UTILIZATION OF NATURAL GAS, OPTIMIZATION OF COGENERATION/ COMBINED CYCLE APPLICATIONS IN CAMPUS ENVIRONMENT

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ABSTRACT

UTILIZATION OF NATURAL GAS, OPTIMIZATION OF COGENERATION/ COMBINED CYCLE APPLICATIONS IN CAMPUS ENVIRONMENT

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A computer program, called “Cogeneration Design" is developed using Visual Basic 6.0, for conceptually designing cogeneration power plants. Design is focused on power plants to be built in university campuses, where there is mainly heating, hot water, electricity and sometimes cooling demands. Middle East Technical University campus is considered as the primary working area.

Before the conceptual design study, detailed information regarding description of the campus, infrastructure, annual electric, water and heat demand covering last 10 years, properties of existing heat plant including natural gas expenses and specifications of the steam distribution pipes and electricity grid are collected and examined in detail.

Throughout the thesis, eight different natural gas fired cogeneration power plant designs are developed regarding different gas turbine and steam turbine configurations, for METU Campus, considering the Campus' properties described above, by using the "Cogeneration Design" program. Then, by means of a thermoeconomic optimization process, cost summary reports are prepared and the feasibility of the designed cogeneration power plants are discussed.

Key Words: Cogeneration, Combined Cycle, Optimization, Natural Gas, Campus Environment

ÖZ

KAMPÜS ÖLÇEĞİNDE KOJENERASYON/ KOMBİNE ÇEVİRİM SANTRALLARI UYGULAMALARI OPTİMİZASYONU VE DOĞAL GAZ KULLANIMI

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Çalışmada, Visual Basic 6.0 kullanılarak kojenerasyon santrallarının kavramsal tasarımına yönelik "Kojenerasyon Tasarımı" isimli bir bilgisayar yazılımı geliştirilmiştir. Tasarım çalışması, ısıtma, sıcak su sağlanması, elektrik sağlanması ve bazan da soğutma taleplerinin birarada bulunduğu üniversite kampüslarında kurulacak santallar üzerinde odaklanmış olup, esas çalışma alanı olarak Orta Doğu Teknik Üniversitesi kampüsü belirlenmiştir.

Kavramsal tasarım çalışmasına geçilmeden önce, ODTÜ kampüsünün tanımı, altyapısı, son on yıla ilişkin yıllık elektrik, su ve ısınma talebi, mevcut ısı santralının doğal gaz giderleri ve buhar dağıtım boru hatları ile elektrik şebekesinin özelliklerine ilişkin bilgi toplanmış ve ayrıntılı biçimde değerlendirilmiştir.

Tez çalışmasında, “Kojenerasyon Tasarımı” yazılımı kullanılarak farklı gaz ve buhar türbini konfigürasyonlarına göre ve ODTÜ kampüsünün yukarıda değinilen özellikleri dikkate alınarak, kampüs için sekiz adet farklı doğal gaz yakıtlı kojenerasyon santralı tasarımı geliştirilmiştir. Daha sonra ise, termoekonomik optimizasyon süreciyle maliyet dökümleri çıkarılmış ve tasarlanan kojenerasyon santrallarının yapılabilirliği tartışılmıştır.

Anahtar Sözcükler: Kojenerasyon, Kombine Çevrim, Optimizasyon, Doğal Gaz, Kampüs Ortamı

To my Family

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NOMENCLATURE

B_f	The exergy content of hydrocarbon fuel (kJ)
B_P	Exergy of process heat (kJ)
COP	Coefficient of performance of the refrigeration cycle
c_{pa}	Constant specific heat of air (kj/kgK)
c_{pg}	Constant specific heat of air+gas mixture (kj/kgK)
Ec	Heat recovered from the boiler (kW_t)
$\dot{E}c$	Heat recovered from the boiler (HRSG) per mass flow rate (kWs/kg)
E_f	Power supplied by fuel (kW)
E_P	Generated electrical power output (kW)
f_a	Fuel/air ratio
g_i	Gibbs function of reactants
g_e	Gibbs function of products
h_1	Enthalpy of refrigerant at the compressor inlet (kJ/kg)
h_2	Enthalpy of refrigerant at the condenser inlet (kJ/kg)
h_3	Enthalpy of refrigerant before the expansion valve (kJ/kg)
h_4	Enthalpy of refrigerant at the evaporator inlet (kJ/kg)
h_A	Enthalpy of Steam at steam turbine inlet (kJ/kg)
h_{B_actual}	Enthalpy of Steam at steam turbine outlet (actual case) (kJ/kg)
h_{B_ideal}	Enthalpy of Steam at steam turbine outlet, if process is assumed to be ideal (kJ/kg)
h_c	Enthalpy of condensate return (kJ/kg)
h_{Cond_exit}	Enthalpy of condensate water at the condenser exit satate (kJ/kg)
h_{econ}	Enthalpy of process steam, at HGSR economizer outlet temperature (kJ/kg)

$h_{fmaxsteam}$ Enthalpy of the saturated steam at maximum temperature and pressure in the Rankine(Steam Turbine) cycle (kJ/kg)

h_f Enthalpy of saturated liquid of process steam, at HGSR pressure and outlet temperature (kJ/kg)

h_i Enthalpy of reactants (kJ/kg)

h_e Enthalpy of products (kJ/kg)

$h_{maxsteam}$ Enthalpy of the steam at maximum temperature and pressure in the Rankine (Steam Turbine) cycle (kJ/kg)

h_{steam_exit} Enthalpy of process steam, at HGSR pressure and outlet temperature (kJ/kg)

h_{steam_inlet} Enthalpy of process steam, at HGSR pressure and inlet temperature (kJ/kg)

h_v Enthalpy of saturated vapour of process steam, at HGSR pressure and outlet temperature (kJ/kg)

m_a Mass flow rate of air (kg/s)

m_{fuel} Mass flow rate of fuel (kg/s)

m_{ref} Mass flow rate of the refrigerant (kg/s)

m_s Steam mass flow rate (kg/s)

m_{tot} Total mass flow rate of gas, air and combustion products (kg/s)

m_{water} Mass flow rate of inlet water to HRSG from process or condenser pump (kg/s)

n Politropic expansion constant for gases

P Power of the cogeneration plant (MW)

P_1 Compressor inlet pressure (Ambiant pressure) for the gas turbine (kPa)

P_2 Compressor outlet pressure (combustion chamber pressure) for the gas turbine (kPa)

P_3 Turbine inlet pressure for the gas turbine (kPa)

P_4 Turbine outlet pressure for the gas turbine (kPa)

P_e Electrical work (power) output of the Power Plant (kW)

$P_{maxsteam}$ Maximum pressure of the Rankine(Steam Turbine) cycle (kPa)

Q_{sh} Heat supplied by hot gases and air at superheater coils(kW)

Q_{econ}	Heat supplied by hot gases and air at economizer coils (kW)
Q_{evap}	Heat supplied by hot gases and air at evaporator coils (kW)
Q_H	Rate of heat disipation from the condensor of the vapour compression refrigeration cycle. (kW)
Q_L	Rate of heat removal from the evaporator of the vapour compression refrigeration cycle. (kW)
Q_P	Useful heat recovered in the HRSG (kW)
$Q_{sh+evap}$	Heat supplied by hot gases and air at superheater+evaporator coils (kW)
r, r_p	Compression ratio of Gas Turbine compressor
r_{ph}	Power to Heat Ratio
s_A	Entropy of steam at steam turbine inlet (kJ/kg.K)
s_{B_ideal}	Entropy of steam at steam turbine outlet, if process is assumed to be ideal (kJ/kg.K)
s_c	Entropy of condensate return. (kJ/kg.K)
SFC	Spesicif fuel consumption (kg/kWh)
T_0	Temperature of the environment
T_1	Compressor inlet temperature for the gas turbine (Ambiant temperature) (K)
T_2	Compressor outlet (combustion chamber inlet) temperature for the gas turbine (K)
T_3	Turbine inlet temperature (combustion chamber outlet temperature) for the gas turbine (K)
T_4	Turbine outlet temperature for the gas turbine (K)
T_e	HRSG outlet temperature of air+gas mixture (K)
T_{evap}	Temperature of steam at evaporator exit (K)
$T_{maxsteam}$	Maximum temperature of the Rankine(Steam Turbine) cycle (K)
$T_{sat@P}$	Saturation temperature of steam at HRSG pressure (K)
v_f	Spesific volume of saturated liquid water (m^3/kg)
W_{comp}	Required turbine work to drive the compressor (kJ/kg)
W_{in}	Isentropic compression power input into the cyle (kW)
W_{pump}	Pump work in the steam turbine cycle (kW)

W_{spec} Usable turbine work for the cycle (kJ/kg)

W_{Steam} Electrical Work (power) output of the steam turbine (kW)

W_{tot} Total gas turbine work (kJ/kg)

Greek Letters

γ Ratio of specific heats

η_c Isentropic efficiency of the compressor

η_t Isentropic efficiency of the gas turbine

η_{cc} Efficiency of combustion chamber

η_m Mechanical efficiency of the gas turbine

h_{elect} Mechanical efficiency of electric production for gas turbine

η_{elect_Steam} Mechanical efficiency of electric production for steam turbine

η_{ST} Steam turbine efficiency

ΔT_{pinch} Pinch point temperature difference (K)

Abbreviations

AAR Ammonia-Water Absorption Refrigeration

CCGT Combined Cycle Gas Turbine

CHCP Combined Heating, Cooling And Power Generation

CHP Combined Heat and Power

GT Gas Turbine

HAT Humid Air Turbine

HP High Pressure

HR Heat Recovery

HRSG Heat Recovery Steam Generator

HTWH High Temperature Water Heater

IP Intermediate Pressure

LHV	Lower Heating Value
LP	Low Pressure
MRA	Absorption Chiller
NG	Natural Gas
SOFC	Solid Oxide Fuel Cells
ST	Steam Turbine
TBC	Thermal Barrier Coatings
TIT	Turbine Inlet Temperature

CHAPTER 1

INTRODUCTION

Cogeneration is one of the best energy production techniques that can be used to maintain the quality, and accessibility in energy production while reducing fuel consumption, thus, representing energy conservation and more efficient use of energy resources.

Whenever a simultaneous demand for power and process heat is needed, co-generation (COGEN) system, or also referred to as combined heat and power (CHP) offers an opportunity which can contribute signifcantly to the efficient use of energy. In other words, COGEN systems are expected to play an important role in solving the global energy and environmental problems that are on issue in the recent years, while reducing emissions of air pollutants and greenhouse gases.

Cogeneration is simply simultaneous production of electrical or mechanical energy and useful thermal energy from a single energy source such as oil, coal, natural or liquefied gas, biomass, or solar. Cogeneration allows the producer to have his own electricity, hot water and steam, if he needs. In this way, a cogenerator reduces the site's total outside purchased energy requirements and this reduction on energy use compared to independent heat and electricity generation, may, in return reduce the total cost of utility service, and also the fuel resources. Also the distribution losses which is an important problem will be decreased.

As it is seen clearly, cogeneration and its facilities offer great advantages and for today's energy production sector, it plays an important role. In this study, a brief description about cogeneration, it’s technical systems, components, facilities and products are given, and a computer program is developed for the basic design of a cogeneration power plant. The user manual which is prepared for the computer program, is given in detail. With the help of this program, an optimization study and a detailed case study is carried out, for the campus of Middle East Technical University. All steps of that process are clearly identified in the present study, together with the thermodynamic approach and formulations for the calculations.

Another topic studied in the thesis work is trigeneration; an even more efficient and environmental friendly process than cogeneration. A trigeneration plant, defined in non-engineering terminology, is most often described as a cogeneration plant with absorption chillers. A well-designed trigeneration plant can achieve up to 10% greater system efficiency than a cogeneration plant of similar size. Trigeneration energy process produces four different forms of energy from the primary energy source, namely, hot water, steam, cooling (chilled water) and power generation (electrical energy). For a better understanding of the trigeneration system, refrigeration processes and related formulations are also given in the present study.

All the plant designs studied here are based on the use of natural gas as the primary energy source. This is because natural gas has a number of advantages in utilization It is a clean, safe, easy to use and easy to deliver type of fuel among all energy sources. These properties of natural gas make it very suitable for the campus environment.

To determine the feasibility of the plants, an economic approach is adopted in the thesis work. This is done by means of including a cost optimization concept into the design process of power plants, considering the investment required and the operation costs. A brief cost analysis is given for each cogeneration and trigeneration case and a long term economic study is carried out with the help of a commercial program called “Thermoflow”.

1.1.General

1.1.1. Gas Turbine Power Cycle Applications

As a means of producing mechanical power, the turbine is the most satisfactory one in many respects. The absence of reciprocating and rubbing members means that balancing problems are few, oil consumption is exceptionally low and reliability is relatively high. Around the turn of the 20th century, steam turbine has become the most important prime mover for electricity generation. It was not long before the Second World War, the need for higher energy resulted a development in the turbine technology, and the hot gases themselves are used to drive the turbine, which then resulted the spread of gas turbine.

Now the gas turbine is used in a wide range of applications. Common uses include power generation plants (electric and heat production), military and commercial aircraft (thrust generation).

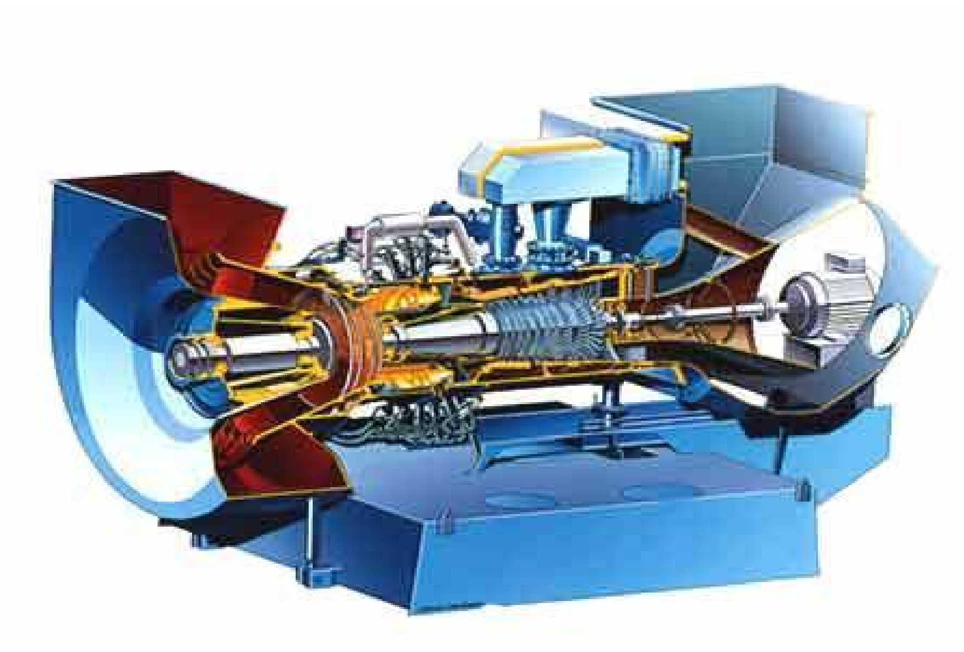


Figure 1.1. A Typical Gas Turbine Engine

1.1.1.1. Simple Cycle

In a simple gas turbine cycle, low-pressure air is drawn into a compressor (state 1) where it is compressed to a higher pressure (state 2). Fuel is added to the compressed air and the mixture is burnt in a combustion chamber. The resulting hot products enter the turbine (state 3) and expand to state 4. This expansion of the hot working fluid produces a great power output from the turbine. Most of the work produced in the turbine is used to run the compressor and the rest is used to run auxiliary equipment and produce power.

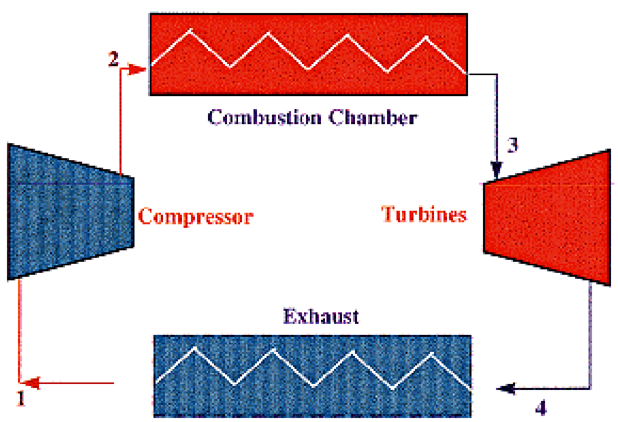


Figure 1.2 Simple Gas Turbine Flow Diagram

Air standard models; The Brayton cycle; provide useful quantitative results for gas turbine cycles. The four steps of the cycle are:

- (1-2) Isentropic Compression
- (2-3) Constant Pressure Heat Addition
- (3-4) Isentropic Expansion
- (4-1) Constant Pressure Heat Rejection

Necessary formulation for gas turbine cycle based on thermodynamic relations and energy equations are given in Part 3.2.1. In these models the following assumptions hold true:

- The working substance is air and treated as an ideal gas before entering the combustion chamber
- The combustion process is modelled as a constant pressure heat addition
- The exhaust is modelled as a constant pressure heat rejection process

In practice, losses occur both in the compressor and in all components of the turbine, which increase the power absorbed by the compressor and decrease the power output of the turbine. On top of this, pressure losses and piping losses occuring throughout the system contribute to reduction of the system efficiency. The maximum air/fuel ratio that may be used is governed by the working temperature of the highly stressed turbine blades, where temperature must not be allowed to exceed a certain critical value. This value depends upon the strength of the materials used in the construction of the turbine. Two most important factors affecting the performance of gas turbine is then, are the component efficiencies and the turbine working temperature. The overall efficiency of the gas turbine also depends on the pressure ratio of the compressor, and the heating value of the fuel indirectly.

It is important to realise that, in gas turbine, process of compression, combustion and expansion take place in different components, which are separate in the sense that they can be designed, tested and developed individually, and then linked together to form a gas turbine unit in various ways. Other components like heat exchangers, intercoolers and extra combustion chambers and extra turbines can be added to fulfill the system requirements for different applications.

Simple gas turbine, when compared to other power cycles, such as coal fired, hydraulic and steam turbine power cycles, is more flexible; the system is convenient for many applications; the power production is more reliable and the equipment is more durable. Also for gas turbine cycle applications, material and maintenance costs are relatively low, delivery time is short and starting and loading periods are fast. The turbine is environmentally profitable and clean.

The lack of efficiency in a simple gas turbine cycle may be overcame by installing cogeneration (CHP) facilities to the system thus utilizing the energy in the exhaust.

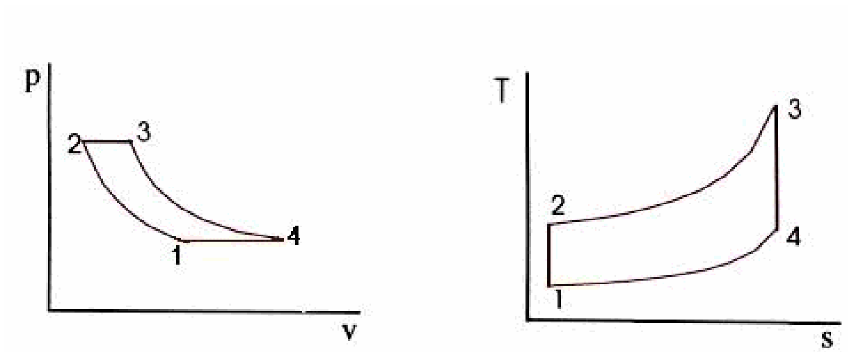


Figure 1.3 Pressure Versus Specific Volume and Temperature Versus Entropy Curves of a Simple Brayton Cycle

1.1.1.2 Combined Cycle

In the gas turbine, practically all the energy not converted to shaft power is available in the exhaust gases for other uses. The only limitation is the exhaust (stack) temperature. The exhaust heat may be used in various ways. If exhaust heat is used to produce steam in a waste heat boiler for a steam turbine, with the object of augmenting the shaft power produced, the system is called combined gas/steam cycle.

This way, combined cycle plants are used for large base-load generating stations and the overall thermal efficiency is increased, and running cost is decreased appreciably with respect to simple cycle gas turbine. Besides, these cycles are less environmentally hostile. [1] Necessary formulation for combined cycle based on thermodynamic relations and energy equations are given in Parts 3.2.2 and 3.2.3.

