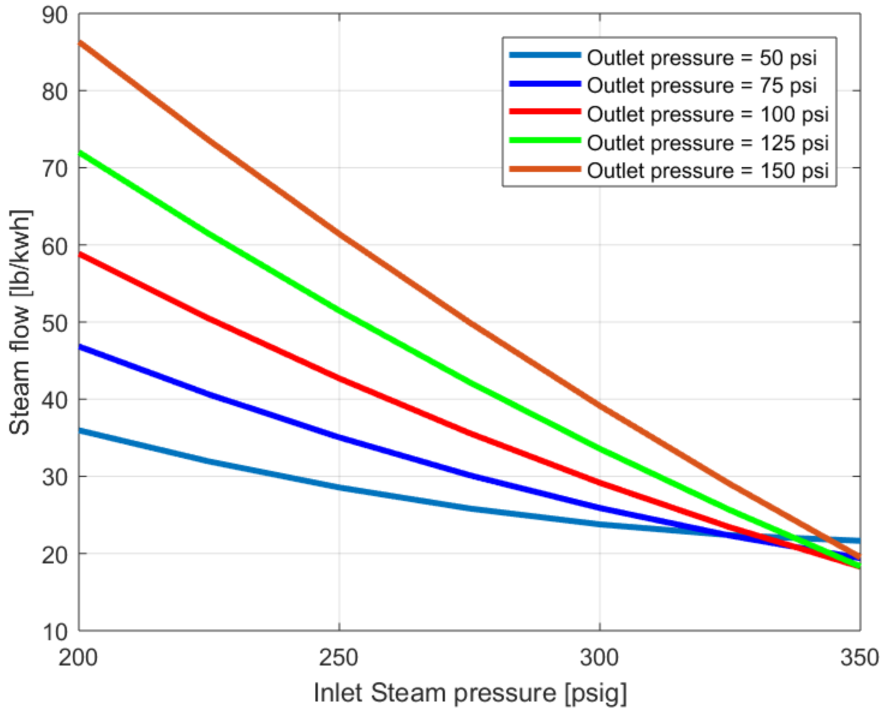


Figure 24: Graphical representation of the *flow-fPSIG* peformance curve

- *Heat Input Ratio* (*HIR-fPLR*).

  This curve modifes the steam consumption as a function of the part load ratio of the turbine. In other words, this curve allows the turbine to modulate its otuput power by calcualting what is the needed steam for running the turbine at a specific electricity production condition. The part load ratio can be defined as the ratio between the power produced at a specific condition and the power produced at full load, as show in (Equation 3.8).

$$\text{PLR} = \frac{\text{Power Produced at Specific Condition}}{\text{Power Produced at Full Load Condition}} \tag{3.8}$$

This curve comes into play when the turbine is not running at full load conditions. The operating range of the turbine is set like previously shown as unit values for the lower and the upper limit.

Thanks to this curve the system is able to provide an acceptable steam rate to the turbine in order to run at different power conditions and it is the reason why it is not needed to use a specific value of *Heat Input Ratio* in the *Basic Spefications* tab.

It is described by a quadratic curve, shown in (Equation 3.9), where Z is the *Heat Input Ratio* and X is the *Part Load Ratio.*

$$Z = \text{a} + \text{b}X^2 + \text{c}X^2 \tag{3.9}$$

With:

- a = 0.12838547;
- b = 0.90049529;
- c = -0.02888074;

A graphical representation of this curve is shown in Figure 25.

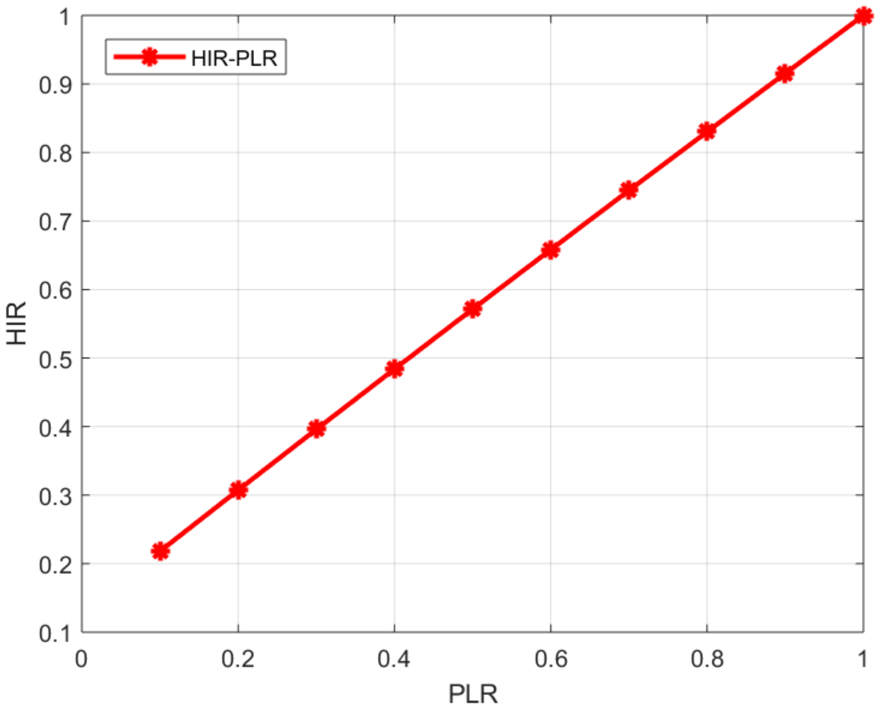


Figure 25: Graphical representation of the *HIR-PLR* peformance curve

As it has been said, the model is not perfect and has its limitations, especially for conditions that are too extreme in terms of pressure (too high and too low). Nonethless, all the parameters that have been used for the simulatins have been an attempt to made the model as reliable as possible in order to have accurate results that approximate the reality in the best possible way. All the parameters that have been chosen and then modified in order to run the model and perform all the needed simulations are described and listed in the next chapter.

3.3.1 Case Study Turbine

It can be useful to summarize the information given in this chapter, to see what are the main characteristics of the turbine that has been used for this work and the simulations. The exact parameters values used to run the model will be detailed in the next chapter. Even though the software is not able to handle steam loops, the building has been considered as a steam system with a specific delivering pressure for the heating system and steam boilers, upon which a steam turbine was installed, in order to produce also electricity. These type of turbines are called topping steam turbines, because they exploit the already existing steam production to extract useful power and produce electricity, realizing a CHP system. The prime mover that has been used for the simulation is a:

- Topping Steam Turbine;
- Impulse Steam Turbine, with a single stage expansion;
- Back-pressure Steam Turbine with an exhaust pressure higher than atmospheric pressure.
- Small size Steam Turbine, not exceeding 2000 kW in capacity.

Examples of manufacturers that sell this type of turbines are: Elliott Group [20] or Dresser-Rand by Siemens [21]. The configuration showed in Figure 13 can be considered a good approximation of the system that will be used for the simulations, where the "building or facility" is represented by the prototype hospital.

CHAPTER 4

SIMULATIONS IN eQUEST®AND RESULTS

In the previous chapter the steam turbine technology has been described and the model of the turbine inside eQUEST®has been explained. This chapter firstly shows how the system and the turbine have been set for the initial simulation, then reports and compares the results obtained for different system's conditions. Many simulations have been done with the aim of optimizing the configuration of the system and its main parameters, in terms for example of efficiency or reduction in source energy consumption, to set a starting point for future works.

4.1 System Configuration

In chapter 2 an overview and a description of the CHP system in eQUEST®was given, explaining and describing the different loops and their objective. In chapter 3 the steam turbine model was evaluated and the main parameters that describe it were assessed. Now the parameters that were used in order to run the model will be shown.

4.1.1 Turbine and Equipment Parameters

Considering that the electricity peak demand of the building, during the summer period, reaches 2000 kW, the steam turbine in the system will not exceed that value. As a first approximation the turbine size selected is 1000 kW, even though the capacity will be changed in order to better optimize the system. Another parameter that needs to be changed is the mechanical efficiency of the turbine. The default value is 10 %, which is too low. A more realistic value of

90 % was used. Finally, the *Maximum Ratio* of the turbine was changed, from 1.1 to 1.0, which means the the maximum power at which the steam turbine can run is the full laod condition (in this case, 1000 kW).

4.1.2 Steam Conditions

Changes were made to the conditions of the steam as well, in terms of pressure and temperature. The outlet pressure has been set to 125 psi. This choice was led by different reasons. First, the turbine must work as a back-pressure turbine, having an outlet pressure above atmospheric conditions. In additon, hospital facilities that use process steam usually work at similar pressures. Finally, in the case of a CCHP (Combined Cooling, Heat and Power), steam driven double absorption chillers should be installed, and they mostly work with a pressure of 125 psi [22]. The outlet pressure will never be changed from now on in every simulation that will be presented, the value of 125 psi will remain constant. The inlet pressure, on the other hand, is a parameter that can be changed in order to better understand what will be the best situation in terms of savings, costs and efficiency. For the first simulation, the inlet pressure was set to 350 psi and the temperature above saturation to 50 F, which corresponds to a temperature of 485 F (being the saturation temperature at 350 psi, 435 F). The goal of having saturated steam is to exploit a larger enthalpy difference with the aim of producing more electricity, trying not to consume too much fuel for icreasing the steam pressure. The inlet pressure is a parameter that will be changed multiple times, 350 psi is just the first trial, even though it was chosen because it makes sense in a realistic application to have such a pressure difference considering the other parameters of the turbine. In fact, given these steam conditions and an output power of 1000 kW, the corresponding isentropic efficiency will be 56.6 % (see section 4.2), which is a realistic value.

In Table VIII there is a summary of the parameters that were changed for the first simulation.

TABLE VIII: **TURBINE PARAMETERS AND STEAM CODITIONS IN eQUEST®FOR FIRST SIMULATIONS**

Parameter	**Value**
Mechanical Efficiency (%)	90
Minimum Ratio	0.0
Maximum Ratio	1.0
Inlet Pressure (psig)	350
Inlet Temperature above Saturation (F)	50
Outlet pressure (psig)	125
Boiler Efficiency (%)	80

In Table VIII is also reported the efficiency of the steam boiler which was left as default value at 80 %.

4.2 Control Management of the System

As already mentioned, with these specific conditions, the isentropic efficiency of the turbine is 56.6 %, which corresponds to an overall efficiency calculated by eQUEST®with (Equation 3.5), of 51 %. However, the electrical efficiency of the steam turbine as it was explained in chapter 2, is not very high, because electricity is a byproduct of heat generation. In particular, given this conditions, we can calculate a fuel consumption of 35.5 MMBtu/hr. Therefore, through a series of steps it is possible to calculate an electrical efficiency of 10.1 % which is in line with the average values of existing steam turbines.