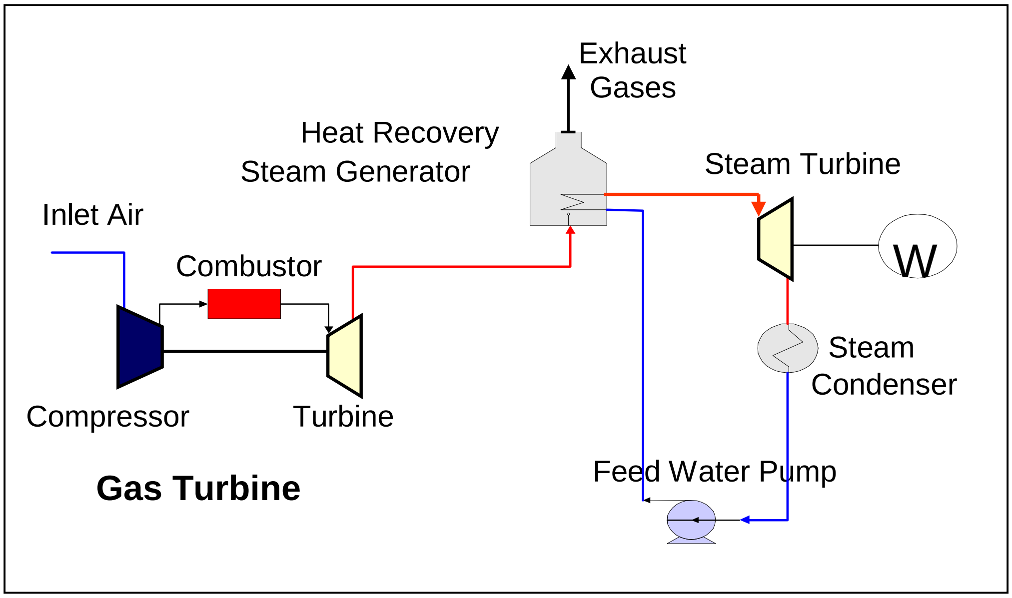


Figure 1.4 Combined Power Plant Illustration

1.1.1.3 Cogeneration Power Cycle

If the exhaust heat is used to produce hot water or steam for any purpose, or cold water/ice on site, the system is referred to as a cogeneration plant or CHP. In other words, with a cogeneration plant design exhaust gases are not allowed to escape until excess thermal energy has been recovered, this means, the "waste" heat that would be "lost up the stack" is captured.

A unique type of heat exchanger or heat recovery - high temperature water heater (HR-HTWH) is placed in line with the exhaust of a gas turbine, producing significant amounts of high temperature water (or steam) by using the hot exhaust gases in place of a boiler flame. Commercially available natural gas, which is mainly methane and mercaptan, is typically used as the primary fuel source.

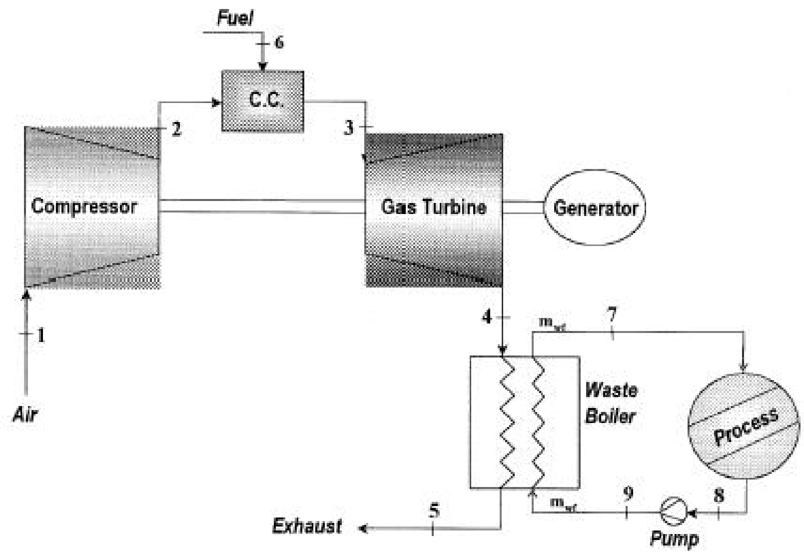


Figure 1.5 Cogeneration Power Plant Schematic. [2]

The use of clean-burning natural gas, coupled with the efficiency inherent in a cogeneration design, yields an independent power plant which produces electricity and hot water or steam at high efficiency while producing negligible emissions.

For a cogeneration cycle the formulation is given in Parts 3.2.2 and 3.2.3.

1.1.2. Cogeneration Systems

Cogeneration, simply is an opportunity to control and reduce energy costs by investing in a highly efficient, on-site power plant. Simply a cogeneration system takes heat that would normally be wasted and uses it to satisfy some or all of the thermal energy requirement. In this way, a cogenerator reduces the site's total outside purchased energy requirements and this reduction of energy use compared to independent heat and electricity generation, may, in return reduce the total cost of utility service, and also the unit fuel cost.

Like other investment opportunities, cogeneration may not be suitable for all cases. Cogeneration is a wise investment for the right combination of fuel and electric costs coupled with the energy user's ability to use the forms of energy that a cogeneration system can produce. When there is the right combination of factors, it is possible to reduce the annual energy cost by 33% to 50%. If the end user cannot effectively utilize the 'lost' heat, cogeneration may not be economical. For instance; for some cogeneration systems, investment payback period could be high, due to high investment cost and sometimes high fuel price. Also, price of excess electricity sold to grid maybe often low and cost of grid connection might be high. [2]

Cogeneration systems have been designed and built for many different applications. Large-scale systems can be built on-site at a plant, or off-site.

A large-scale application of cogeneration is for district heating. Many colleges and municipalities, which have extensive district heating and cooling systems, have cogeneration facilities. Some large cogeneration facilities are built primarily to produce power. They produce only enough steam to meet the requirements. Many utilities have formed subsidiaries to own and operate cogeneration plants. These subsidiaries are successful due to the operation and maintenance experience that the utilities provide them. They also usually have a long-term sales agreements lined up before the plant is built.

Cogeneration systems are also available to small-scale users of electricity. Small-scale packaged or "modular" systems are being manufactured for commercial and light industrial applications. Modular cogeneration systems are compact, and can be manufactured economically. These systems, ranging in size from 20 kilowatts (kW) to 650 kW produce electricity and hot water from engine waste heat. Several companies also attempted to develop systems that burn natural gas and fuel oil for private residences. These home-sized cogeneration packages had a capacity of up to 10 kW, and were capable of providing most of the heating and electrical needs for a home several fuel call manufacturers are targeting residential and small commercial applications. [3]

On the other hand, cogeneration provides several environmental benefits by making use of waste heat and waste products; air pollution is a concern any time fossil fuels or biomass are burned. Water pollution is also lessened by cogeneration systems.

One of the biggest advantages of cogeneration is that, the necessary energy of any type may be produced anytime, in any quantity, in other words, created independency for energy. Good examples for this advantage are, group of residences that are far from towns, industrial regions and university campuses.

1.1.3. Cogeneration In Universities

University campuses are places where heat demand is quite high during the semesters, and comparatively low during summer holidays, as well as the electrical energy need. Usually the ratio of heat demand to electrical demand is relatively high, and to meet the heating requirements for the campus, there will always be excess electrical energy, which should be sold to distribution company.

In other words, for cogeneration to be a reasonable and profitable choice for a university, a careful investigation is to be made, and campuses' heat and electric demands are to be clearly specified.

Like most of the other possible areas, in a campus, cogeneration would be the most cost-effective means of producing heat and electrical energy as well as the most realistic mechanism for controlling electrical energy costs. The university will benefit from the reduced CO_2 emissions arising globally from the independent generation of power as well as by the virtue of reduced electricity costs.

There are some important parameters to be examined concerning energy and power production in campus.

There are quite a lot of universities all around the world, which use the opportunity of cogeneration. The detailed list for these universities is given in Appendix A.

1.1.4. Natural Gas and Its Utilization

Natural gas is a vital component of the world's supply of energy. It is one of the cleanest, safest, and most useful of all energy sources. While commonly grouped in with other fossil fuels and sources of energy, there are many characteristics of natural gas that make it unique.

Natural gas is colorless and odorless in its pure form. It is combustible, and when burned it gives off a great deal of energy. Unlike other fossil fuels, natural gas is clean burning and emits lower levels of potentially harmful byproducts into the air. Energy requirement, to heat residences, cook food, to be used in the transportation sector and to generate electricity has elevated natural gas to such a level of importance in our society, in our lives and of course, in industry.

Natural gas is a combustible mixture of hydrocarbon gases, which is formed primarily of methane, it can also include ethane, propane, butane and pentane.

The composition of natural gas can vary widely, but below is a chart, in Table 1.1, outlining the typical makeup of natural gas before it is refined.

Table 1.1 Typical Composition of Natural Gas [4]

Methane	CH_4	70-90%
Ethane	C_2H_6	0-20%
Propane	C_3H_8	0-20%
Butane	C_4H_{10}	0-20%
Carbon Dioxide	CO_2	0-8%
Oxygen	O_2	0-0.2%
Nitrogen	N_2	0-5%
Hydrogen Sulphide	H_2S	0-5%
Rare Gases	A, He, Ne, Xe	trace

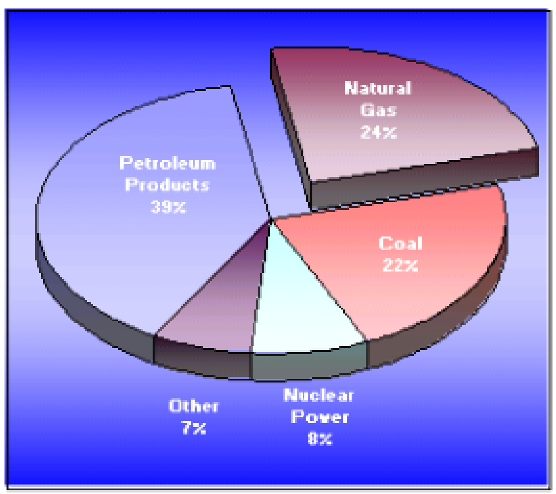


Figure 1.6 Total Energy Consumed in U.S., 2000 [4]

After refining, the clean natural gas is transmitted through a network of pipelines of thousands of miles. From these pipelines, natural gas is delivered to its point of use. For natural gas, the cycle begins at gas wells, where gas is extracted from the ground. After processing, the gas is compressed and distributed through pipelines-processes that consume a small amount of energy. According to the U.S. Department of Energy

(DOE), the overall efficiency of natural gas from source to end-user is about 91%. In other words, more than 9 out of 10 units of the primary energy taken from the ground actually reach the appliance. [5]

For the usage of natural gas as the firing fuel in power plants, the combination of high efficiency and low emissions at each point along the energy cycle lead to economic and environmental superiority. This is true in most cases, regardless of the application and competing fuel source. But as can be clearly seen in Figure 1.7, natural gas is mostly used in industrial applications.

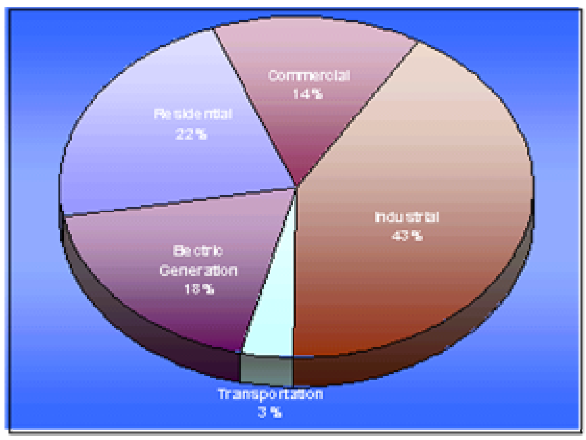
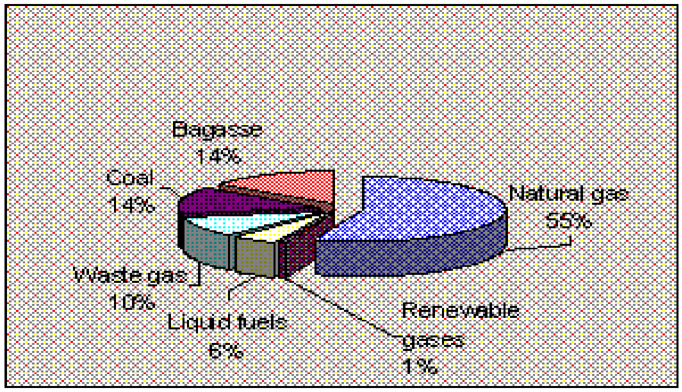


Figure 1.7 Natural Gas Use by Sector in U.S., 2002 [4]

To most consumers, natural gas is an invisible fuel. The pipeline and the product are transported underground and out of sight. Besides, natural gas service is reliable. The resource base is ample, the delivery system is efficient and expanding rapidly, the restructuring of the industry and increased reliance on market forces have improved service, and contracts can be written to meet an individual customer's needs.

Like in other energy production facilities, natural gas is the most used fuel in all kind of cogeneration applications; this can be seen in Figure 1.8. When all types of environmental impacts are considered, natural gas stands out as a superior energy form.



1.8 Fuel Types Used in Cogeneration, 2002 [5]

1.1.5. Trigeneration

Trigeneration process is an alternative design to further increase the efficiency in CHP generation, by having a bigger yield from the processes, resulting a lower consumption of the natural resources and a more economic performance. Trigeneration can be used in many industrial processes where there is a simultaneous need for electric power, heat and refrigeration at low temperatures.

Trigeneration, also referred to as district energy, achieves a higher efficiency and smaller environmental impact than cogeneration. The installation of a trigeneration plant can achieve up to 10% greater system efficiency than a cogeneration plant of similar size. A trigeneration plant is often described as a cogeneration plant that has added absorption chillers (MRA), which take the waste heat a cogeneration plant would have wasted, and convert this free energy into useful energy in the form of chilled energy. [6]

Best examples where trigeneration is used are the food and chemical industries. Nowadays, high fuel prices as well as the ecological implications of fuel consumption, give an impulse to energy technologies that better use the primary energy sources. This is why integrated production of utilities should be considered when designing a new production plant. The number of trigeneration system installations (electric generator, heat generator and absorption refrigeration plant) is increasing. [7]

In this study, trigeneration will be discussed as an alternative way of efficient and effective energy production in the following chapters. Necessary formulation for trigeneration cycles based on thermodynamic relations and energy equations are given in Part 3.3.

1.2. Literature Survey

Carlo Carcasci and Bruno Facchini, (2000), [1] presented the significance of research efforts which are currently centered on developing advanced gas turbine systems for electric power generation applications. They studied high efficiency gas+ steam combined cycles, proposed two innovative gas turbine technology applications for combined cycle applications. They have also presented two gas+steam combined cycles using thermodynamic analysis and a combined cycle with three pressure levels with reheat heat recovery boiler, used with two different gas turbine technologies (high pressure ratio and reheat against ``H'' technology). This analysis constitutes a comparison, not only between two different constructive solutions but also between two different gas turbine (GT) techniques (reheat and GT steam cooling) and technology applications (a consolidated and an advanced gas turbine technology) applied to a combined cycle. The analysis of the simulation results, based only on the thermodynamic analysis, indicates that both ``H'' technology and reheat lead to a relevant increase of performance in terms of specific work and efficiency (57.5% for the Alstom GT26 and 60% for the General Electric MS9001H). A supplementary simulation shows that the combination of the two technologies leads to a net efficiency increase of about 62%.

J.Luz-Silveira, A.Beyene, E.M.Leal, J.A.Santana and D.Okada (2000) [2], analysed replacement of an equipment of a cogeneration system by a thermo economic analysis method, based on the First and the Second Law of Thermodynamics. The cogeneration system consists of a gas turbine linked downstream to a waste boiler. The electrical demand of the campus is approximately 9 MW, but the COGEN system generates approximately one third of the university requirement as well as 1.764 kg/s of saturated steam (at 0.861 Pa), approximately, from a single fuel source. They showed by the energy-economic study that, the best system, based on pay-back period and based on the maximum savings (in 10 years), was the system that used the gas turbine ‘‘M1T-06’’of Kawasaki Heavy Industries and the system that used the gas turbine ‘‘CCS7’’of Hitachi Zosen, respectively. The exergy economic study showed that the best system, which has the lowest EMC, was the system that used the gas turbine ‘‘ASE50 ’’of Allied Signal.

Arif Hepbaşli, Nesrin Özalp (2002), [3] dealt with many aspects of the implementation of CHP studies in Turkey. An application of a ceramic factory located in İzmir, Turkey, with a total installed capacity of 13 MWe is also presented an discussed. They examined COGEN systems that installed during last few years in Turkey. They came up to some facts that; Most autoproducers work in real COGEN mode, utilizing heat to a significant extent. Simple cycle generation was permitted for one year only; sometimes costs for COGEN investments are 5 –6% higher than the simple cycle; Turkish fuel prices persist high while selling price of surplus electricity low; NG shortages forced many autoproducers to resort to less attractive fuels in combination with NG, or to discard the NG option all together; and 17 % of total autoproduction capacity generated on mixed fuel.

Joel Hernandez-Santoyo, Augusto Sanchez-Cifuentez (2003) [6] presented the design of a system of trigeneration, which is an alternative way for energy improvement to use in cogeneration systems. Savings are observed by the decrease of the fuel fed to the turbo generation equipment. A regenerative-cycle cogeneration system and a new trigeneration system were studied, showing their benefits as well as the operation criteria for both processes. They came up with the fact that trigeneration is a mean to achieve energy savings in future installation plants for heat and electricity generation.

Piero Colonna, Sandro Gabrielli, [7] studied industrial trigeneration using ammonia-water absorption refrigeration systems (AAR). They mentioned that, in many industrial places, there is a simultaneous need for electric power and refrigeration at low temperatures, like in the food industry and chemical industries. They figured out that, the increase in fuel prices and the ecological implications give an impulse to energy technologies that better exploit the primary energy source, and integrated production of utilities should be considered when designing a new production plant. The answer to the needs; trigeneration systems installations (electric generator and absorption refrigeration plant) are increasing nowadays, and ammonia water absorption refrigeration plants is a good solution, which they dealt, if low temperature refrigeration is needed. They presented the thermodynamic system of trigeneration configurations using commercial software integrated with specifically designed modules. Heat recovery from the primary mover at different temperature levels is analyzed and compared in the study and a simplified economic assessment is also given for one test case (10 MW electric power, 7000 h/year).

G. Ramkiran, K.Ashok Kumar, P.K. Nag (2002) [10], studied a waste heat recovery steam generator, (which) consisting of an economizer, an evaporator and a super heater. The unit produces superheated steam by absorbing heat from the hot flue gases. A general equation for the entropy generation is proposed, which incorporates all the irreversibilities associated with the process. By using suitable non-dimensional operating parameters, an equation for entropy generation number is derived. The effect of various non-dimensional operating parameters, on the entropy generation number is investigated. The results they found provide better understanding of influence of different non-dimensional operating parameters on entropy generation number, which in turn will be useful to optimize the performance of the unit.

Mitsuhiro Fukuta, Tadashi Yanagisawa, Hiroaki Iwata, Kazutaka Tada (2000) [13], discussed the feasibility of a vapour compression/absorption hybrid refrigeration cycle for energy saving and utilization of waste heat. The cycle employs propane as a natural refrigerant and a refrigeration oil as an absorbent. A prototype of the cycle is constructed, in which a compressor and an absorption unit are combined in series. They examined the performance of the cycle, both theoretically and experimentally. They concluded that, although the solubility of the propane with the oil was not enough as a working pair in the absorption unit, the theoretical calculation shows that the hybrid cycle had a potential to achieve a higher performance in comparison with a simple vapour compression cycle by using the waste heat. They also pointed out that, the application of an AHE (absorber heat exchanger) can reduce the heat input to a generator.

J.C. Bruno J. Miquel, F. Castells (1999) [14] studied the integration of absorption chillers in (combined heat and power) CHP plants by using a mathematical programming approach. The aim of this work is to determine the economic viability of the introduction of ammonia absorption chillers in energy systems instead of using the more conventional compression cycles. This procedure selected the best refrigeration alternative taking into account both absorption and compression cycles. They implemented this approach in the computer program “XV”, where the maximum power that can be produced is determined, and tested in an energy plant in the petrochemical complex of Tarragona (Spain). Refrigeration demands to be met were 0 and - 20 °C. The results highlighted the benefit obtained with the simultaneous presence of ammonia absorption cycles and a cogeneration based energy plant.

G.G. Maidment, X. Zhao, S.B. Riffat and G. Prosser, (1999) [16] summarized the results of an investigation concerning the viability of CHP systems in supermarkets. They theoretically investigated the viability of a conventional CHP by using a mathematical model of a typical supermarket, and demonstrated that a conventional CHP system might practically be applied. They also showed that, compared to the traditional supermarket design, the proposed CHP system would use slightly less primary energy, and the running costs would be significantly reduced. They (have) calculated an attractive payback period of approximately 4 years. However, it was also pointed out in the study that, despite these advantages a considerable quantity of heat would be rejected to atmosphere with this system, and this was because the configuration was utilizing the heat mainly for space heating, only for part of the year. To increase the utilization period, they proposed a novel CHP/absorption system. This configuration, driving an absorption chiller that refrigerates propylene glycol to 10°C for cooling the chilled-food cabinets, provided a continuous demand for the waste heat. According to the results of the study, such solution was theoretically applicable and the system was extremely efficient.

P.A. Pilavachi (2000) [18], prepared an overview of power generation with gas turbine and (combined heat and power) CHP systems. He also presented the European Union strategy for developing gas turbines and CHP systems. Ways to improve the performance of several types of gas turbine cycle, which will be a major objective in the coming years, are briefly discussed. The targets set forth were combined cycle efficiencies above 60%, industrial gas turbine system efficiencies of at least 50% and small gas turbines efficiencies above 35% and designs for the use of fuels with less than 25% heating value of that of natural gas. The main CHP targets are the reduction of the overall costs and the development of above 40 kW biomass-fired systems.