Considering as steam conditions entering the turbine a pressure of 350 psi and a temperature of 485 F, the enthalpy of the high pressure steam from saturation tables is 1240 Btu/lb. While, for the outlet condition, having a pressure of 125 psi and saturated steam, the enthalpy is 1193 psi. Finally, the enthalphy of the exiting steam in isentropic conditions can be found on the Mollier Table following an isentropic expansion (vertical line) until the desired pressure, and it results in 1157 Btu/lb. As a consequence, the isentropic efficiency can be calculated with (Equation 3.1) as shown in (Equation 4.1).

$$\eta_{\mathrm{is}} = \frac{\mathrm{h_{in}} - \mathrm{h_{out}}}{\mathrm{h_{in}} - \mathrm{h_{out,is}}} = \frac{1240 - 1193}{1240 - 1157} \cdot 100 = 56.6\,\% \tag{4.1}$$

In the following set of equations the calculation of the electrical efficiency is shown. However, another enthalpy value must be extrapolated from saturation tables in order to perform this calculation, the enthalpy of the saturated liquid at 125 psi, which is 868 Btu/lb.

$$G = \frac{\mathrm{P_{out}}}{\mathrm{h_{in}} - \mathrm{h_{out}}} = \frac{1000 \cdot 3412.142 \cdot \frac{1}{0.95}}{1240 - 1193} = 76.4 klb/hr \tag{4.2}$$

$$\mathrm{Fuel_{input}} = \frac{(\mathrm{h_{in}} - \mathrm{h_{out}}) \cdot \mathrm{G}}{\eta_{\mathrm{boiler}}} = \frac{(1240 - 868) \cdot 76.4 \cdot 10^3}{0.8} = 35.5 MMBtu/hr \tag{4.3}$$

$$\text{Useful Power} = \frac{\mathrm{P_{out}}}{\eta_{\mathrm{gen}}} \cdot (3412.142 \frac{Btu}{kWh}) = \frac{1000}{0.95} \cdot 3412.142 = 3.60 MMBtu/hr \tag{4.4}$$

$$\eta_{\mathrm{electrical}} = \frac{\text{Useful Power}}{\mathrm{Fuel_{input}}} = \frac{3.60}{35.5} \cdot 100 = 10.1\,\% \tag{4.5}$$

Where:

- G is the steam flow expressed in (klb/hr);
- P_{out} is the net output power of the steam turbine in (Btu/hr);
- η_{boiler} is the efficiency of the boiler, equal to 80 %;
- η_{gen} is the efficiency of the generator connected to the turbine, which was assumed to be 95 %.

Considering the fact that the electric efficiency is not very high for steam turbines in general and in this specifc case as well, an important decision was made concerning the management of the system. Instead of tracking the electric load, like most CHP systems, it was decided to follow the thermal load. This means that the turbine will be driven by the thermal load and it will have the objective of covering the thermal demand of the hospital. As a consequence, electricty becomes a secondary product and the electric production, will be used to partly satisfy the electric load of the building. From an economic standpoint, it is probably not the most efficient solution, due to the higher cost of electricity compared to natual gas, however the economic analysis of the system will not be performed in this thesis, focusing more on the energy analysis.

4.3 First Simulations Result

With all the parameters described in previous chapters and subsections the first simulation on the sofwtare was performed. In Figure 26 and Figure 27 the thermal load with the recoverable and recovered heat and the electric load with the electric production of the turbine are shown respectively.

From Figure 26 is clear that the turbine is following the thermal load, as the heat production higher during the winter season, like the load, and lower during the summer one. The red line represents the thermal load of the building that the turbine is trying to satisfy, while the green line represents the heat that was recovered by the recovery loop and supplied to the building. The blue line, instead, is the recoverable heat which allows us to understand that some heat is actually wasted. Even though the steam turbine was set to follow the thermal load, it is not able to modulate properly during summer when the output power is around 20 % of the capacity and it produces more heat than it is actually needed by the building for the hot water demand. As it was already mentioned, the model is probably not working in the most efficienct way and the turbine is not able to run at low part load conditions. The satisfaction of the electric load, which in our case is a secondary goal of the system, is clearly not excellent. In

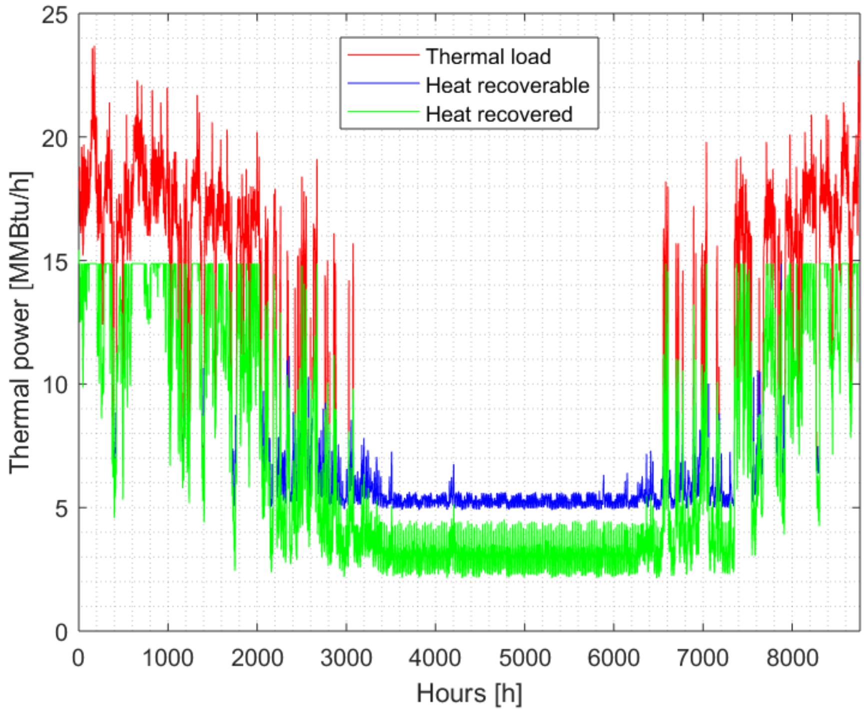


Figure 26: Thermal load of the building with the recoverable and the actual recovered heat by the 1000 kW turbine

particular because the electric peak demand occurs during summer, that corresponds to the lowest electric production of the steam turbine.

To better understand how much the overall system was improved, the duration curves of the thermal load and electric load were constructed, comparing the baseline case and the steam turbine system. The curves are represented in Figure 28 and Figure 29.

Both loads have decreased, in particular the thermal load which was almost totally covered.

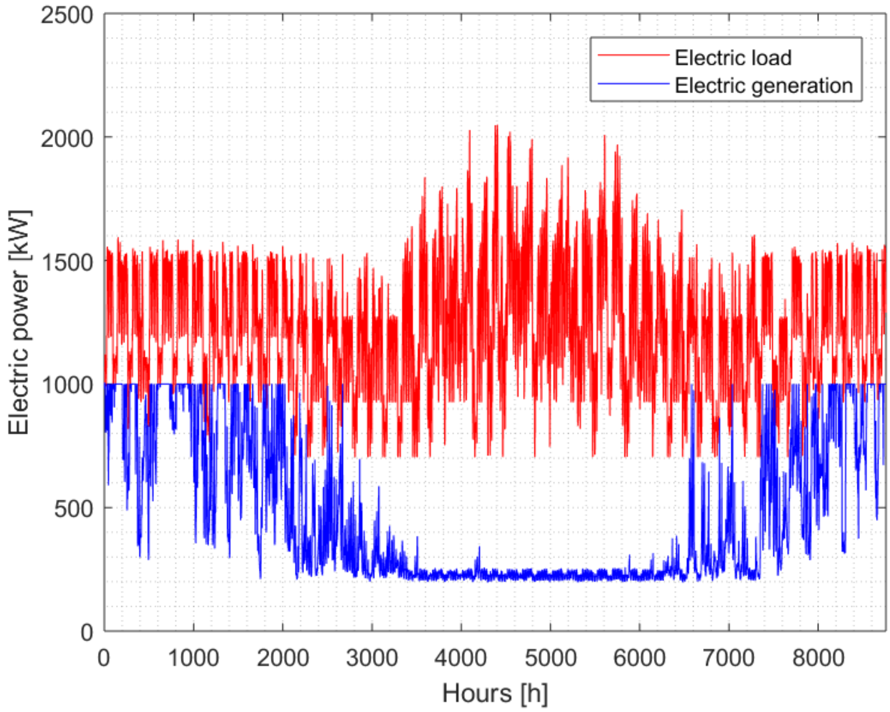


Figure 27: Electric load of the building with the electricity produced by the 1000 kW Turbine

Another useful curve that can be shown is the duration curve for the fuel consumption in both the baseline case and the new condition. If adding the steam turbine allows to reduce the thermal and electric load by the recover heat and production of electricity respectively, the fuel consumption clearly will be higher in order to run the steam boiler. The duration curve for the fuel consumption is shown in Figure 30. As expected, the fuel consumption is increased after the installation of the turbine. The new condition is represented by the blue curve.

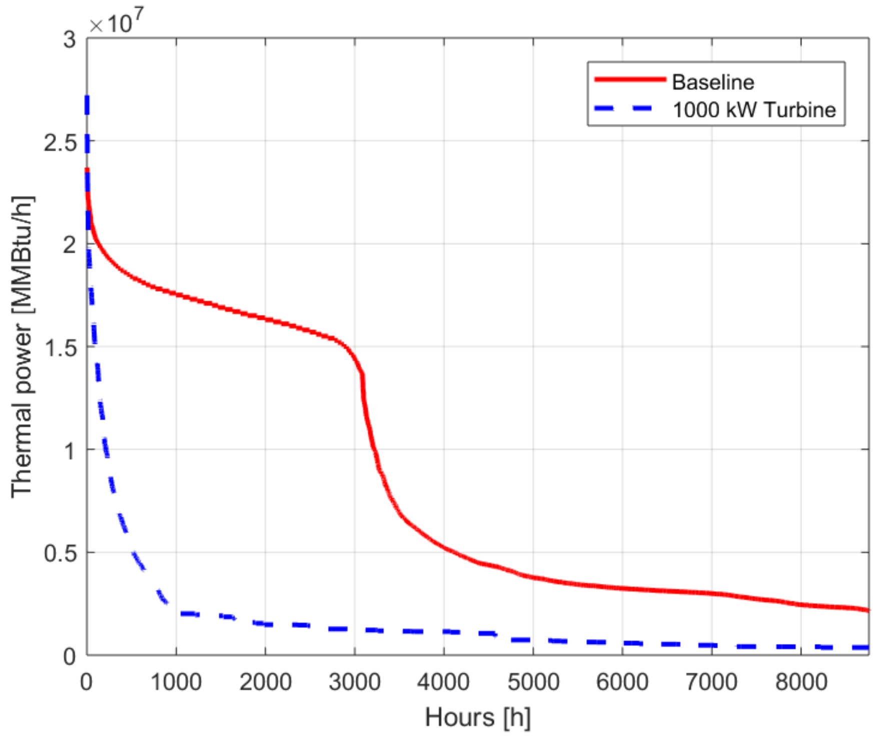


Figure 28: Duration curves for the thermal laod in baseline and 1000 kW turbine cases

It is important to add that even though the previous figures show some improvements to the system, in order to evaluate the real benefits of the CHP application other analysis and calculations should be done. Starting from calculating the efficiency of the system, which will be done in the next subsection or by performing an economic analsis, based on the investement costs and energy savings.