Yousef S.H. Najjar (1999) [19] reviewed ten research investigations in the field of gas turbine cogeneration in power and industry are reviewed that carried out by himself and his associates during the last ten years. He came up with below facts: The worldwide concern about cost, environment and quick availability to meet continuous load growth will continue to enhance the adoption of gas turbine engines in power systems; The escalating interest in efficient use of energy will support the adoption of cogeneration with simultaneous production of power and thermal energy; Cogeneration with gas turbines utilizes the engine's relative merits and boosts its thermal efficiency even at part load, with consequent high acceptance in power and industry and a multitude of research works utilizing gas turbine engines with steam, hydrogen and refinery gases predicted superior performance and economic feasibility of these cogeneration systems

Yousef S.H.Najjar (1996) [20] dealt with enhancement of performance of gas turbine engines by inlet air cooling and cogeneration system. He pointed out the fact that, the efficiency of the gas turbine would decrease with increasing ambiant temperature. Thus he considered improving the efficiency by adding an inlet air precooler of ammonia water absorption chiller. A heat recovery boiler was used to recover the exhaust heat before entering the generator of the chiller. He studied the power output, efficiency and specific fuel consumption of the system and compared it with the simple cycle. Results showed that the combined system achieves gains in the power, efficiency and SFC about 20%, and that the system is viable.

F.J.Wang,J.S.Chiou [21] examined many simple cycle gas turbine generation sets (GENSETs) that were originally designed to serve as peak load units which are forced to operate continuously during the entire summer season, due to the serious power shortage in Taiwan. They have seriously considered converting those GENSETs (which have the advantage of fast start up, but suffer from low power output and thermal efficiency at high ambient temperature) into more advanced cycle units with higher efficiency and higher output. Among many proven technologies, like inlet air cooling, intercooling regeneration, reheating, steam-injection gas turbine (STIG) etc., they found that STIG was one of the most effective technologies in boosting both the output capacity and thermal efficiency. The results from computer simulation indicated that the retrofitting of existing GE Frame 6B simple cycle unit into STIG cycle could boost the output from about 38 to 50 MW, while the generation efficiency could be increased from about 30% to 40%.

E. Bilgen (2000) [22] presented an exergetic and engineering analyses as well as a simulation of gas turbine-based cogeneration plants consisting of a gas turbine, heat recovery steam generator and steam turbine. The exergy analysis was based on the first and second laws of thermodynamics and the engineering analysis was based on both the methodology of levelized cost and the pay back period. Two cogeneration cycles, one consisting of a gas turbine and the other of a gas turbine and steam turbine and process to produce electricity and heat were analyzed. Based on these analyses, an algorithm was developed for thermodynamic performance and engineering evaluation of combustion gas turbine cogeneration systems. Simulation results of gas turbine systems with cogeneration showed good agreement with the reported data.

M. Tuma, J. Oman, M. Sekavcnik (1999), [23], studied and discussed the equations for determination of the overall energy and exergy efficiency of a combined gas-steam cycle process. The cogeneration in the gas and in the steam cycle, the reduction of power due to increasing the heat flow in the steam process, the supplementary firing at the gas turbine exhaust, the heat recovery boiler efficiency as well as the heat exchanger efficiency in the gas and steam cycle were taken into consideration. The presentation of the results was based on the thermodynamic derivation, calculations of a typical example, and graphic diagrams.

Sergio Augusto Araujo da Gama Cerqueira, Silvia Azucena Nebra (1999) [24] studied the design and operation analysis and optimization of complex thermal plants, in terms of both thermal and economical variables. Several different thermo economical methodologies are presented in the literature, each one based upon different principles and therefore presenting somewhat different results. In this article, four methodologies developed by diverse authors are applied to a simple gas turbine cogeneration system. The results were compared and the importance of the division of exergy into mechanical and thermal components, and the allocation of the cost of external irreversibility were discussed.

Yong-Ho Kwon, Ho-Young Kwak and Si-Doekoh (2000) [27] studied exergoeconomic analysis of gas turbine cogeneration, using the annualised cost of a component on the production cost in 1000 kW gas-turbine cogeneration system by utilizing the generalized exergy balance and cost-balance equations. Comparison between typical exergy-costing methodologies were also added by solving a predefined cogeneration system, CGAM problem. They found that the cost of products were crucially dependent on the change in the annualized cost of the component whose primary product was the same as the system's product. On the other hand, the change in the weighted average cost of the product was proportional to the change in the annualized cost of the total system.

Flavio Guarinello Jr, Sergio A.A.G. Cerqueira and Silvia A. Nebra (2000) [28], applied thermoeconomics concepts to a projected steam injected gas turbine cogeneration system, which aims at meeting the thermal and electrical demands of an industrial district sited in Cabo (Brazil). The power plant was evaluated on the basis of the First and Second Laws of Thermodynamics. A thermoeconomic analysis, using the Theory of Exergetic Cost, was performed in order to determine the production costs of electricity and steam. Two hypothetical operational conditions, concerning the level of electric power generation, were considered.

M.Liszka, G.Manfrida, A. Ziebik (2002) [29] dealt with modernization of an (industrial combined heat and power) CHP plant located in a medium capacity steelworks industrial site. It was proposed to couple the existing power plant with a new gas turbine unit fired with Corex export gas which is a cold, low Btu by-product of the Corex process for pig iron production. In the paper, the idea was to select the right distribution of heating surfaces in the heat recover steam generator (HRSG) connected to a previously selected gas turbine and to the existing bottoming cycle, in order to maximize the efficiency and economical profits of the whole plant. The study was performed using several simulation tools: a complete simulation of the system by means of engineering equation solver and a dedicated Fortran language code capable of performing all energy balances. For the correct design of the HRSG, a pinch analysis was applied. The results of the economic optimization (they have made,) demonstrated that there was a good opportunity for performance improvement using a multi pressure HRSG with optimized operating parameters.

CHAPTER 2

SYSTEM DESCRIPTION

2.1. Cogeneration System Description and Technology

A typical cogeneration system consists of a prime mover where fuel is converted to mechanical power and heat, steam turbine, or combustion turbine that drives an electrical generator, a waste heat exchanger (heat recovery system) that recovers waste heat from the engine and/or exhaust gas to produce hot water or steam, a heat rejection system, an electrical and mechanical interconnection between the cogenerator and the energy user, and a control system.

In a cogeneration system, the engine is usually used to drive an electric generator. The fuel is converted to electricity at en efficiently ranging from 25% to 30%. However, unlike the central power plant, a cogeneration plant should be located near a user of heat whose requirement will be satisfied by the heat rejected in the engine exhaust and cooling water. Cogeneration system design represents a balance between a number of technical and economic factors. Choice of the prime mover, availability of spare parts and the condition of existing utility and mechanical/electrical system, reliable service are the most important factors that are to be considered.

Below, the components for a cogeneration system are discussed in detail.

2.1.1. Prime Mover

The common feature in all cogeneration systems is the prime mover. It is the hearth of the cogeneration system which converts fuel into mechanical energy. The choice of the prime mover depends on the sites' heating and operating requirements, equipment availability and fuel availability.

There are four principal types of CHP scheme, according to the prime mover choice: steam turbine, gas turbine, combined cycle systems and reciprocating engines.

For the steam turbine type, steam at high pressure is generated in a boiler. In back pressure steam turbine systems, the steam are wholly or partly used in a turbine before being exhausted from the turbine at the required pressure for the site. In pass-out condensing steam turbine systems, a proportion of the steam used by the turbine is extracted at an intermediate pressure from the turbine with the remainder being fully condensed before it is exhausted at the exit. The boilers used in such schemes can burn a wide variety of fuels including coal, gas, oil, and waste-derived fuels. With the exception of waste-fired schemes, steam turbine plant has often been in service for several decades. Steam turbine schemes capable of supplying useful steam have electrical efficiencies of between 10 and 20 %, depending on size, and thus between 70 % and 30 % of the fuel input is available as useful heat. Steam turbines used in CHP applications typically range in size from a few MW_e to over 100 MW_e.

Gas turbine systems, are the ones where fuel (gas, or gas-oil) is combusted in the gas turbine and the exhaust gases are normally used in a waste heat boiler to produce usable steam, though the exhaust gases may be used directly in some process applications. Gas turbines range from 30kWe upwards, achieving electrical efficiency of 23 to 30 % (depending on size) and with the potential to recover up to 50 % of the fuel input as useful heat. They have been common in CHP since the mid 1980's. The waste heat boiler can include supplementary or auxiliary firing using a wide range of fuels, and thus the heat to power ratio of these schemes can vary.

For the combined cycle systems, the plant comprises more than one prime mover. These are usually gas turbines where the exhaust gases are utilized in a steam generator, the steam from which is passed wholly or in part into one or more steam turbines. In rare cases reciprocating engine may be linked with steam turbines. Combined cycle is suited to larger installations of 7 MW_e and over. They achieve higher electrical efficiency and a lower heat to power ratio than steam turbines or gas turbines. Recently installed combined cycle gas turbine (CCGT) schemes have achieved an electrical efficiency approaching 50 per cent, with 20 per cent heat recovery, and a heat to power ratio of less than 1:1.

Reciprocating engine systems range from less than 100 kW_e up to around 5 MW_e, and are found in applications where production of hot water (rather than steam) is the main requirement, for example, on smaller industrial sites as well as in buildings. They are based on auto engine or marine engine derivatives converted to run on gas. Both compression ignition and spark ignition firing is used. Reciprocating engines operate at around 28 to 40 % electrical efficiency with around 50 % to 33 % of the fuel input available as useful heat. Reciprocating engines produce two grades of waste heat: high-grade heat from the engine exhaust and low grade heat from the engine cooling circuits.

Lastly, an emerging technology that has cogeneration possibilities is the fuel cell. A fuel cell is a device that converts hydrogen to electricity without combustion. Heat is also produced. Most fuel cells use natural gas (composed mainly of methane) as the source of hydrogen. The first commercial availability of fuel cell technology is the phosphoric acid fuel cell, which has been on the market for a few years. There are about 50 installed and operating in the United States. Other fuel cell technologies (molten carbonate and solid oxide) are in early stages of development. Solid oxide fuel cells (SOFCs) may be potential source for cogeneration, due to the high temperature heat generated by their operation. [8]

Among these prime movers, the gas turbine engine will be considered to be the prime mover for this study. It consists of a compressor, thermal device that heats the working fluid, a turbine, a control system, and auxiliary equipment. Relevant information about the gas turbine is given in Chapters 1 and 2.

2.1.2. Heat Recovery Steam Generator

The heat recovery steam generator, or HRSG, takes the hot exhaust gases from the turbine and water from the waste water treatment plant to produce steam.

HRSG comes in numerous shapes, designs, configurations, arrangements, etc. In this study, to simplify the discussion; the type of HRSG to be examined is commonly what may be referred to as a water tube (as opposed to a fire tube) type heat recovery unit. This refers to the process fluid, i.e., the steam or water being on the inside of the tube with the products of combustion being on the outside of the tube. The products of combustion are normally at or close to atmospheric pressure, therefore, the shell side is generally not considered to be a pressure vessel. The two types of HRSG configurations can be seen in Figures 2.1 and 2.2.

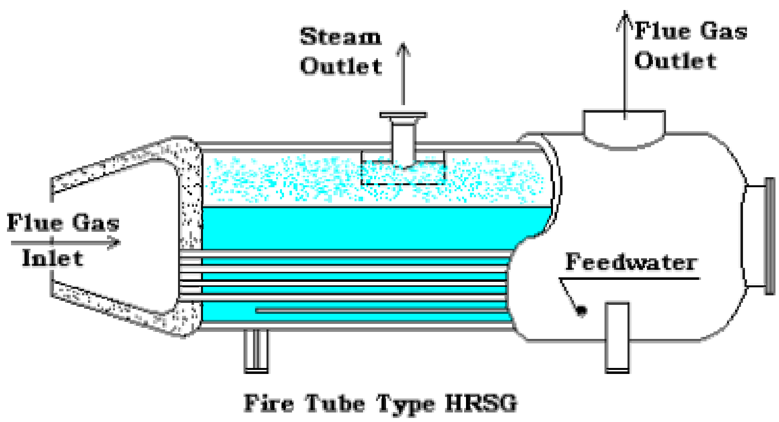


Figure 2.1 Fire Tube Type HRSG [9]

In the design of an HRSG, the first step normally is to perform a theoretical heat balance to give us the relationship between the tube side and shell side process. Before computing this heat balance, the tube side components which will make up the HRSG unit should be decided. The three primary coil types that may be present, are evaporator, superheater, economizer, which are discussed below. Also there may be some other extra firing units like preheaters, for increasing the total efficiency of the power plant.

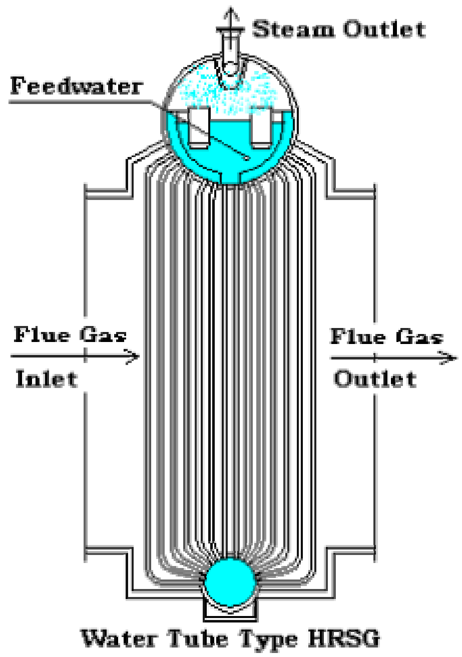


Figure 2.2 Water Tube Type HRSG [9]

2.1.2.1. Components of HRSG

2.1.2.1.1. Evaporator Section

The most important component of HRSG is the evaporator section. An evaporator section may consist of one or more coils. In these coils, the effluent (water), passing through the tubes is heated to the saturation point for the pressure it is flowing.

2.1.2.1.2. Superheater Section

The superheater section of the HRSG is used to dry the saturated vapor being separated in the steam drum. In some units it may only be heated to little above the saturation point where in other units it may be superheated to a significant temperature for additional energy storage. The superheater section is normally located in the hotter gas stream, in front of the evaporator.

2.1.2.1.3. Economizer Section

The economizer section, sometimes called a preheater or preheat coil, is used to preheat the feedwater being introduced to the system to replace the steam (vapor) being removed from the system via the superheater or steam outlet and the water loss through blow down. It’s temperatures are both close to the saturation temperature for the system pressure, the amount of heat that may be removed from the flue gas is limited due to the approach to the evaporator, known as the pinch which is discussed below, whereas the economizer inlet temperature is low, allowing the flue gas temperature to be taken lower.

2.1.2.1.4. Boiler Drums

The steam and water separation is achieved by means of a separator installed in the upper part of the drum. Other purposes of the drum are; insuring a good mixing of feed water and boiler water and constituting a water reserve required for the controlled circulation system.

2.1.2.2. Types and Configurations of HRSG According to Evaporator Layouts

The evaporator section type is very important since it generally defines the overall configuration of the HRSG unit. Five general types according to evaporator layout for HRSG are described below.

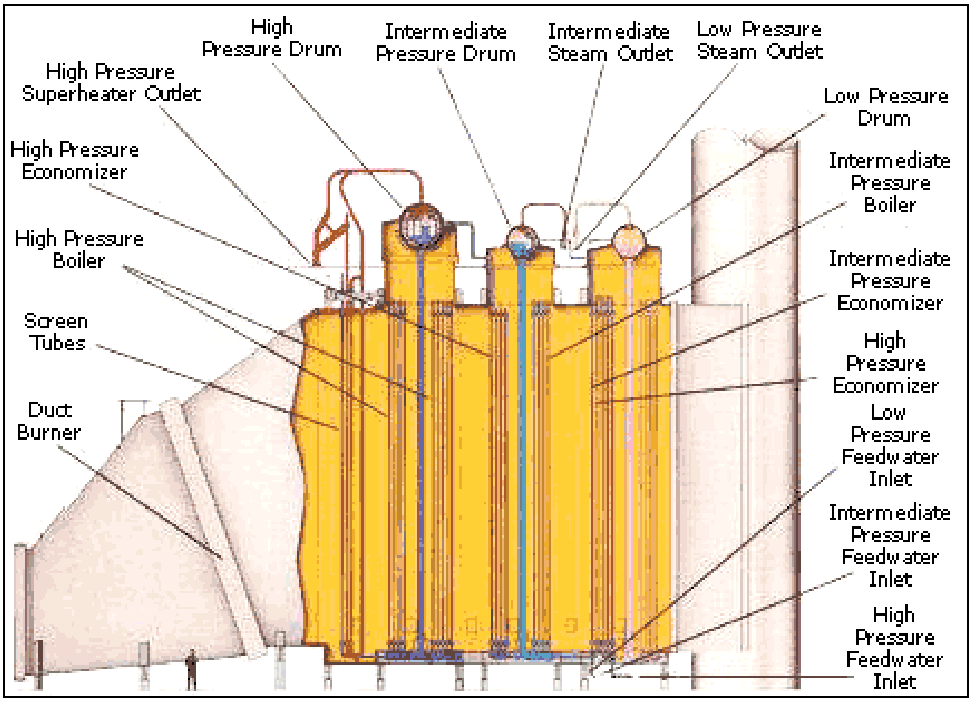


Figure 2.3 A general view of a HRSG [9]

2.1.2.2.1 D-Frame Evaporator Layout

This configuration is very popular for HRSG units recovering heat from small gas turbines and diesel engines. It is a very compact design and can be shipped totally assembled. It is limited, however, since the bent tube arrangement quickly causes the module to exceed shipping limitations for units having a large gas flow. A schematic of the D-frame evaporator layout can be seen in Appendix B, Figure B.1.

2.1.2.2.2. O-Frame Evaporator Layout

Being the most well known one, this configuration has been used for more years than any of the others. It has the advantage of the upper header being configured as the steam separation drum. Or, the upper header can be connected to the steam drum by risers, allowing more than one O-frame evaporator to be connected to the same steam drum, resulting in shippable modules being able to handle very large gas flows. A schematic of the O-frame evaporator layout can be seen in Appendix B, Figure B.2.

2.1.2.2.3. A-Frame Evaporator Layout

This configuration is simply a variation of the O-frame evaporator. It is popular for services with a large amount of ash, since the center area between the lower drums could be configured as a hopper to collect and remove solid particles. A schematic of the A-frame evaporator layout is given in Appendix B, Figure B.3.

2.1.2.2.4. I-Frame Evaporator Layout

This configuration is also popular among the Evaporator designs. This type module can be built in multiple axial modules or in multiple lateral modules, allowing it to be designed to accept any gas flow. There are numerous variations of this design where tube bundles may contain one, two, or three rows of tubes per header. It is also, normally, more economical to manufacture, ship and field construct. A schematic of the I-frame evaporator layout can be seen in Appendix B, Figure B.4.

2.1.2.2.5. Horizontal Tube Evaporator Layout

The horizontal tube evaporator is used, not only for heat recovery from gas turbine exhaust, but also for recovery from flue gases in refinery and petrochemical furnaces. It has similar size limitations due to shipping restrictions similar to the O-frame modules. It is generally a less expensive unit to manufacture than the other configurations. If it is a natural circulation design with large tubes, such as in some CO boilers, or very long tubes, special consideration needs to be given to assure all tubes are provided with sufficient effluent. A schematic of the horizontal tube evaporator layout is given in Appendix B, Figure B.5.

2.1.2.3. Types and Configurations of HRSG According to Superheater Layouts

Superheater designs would normally follow along with the evaporator type that is being used. Schematics of three basic superheater designs, namely horizontal tube, vertical tube, and I-frame type can be found in Appendix B, Figure B.6, B.7 and B.8 respectively. The horizontal tube design is normally used for the D-frame evaporator if gas flow is vertical up at the outlet. This horizontal design would be expected to be used also on a horizontal evaporator design. The vertical tube design is generally used with the A-frame or O-frame evaporator and with the D-frame if the gas exits horizontally. The I-frame superheater would be used with the I-frame evaporator, but may also be used with the other evaporator designs.

2.1.2.4. Types and Configurations of HRSG According to Economizer Layouts

Economizer designs normally follow along with the evaporator type that is being used and be similar in design to the superheater. The configurations would be similar to the ones shown in the Appendix B, for the superheaters.

2.1.2.5. Arrangement of Coils

The superheater, would be in the hottest part of the gas stream since this is where it would take the least amount of surface to exchange the heat, and would allow a stepped heat recovery for maximum heat exchange. The curve below in Figure 2.4, shows this relationship between the heat given up, and the three primary coils found in an HRSG.