As it was mentioned before, the steam turbine is not able to modulate its power below certain values during the summer period, therefore, it produces more thermal energy than

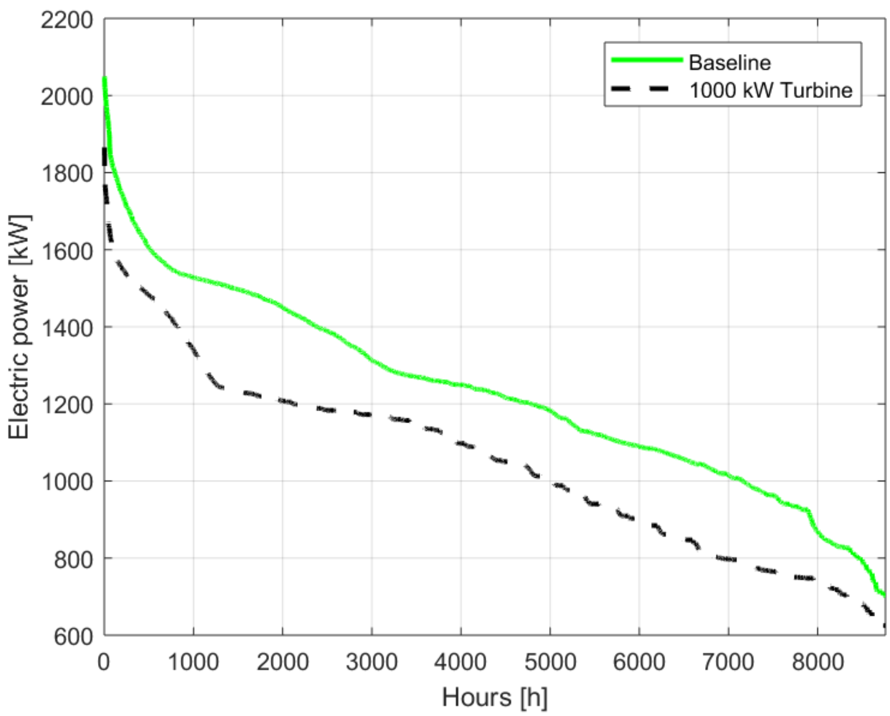


Figure 29: Duration curves for the elctric load in baseline and 1000 kW turbine cases

needed in the form of waste heat. In addition, it was seen that during the winter season, when the turbine is running at full laod conditions, the electricity production exceeds the electric load. This means that there is a surplus in electricity which will be sold to the grid. Therefore, both heat and electricity production over the entire year exceed their rispective loads in some periods. It would make sense, to install storages for helping both situations for load shaping. A hot thermal storage could be used in order to store the waste heat during summer and re-use it later. An electric storage, in the form for example of a battery pack, could be used to store the

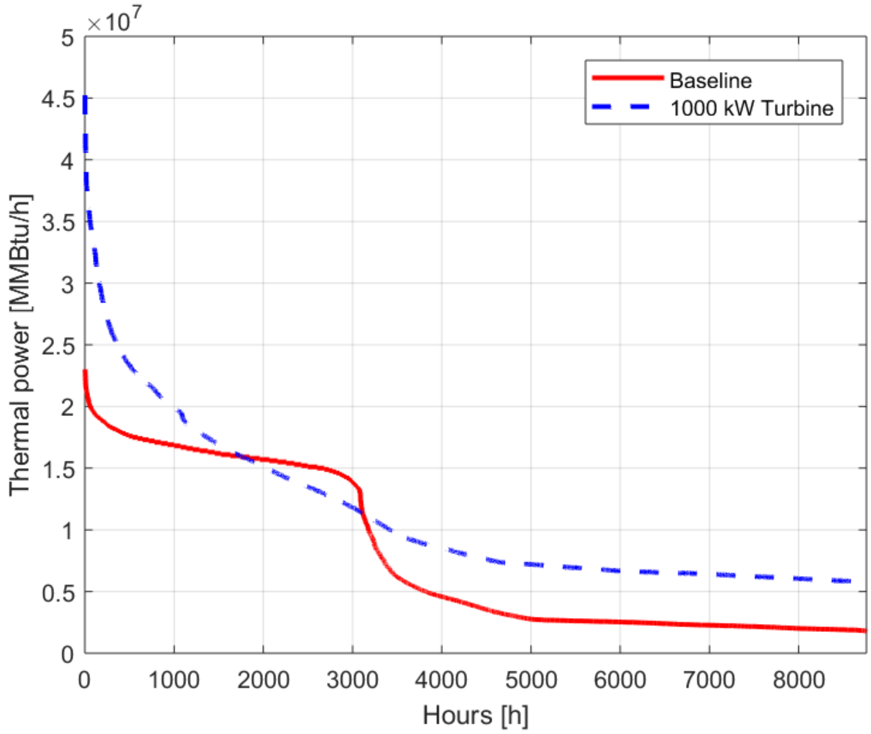


Figure 30: Duration curves for the fuel consumption in baseline and 1000 kW turbine cases

surplus electricity during the peak thermal laod hours of the winter and re-use that electricity at a later stage instead of selling the surplus back to the grid.

In Table IX the results of this first simulations are summarized, highlighting the full load hours during the entire year and how much heat and electricity are "wasted".

TABLE IX: **RESULTS OF THE SIMULATION WITH A 1000 kW TURBINE**

Parameter	**Value**
Full-load hours	1,081
Hours at surplus heat production	5,395
Hours at surplus electricity production	75
Waste heat (MMBtu)	11,328
Electricity sold to the grid (kWh)	1,358.4

4.4 Steam Turbine Size Optimization

Only one size of the steam turbine was analyzed so far. Thus, to perform a better analysis different sizes should be evaluated. Starting from 500 kW up to 1500 kW, five capacities were evaluated: 500 kW, 750 kW, 1000 kW, 1250 kW, 1500 kW. For this optimization process, the inlet steam pressure that was used is 300 psi, all the other paramenters were kept constant. The analysis and comparison between different steam conditions will be performed in the next section.

For the sake of clarity, only the graphs of the heat recovered and the electricity produced for all the turbine sizes are reported below. In Appendix C, it will be possible to see also the other graphs, concerning the duration curves of the thermal load, the electric load and the fuel consumption.

In Figure 31, Figure 32 and the results for the 500 kW turbine are shown. In Figure 33 and Figure 34 the graphs of the 750 kW are represented. For the 1250 kW the results are shown in Figure 35 and Figure 36. Finally, Figure 37 and Figure 38 are for the 1500 kW turbine.

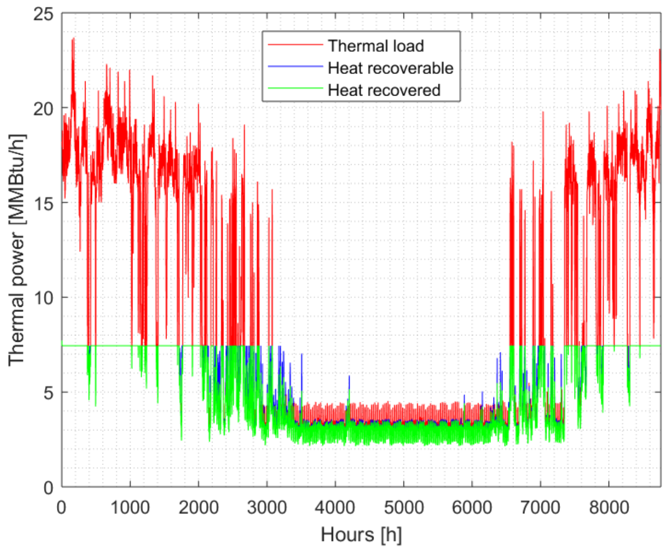


Figure 31: Thermal load of the building with the recoverable and the actual recovered heat by the 500 kW turbine

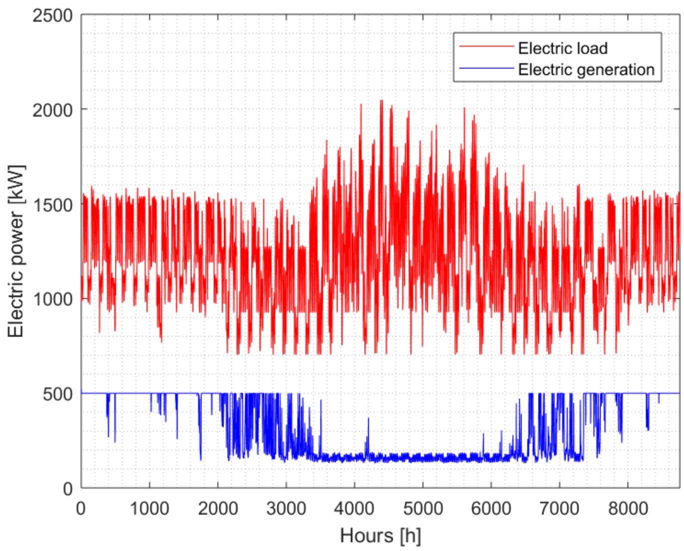


Figure 32: Electric load of the building with the electricity produced by the 500 kW Turbine

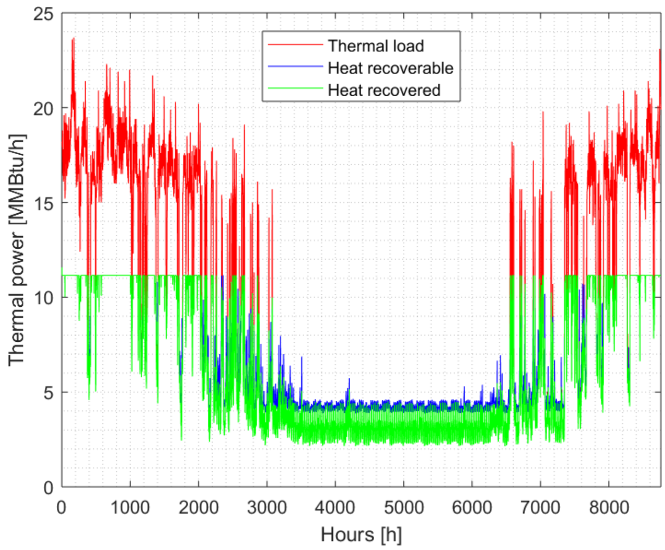


Figure 33: Thermal load of the building with the recoverable and the actual recovered heat by the 750 kW turbine

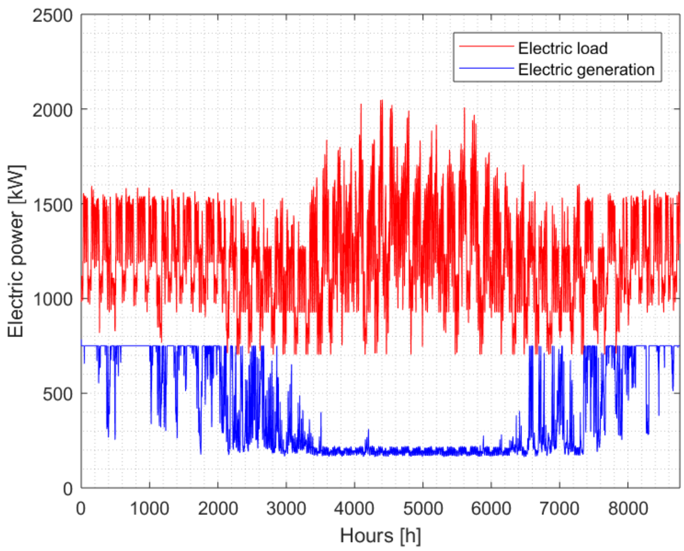


Figure 34: Electric load of the building with the electricity produced by the 750 kW Turbine

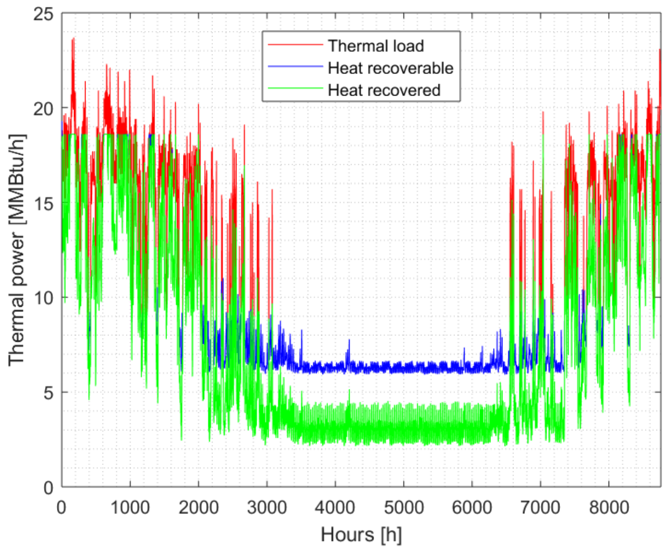


Figure 35: Thermal load of the building with the recoverable and the actual recovered heat by the 1250 kW turbine

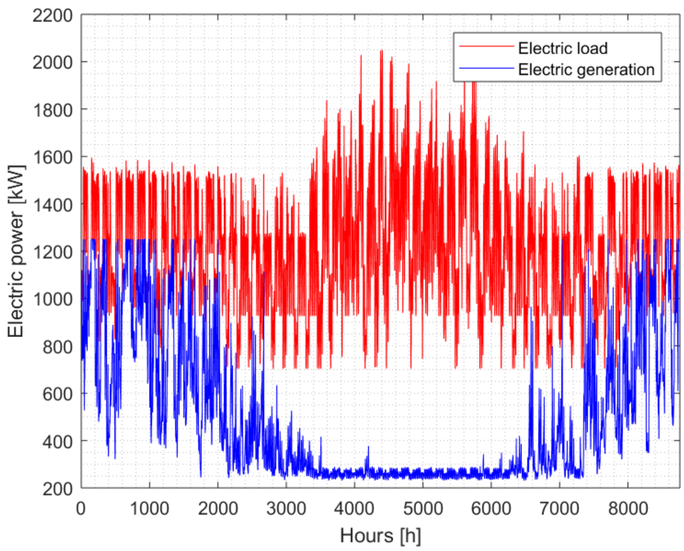


Figure 36: Electric load of the building with the electricity produced by the 1250 kW Turbine

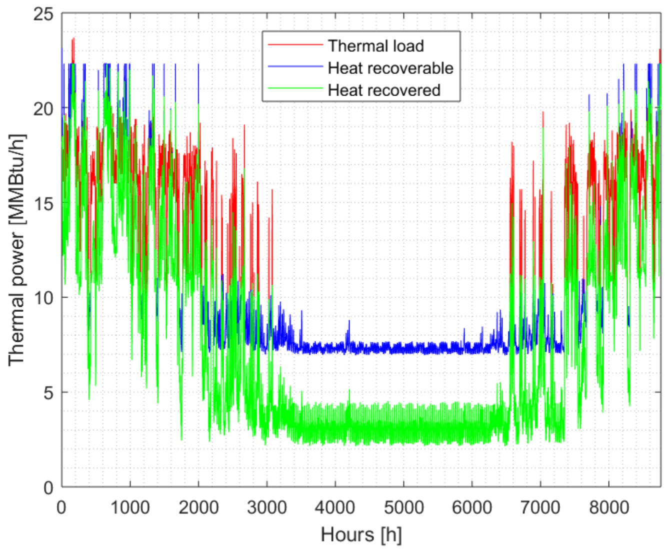


Figure 37: Thermal load of the building with the recoverable and the actual recovered heat by the 1500 kW turbine

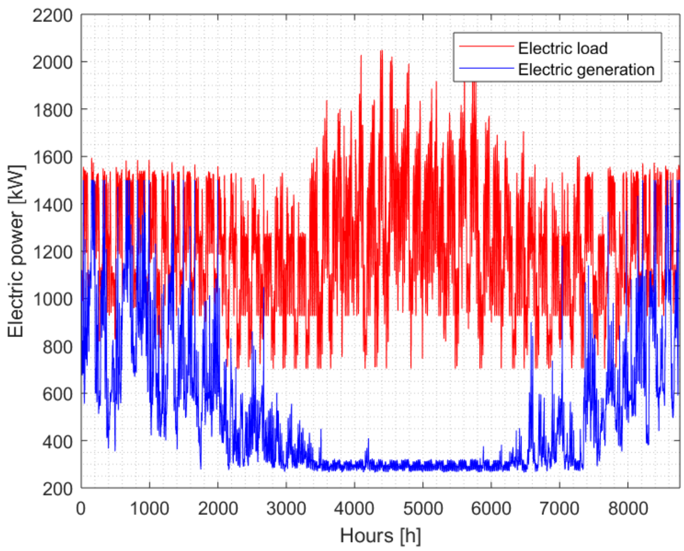


Figure 38: Electric load of the building with the electricity produced by the 1500 kW Turbine

Clearly, a turbine with a capacity of 500 kW runs at full load a higher number of hours throughout the year (3605 hours) than the larger ones. The heat recovered and electricity produced are much lower, however, considering the smaller capacity, there is no electricity surplus and the wasted heat is not as much as the case with a bigger steam turbine.

It is noiticeable that, every time the steam turbine size is increased, the thermal and electric production increase. For the larger steam turbines, the number of full load hours is lower but the waste in heat and electricity increases.

In Table X the results of these simulations are summarized in terms of full load hours and waste, including also the 1000 kW turbine.

TABLE X: **COMPARISON OF THE STEAM TURBINE SIZES**

Parameter	500 kW	750 kW	1000 kW	1250 kW	1500 kW
Full-load hours (hrs)	3,605	2,371	1,081	394	171
Hours at surplus of heat production (hrs)	4,342	5,251	5,395	5,620	5,942
Hours at surplus of electricity production (hrs)	0	0	75	355	323
Recoverable wasted heat (MMBtu)	2,186	6,363	11,328	16,854	23,023
Electricity sold to the grid (MWh)	0	0	1.358	43.30	63.95

As Table X shows and as it was already mentioned, the full load hours increase with the capacity of the steam turbine, as well as the hours at surplus of electricity and heat production.

A further analysis, to better understand what the best size for the steam turbine is, in this specific system, may include the evaluation of the efficiency. Using (Equation 3.4), the efficiency of the overall CHP system was calculated for all the system configurations. In addition, two more steam turbine sizes have been added to the efficiency analysis, 1750 kW and 2000 kW.

The parameters needed to compute the CHP efficiency are: the electric production (E), the total heat recovered (Q) and the total fuel consumption (F). In Table XI the results for all the steam turbine sizes are shown.

TABLE XI: **CHP EFFICIENCY FOR DIFFERENT STEAM TURBINE SIZES**

Steam Turbine Size	**E (kWh)**	**E (Btu)**	**Q (Btu)**	**F (Btu)**	**Efficiency (%)**
500 kW	1.98×10^6	6.74×10^9	6.07×10^{10}	8.60×10^{10}	78.4
750 kW	2.11×10^6	7.20×10^9	6.35×10^{10}	9.63×10^{10}	73.4
1000 kW	2.27×10^6	7.73×10^9	6.63×10^{10}	1.11×10^{11}	66.9
1250 kW	2.48×10^6	8.46×10^{10}	7.09×10^{10}	1.27×10^{11}	62.3
1500 kW	2.73×10^6	9.30×10^9	7.46×10^{10}	1.45×10^{11}	57.9
1750 kW	2.98×10^6	1.02×10^{10}	7.48×10^{10}	1.63×10^{11}	52.2
2000 kW	3.24×10^6	1.11×10^{10}	7.48×10^{10}	1.81×10^{11}	47.5

The highest efficiency was found for the 500 kW with a value of 78.4 %, which is similar to real values for CHP systems using small size steam turbines. In Figure 39 the efficiency values calculated are shown together with the efficiency trend.

Looking at the trendline is noticeable that when the steam turbine size increases, the efficiency decreases. It is pretty understandable considering that, as it was previously said and shown, when the turbine becomes bigger the fuel needed for running the boiler increases and

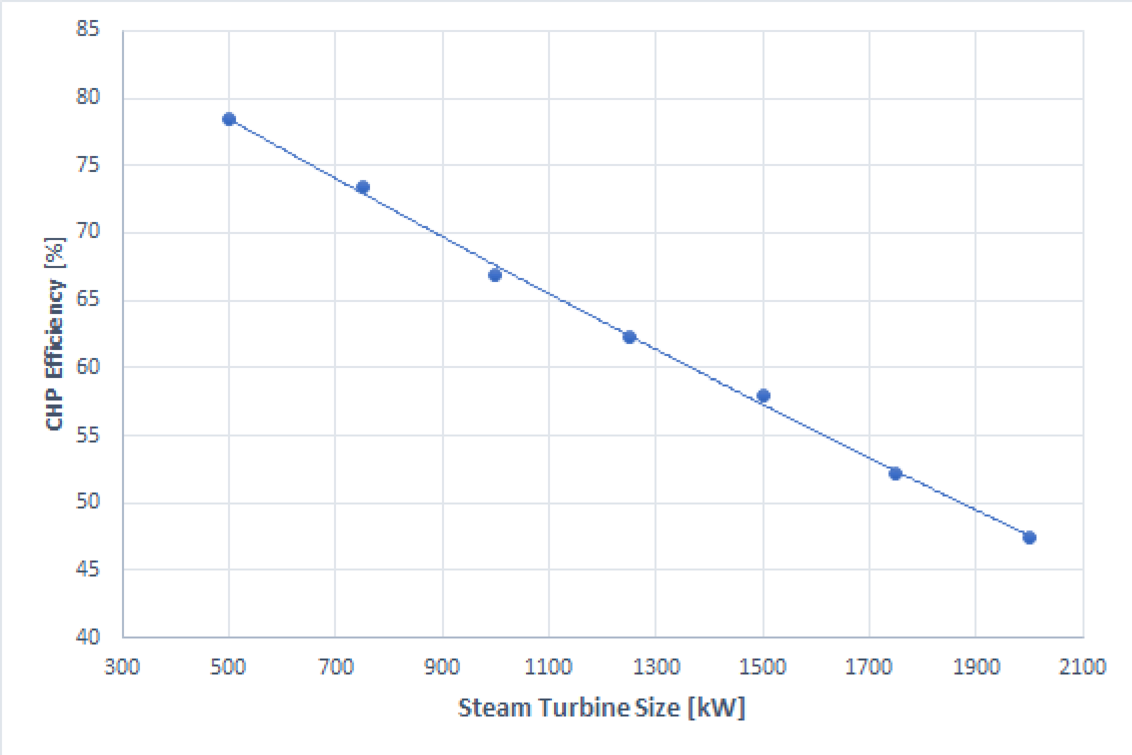


Figure 39: CHP Efficiency for different Steam Turbine sizes at 300 psi inlet steam pressure

the waste in electricity and heat increases as well. The line is reaching a maximum point going to a smaller turbine size, which corresponds to the efficiency of the boiler, which is the highest possible value that can be reached. Considering that the goal of this analysis was to find an optimum point, using the CHP efficiency is not the best solution. In fact, what this results are saying is that the best condition is the one in which we do not install a turbine (0 kW turbine).