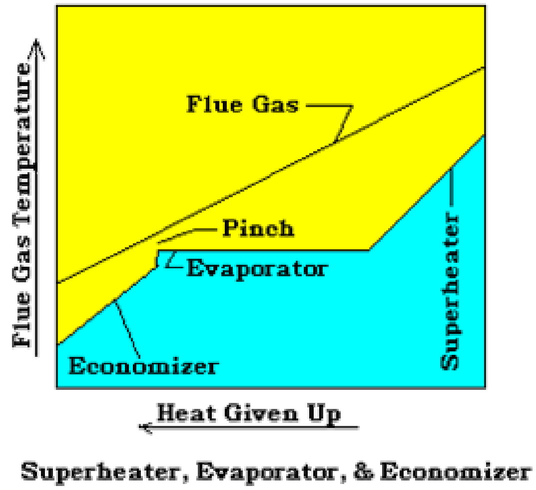


Figure 2.4 Relationship Between Heat Given Up and Three Primary Coils

In viewing this generalized sketch that shows the relationship between the heat absorbed and the heat given up, it is important to consider the area referred to as the "pinch" at the evaporator outlet. At a very high inlet temperature, there may be a critical approach temperature occur at the economizer inlet, and going the other way, at a lower inlet temperature, this may occur at the superheater outlet.

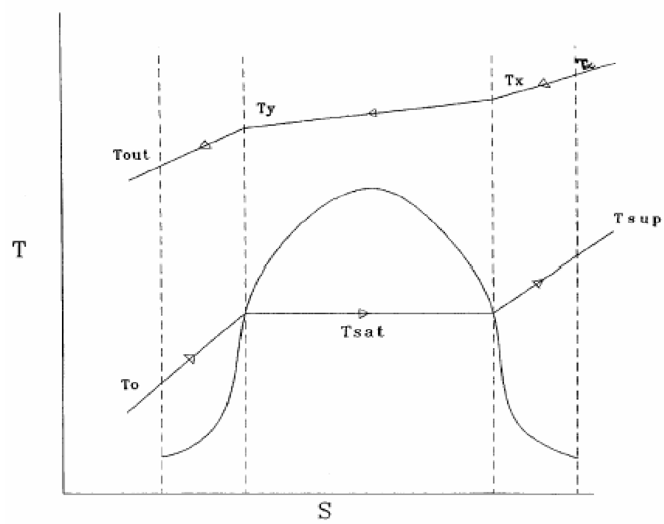


Figure 2.5 T S Diagram of Waste Heat Recovery Boiler [10]

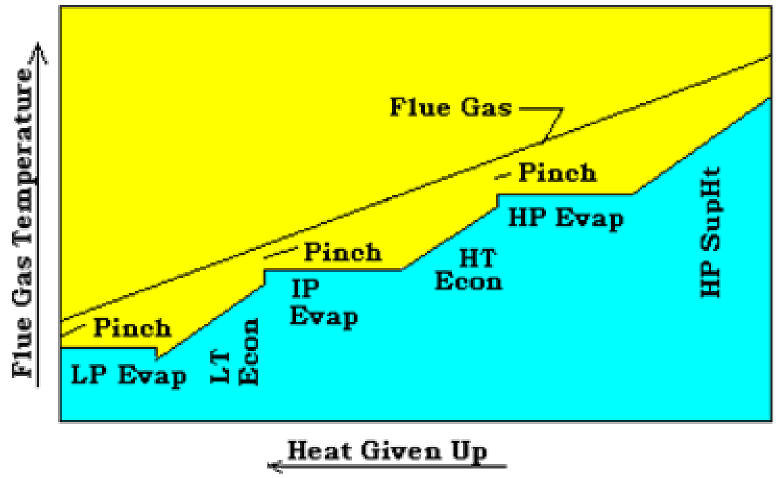


Figure 2.6 Relationship Between Heat Given Up and 3 Pressure Levels for Coils in HRSG

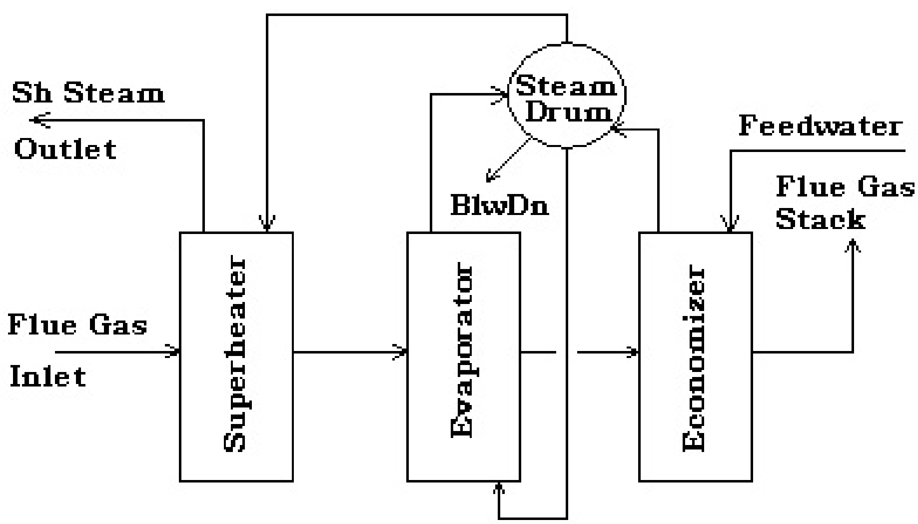


Figure 2.7 Single Pressure Flow Schematic for HRSG

Modern HRSG units are not always such simple. The components can and are placed in many configurations to achieve desired results. The range of arrangements that the coils may be placed, is only limited by the users necessities and the constraints of the temperature approaches.

2.1.2.6. Evaporator Pinch Design

Pinch point is the difference between the gas temperature leaving the evaporator and the saturation temperature, while approach point is the difference between the water temperature leaving the economizer and saturation temperature. The evaporator pinch, or approach temperature, is what limits the amount of heat that can be recovered in most HRSG designs.

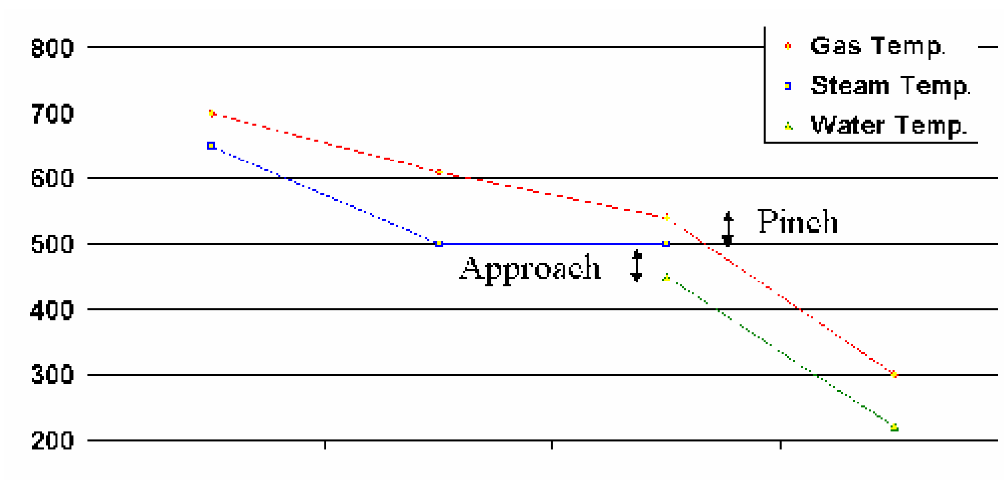


Figure 2.8 Approach and Pinch Point Illustrations

For many general purpose HRSG's such as those found in refineries and chemical plants, a pinch of 30 °C provides an economical design with a realistic payout. But in the more competitive markets of combined cycle or co-generation plants, it is common to see pinch points drop to 15 °C. And as a practice, a 25 °C pinch design for these HRSG's should be considered. [9]

2.1.3. Steam Turbine

Steam turbine is an excellent prime mover to convert heat energy of steam to mechanical energy. It is one of such well-known prime movers as gasoline engines, diesel engines, gas turbines, jet engines, etc.

All steam engines, whether turbines or not, are designed to extract energy from high pressure steam and convert it into motion by allowing the steam to expand. For the turbine designs, steam is allowed to expand gradually through more than one set of blades, for attaining much higher efficiencies compared to a single step expansion. The steam expands through successive rings of moving blades on a shaft and fixed blades in a casing, producing purely rotary movement.

When coupled to an electric generator, steam turbine is one of the most important means of producing bulk electric power in the world.

Though the steam turbine was later put to other uses, most notably in marine propulsion, its first purpose is to generate useful electrical power.

The modern steam turbine may have three stages. The high-pressure section has small blades. They are small because the incoming steam has very high energy at very high temperature (about 1200 K). After the steam passes through the high pressure section, it is sent back to the boiler to be reheated to 1,000 degrees. The steam is then sent to the next section of the turbine, called the intermediate pressure section. The blades here are larger than those in the high-pressure section.

After passing through this section, the steam is sent to the low-pressure section of the turbine. Because most of the energy has already been removed from the steam, the blades here are the largest in the turbine. The steam exits the turbine through the bottom, where it is condensed back into water. From there it is sent back to the boiler to be made into steam again.

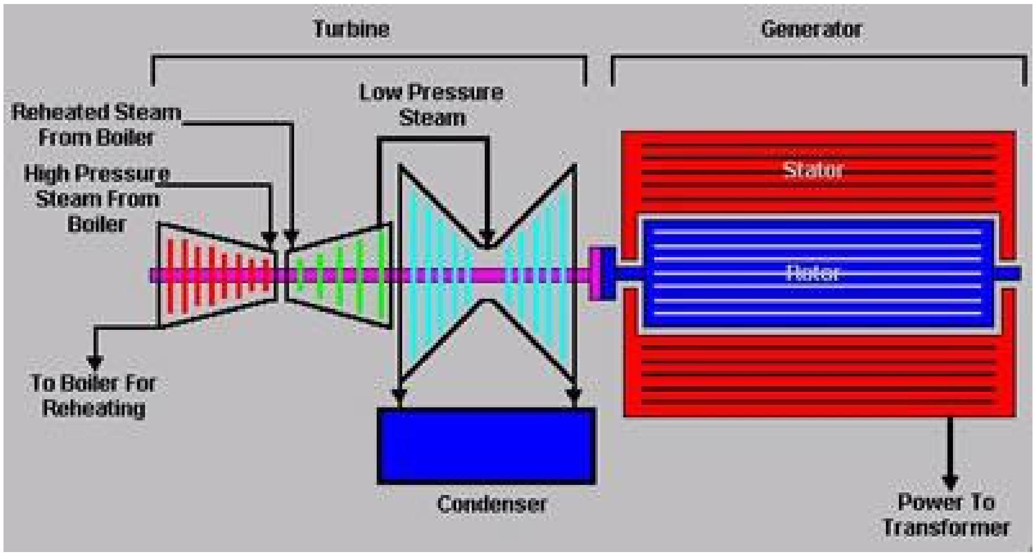


Figure 2.9 Steam Turbine Overview [11]

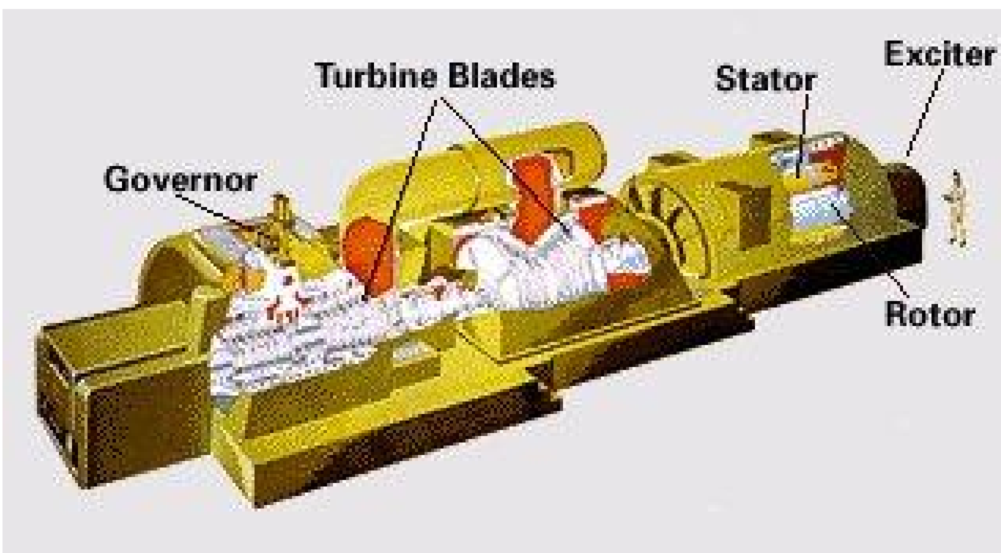


Figure 2.10 Steam Turbine Schematics

The steam turbine is often used in a combined heat and power generation process: the turbine drives a machine at the same time that, steam extracted from the machine is used to supply district heating and/or process steam networks.

In this way, the primary energy can be utilized optimally, which contributes to the conservation of natural resources and increases the economy of the system. [11]

General usages for the steam turbine are refineries, steel making and casting, metal working, paper manufacturing, cement production, food processing, wood processing, textile industry, cogeneration, sugar production, district heating.

2.1.4. Other Components of a Cogeneration System

Other Components of a Cogeneration System are; deareator which is used for providing the boiler with suitable water to produce steam free of impurities, demineralized water tanks for water storage, feed water pumps, process pumps, condenser which is used for collecting the returned condensed steam and heating the make-up water supply, electric generator, valves, oil lubrication system, blowers, raw water tanks and supplementary thermal devices.

2.2. Products of Cogeneration

A cogenerator, besides electrical energy, and heat, can provide compressed air for process use or cold air to be used for refrigeration processes. For refrigeration; recovered heat may be used in an absorption chiller, or electrical energy may be used to drive a compressor for compresion cooling.

Heat produced is usually used in a large-scale application of cogeneration, i.e: district heating. Many colleges and cities, which have extensive district heating and cooling systems, have cogeneration facilities. There are also home-sized cogeneration packages, which are capable of meeting most of the heating and electrical needs of a home.

Besides, pressurized steam may be used by textile, or paper production industries, for some special processes.

Below, heating and refrigeration processes of cogeneration and related products are clearly identified.

2.3. Cogeneration Processes

2.3.1. Heating Processes

Hot water or steam produced in the HRSG and/or steam turbine is used for any kind of heating processes, including district heating. The most common application for cogeneration is electric and heat production on site, since for other processes, extra equipment and further investment is required, as well, pay back period with a refrigeration system is longer when compared to heating systems.

For heating, water, or steam at any pressure can be taken at any stage of the steam turbine, or from the HRSG, directly given to radiation systems. The condensate, which returns from heating process, passes from the deaerator and is pumped back to steam generator pressure, with the make up water.

2.3.2. Refrigeration Processes

Refrigeration is the removal of heat from a substance or space so that temperature lower than that of the natural surroundings is achieved.

Refrigeration may be produced by:

- thermoelectric means
- vapour compression systems
- vapour absorption systems

- expansion of compressed gases
- throttling or unrestrained expansion of gases.

In this study, mainly vapour absorption and vapour compression systems are analysed, and water compression system is formulised, in Part 3.3.

2.3.2.1. Vapour-Compression Refrigeration Systems

Vapour compression systems are most commonly used mechanical refrigeration systems. Here, cooling is accomplished by evaporation of a liquid refrigerant under reduced pressure and temperature. There are four basic components: a compressor, a condenser (where we reject the heat), an expansion valve (throttling), and an evaporator (where we absorb the heat).

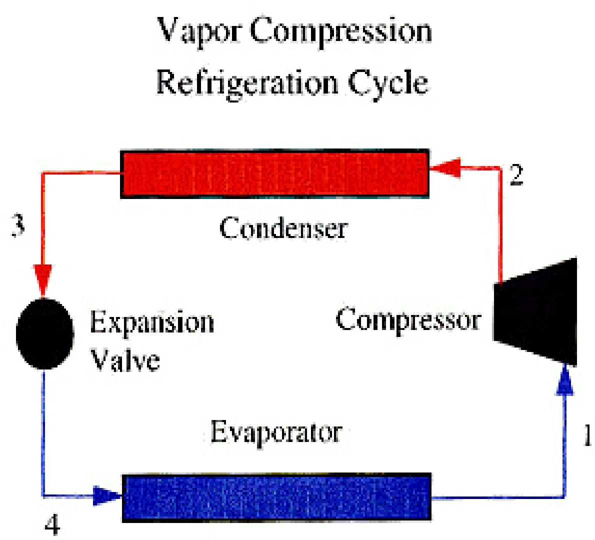


Figure 2.11 Vapour Compression Refrigeration Cycle

The vapour refrigerant enters the compressors at state 1 where the temperature and pressure are elevated by mechanical compression (state 2). The vapour condenses at this pressure, and the resultant heat is dissipated to the surrounding while it flows into the outdoor coil known as the condenser. The high pressure liquid (state 3) then passes through an expansion valve through which the fluid pressure is lowered and the refrigerant is cooled to the point where it returns to a liquid state. The low-pressure cool, liquid refrigerant fluid enters the evaporator at state 4 where it changes state from liquid to vapour by absorbing heat from the refrigerated space, and re-enters the compressor. The whole cycle is repeated.

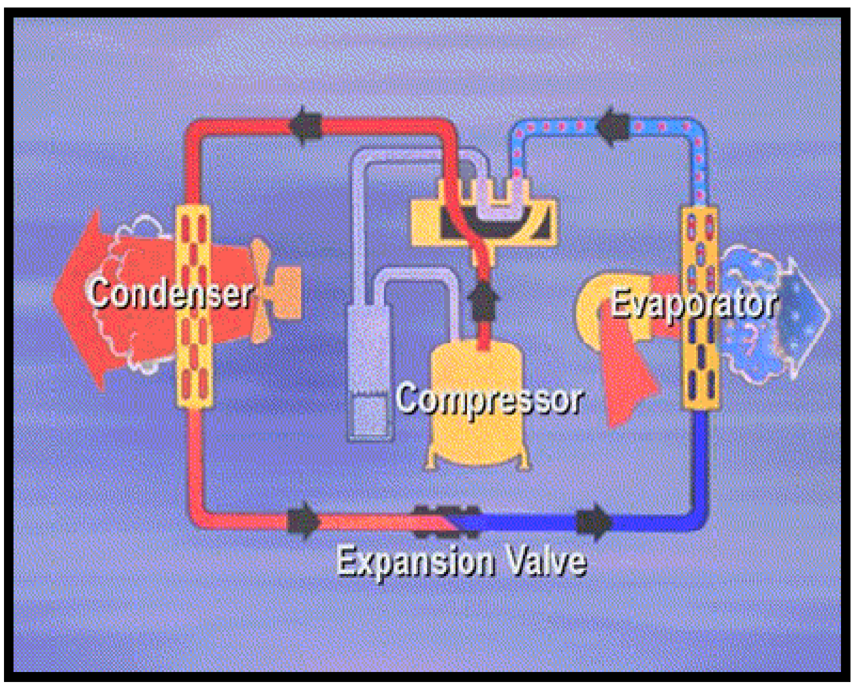


Figure 2.12 Vapour Compression Refrigeration Cycle

2.3.2.1.1. The Working Fluids

There are several working fluids available for use in refrigeration cycles. Four of the most common working fluids are available in R-12, R-22, R-134, and ammonia. (Nitrogen is also available for very low temperature refrigeration cycles.)

2.3.2.1.2. Basics of Vapor-Compression Refrigeration Cycles

Compression refrigeration cycles, in general, take advantage of the idea that highly compressed fluids at one temperature will tend to get colder when they are allowed to expand. If the pressure change is high enough, then the compressed gas will be hotter than source of cooling (outside air, for instance) and the expanded gas will be cooler than the desired cold temperature.

Vapour-compression refrigeration cycles specifically have two additional advantages. First, they exploit the large thermal energy required to change liquid to vapour so great amount of heat can be removed of air-conditioned space. Second, the isothermal nature of the vaporisation allows extraction of heat without raising the temperature of the working fluid to the temperature of what is being cooled. This is a benefit because the closer the working fluid temperature approaches that of the surroundings, the lower the rate of heat transfer. The isothermal process allows the fastest rate of heat transfer.

The cycle operates at two pressures, P_{high} and P_{low}, and the state points are determined by the cooling requirements and the properties of the working fluid. Most coolants are designed so that they have relatively high vapour pressures at typical application temperatures to avoid the need to maintain a significant vacuum in the refrigeration cycle. The T-s diagram for a vapour-compression refrigeration cycle is shown below, in Figure 2.13.