Considering that the CHP efficiency trend has an unpredicted behavior or more in general is not useful for the optimization purpose, another way for evaluating the performance of the CHP system is considering the percentage reduction in source energy consumption (PRSEC). The PRSEC allows to estimate the reduction in primary energy source consumption produced by the installation of the cogeneration plant. In order to do that, the baseline system production is compared to the consumption of the new layout. The baseline condition is the original one, when electricity was bought from the grid and heat was generated by the on-site boilers. The PRSEC formula is shown in (Equation 4.6).

$$\mathrm{PRSEC} = \frac{\mathrm{SE}_{\mathrm{CONV}} - \mathrm{SE}_{\mathrm{CHP}}}{\mathrm{SE}_{\mathrm{CONV}}} = 1 - \frac{\mathrm{SE}_{\mathrm{CHP}}}{\mathrm{SE}_{\mathrm{CONV}}} \tag{4.6}$$

Where:

- SE_{CONV} is the Source Energy Consumption for the Conventional Production;
- SE_{CHP} is the Source Energy Consumption due to the CHP installation.

Thank to the ANSI/ASHRAE Standard 105-2014:Standard Methods of Determining, Expressing and Comparing Building Energy Performance and Greenhouse Gas Emissions [23] the source energy consumption analysis was perforemd. This standard provides an approach for estimating the energy performance of buildings and their GHG emissions associated with the building operations. In order to calculate the PRSEC the Standard provides specific conversion factors for every energy source. By knowing these factors and the effective consumption of the bulding, the source energy cnsumption can be easily calcualted.

Based o the area of the country the facility is located there are different conversion factors. They depend on where the energy consumption occurs. The reason for this differentiation is due to the fact that the efficiency of the production and delivery change from state to state. The employed technology the grid efficiency, the region of provisions are some of the features that are taken into account for the evaluation of the conversion factors. The standard divides the country in 26 regions. Illinois is part of the region called RFCW where the conversion factors are:

- 3.29 for electricity;
- 1.09 for natural gas.

There are two possible approaches to calculate the PRSEC index. The first one analyzes the overall facility, taking into consideratin all the equipment for the operation of the building. By doing so, the percentage reduction in soruce energy consumption of the entire facility due to the installation of the CHP system will be estimated. It is influenced not only by the performance of the CHP itself and the prime mover, but also by all other equipment of the system such as the boilers or the chillers. If following this approach the equation of the PRSEC becomes like (Equation 4.7), where the numerator represents the source energy consumption of the facility with the CHP system, while the denominator represents the fuel source energy consumption of the baseline facility, without the CHP installation.

$$\mathrm{PRSEC} = 1 - \frac{(\mathrm{SE_{F.B.}} + \mathrm{SE_{F.PM.}} + \mathrm{SE_{E.P}})_{\mathrm{CHP}}}{(\mathrm{SE_{F.B.}} + \mathrm{SE_{E.P.}})_{\mathrm{CONV}}} = \tag{4.7}$$

$$= 1 - \frac{(\mathrm{F_B} \cdot \mathrm{AF_{NG}} + \mathrm{F_{PM}} \cdot \mathrm{AF_{NG}} + \mathrm{E_{pur.}} \cdot \mathrm{AF_E})_{\mathrm{CHP}}}{(\mathrm{F_{PM}} \cdot \mathrm{AF_{NG}} + \mathrm{E_{pur.}} \cdot \mathrm{AF_E})_{\mathrm{CONV}}} \tag{4.8}$$

Where:

- $SE_{F.B.}$ is the Source Energy associated with the fuel for the boilers;
- $SE_{F.P.M.}$ is the Source Energy associated with the fuel for the prime mover;
- $SE_{E.P.}$ is the Source Energy associated with the electricity purchased;
- F_B is the fuel consumption for the boilers;
- $F_{P.M.}$ is the fuel consumption for the prime mover;
- $E_{pur.}$ is the electricity purchased from the grid;
- AF_{NG} is the ASHRAE factor for natural gas;
- AF_E is the ASHRAE factor for electricity purchased.

The second approach for calculating the PRSEC focuses just on a smaller part of the system rather than the whole plant. Thus, instead of measuring the performance of the whole system, this second approach represents the calculation of the performance of just the CHP system. For this approach the PRSEC becomes the $PRSEC_{os}$, which states for *on-site*. The equation for calculating it becomes the ratio of the source energy needed by the CHP system for its operation to the source energy needed for producing the same electrical and thermal output in the conventional plant without CHP. In other words it represents the percentage reduction of source energy consumption per unit of on-site energy produced by the CHP plant. This apporach wants to measure the CHP performance without including the other equipment of the facility. In (Equation 4.9) the formula for its estimation is shown.

$$\mathrm{PRSEC_{os}} = 1 - \frac{(\mathrm{SE_{F.P.M.}})_{\mathrm{CHP}}}{(\mathrm{SE_{Q_{rec.}}} + \mathrm{SE_{E_{PM}}})_{\mathrm{CONV}}} = \tag{4.9}$$

$$= 1 - \frac{(\mathrm{F_{PM}} \cdot \mathrm{AF_{NG}})_{\mathrm{CHP}}}{(\mathrm{F_B} \cdot \mathrm{AF_{NG}} + \mathrm{E_{P.M.}} \cdot \mathrm{AF_E})_{\mathrm{CONV}}} = \tag{4.10}$$

$$= 1 - \frac{(\mathrm{F_{PM}} \cdot \mathrm{AF_{NG}})_{\mathrm{CHP}}}{(\frac{\mathrm{Q_{rec.}}}{\eta_{\mathrm{Boiler}}} \cdot \mathrm{AF_{NG}} + \mathrm{E_{P.M.}} \cdot \mathrm{AF_E})_{\mathrm{CONV}}} \tag{4.11}$$

Where:

- $\mathrm{Q_{rec.}}$ is the recovered heat from the prime mover;
- η_{Boiler} is the efficiency of the boilers;
- $\mathrm{E_{P.M.}}$ is the electricity produced by the prime mover;
- $\mathrm{F_{P.M.}}$ is the fuel consumption for the prime mover;
- $\mathrm{F_B}$ is the boiler fuel consumption to replace the recovered heat only;
- $\mathrm{SE_{Q_{rec.}}}$ is the source energy associated with the fuel for the boilers to replace the recovered heat only;
- $\mathrm{SE_{F.P.M.}}$ is the source energy associated with the fuel for the prime mover;
- $\mathrm{SE_{E_{PM}}}$ is the source energy associated with the electricity produced by the gas turbine;
- $\mathrm{AF_{NG}}$ is the ASHRAE factor for natural gas;
- $\mathrm{AF_E}$ is the ASHRAE factor for the electricity purchased.

Despite having two possible approaches for calculating the percentage reduction of source energy consumption, only one of them will be assessed. Only the PRSEC will be calculated, so that it will be possible to evaluate the performance of the overall system and have a clearer understanding of the plant as a whole. Therefore, in Table XII are reported the efficiency results in terms of percentage reduction in source energy consumption (PRSEC) for all the steam turbine sizes, while Figure 40 the trendline of the PRSEC is shown for several steam turbine size.

TABLE XII: **PRSEC RESULTS FOR DIFFERENT SIZES**

Size (kW)	$\mathbf{E_{pur,c}}$ **(kWh)**	$\mathbf{E_{pur,c}}$ **(Btu)**	$\mathbf{F_{CHP}}$ **(Btu)**	$\mathbf{F_b}$ **(Btu)**	$\mathbf{E_{pur,b}}$ **(Btu)**	**PRSEC (%)**
500	7.92×10^6	2.70×10^{10}	8.77×10^{10}	1.08×10^7	3.67×10^{10}	5.8
750	7.09×10^6	2.42×10^{10}	9.30×10^{10}	1.08×10^7	3.67×10^{10}	7.6
1,000	6.65×10^6	2.27×10^{10}	9.74×10^{10}	1.08×10^7	3.67×10^{10}	7.7
1,250	6.50×10^6	2.22×10^{10}	1.03×10^{11}	1.08×10^7	3.67×10^{10}	5.7
1,500	6.38×10^6	2.18×10^{10}	1.09×10^{11}	1.08×10^7	3.67×10^{10}	2.7

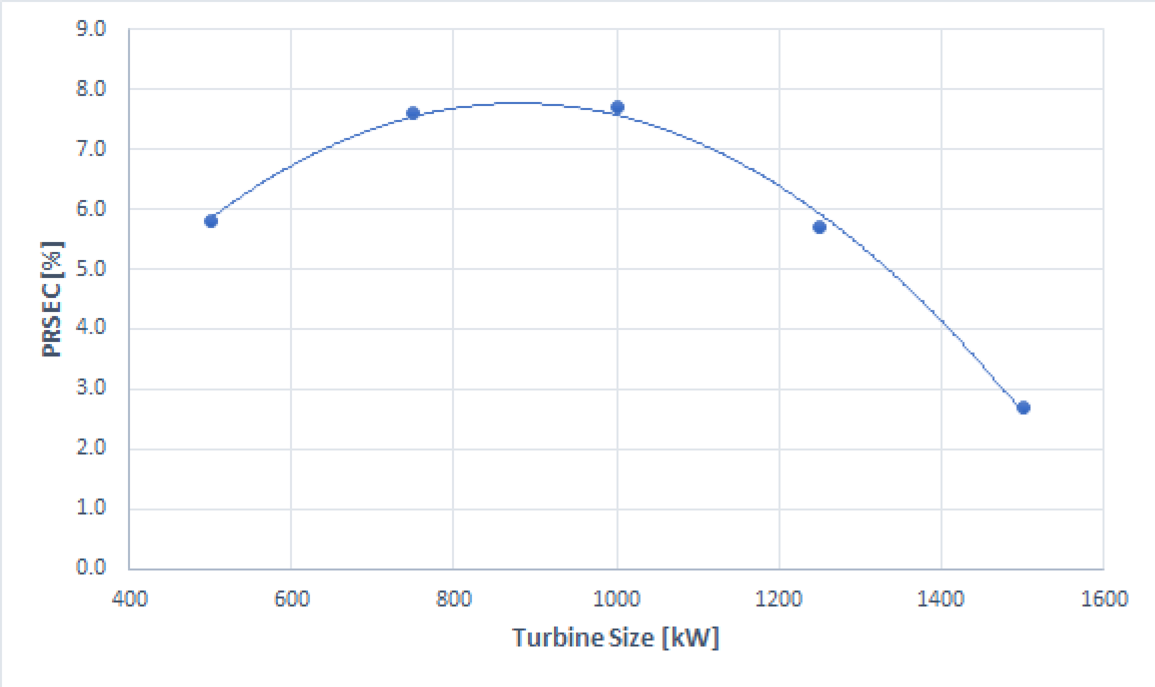


Figure 40: PRSEC trend for different Steam Turbine sizes at 350 psi inlet steam pressure

The table and the figure show that the highest value of PRSEC is obtained for the 1000 kW turbine. It means that the layout with a 1000 kW ensures the best reduction in terms of source energy consumption. The trend for smaller sizes is reasonable because in the case of a smaller turbine the amount of electricity produced and heat recovered is lower. Going to a zero kW turbine, the percentage reduction tends to zero, because it represents the conventional plant (baseline). For a larger steam turbine the heat recovered and produced electricity are higher, the latter in particular will replace the more expansive purchase from the grid in terms of source energy. However, when the steam turbine size is too big the system is oversized and there are waste electricity and waste heat which result in lower efficiencies. The PRSEC trend, being parabolic, is probably more similar to the behavior that one would expect for a CHP system, unlike it happened with the CHP efficiency.

4.5 Steam Conditions Optimization

After having simulated and compared different steam turbine sizes, the steam conditions will be addressed. In particular, considering that the steam exhaust pressure will remain constant, as it was already explained, what will be changed is the steam pressure at the inlet of the turbine.

Previous simulatins have been performed with an inlet pressure of 300 psi and 350 psi. They are reasonable values taken as an average from multiple examples found online with an exhaust pressure of 125 psi and a small size steam turbine.

The pressure range that has been considered goes from 300 psi to 400 psi. Larger pressure values could have been evaluated, but eQUEST®has some limitations in terms of pressure. In particular the software's guide states that if the pressure exceeds 350 psi, since we are dealing with hot water loops that simulates steam loops' behavior, the algorithm through which the software simulates the building performance could result in errors. Below 300 psi, the pressure becomes too low for running the turbine in the proper way, given the outlet pressure of the steam.

As it was shown previously, the CHP efficiency is not the best parameter to optimize the system because it actually does not allow to obtain an optimum point. On the contrary, it states that the turbine should not be installed in the first place to reach the highest value of efficiency. For this reason, the CHP efficiency will not be used for the optimization of the steam pressure.

The Percentage Reduction of Energy Source Consumption (PRESC) will be used to see which inlet pressure allows to reach the higher efficency in eQUEST®. It should be more accurate in order to understand if cogeneration is bringing the right improvements to the baseline system in terms of source energy consumption. The analysis that was performed for an inlet steam pressure of 350 psi was repeated for other pressures, in particular the ones shown are 300 psi, 330 psi and 400 psi. The intermediate values were not displayed because the objective is to highlight noticeable changes in the curves. In Figure 41 these four conditions are shown.

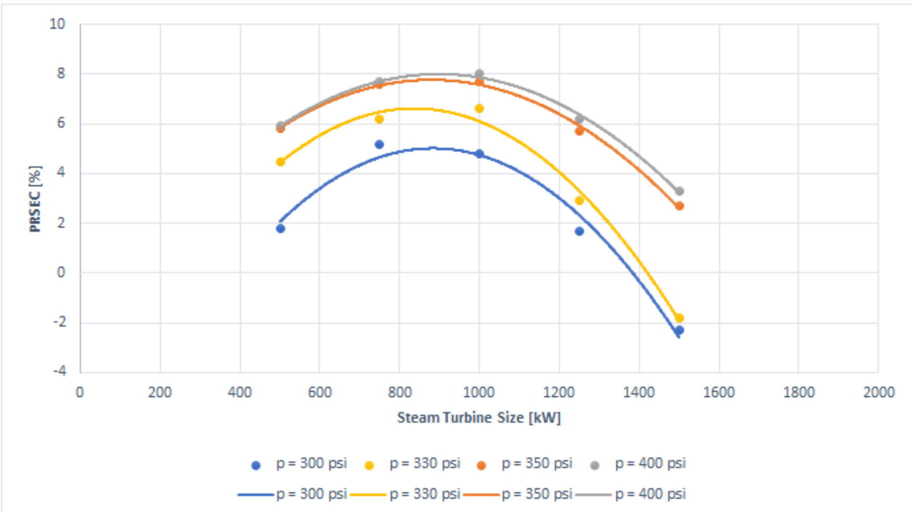


Figure 41: PRSEC trendlines for different steam pressures and steam turbine size

The trend of the percentage reduction in source energy consumption is similar for all the steam pressure conditions. It has a parabolic trend, decreasing for larger turbine sizes and when the steam turbine size tends to zero, the PRSEC tends to zero as well. Concerning the right side of the graph, when the size is increasing, it is important to remember that as the steam turbine gets larger, the amount of waste heat and electricity increases, affecting the efficiency of the system and as a consequence there is not an improvement in terms of source energy consumption. The overall value is not very high, reaching a maximum of 8 %. Electricity has a higher conversion factor than natural gas, therefore, considering that the turbines are following the thermal load and that steam turbines in general do not have a high electrical efficiency, the reduction in source energy consumption does not reach great values. In addition, for a very large turbine with a low inlet pressure the percentage reduction becomes negative, which means that the CHP implementation is not improving the system in terms of source energy consumption. Finally, it is noticeable that from 300 psi to 400 psi the PRSEC values do not change very much.

4.6 Final results Comparison

Previous sections focused on the optimization of the steam turbine size and the steam pressure in terms of efficiency of the CHP system and source energy reduction.

The results showed that the the CHP efficiency has a trend that is not useful for optimizing the system. In particular, the trend is descendant and the highest efficiency is reached when the turbine is just 500 kW. It means that the best solution would be remove the steam turbine in order to obtain the best result.

However, when considering the percentage reduction in source energy consumption, even though the PRSEC values for an inlet pressure of 400 psi are the best, the increase is not considerably higher, leading to think that increasing the pressure over 350 psi is not worth consideration. Therefore, the very first condition that was evaluated for the simulations ends up being, if not the optimum one, the one that is best suited for this application.

A final comparison that can be done in order to understand how all the results obtained and showed in previous sections are related to one another, could be considering the electricity produced in every simulation performed over the entire year. The steam turbine has been set to track the thermal load, which means that electricity is a secondary output of the system. Nonetheless, it is very important to produce or "recover" has much electricity as possible in order to reduce the costs associated with its purchasing from the grid. In Figure 42 this final comparison is shown.

From Figure 42 it is noticeable that the electricity produced over the entire year by the steam turbine in MWh has been plotted against the steam turbine size in kW for all the previously mentioned inlet steam pressures. With the increase of the steam pressure the curves tend to go higher, being the electricity produced more. For the highest steam pressures, the trend tends to a flat line when the steam turbine size is larger. It means that electricity is being wasted. The turbine is following the thermal load and producing too much electricity, thus selling it to the grid. On the other hand for the lowest pressures, the trendline is increasing even for larger steam turbine sizes. The reason being the fact that the turbine is not running at full capacity, even when the load would require it, because the pressure difference is too low and

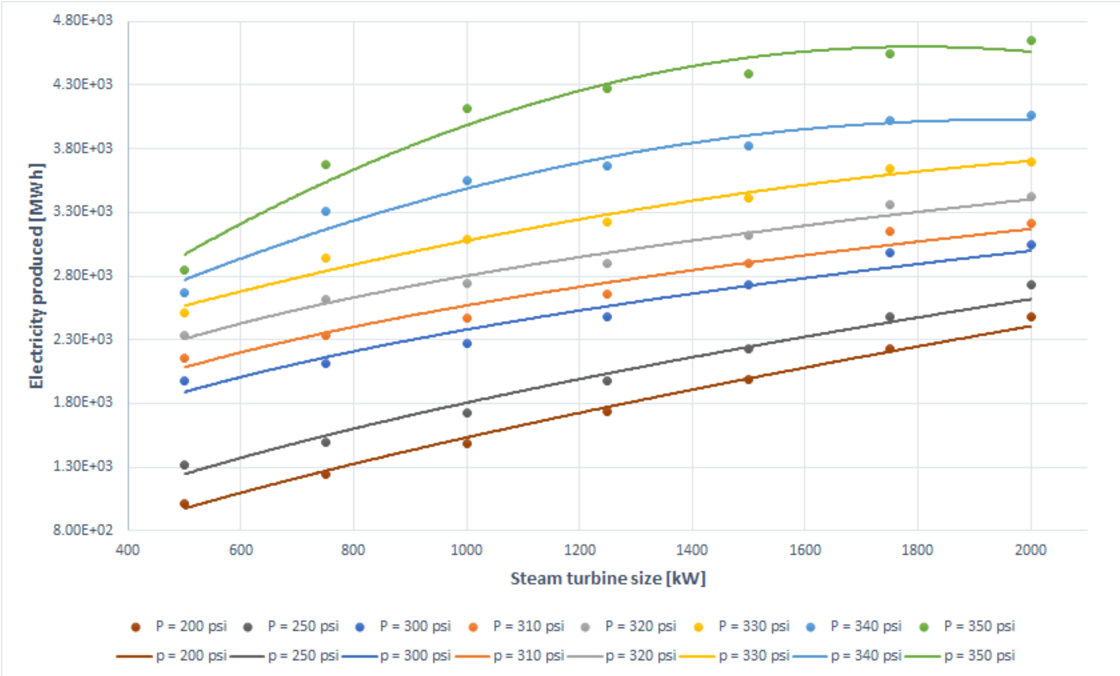


Figure 42: Electricity produced versus steam turbine size

as a consequence the enthalpy difference as well. Thus, even for a steam turbine with a larger capacity the electricity production will keep increasing.

The previous graph could be very useful if someone is considering to install a topping steam turbine in a similar building with a steam system and wants to know how to size the turbine. By knowing the expected load of the building over the entire year and the working pressure, drawing a horizontal line corresponding to the electricity production, where this line intercepts the working pressure of the system it is possible to know what is the resulting size needed for the steam turbine.

CHAPTER 5

CONCLUSIONS

The goal of this work was to analyze a steam turbine driven cogeneration system for a prototype hospital facility using the free software eQUEST®, checking if the model was appropriate and evaluating the energy performance of the building. Starting from some control strategies, such as setting the turbine to follow the thermal laod, some parameteres, like the exhaust steam pressure being fixed to a certain value (125 psi), all the simulations were performed.

The results highlighted the fact that the software has some limitations. The first one regarding the use of steam turbines themselves. In fact, eQUEST®does not simulates steam loops therefore all the loops in the plant are hot water loop, treating the one connected to the boilers as a steam loops. In addition, the model is able to simulate a steam turbine but it is not too much accurate. It was seen that even though the turbines were following the thermal load, they were not able to modulate their power in the proper way for part load performances in the range of 10 % - 20 % of the total capacity, causing the production of waste heat. This influenced the calculation of the CHP efficiency of the plant and its trend, especially when the pressure of the steam was increased.

Further improvements to this work can start from creating a new model of the steam turbine, by looking at real turbines on the market, taking into consideration the fact that small size turbines would be required and what efficiencies curves the software requires by the user. Some examples of manufacturers are Elliott Group or Dresser-Rand by Siemens.

In addition, kwnowing which turbine size resulted in the best efficiencies results, in particular considering the percentage reduction in source energy consumption (PRSEC), an economic analysis could be carried out in future theses works. The results show that the most favourable condition is represented by a 1000 kW steam turbine with a inlet pressure of 350 psi and an outlet pressure of 125 psi. This condition is also the first one that was evaluated. It seems like the most accurate choice even though some other simulations showed a higher CHP efficiency. Initial costs of the purchased turbine and future savings could be evaluated, with the aim of calculating the net present value over the project lifetime and the payback time.