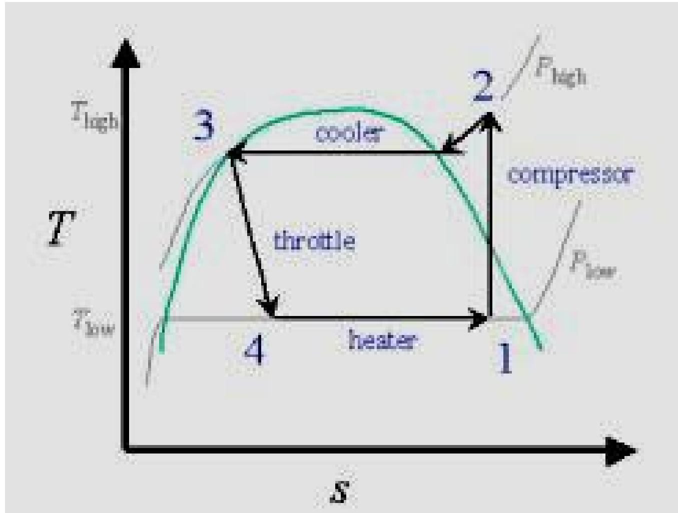


Figure 2.13 Vapour-Compression Refrigeration Cycle T-s Diagram

The cooler (also known as the condenser) rejects heat to the surroundings. Initially, the compressed gas (at S1) enters the condenser where it loses heat to the surroundings. During this constant-pressure process, the coolant goes from a gas to a saturated liquid- vapour mix, then continues condensing until it is a saturated liquid at state 2. Potentially, it can be cooled even further as a subcooled liquid, but there is little gain in doing so because already so much energy has been removed during the phase transition from vapour to liquid.

The working fluid absorbs heat from the surroundings. Since this process involves a change of phase from liquid to vapour, this device is often called the evaporator. This is where the useful "function" of the refrigeration cycle takes place, because it is during this part of the cycle that, heat is absorbed from the area trying to be cooled. For an efficient air conditioner, this quantity should be large compared to the power needed to run the cycle.

The usual design assumption for an ideal heater in a refrigeration cycle is that, it is isobaric (no pressure loss is incurred from forcing the coolant through the coils where heat transfer takes place). Since the heating process typically takes place entirely within the saturation region, the isobaric assumption also ensures that the process is isothermal. [12]

2.3.2.2. Vapour Absorption Refrigeration Systems

Another refrigeration application is vapor -absorption systems, where the compressor is replaced by an absorber-generator-pump assembly in which the refrigerant is absorbed into water as heat is removed. The liquid refrigerant-water solution is pumped and heated to drive off the refrigerant vapor and is then sent back into the refrigeration system. [13]

2.3.2.2.1. Schematics of Vapor Absorption Refrigeration Systems

Below, a vapour absorption refrigeration system, which uses water as the absorbent and ammonia as the refrigerant, is examined as to clearly illustrate the refrigeration cycle. [14]

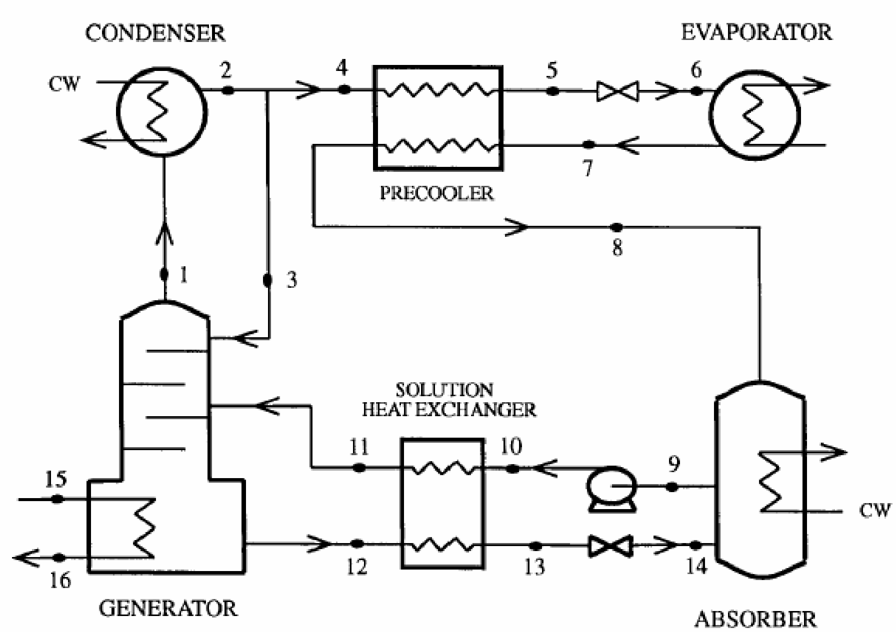


Figure 2.14 A single Stage Ammonia Absorption Chiller.

In the cycle strong solution refers to a solution with a high refrigerant content, while a weak solution (is) means a solution with a low content of refrigerant.

To reduce the water content in the refrigerant flow, a distillation column is typically used. The saturated liquid solution leaving the absorber (9), is pumped to the inlet of the distillation column, by the solution pump. In order to minimize the input (15) of high level energy as steam, the saturated weak solution leaving the generator (12) exchanges heat with the solution coming from the absorber (10).

The subcooled weak solution (13) exiting the exchanger is throttled to the absorber pressure (low pressure) and the two phase solution is brought into contact with the refrigerant vapor in the absorber (8). The heat generated in the absorption process (Qa) is rejected to cooling water, which will be later used also as a medium temperature sink in the condenser. The saturated liquid solution rich in refrigerant (9), the strong solution, leaves the absorber and starts again the solution circuit.

The refrigerant leaving the top of the distillation column (1) follows the same path as the refrigerant of a compression machine. It enters the condenser where the steam is condensed by rejecting heat (Qc) to a medium temperature sink (cooling water). To improve the performance of the system a condensate precooler is included. This heat exchanger subcools the saturated liquid refrigerant leaving the condenser (4) by pre-heating the evaporator outlet (7). Provided that stream (7) is usually kept at the required temperature if the enthalpy content of liquid refrigerant (5) is reduced by the precooler the performance of the evaporator is increased, because a higher mean temperature difference between the refrigerant and the chilled water is reached. This benefit overcomes the inconvenience of a higher rejected heat in the absorber, and the extra pressure drop caused by this heat exchanger in real machines. The evaporation of refrigerant takes place at low pressure using the heat released by the water to be chilled (Qev). The steam generated in this process (7) flows to the subcooler and finally to the absorber to dilute the weak solution.

It is assumed that, pressure changes are only significant in valves and pumps, and the heat loss to the surroundings is negligible. The states at the outlet of the throttle or expansion valves are calculated assuming isoenthalpic expansion and by applying the corresponding balances. Also at the outlet of the evaporator two phases are allowed. [14]

2.3.3. Trigeneration Systems (Both Heating and Refrigeration)

Trigeneration is more efficient and environmentally friendly than cogeneration. A well-designed trigeneration plant can achieve up to a 10% greater system efficiency than a cogeneration plant of similar size.

A trigeneration plant, defined in non-engineering terminology, is most often described as a cogeneration plant that has added absorption chillers - which takes the "waste heat" a cogeneration plant would have "wasted," and converts this "free energy", into useful energy in the form of chilled water.

The trigeneration energy process produces four different forms of energy from the primary energy source, namely, hot water, steam, cooling (chilled water) and power generation (electrical energy).

Trigeneration has also been referred to as CHCP (combined heating, cooling and power generation), this option allows having greater operational flexibility at sites with demand for energy in the form of heating as well as cooling. This is particularly relevant in tropical countries where buildings need to be air-conditioned and many industries require process cooling.

Since many industries and commercial buildings need combined power and heating and cooling, trigeneration plants have very high potentials for industrial and commercial applications - with the associated energy and economic savings inherent with trigeneration. [15]

2.3.3.1. Refrigeration Cycle Selection For a Trigeneration Plant

Cycle selection is not an easy task, as it is conditioned by many factors. Some factors are due to the type of cycle. For example, absorption chillers are usually driven by low cost and low temperature waste heat. Compression chillers on the other hand are the most efficient, and also have the lowest capital cost due to the fact that a lower number of pieces of equipment are required, but nevertheless high quality primary energy will be consumed, with the potential increase in operation costs.

Cycle selection is also conditioned by a very high number of factors external to the cycle itself. These factors may include availability of electricity, coming from the general grid or locally generated, steam raised in a conventional or cogeneration plant, the availability of waste heat, and also factors involving the renewal or enlargement of existing equipment units. Therefore, due to the variety and great number of parameters involved in many cases in the integration of absorption cycles it is very difficult to generalize. Thus, to select the most suitable refrigeration cycle for a given refrigeration load, it is necessary to model the performance of each cycle, and to take into account the interactions between the energy system and the considered cycles, optimizing the performance of the global plant. [13]

In the present study, the most important factor considered for selecting either the vapor compression or vapor absorption cycle was according to the excess product of the plant. If there is much of excess steam on site, absorption cooling would be the one to choose; but if there is always excess electricity produced on site, it would be feasible to use the compression cooling for refrigeration. Below, comparison of the two different refrigeration systems is done in Table 2.1.

Table 2.1 Comparison Between Compression and Absorption Chillers [16]

Vapour Compression Characteristics	Vapour Absorption Characteristics
Efficient operation	Poor efficiency
Consume Electrical Energy	Consume Steam Energy
Consume electrical energy	Consume steam energy
Typically noisy	Quiet operation
AC operation	AC/DC power
Higher operation cost	Low operation cost
Low capital cost	High capital cost

CHAPTER 3

THEORY AND GOVERNING EQUATIONS

3.1. Important Parameters in Power Production and Cogeneration

CHP involves two (essentially equivalent) products (electricity and heat), which are generated simultaneously from one and the same high-temperature process medium (generally steam or flue gas), the exergy of which is used primarily to generate power. The remaining exergy and latent condensation heat of the waste steam or residual energy of the flue gas is used as heat.

As a physical process, CHP may operate with any fuel, as the fuel merely generates the high-temperature heat on which the combined process is based; in other words, CHP is fuel-neutral.

Use of the heat requires the existence of corresponding heat demand or heat sink. Transfer of all the heat to this heat sink (usually district heating pipeline water) requires a propelling gradient, i.e. the temperature difference of a heat exchanger. The temperature of the available heat sink in conjunction with this temperature difference, acting as a minimum process temperature, thus determines the achievable degree of utilization of the heat on offer.

On the other hand, the maximum process temperature is crucial in determining the thermodynamic quality or efficiency in the generation of the product - electricity: the degree of efficiency or utilization ratio of the power generated or the electricity yield of a CHP process.

The sum of the products electricity and heat generated in relation to the quantity of fuel used, provides the total utilization ratio of the fuel. This utilization ratio is an important quality criterion of CHP.

The achievable utilization ratio for the use of solid fuels is 80 to 85% lower than for gaseous or liquid fuels.

The ratio of the products electricity and heat generated in the CHP process is known as the power-to-heat ratio**.** It constitutes a further quality criterion of CHP, in addition to the utilization ratio. The power-to-heat ratio increases with the degree of efficiency of the electricity generated by the technology used and decreases as the temperature of the heat product required rises.

These two CHP quality criteria the power-to-heat-ratio and the utilization ratio may differ, depending on the technology used, specific properties of the fuel and the thermodynamic value of the heat product.

Typical ranges of power-to-heat ratios for different applications are as follows [17]:

- Waste incineration: 0.2 – 0.3
- Backpressure, extraction backpressure, extraction-condensing, uncontrolled extraction condensing (industry): 0.3 – 0.5
- Backpressure, extraction backpressure, extraction-condensing, uncontrolled extraction condensing (district heating): 0.4 – 0.6
- Gas turbine with waste heat boiler: 0.4 – 0.7
- Block heat and power plant: 0.5 – 0.9
- Gas and steam: 0.7-1.2

CHP is simply a very efficient process for meeting existing parallel demands for electricity and heat. Efficient in the sense of "energy-saving".

For increasing gas turbine and therefore CHP efficiency, some modifications on the system and cycle may be implemented. One of them is increasing the turbine inlet temperature (TIT) up to the metallurgical limit set by the material of the turbine blades and last stage turbine stress level. This TIT increase has been achieved by the development of better materials including ceramics or thermal barrier coatings (TBC), and by blade cooling techniques frequently based on bleed air or steam flowing through complex internal passages (for small turbines of say less than 100 kW, the turbine blade geometry makes cooling very difficult, and for these units it will be necessary to use ceramic components). For large utility-size machines with the next generation of engines, TIT's will increase above 1500 °C.

Another modification of the gas turbine cycle is to recover the exhaust energy partially in a heat exchanger of a recuperative cycle. A recuperator is a heat exchanger located in a gas turbine exhaust. It enables waste heat to be transferred from the exhaust to the combustor inlet air, hence partially replacing fuel. It will reduce specific fuel consumption compared to a conventional gas turbine cycle, while ensuring exhaust temperature is still suitable for CHP. Heat recovery schemes (recuperators or regenerators) are the most important ways of increasing the efficiency of the power generation process by more than 40%; they also result in lower levels of pollution for a given output of electricity.

Humid air turbine cycle may be used for a more efficient cycle. The main innovation of the humid air turbine (HAT) cycle is that steam is produced along the airflow, thus eliminates the heat recovery boiler. It consists of an inter-cooled gas turbine cycle having an air+water mixing evaporator before the combustion chamber and an exhaust gas recovery system. The efficiency and power output are increased while the NO_x is reduced. The system has two cooling stages after the compression stages, the mixing evaporator, the surface recuperator between the mixture and the exhaust gases, and the economiser before the gas discharge [18]

The performance of turbines is adversely affected by high ambient temperatures. Several means of reducing the turbine inlet temperature (off-peak water chiller and ice storage and absorption refrigeration systems) have been proposed as a means of increasing turbine output. The energy in the turbine exhaust has the potential of producing additional cooling beyond that required to reduce the inlet temperature. The excess cooling available from the system could be used to provide chilled water for air-conditioning adjacent buildings or for industrial processes. [19], [20]

A regenerative cycle may also be used for improving gas turbine efficiency. A regenerator is installed after the compressor, recovering some of the energy in the exhaust gases, adding to compressed air before entering the combustor, increasing the heat input of the cycle, and thus the recovered heat, and the steam energy [21]

3.2. Thermodynamic Analysis of Combined Cogeneration Cycle

Formulations for combined/ cogeneration cycle are based on first and second laws of thermodynamics. The constant specific heats of air and air+gas mixture are assumed to be constant, with the values respectively; c_{pa}=1005 kj/kgK and c_{pg}=1148 kj/kgK. Also isentropic efficiencies of the gas turbine and the compressor are assumed to be constant in all cases. Below, detailed formulations are given for all cases examined in the current study.

3.2.1. Analysis of a Simple Gas Turbine:

For equations from 3.1 to 3.9, refer to figure 1.2 on page 4.

Compressor pressure and temperature ratios:

$$r_p = P_2 / P_1 = P_3 / P_4 \tag{3.1}$$

$$T_2 / T_1 = r^{(g-1)/g} = T_3 / T_4 \tag{3.2}$$

Specific work output for the gas turbine:

$$W_{spec} = c_{pg} \cdot (T_3 - T_4) - c_{pa} \cdot (T_2 - T_1) \tag{3.3}$$

Compressor outlet temperature:

$$T_2 = T_1 + \frac{T_1}{h_c} \cdot \left[r_p^{\frac{g-1}{g}} - 1 \right] \tag{3.4}$$

Turbine inlet temperature:

$$T_3 = T_4 \cdot \frac{1}{1 - h_t \cdot \left[1 - \left(1 / r_p \right)^{\frac{g-1}{g}} \right]} \tag{3.5}$$

Combustion chamber calculations:

Mass flow rate of air+gas mixture (assumed to be constant, i.e. no bleed):

$$\dot{m}_{tot} = \frac{P}{W_{spec} \cdot h_m} \tag{3.6}$$

$$\dot{m}_{tot} = \dot{m}_a + \dot{m}_{fuel} = \dot{m}_a \cdot (1 + f_a)$$

$$\text{where } f_a = \frac{\dot{m}_{fuel}}{\dot{m}_a} \tag{3.7}$$

Specific fuel consumption calculation:

$$SFC = \frac{f_a * 3600}{W_{spec}} \tag{3.8}$$

Power supplied by fuel in the combustion chamber:

$$E_f = h_{cc} \cdot m_{tot} \cdot c_{pg} \cdot (T_3 - T_2) \tag{3.9}$$

3.2.2. Analysis of Combined/ Cogeneration Cycle Gas Turbine (Without Steam Turbine)

The illustration of the cycle can be seen in Appendix C, Figure C.1.

Heat released from combustion gases in HRSG:

$$\dot{E}c = C_{pg} \cdot (T_4 - T_e)$$
$$Ec = \dot{m}_{tot} \cdot C_{pg} \cdot (T_4 - T_e) \tag{3.10}$$

Pinch point design formulation:

$$\Delta T_{pinch} = T_{evap} - T_{sat\,@\,p} \tag{3.11}$$

HRSG heat equations for steam side:

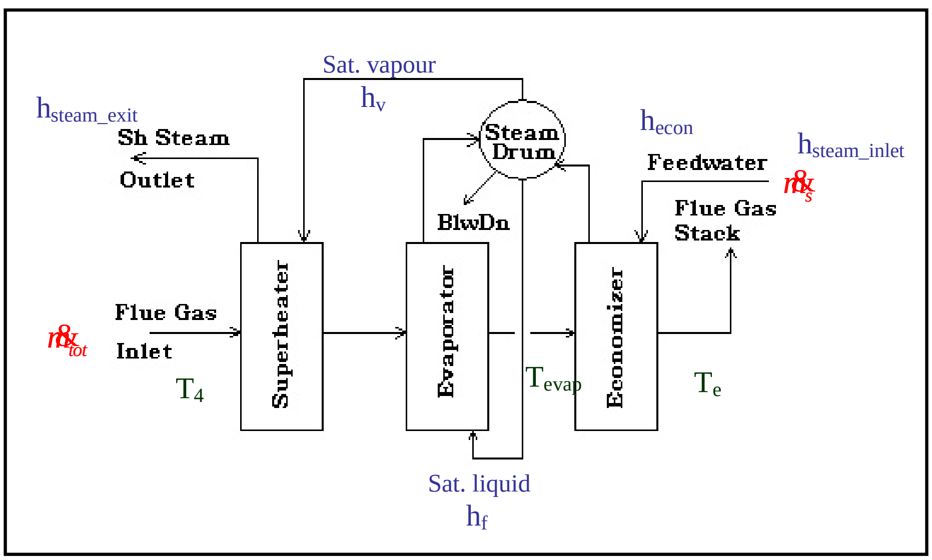


Figure 3.1 Schematics for HRSG Heat Equations For No Steam Turbine Case

Superheater heat balance:

$$Q_{SH} = \dot{m}_s \cdot (h_{steam\_exit} - h_v) \tag{3.12}$$

Evaporator heat balance:

$$Q_{evap} = \dot{m}_s \cdot (h_v - h_f) \tag{3.13}$$

Economizer heat balance:

$$Q_{econ} = \dot{m}_s \cdot (h_{econ} - h_{steam\_inlet}) \tag{3.14}$$

HRSG heat equations for air side:

Superheater+evaporator heat balance:

$$Q_{SH+evap} = \dot{m}_{tot} \cdot c_{pg} \cdot (T_4 - T_{evap}) \tag{3.15}$$

Economizer heat balance:

$$Q_{econ} = \dot{m}_{tot} \cdot c_{pg} \cdot (T_{evap} - T_e) \tag{3.16}$$

HRSG heat balance at superheater+evaporator part:

$$\dot{m}_s \cdot (h_{steam\_exit} - h_f) = \dot{m}_{tot} \cdot c_{pg} \cdot (T_4 - T_{evap}) \tag{3.17}$$

HRSG heat balance at economizer part:

$$\dot{m}_s \cdot (h_{sat} - h_{steam\_inlet}) = \dot{m}_{tot} \cdot (T_{evap} - T_e) \cdot c_{pg} \tag{3.18}$$

Heat recovered by steam flow:

$$Q_{sh} = \dot{m}_s \cdot (h_{steam\_exit} - h_f) \tag{3.19}$$

Power to heat ratio:

$$r_{ph} = P_e / Q_P \quad \text{Where} \quad P_e = W_{spec} \cdot \dot{m}_{tot} \tag{3.20}$$

3.2.3. Analysis of Combined/ Cogeneration Cycle Gas Turbine (With Steam Turbine)

3.2.3.1. Non Condensing Steam Turbine:

The illustration of the cycle can be seen in Appendix C, Figure C.2.