Finally, the last graph shown in Figure 42, as it was already mentioned could be used to size a turbine in an already existing steam system located in a similar building type.

APPENDICES

Appendix A

HOSPITAL

A.1 First floor

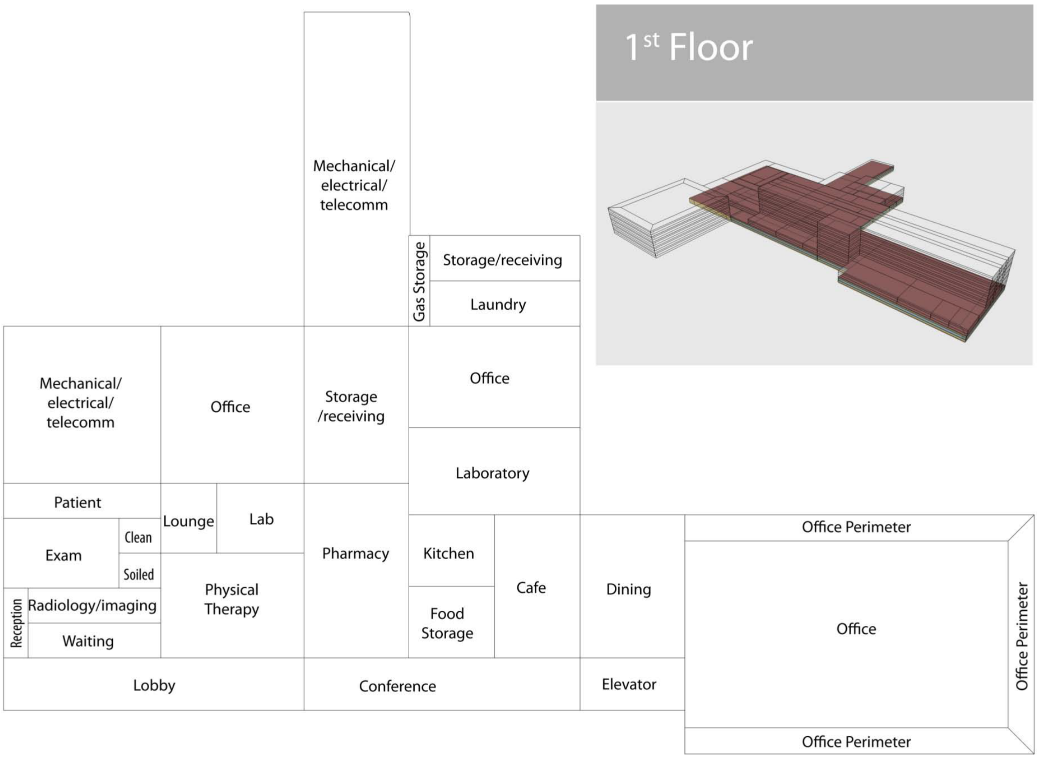


Figure 43: Map of the hospital's first floor

Appendix A (continued)

A.2 Second floor

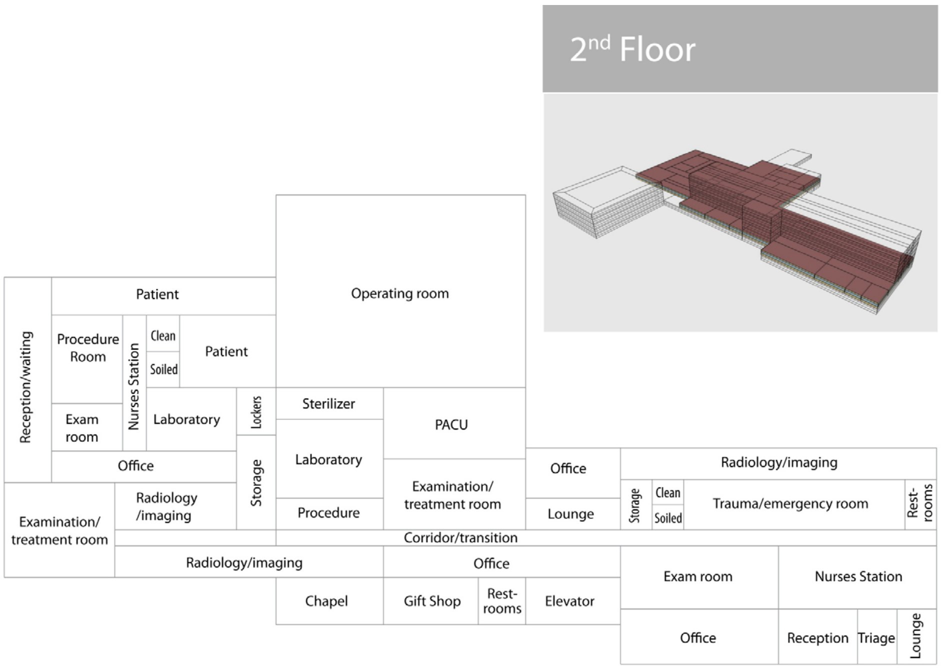


Figure 44: Map of the hospital's second floor

Appendix A (continued)

A.3 Third floor

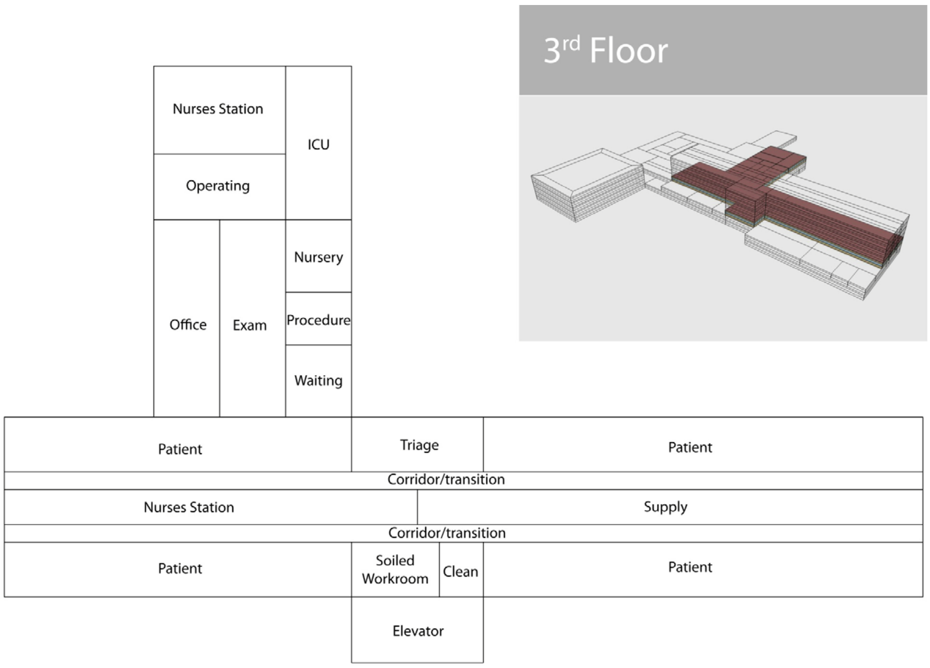
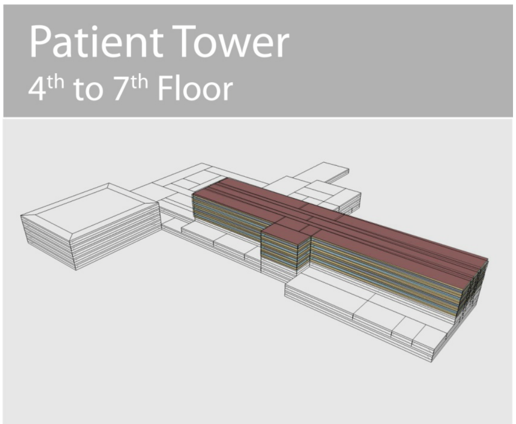


Figure 45: Map of the hospital's third floor

Appendix A (continued)

A.4 Patient Tower (PT)



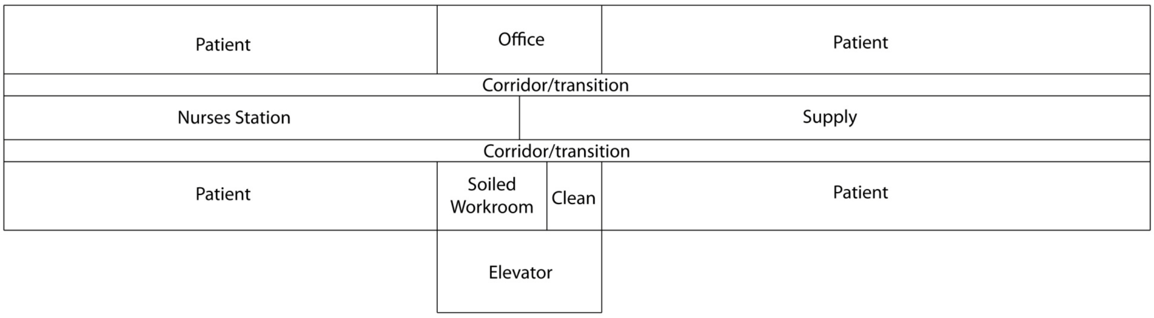


Figure 46: Map of the Patient Tower's third to seventh floors

Appendix A (continued)

A.5 Medical office building (MOB)