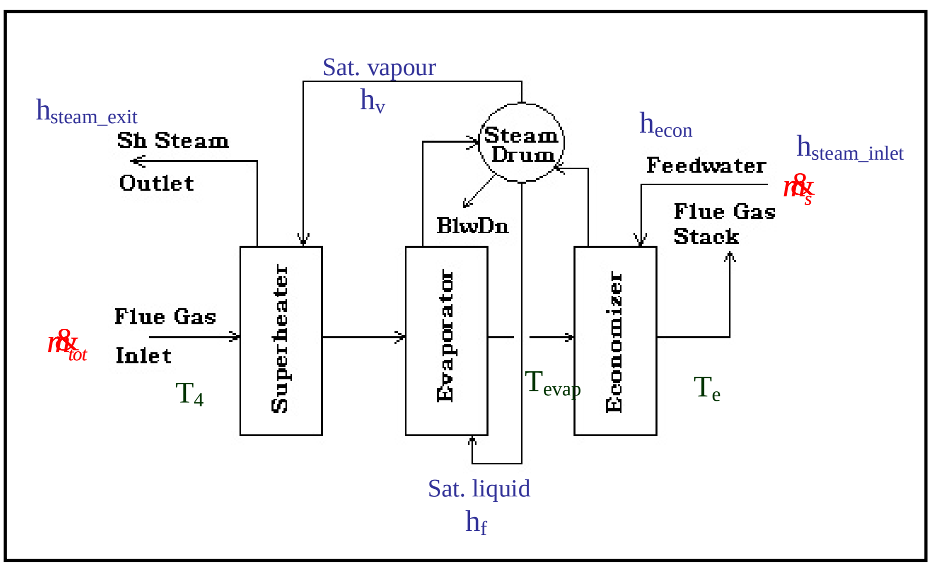


Figure 3.2 Schematics for HRSG Heat Equations For Steam Turbine Case

Power to heat Ratio:

$$r_{ph} = W_e / \dot{m}_s \cdot (h_{steam\_exit} - h_{steam\_inlet})$$

Where $P_e = W_{spec} \cdot \dot{m}_{tot}$ (3.21)

Heat released from combustion gases in HRSG:

$$\dot{E}c = C_{pg} \cdot (T_4 - T_e)$$
$$Ec = \dot{m}_{tot} \cdot C_{pg} \cdot (T_4 - T_e) \qquad (3.22)$$

Pinch point design formulation:

$$\Delta T_{pinch} = T_{evap} - T_{sat\,@\,p} \qquad (3.23)$$

HRSG heat equations for steam side:

Superheater heat balance:

$$Q_{SH} = \dot{m}_s \cdot (h_{steam\_exit} - h_v) \qquad (3.24)$$

Evaporator heat balance:

$$Q_{evap} = \dot{m}_s \cdot (h_v - h_f) \qquad (3.25)$$

Economizer heat balance:

$$Q_{econ} = \dot{m}_s \cdot (h_{econ} - h_{steam\_inlet}) \qquad (3.26)$$

HRSG heat equations for air side:

Superheater+evaporator heat balance:

$$Q_{SH+evap} = \dot{m}_{tot} \cdot c_{pg} \cdot (T_4 - T_{evap}) \tag{3.27}$$

Economizer heat balance:

$$Q_{econ} = \dot{m}_s \cdot c_{pg} \cdot (T_{evap} - T_e) \tag{3.28}$$

Total electrical power output of GT and ST:

$$W_e = \dot{m}_{tot} \cdot W_{Spec} \cdot \eta_{elect} + \dot{m}_s \cdot W_{Steam} \cdot \eta_{elect\_Steam} - W_{pump} \tag{3.29}$$

Pump work:

$$W_{pump} = m_{water} \cdot v_f \cdot (P_{\max} - P_{\min}) \tag{3.30}$$

HRSG heat balance:

$$\dot{m}_{tot} \cdot c_{pg} \cdot (T_4 - T_e) = \dot{m}_s \cdot (h_{steam\_exit} - h_{steam\_inlet}) \tag{3.31}$$

Steam turbine efficiency:

$$\eta_{ST} = \frac{W_{Steam}}{h_{in} - h_{out\_ideal}} = \frac{(h_A - h_{B\_actual})}{(h_A - h_{B\_ideal})}, \quad s_A = s_{B\_ideal} \text{ for steam turbine work} \tag{3.32}$$

3.2.3.2. Condensing Steam Turbine

The illustration of the cycle can be seen in Appendix C, Figure C.3.

Power to heat ratio:

$$r_{ph} = W_e / \dot{m}_s \cdot (h_{steam\_exit} - h_{steam\_inlet})$$

Where $P_e = W_{spec} \cdot \dot{m}_{tot}$ (3.33)

Heat released from combustion gases in HRSG:

$$\dot{E}c = C_{pg} \cdot (T_4 - T_e)$$
$$Ec = \dot{m}_{tot} \cdot C_{pg} \cdot (T_4 - T_e) \tag{3.34}$$

Pinch point design formulation:

$$\Delta T_{pinch} = T_{evap} - T_{sat\,@\,p} \tag{3.35}$$

HRSG heat equations for steam side:

Superheater:

$$Q_{SH} = \dot{m}_s \cdot (h_{steam\_exit} - h_v) \tag{3.36}$$

Evaporator:

$$Q_{evap} = \dot{m}_s \cdot (h_v - h_f) \tag{3.37}$$

Economizer:

$$Q_{econ} = \dot{m}_s \cdot (h_{econ} - h_{steam\_inlet}) \tag{3.38}$$

HRSG Heat equations for air side:

Superheater+evaporator:

$$Q_{SH+evap} = \dot{m}_{tot} \cdot c_{pg} \cdot (T_4 - T_{evap}) \tag{3.39}$$

Economizer heat balance:

$$Q_{econ} = \dot{m}_s \cdot c_{pg} \cdot (T_{evap} - T_e) \tag{3.40}$$

Total electrical work output of GT and ST:

$$W_e = \dot{m}_{tot} \cdot W_{Spec} \cdot \eta_{elect} + \dot{m}_s \cdot W_{Steam} \cdot \eta_{elect\_Steam} - W_{pump} \tag{3.41}$$

Pump work:

$$W_{pump} = m_{water} \cdot v_f \cdot \left(P_{\max} - P_{\min}\right) \tag{3.42}$$

HRSG heat balance:

$$\dot{m}_{tot} \cdot c_{pg} \cdot (T_4 - T_e) = \dot{m}_s \cdot (h_{steam\_exit} - h_{steam\_inlet}) \tag{3.43}$$

Steam turbine efficiency:

$$\eta_{ST} = \frac{W_{Steam}}{h_{in} - h_{out\_ideal}} = \frac{(h_A - h_{B\_actual})}{(h_A - h_{B\_ideal})}\ , \quad s_A = s_{B\_ideal} \ \text{ for steam turbine work} \tag{3.44}$$

Condensator heat exchange:

$$Q_{cond} = \dot{m}_{water} \cdot (h_{B\_actual} - h_{Cond\_exit}) \tag{3.45}$$

3.3. Analysis of a Triple Cycle (Cogeneration With Refrigeration)

Vapor Compression Refrigeration Cycle:

The equations from 3.46 to 3.50 refer to Figure 2.11 on page 39 and Figure 2.13 on page 41.

Rate of heat removal

$$Q_L = \dot{m}_{ref} \cdot (h_1 - h_4) \tag{3.46}$$

Power input to the cycle (isentropic compression work):

$$W_{in} = \dot{m}_{ref} \cdot (h_2 - h_1) \tag{3.47}$$

Rate of heat disipation:

$$Q_H = \dot{m}_{ref} \cdot (h_2 - h_3) = Q_L + W_{in} \tag{3.48}$$

Isenthalpic expansion:

$$h_3 = h_4 \tag{3.49}$$

Coefficient of performance of the cycle:

$$COP = \frac{Q_L}{W_{in}} \tag{3.50}$$

Trigeneration electric equivalent efficiency:

$$h_{el\_equiv} = \frac{(W_{spec} + W_{in})}{\dot{E}_c} \tag{3.51}$$

3.4. Exergy Analysis

Thermodynamic formulation for the presented cycles are given above. To have a better insight into the thermodynamic performance, before considering economics, exergy and exergy analysis should be defined.

Second Law of Thermodynamics tells that the quality of energy is degraded every time energy is used in any process. In any process, loss can be defined in terms of entropy generation or exergy destruction. Entropy is the quality to measure the spontaneous dispersal of energy: how much energy is spread out in a process, or how widely spread out it becomes – as a function of temperature. The "energy quality" can be named "Exergy". Exergy analysis which may be called availability analysis, consists of using first and second laws together.

The amount of energy in the universe remains constant (First Law), but exergy is constantly used up (Second Law). In the end (very long time from now), all the available exergy is used up in the universe, and no processes can run.

The thermodynamic performance based on the first law efficiency is defined as fuel utilization efficiency:

$h = \frac{W_e + Q_p}{E_f}$ W_e is the electrical energy , Q_p is thermal energy E_f is Energy of a hydrocarbon fuel. (3.52)

$E_f = \sum_p n_e \cdot h_e - \sum_r n_i \cdot h_i$ h_i and h_e are enthalpies of reactants (shown with r) and products (shown with p) (3.53)

The exergy efficiency or fuel utilization efficiency may be calculated in the same way as the energy efficiency for a gas+steam cycle plant:

$$h = \frac{W_e + B_p}{B_f} \tag{3.54}$$

where W_e is work, hence considered all exergy, B_p is the exergy content of process heat produced and B_f is the exergy content of fuel input In gas or gas steam turbine plants, the fuel is very often natural gas, and in this case, its lower heating value is a little higher than its exergy

The exergy content of the process heat produced is evaluated as:

$$B_p = m_s \cdot \left[(h_s - h_c) - T_0 \cdot (s_s - s_c) \right] \tag{3.55}$$

where s is the entropy of the produced steam, h_c is the enthalpy and s_c is the entropy of condensate return. The first part represents the energy of the process heat.

The exergy content of hydrocarbon fuel is B_f. g_i, g_e are the Gibbs functions of reactants and products:

$$B_f = \left(\sum_r n_i \cdot g_i - \sum_p n_e \cdot g_e \right) + R \cdot T_0 \cdot \ln\left(\frac{y^{y1}{}_{O2}}{y^{x1}{}_{CO2}\, y^{x2}{}_{H_2O}} \right) \tag{3.56}$$

To have a better assessment, a useful ratio; process heat exergy factor should be defined. It is expressed as the exergy energy ratio of the process heat flow:

$e_P = \frac{B_p}{Q_P} = \frac{T_f - T_0}{T_f}$ where T_f is effective temperature of the combustion chamber (maximum temperature of the cycle), T_0 is the ambient temperature. (3.57)

[22, 23]

CHAPTER 4

THERMOECONOMIC OPTIMIZATION AND FEASIBILITY ANALYSIS

In the analysis of cogeneration systems, it is important to consider the first and second laws of thermodynamics, together with an engineering methodology, cost evaluation and economics.

4.1. Cogeneration Economics, Financing and Investment for Power Plants

4.1.1. Economics of Cogeneration

Cogeneration is the sequential production of two or more forms of useful energy from a single heat source. Waste is recovered and converted into hot water or steam to meet building or process heating or cooling requirements. The high efficiencies of these components, combined with the use of a low cost fuel, result in significant energy cost savings. Cogeneration sites stabilize energy costs by producing most of the sites electricity, thereby shielding users from the potential volatility of the electricity market.

Many factors can influence the cost effectiveness of cogeneration. For instance, efficient operation relies on gas production levels and wise use of any available gas storage capacity. If on-site energy needs decrease, the economics of cogeneration change as well. Cogeneration plants may find themselves selling excess electricity to the local utility for less than it costs to produce, or venting unneeded thermal energy. If on-site energy needs rise, managers should examine the benefits of increasing capacity or changing the mix of cogeneration and purchased power.

Manufacturers constantly improve combustion technologies and auxiliary cogeneration equipment. Lean-burn technology, for example, allows some facilities to increase generation capacity while staying within legal emission limits. These advances can change the economics of cogeneration. By checking with vendors periodically, managers can keep informed about technological improvements and new implementation strategies that will help them manage their facilities more effectively

Electric utility deregulation has significantly changed the electricity market, allowing for new contractual arrangements and pricing structures. These will affect cogeneration strategies, since cogeneration plants may be able to buy less expensive electricity, sell excess power to other consumers, or power to outlying facilities. By carefully researching recent changes, plant managers can re-evaluate their role as energy users and producers and better understand the regulatory and economic impacts of the evolving market. [24,25]

4.1.2. Financing and Investment for Power Plants

Many factors influence the system design selection process. One of the most important considerations is the power-to-heat ratio (r_{ph}). Each system type presents slightly different thermodynamic considerations, which in turn affects how economically the system can meet a given thermal/electric load. According to the r_{ph}, system configuration is determined as shown in Table 4.1 and with increasing r_{ph}, complexity of the system, thus the installation and construction costs increase, which can be seen in Table 4.2. A facility's power-to-heat ratio can easily be calculated. Corresponding calculations are given in Part 3.2.2, equation 3.20.

Table 4.1 Power-to-Heat Ratio (r_{ph}) According to Systems [26]

Power-to-heat ratio (rph)	**System to Consider**
0.5-1.5	Diesel Engines
1-10	Gas Turbines
3-20	Steam Turbines

Generalizing potential system installation costs can be difficult due to the many design and site conditions that tend to be unique to each installation. However, some general pricing guidelines can be used for comparison. In general, system size is the biggest pricing issue. For example, a 1-MW gas turbine generator/HRSG system could be installed for $1500 per kW where a larger system of 5 MW could cost as little as $600 per kW. These figures reflect relatively simple installation conditions and costs for system engineering and design. Providing a building to house the CHP system, routing electrical conductors a large distance from the new generator to the existing utility point of entry and upsizing on-site gas distribution piping to accommodate increased gas consumption needs, increase the complexity and cost of the system. In Appendix D.1, industry pricing factors for simple cycle and combined cycle power plants can be found.

Table 4.2 System Installation Costs [26]

System Type	**Installed Cost($/kW)**
GT, Generators, HRSG	500-700
ST, Generators	600-1500
Reciprocating Engines, Generators, HRSG	800-2000

In Appendix D.2, industry price levels for simple cycle and combined cycle power plants can be found.

CHP system costs include a considerable maintenance component. Maintenance liabilities vary widely by system type. Maintenance activities include routine preventive maintenance (lubrication, filters, coolant, etc.), bearing maintenance, and periodic overhauls. Most manufacturers offer comprehensive maintenance service agreements at a specified cost per kilowatt-hour generated, which can be seen in Table 4.3.

Table 4.3 Maintenance Costs [26]

System Type	**Expected Maintenance Cost (cent/kWh)**
Steam Turbines	0.1-0.25
Gas Turbines	0.25-0.60
Reciprocating Engines(120-900 rpm)	0.7-1.0
Reciprocating Engines(900-1200 rpm)	1.0-1.20
Reciprocating Engines(1200-1800 rpm)	1.20-1.50

In some cases, when a poor "natural" thermal/electric load match exists (usually due to low thermal (heating) loads during summer months in northern climates), an "artificial" steam load can be created by installing absorption chillers, steam turbine driving centrifugal chillers, or electric motors. In addition to improving the thermal/electrical load (and the system thermal efficiency), the electric summer peak load can be reduced as well as the associated installed power capacity requirement and initial cost. [26, 27]

4.1.3. Cost Summary For Power Plants

Below, a detailed cost analysis is given for a cogeneration power plant, in two parts; investment and annual operational cost. [28]

1 Investment:

1.1 Specialized Equipment:

GT package, ST package, HRSG, Condenser

1.2 Other Equipment:

Pumps, Cooling Tower, Auxilary Heat Exchangers

1.3 Civil costs:

Site work, Excavation and Backfill, Concrete, Roads, parkings, walkways

1.4 Mechanical costs:

Equipment erection and assembly, Piping systems, mechanical installation

1.5 Electrical cost:

Wiring, ductwork, Cable Trays, including all installation, switchgear

1.6 Buildings and Structure

1.7 Engineering and Plant Start up

1.9 Contructor's price, project design

1.9 Price of land

1.10 Other construction expense

1.11 Import, Customs Clearance Fees, Insurance

2. Annual Operational Cost

2.1 Main fuel cost (Natural Gas)

2.2 Auxilary fuel cost

2.3 Water cost

2.4 Operations goods

2.5 Labour Cost,Personel:

Social Pension Funds-employer’s share, Social Security

2.6 Maintenance

2.7 Depreciation

2.8 Fixed expenditure

2.9 General expenditure:

Communication, health and safety

2.10 Finance, marketing and sales, legal costs, taxes etc.

4.2. Thermoeconomic Optimization and Feasibility of the Project

In the thesis, analytic methodology given in Chapter 3 with the governing equations, is combined with the engineering expertise necessary to interpret computer-generated estimates and an economic study. Feasibility for the projects is decided considering all these factors, explained below. (Flow for the thermoeconomic study)

- The user's historic electricity, heat and/or natural gas consumption profiles are evaluated, considering the heat, steam, or air-conditioning needs.

- Site properties like the ambient temperature and pressure, relative humidity etc are defined.

- Demand curves are built.

- These curves are best fitted with supply curves, depending on the users desired capacity, considering the available sizes for gas turbine/ steam turbine configurations in the market.

- Alternative configurations using only gas turbine and HRSG; only steam turbine and HRSG; gas turbine, steam turbine and HRSG and additional refrigeration units to all these previously described configurations are prepared, and presented to the user, considering the related construction and material costs.

- Among configurations, the user is guided to find the best system; considering his requirements. If in any case the capacity is insufficient, or one or more needs can not be efficiently produced, or if the gas turbine can not serve the requirements, user is informed and again guided through.

- Then, the cogeneration design and performance is determined, all important parameters for the gas and/or steam cycles are calculated by a thermodynamic optimization process. Some important relations for this process is given in the following section; 4.3.

- Energy requirements for one year of operation, with the cogeneration system following the planned electricity load are evaluated, short term costs are calculated in more detail.

- Lastly, a long-term economic evaluation is performed.

This study shows how feasible the project is. Thermoeconomic study for METU Campus is given in the following chapter, Chapter 6, with different system configurations and sizes. All the configurations are discussed through out, in Chapters 6 and 7, considering feasibility of each system. [29]

4.3. Optimization for Cogeneration Power Plants

The relations between the input and output parameters for optimal power production are considered in this section. The results are expressed in Figures 4.1 to 4.11.

In gas turbine power production, one of the most important parameter to increase the gas turbine and therefore CHP efficiency is increasing the turbine inlet temperature (TIT) up to the metallurgical limit set by the material of the turbine blades and last stage turbine stress level. Thus, it is the first parameter to be studied. In Figure 4.1, the variation of cycle efficiency with TIT is shown. Here, ambient conditions are taken as 300 K and 92 kPa and the design point for the plant composed of a gas turbine and HRSG is 20 MW. This result is also similar for the gas turbine and steam turbine case. Compressor pressure ratio is chosen to be 14.

For all of the calculations, corresponding combustion chamber efficiency is taken as 98%; turbine isentropic efficiency as 89 %; compressor isentropic efficiency as 89 % and the LHV of the fuel as 36700 kJ/kg.

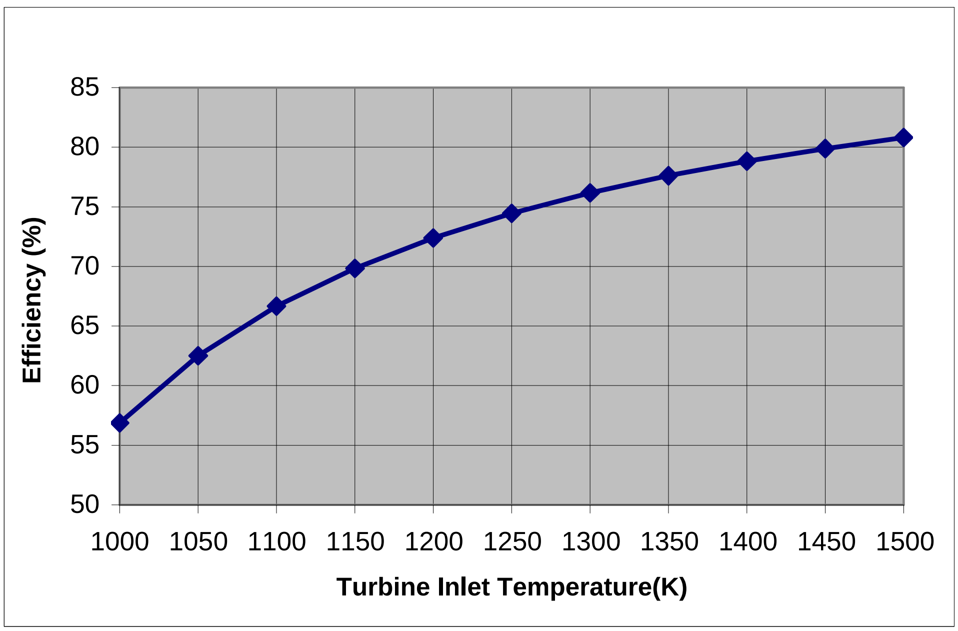


Figure 4.1 Overall (Gross) Efficiency vs TIT

Another important parameter for efficient operation is the ambient temperature. The performance of turbines is adversely affected by high ambient temperatures. Several means of reducing the turbine inlet temperature (off-peak water chiller and ice storage and absorption refrigeration systems) are proposed as means of increasing turbine output. The energy in the turbine exhaust has the potential of producing additional cooling with the help of an absorption cycle for reducing the inlet temperature. For Figure 4.2, results are again calculated at 300 K and 92 kPa, with an optimal total plant capacity of 20 MW, gas turbine, steam turbine and HRSG configuration. Compressor pressure ratio is taken as 14.

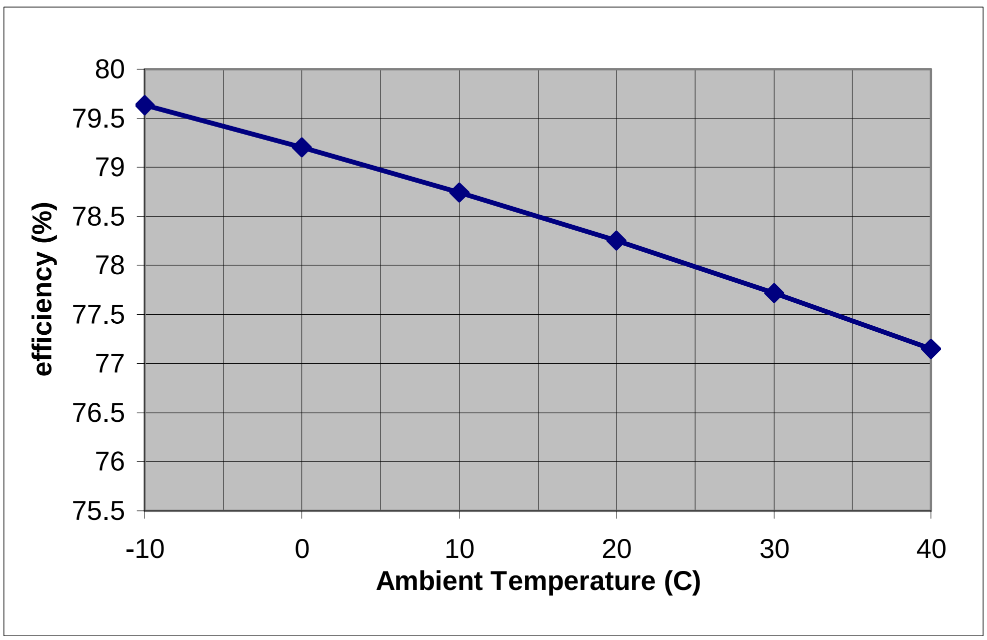


Figure 4.2 Overall (Gross) Efficiency vs Ambient Temperature

Fixed Power Plant Capacity Calculation:

For power plant calculations, when the capacity is fixed, specific electrical power output for the simple/combined cycle first increases, then decreases with increasing pressure ratio, as can be seen in Figure 4.3. This is an important result to decide on the electricity and heat capacities.

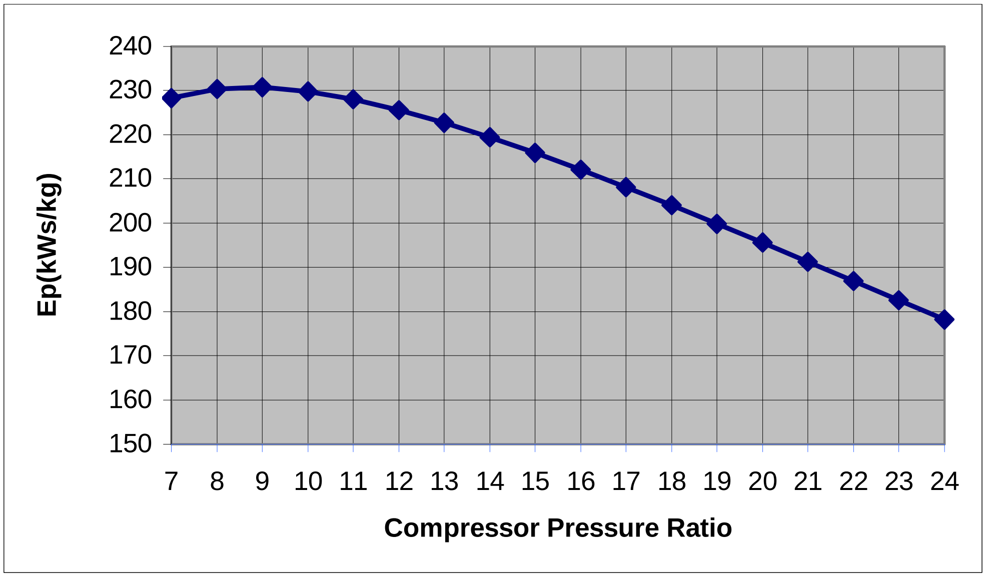


Figure 4.3 Specific Electric Power Output vs Pressure Ratio

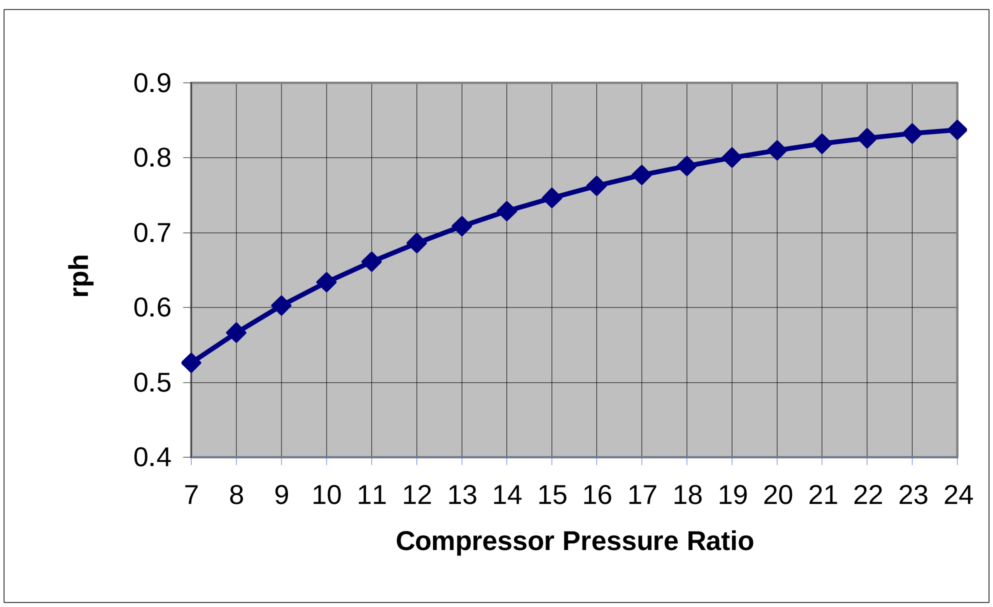


Figure 4.4 Power to Heat Ratio vs Compressor Pressure Ratio

In Figure 4.4 for a fixed capacity, power to heat ratio versus pressure ratio of the compressor behaviour can be seen. Turbine inlet temperature is assumed to be constant, and the rest of the variables are same as before.

For the same cycle, power to heat ratio versus turbine inlet temperature can be seen in Figure 4.5. Compressor pressure ratio is assumed to be constant, which is 14, and the rest of the variables are same.

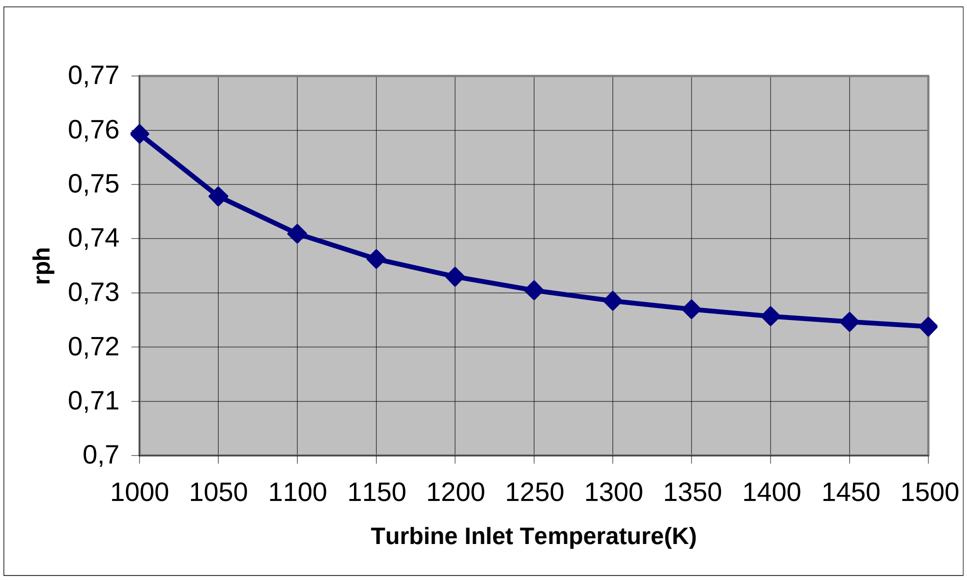


Figure 4.5 Power to Heat Ratio vs TIT

Figure 4.6, represents the relation between electrical power output and turbine inlet temperature for the previous cycle.

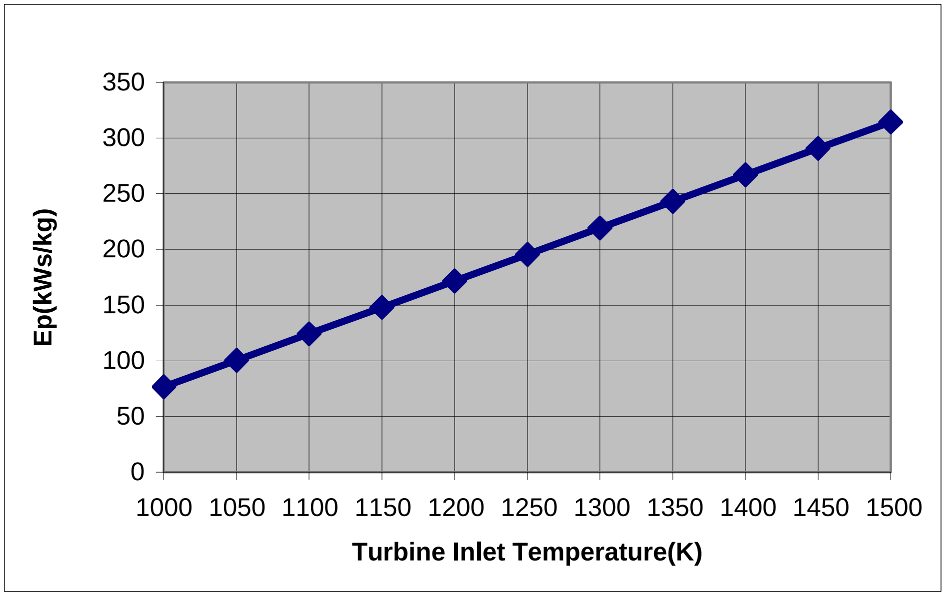


Figure 4.6 Specific Electrical Power Output vs TIT

Fixed Electrical Power Calculations:

Following graphs (Figure 4.7 and 4.8) show the relations between some important parameters when the program is run at a fixed (constant) electrical power output. Calculations are done, and the results are found in an environment of 300 K and 92 kPa. Turbine inlet temperature is assumed to be constant, 1300 K and electrical power output of the plant is 7 MW.

The relations between specific electric output and compressor pressure ratio for the simple/combined cycle is given in Figure 4.7: Specific electric output increases first, then decreases with increasing pressure ratio. This is an important relation for deciding on the electricity and heat capacities. The figure shows a similar behaviour as in Figure 4.3. Power to heat ratio versus pressure ratio also shows the same behavior that as in Figure 4.4.

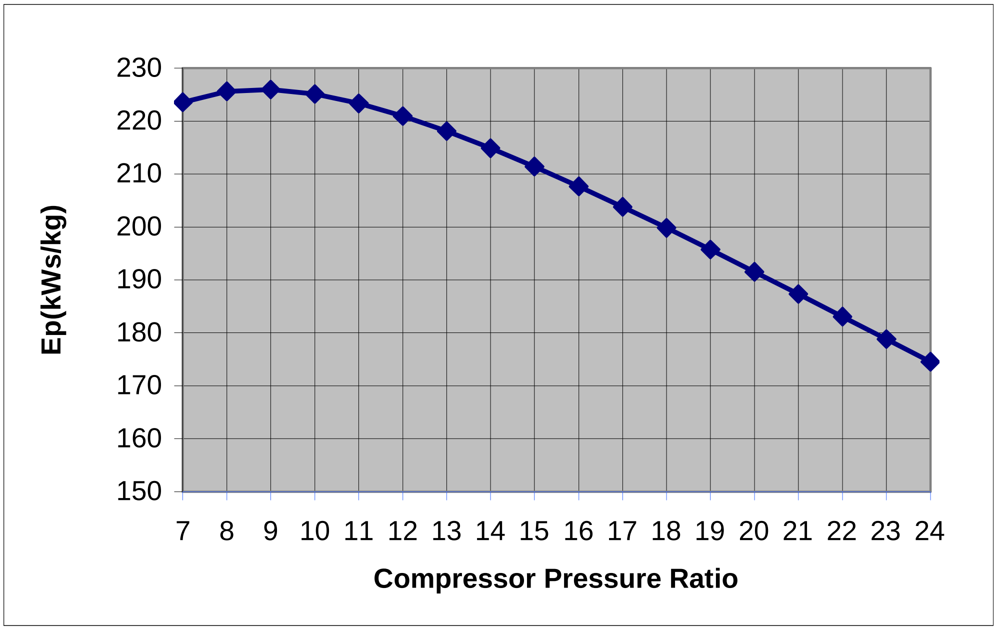


Figure 4.7 Electrical Power Output vs Compressor Pressure Ratio

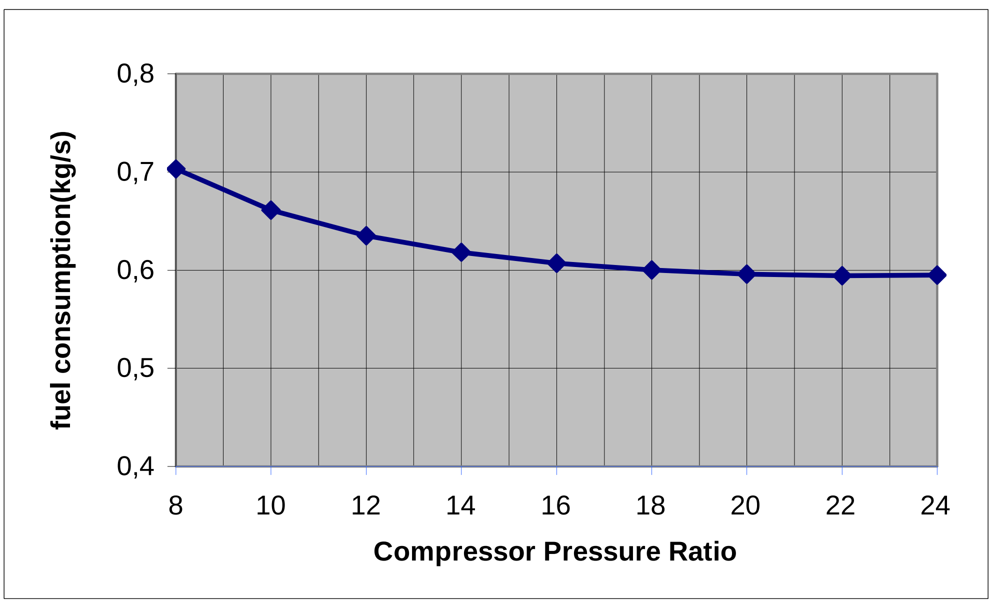


Figure 4.8 Fuel Consumption vs Pressure Ratio

The fuel consumption versus pressure ratio, at same conditions is given in Figure 4.7. As seen in Figure 4.8, fuel consumption decreases until an optimum value of pressure ratio, then it remains constant.

Figure 4.9 gives the relation between electrical power output and turbine inlet temperature. Here, the ambient conditions are same as before. Compressor pressure ratio is 14, and the fixed electrical power output of the plant is 7 MW. As can be seen, electrical output increases with increasing TIT.

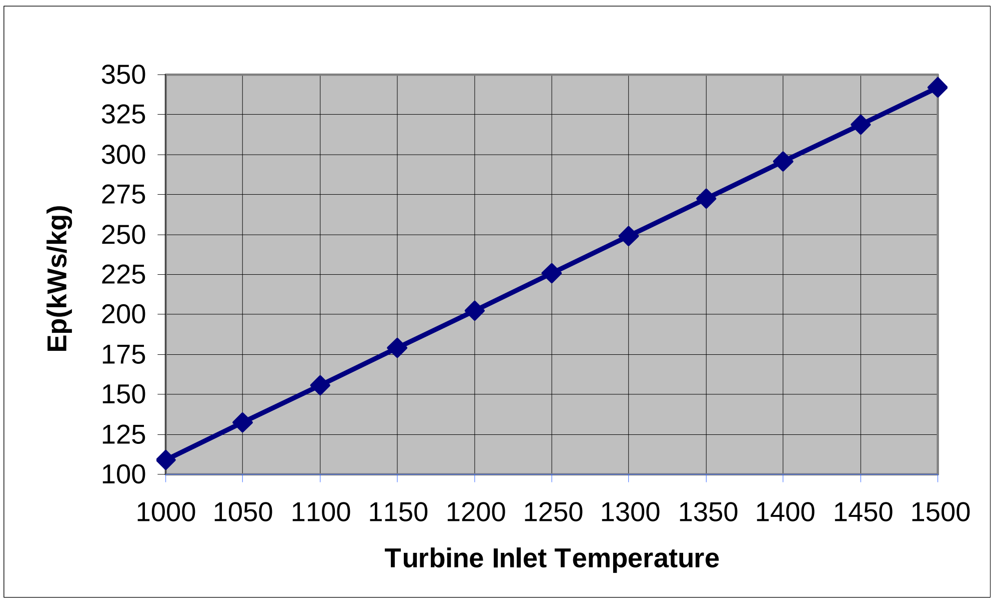


Figure 4.9 Specific Work Output vs TIT

Figures 4.10 and 4.11 show the relation between optimum plant capacity with turbine inlet temperature and pressure ratio.

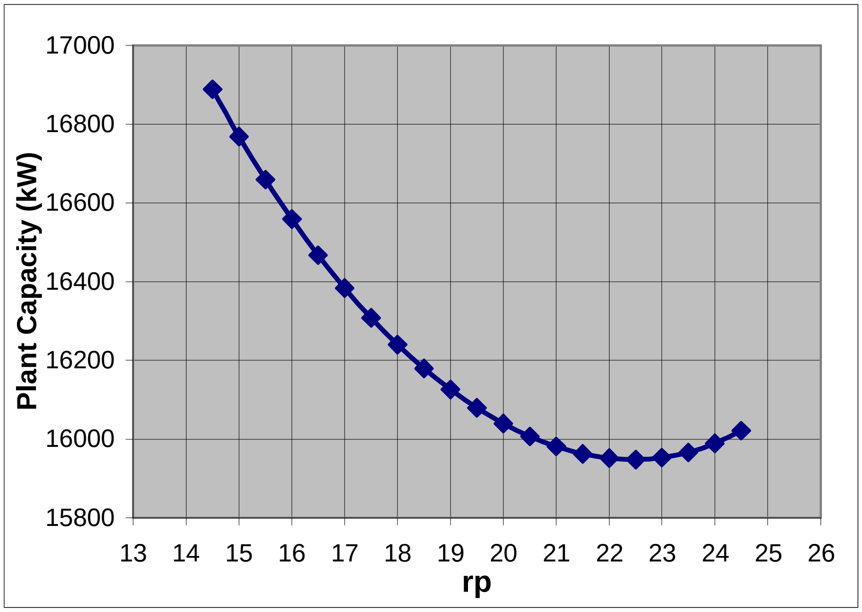


Figure 4.10 Plant Capacity vs Pressure Ratio

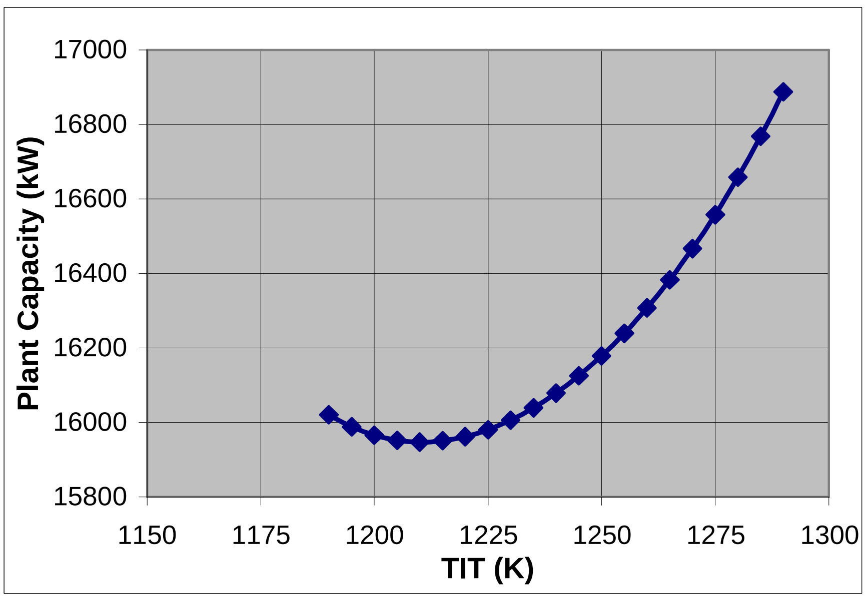


Figure 4.11 Plant Capacity vs TIT

CHAPTER 5

DESCRIPTION OF THE PROGRAM

5.1. Introduction

"Cogeneration Design" is a computer program for conceptually designing cogeneration power plants. The user inputs numerical values, as the design point, and the program computes heat and mass balance, system performance, and gives outputs for an optimal design such as electric and heat (steam) production. "Cogeneration Design" is easy to use, and it takes a few minutes to finish the conceptual design of a cogeneration plant. The user adjustable inputs help to create a wide range of design projects.