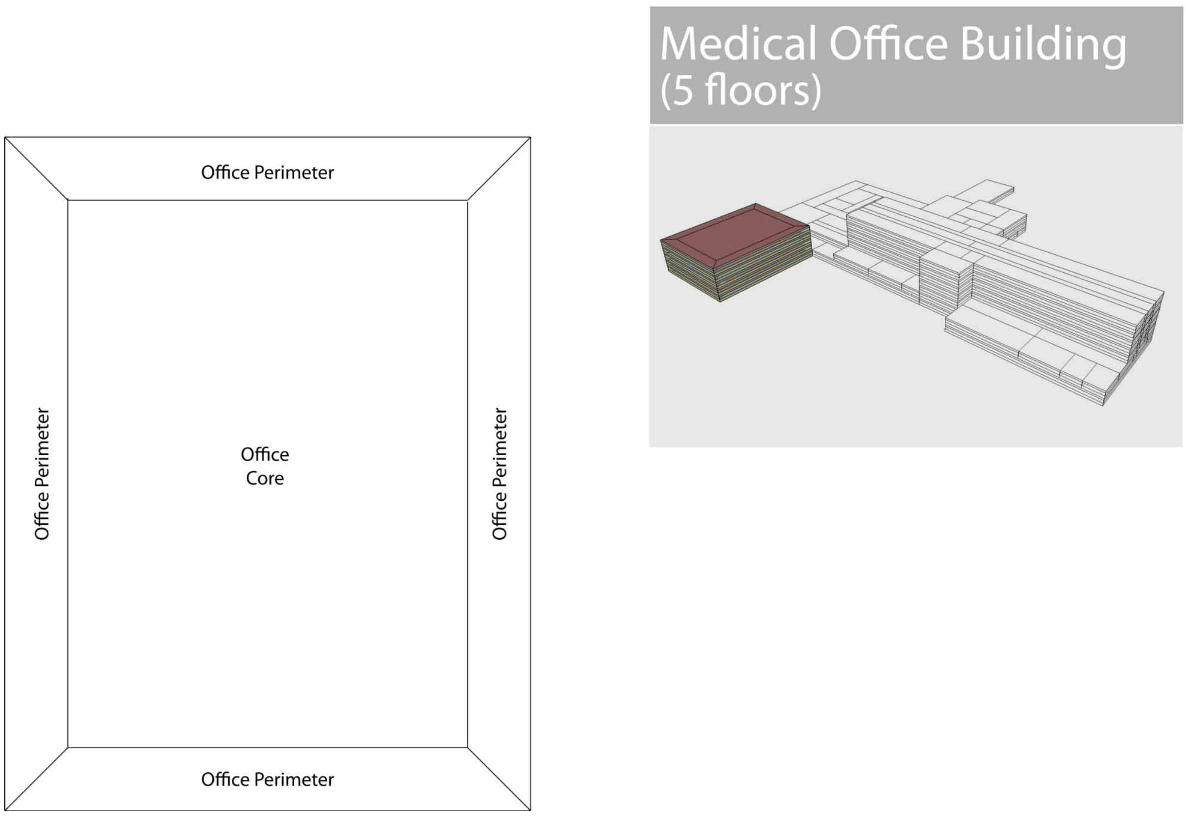


Figure 47: Map of the five floors of the Medical Office Building

Appendix B

INSTRUCTIONS TO MODEL A STEAM TURBINE DRIVEN CHP SYSTEM IN eQUEST®

Step-by-step instructions

- Open the file containing the baseline model and to to *Building Creation Wizard* on *WS1*, rename the file in order to identify the new model and save. In this way, the baseline model can be preserved for later use.
- Go to the *"Mode"* tab in the Toolbar and choose *"Detailed Data Edit"*. At the top of the toolbar select the " *Water-Side HVAC* tab.
- **Input of the Electric Generator (Steam Turbine):**
  - In the Component Tree, at the lef side of the screen, right clik on the folder named *"Electric Generators"* and then click on *"Create Electric Generators"*.
  - In the next screen set the *"Electric Generator Name* and under *"Electic Generator Type"* select *"Steam Turbine"*.
  - Next, in the *"Required Electric Generator Data"* screen, specify the *"Generator Capacity"* in kW and for the *"Steam Loop"* select the *"Hot Water Loop"*.
  - After that, the *"Basic Specifications"* tab set the main electric generator properties.
  - In *"Mechanical Efficiency"* modify the value from 0.1 to 0.9.
  - In the *"Maximum Ratio"* change the value from 1.1 to 1.0.

Appendix B (continued)

- Preserve all the other parameters in this tab, assuring that the *"Capacity"* is the same one that was previously inserted.
- Move to the *"Loop Attachments"* tab and under *"Exhaust Recovery Loop"* select the *"Hot Water Loop"*.
- On the right side of the screen, assure that the *"Steam Loop"* is the one that was selected before and modify the steam paramters.
- For *"Entering Steam Pressure"* use 350 psi, for the *"Superheat Above Sat."* use 50 F, for the *"Exhaust Steam Pressure"* use 125 psi.
- Preserve all the other default parameters.

- **Electric Generator Control:**
  - In the component tree right click on " *"Equipment Controls"* and then *"Create Equipment Control"*.
  - Create from scratch and select *"Electrical"* as the type of control.
  - Chose the meter to which the generator is attached to.
  - Under the *"Load Range 1"* go to *"Generator Name"* deop down menu and search for the turbine that was created. Input the capacity under *"Max Load"*.
- **Domestic Hot Water Load into the Hot Water Loop**
  - From the toolbar go to *"Water-Side HVAC"* and open with a double click the *"DHW plant 1 loop (1)"* from the component tree.
  - Under *"Process/DHW loads"*, copy the value of the *"Process Flow"*.

Appendix B (continued)

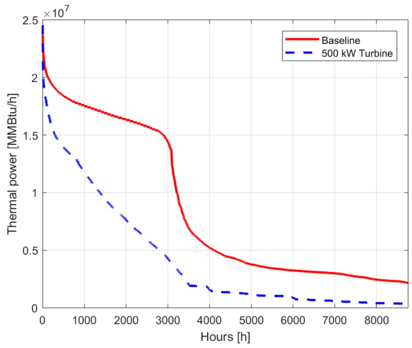
  - Now open the *"Hot Water Loop"* section and select *"Process/DHW loads"*. Using (Equation 2.3) calculate the *"Process Load"* and insert it in *"Process Flow"*.
  - For *"Process Load Schedule"* select *"DHW Eqp NRes Sch"*.

- **Changing the Control Strategy**
  - In the component tree open the electric meter (*"EM1"*) that is attached to the electric generator.
  - In *"Basic Specifications"* look for the *"Track Mode"* tab under *"Cogeneration"*.
  - Change it from *"Track Electric Load"* to *"Track Thermal Load"*.

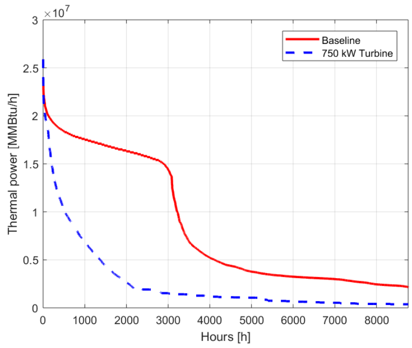
Appendix C

DURATION CURVES OF THE BUILDING LOADS

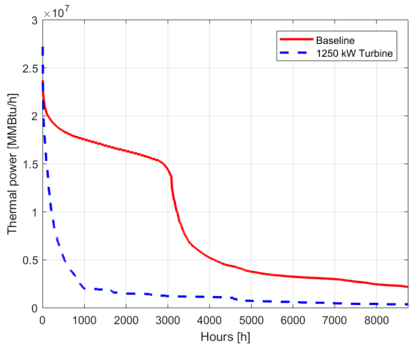
C.1 Thermal Load



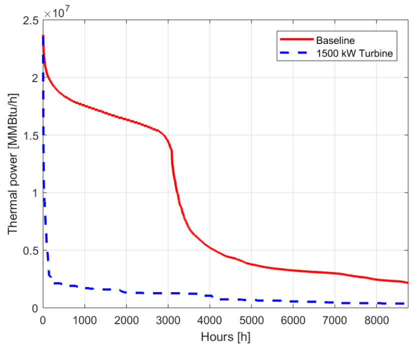
(a) 500 kW Steam Turbine



(b) 750 kW Steam Turbine



(c) 1250 kW Steam Turbine

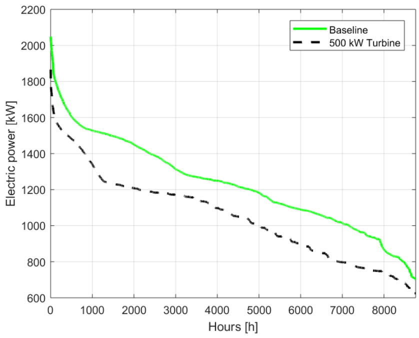


(d) 1500 kW Steam Turbine

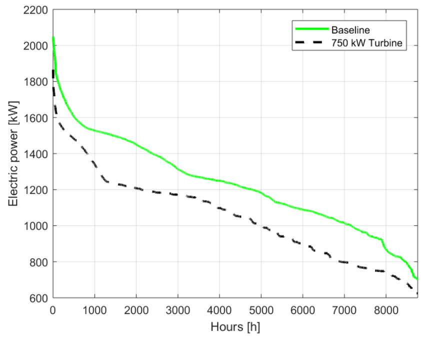
Figure 48: Duration curves for the thermal load in the baseline case and with different turbines

Appendix C (continued)

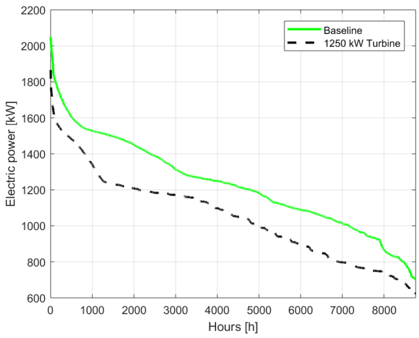
C.2 Electric Load



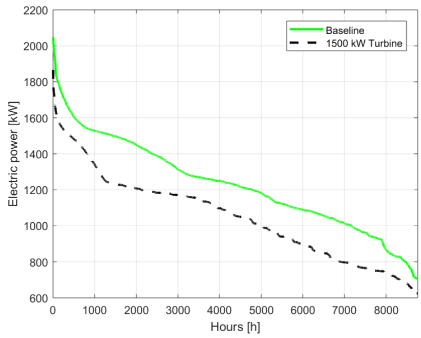
(a) 500 kW Steam Turbine



(b) 750 kW Steam Turbine



(c) 1250 kW Steam Turbine

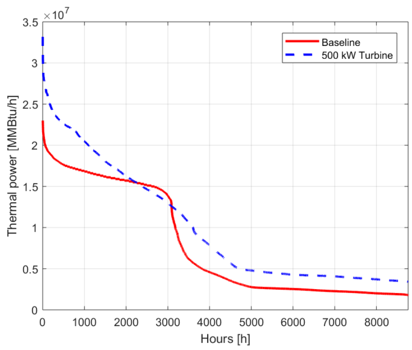


(d) 1500 kW Steam Turbine

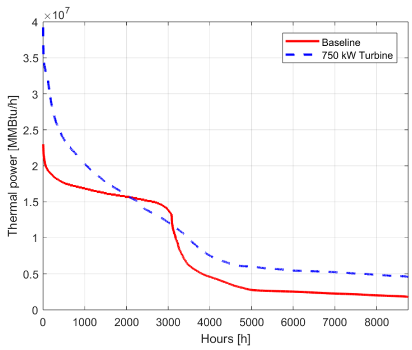
Figure 49: Duration curves for the electric load in the baseline case and with different turbines

Appendix C (continued)

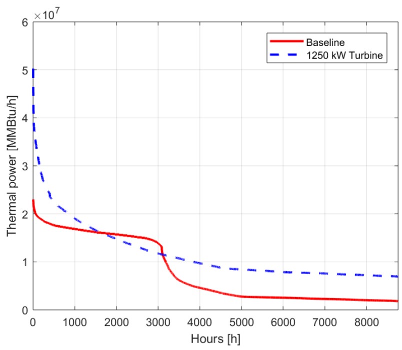
C.3 Fuel Consumption



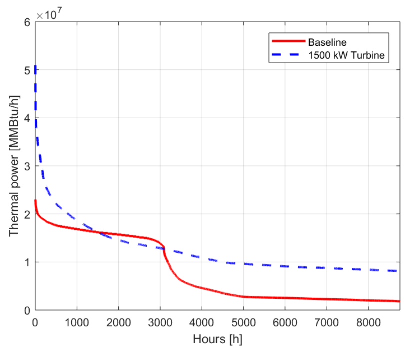
(a) 500 kW Steam Turbine



(b) 750 kW Steam Turbine



(c) 1250 kW Steam Turbine



(d) 1500 kW Steam Turbine

Figure 50: Duration curves for the fuel consumption in the baseline case and with different turbines

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Fall 2018	Italian scholarship for students of the TOP-UIC project
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