Code of the "Cogeneration Design" program is written using Visual Basic 6.0. Number of forms and modules are prepared for the program. Forms are the interactive windows created for user to read or input necessary values during design procedure. Modules are the subprograms that are called when a specific task has to be carried out more than once during the run.

5.2. Flow Chart of the Program

In Figure 5.1, the basic flowchart of the cogeneration design program is given. Algorithm of the program is easy to understand and to follow. For each design step, up to 500 iterations may be done. The program always helps the user to input parameters in a sensible way, that means, there are always allowable ranges for all input values. Whenever a solution can not be obtained, program warns the user, giving a reason for the unfeasible situation, and offers solutions, if possible.

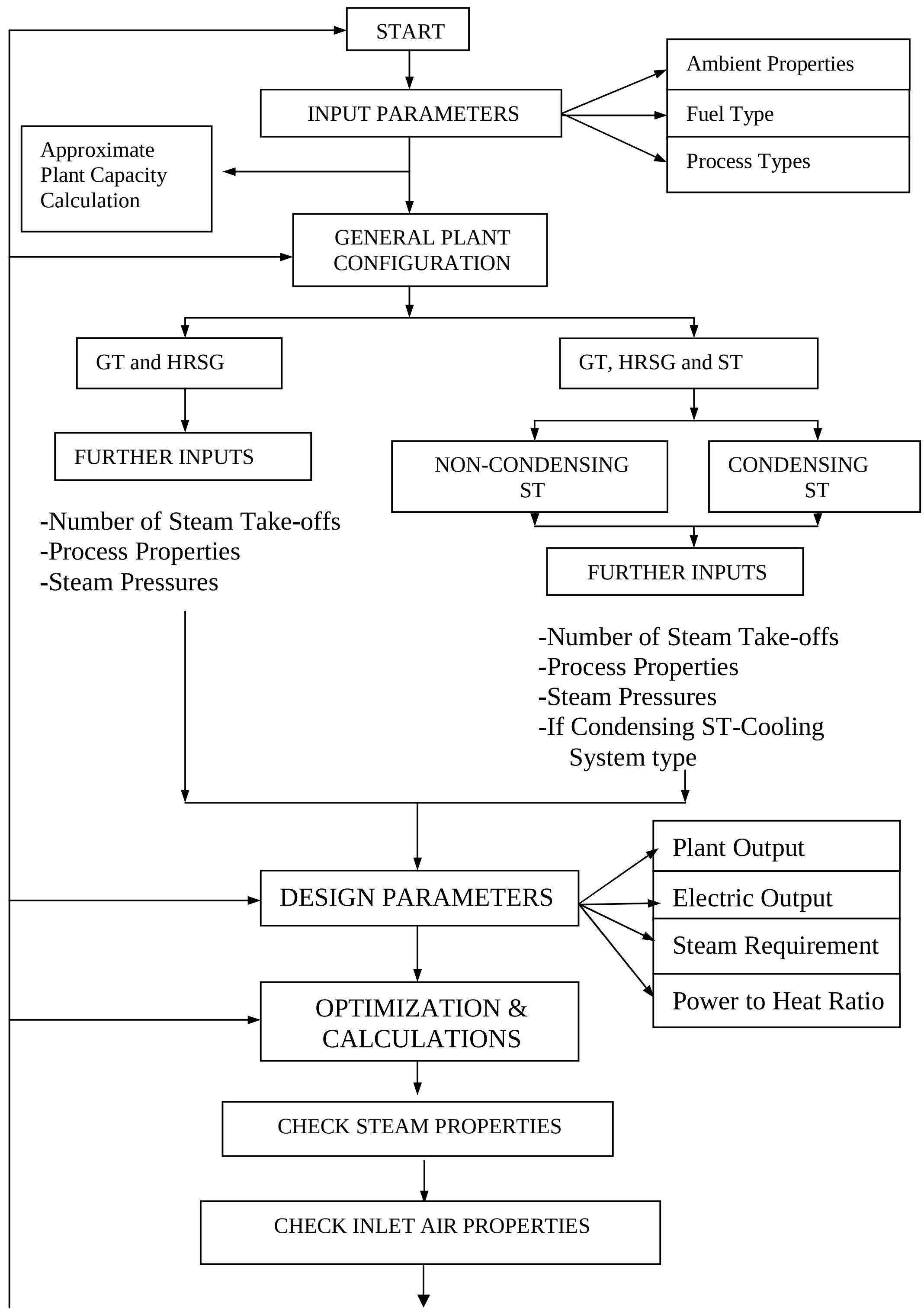


Figure 5.1 Flow Chart of the Program

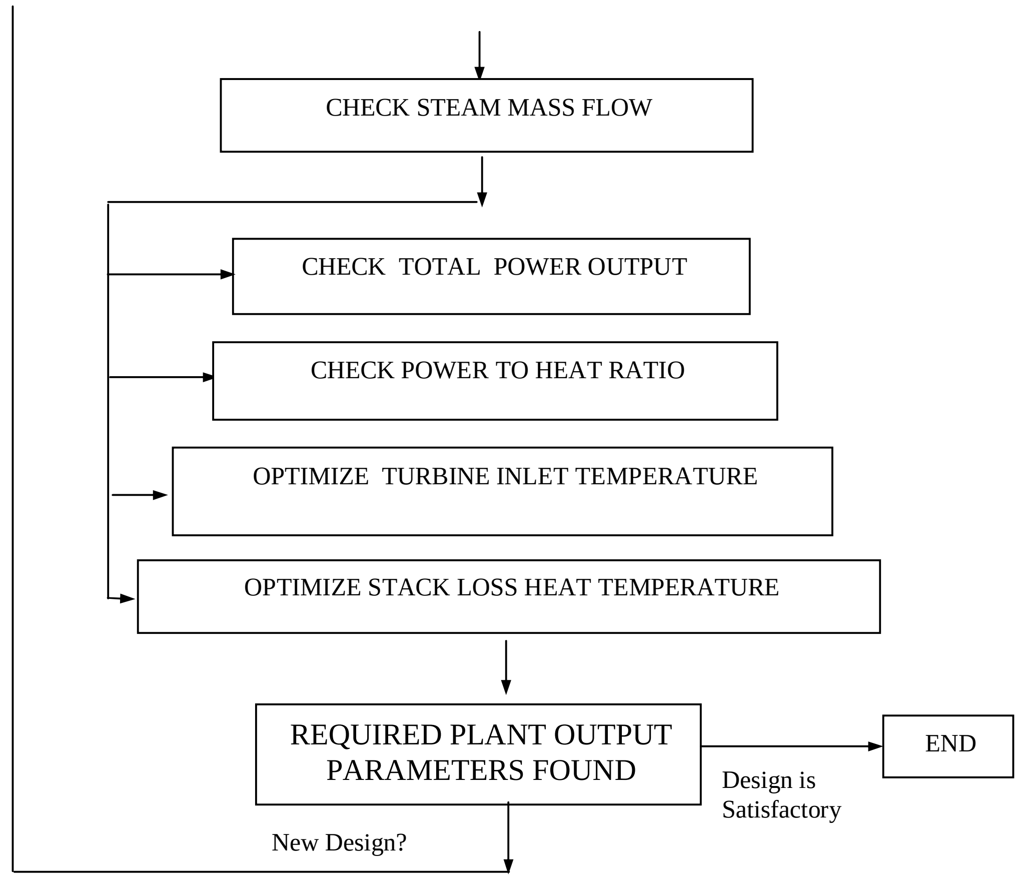


Figure 5.1 Flow Chart of the Program (Cont.)

5.3. Start up of The Program

On the first screen of the program, which is illustrated in Figure 5.2, user has to define the problem, by simply choosing among three options which include the range of his probable inputs. He has to define the fuel type, outside temperature, attitude or the ambient pressure where the plant will be built, average calorific value of the planned fuel, approximate plant output and will choose the general plant configuration. The program will always help the user to be in a reasonable range of properties by error messages and pop up notes, as mentioned before.

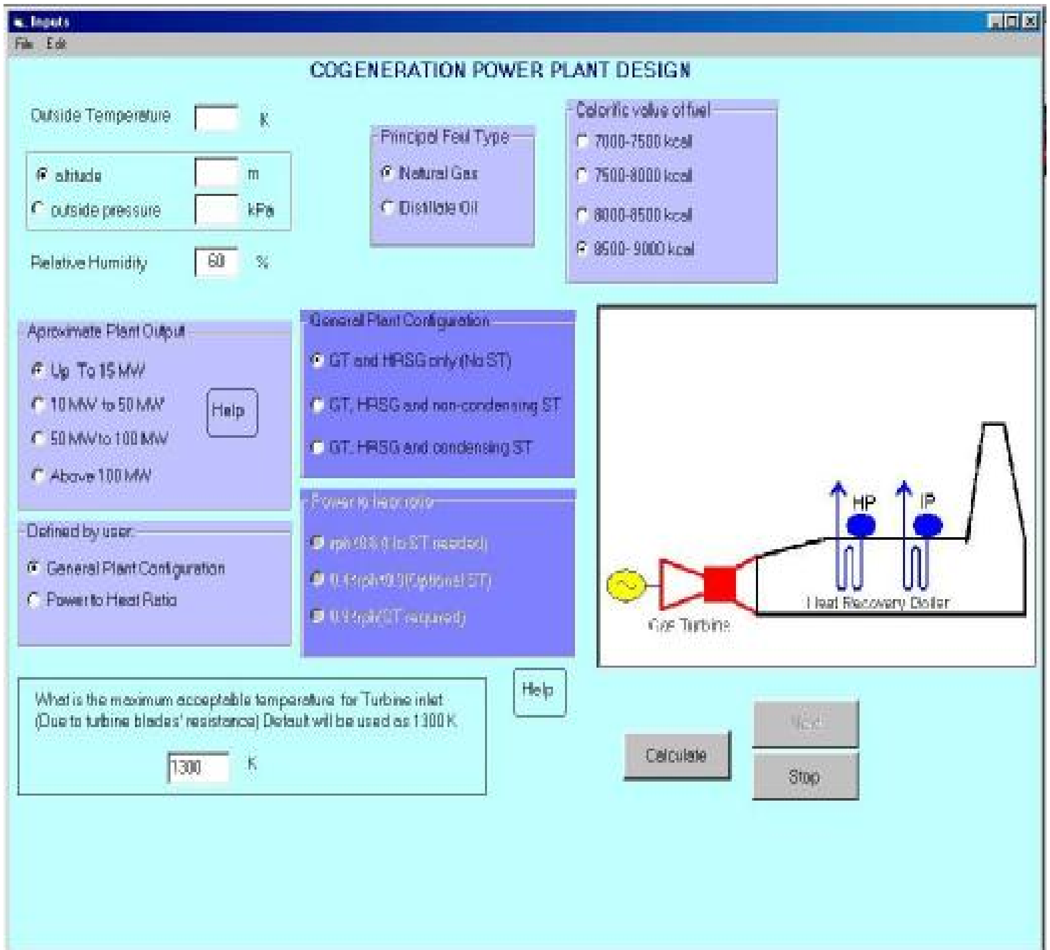


Figure 5.2 Cogeneration Design Program Start up Form

If user needs help for choosing an approximate value for plant output, the following screen, given in Figure 5.3 can be used for calculating the reasonable output range for the user. To find the approximate electrical power, the user has to specify the annual electric consumption in kWh, as well as the operating availability, which is the availability of the plant working hours excluding probable shut down period of the plant due to maintenance and some external errors. Another important factor is the generation factor for the plant, which is the measure for the amount of energy that a plant could generate during the time considered. After these parameters are input, the program first calculates the total hours of work for the plant, annually and the capacity due to electrical consumption. In a similar way, thermal capacity for the plant, considering the annual natural gas consumption can be determined. Among the consumption values, one with the higher value is to be considered as the approximate plant capacity.

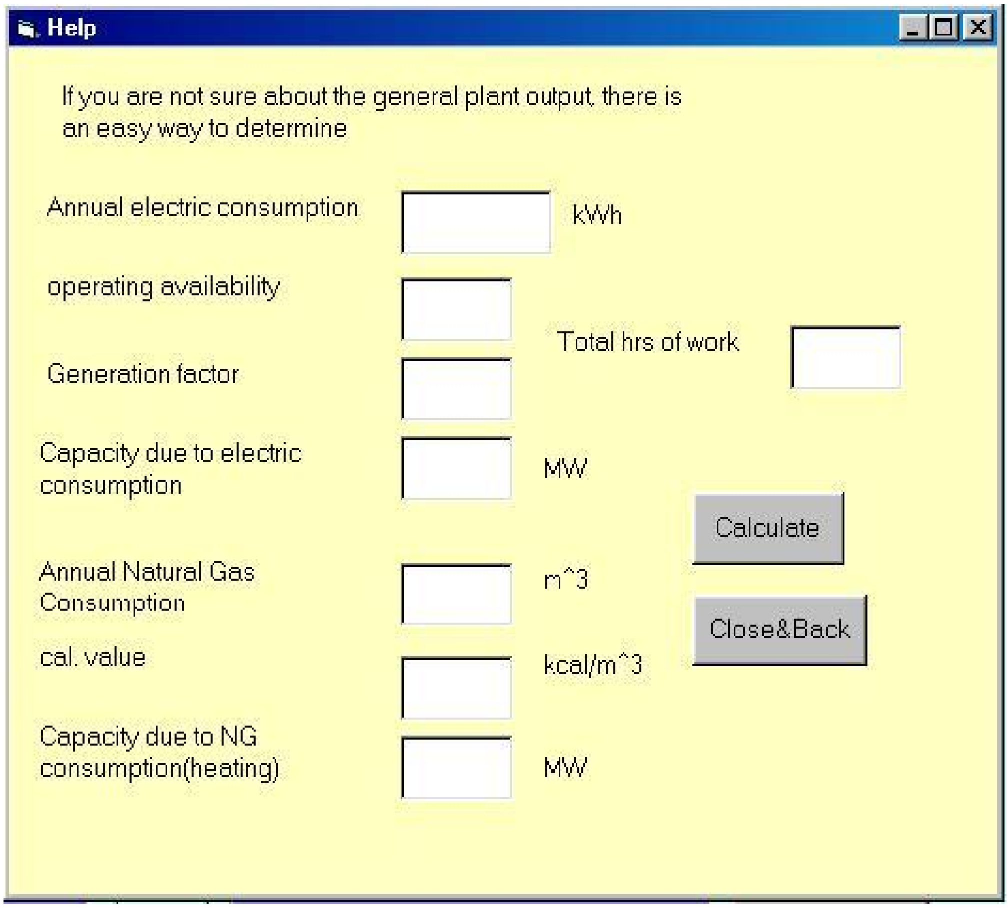


Figure 5.3 Cogeneration Design Program Start up, Help Form:1

If user has any problems in defining the general plant configuration, the help screen in Figure 5.4 above will occur.

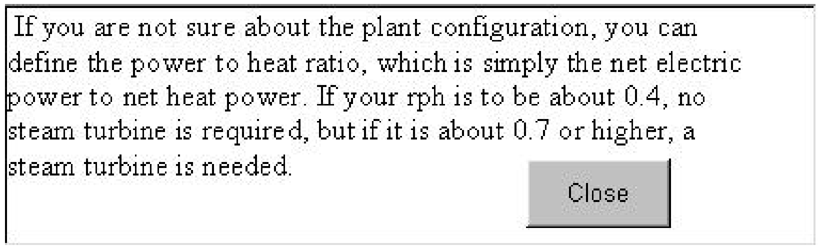


Figure 5.4 Cogeneration Design Program Start up, Help Form:2

After user defines the mentioned inputs, the second window, for the further details concerning the type and properties of the processes will occur. There the decision of using a steam turbine or not will appear. This means that, there are two cases, among which the second one also has two, adding up to three possible configurations:

1- Gas Turbine and HRSG (Heat Recovery Steam Generator) Only (No Steam Turbine)
2- Gas Turbine, HRSG and Non-condensing Steam Turbine
3- Gas Turbine, HRSG and Condensing Steam Turbine

After all data is input, user will push the “next” button to proceed.

5.4. Determination of The Process Type and Properties

“If Gas Turbine and HRSG Only (No Steam Turbine)” is chosen, user is up to the form given in Figure 5.5. If the second or third item is chosen -which means that the system will be gas turbine-HRSG and steam turbine this time- condensing or non condensing, user ends with the given form Figure 5.6. In both forms, number of steam take offs from the HRSG has to be determined. There is only HP (high pressure) steam for process use and to be used in the steam turbine, if one take off is chosen, there are HP and IP (intermediate pressure) steams to be used if two take offs are chosen, or there are all HP, IP and LP (low pressure) steams for process use, or to be used in the steam turbine, if three take offs are chosen.

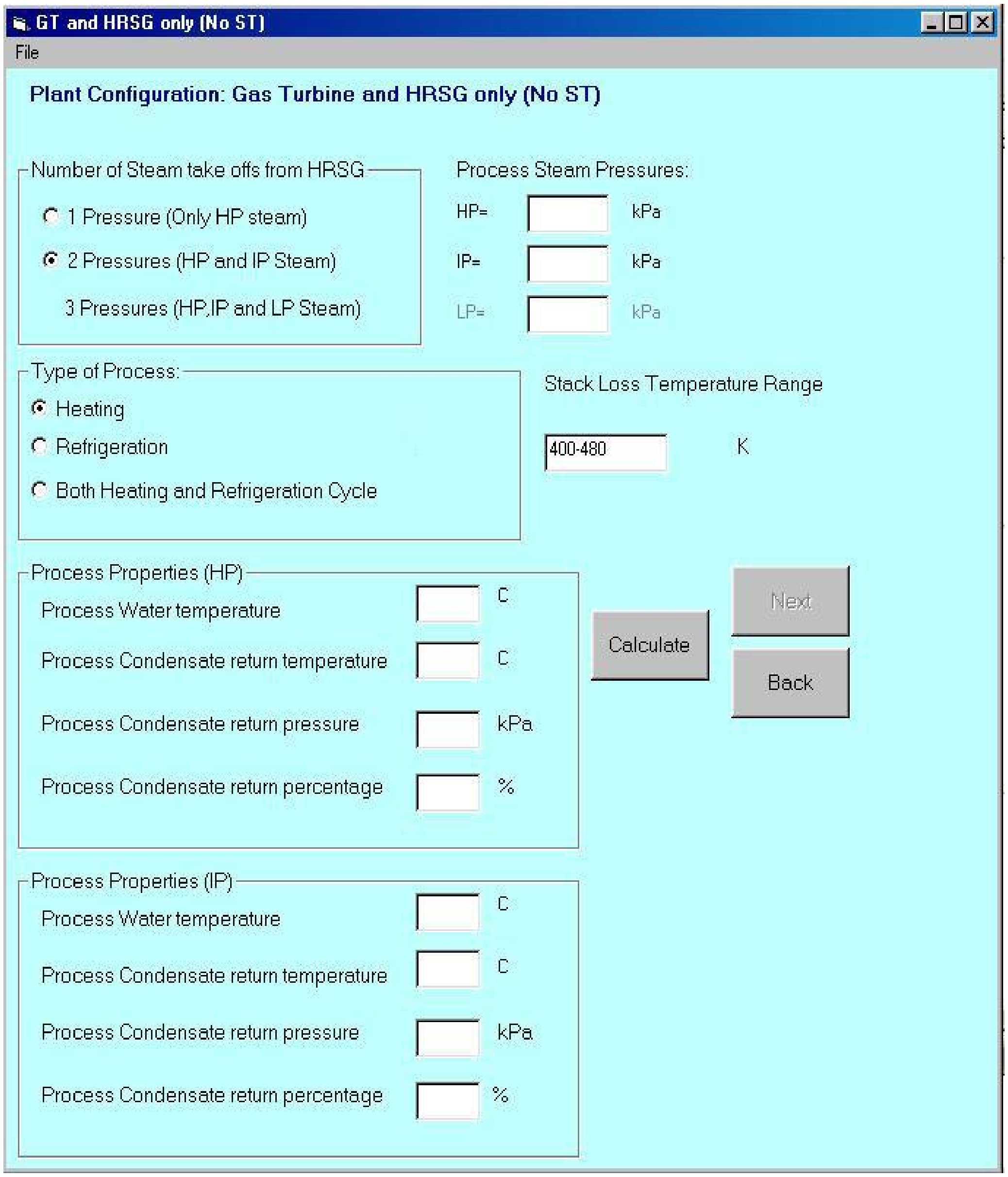


Figure 5.5 Cogeneration Design Program Form-2

For each steam take off, process steam pressure is also to be determined by the user, upon the process type, for example, for heating, refrigeration or both processes (trigeneration). Process steam (water) is the steam which will be used as heating medium, for example for district heating in radiators, or will be used for cooling in an absorption cooling unit, or any other industrial application where steam is to be used. For heating or refrigeration, the following properties are needed to solve for the heat balance; process water temperature, condensate return temperature and pressure and return percentage.

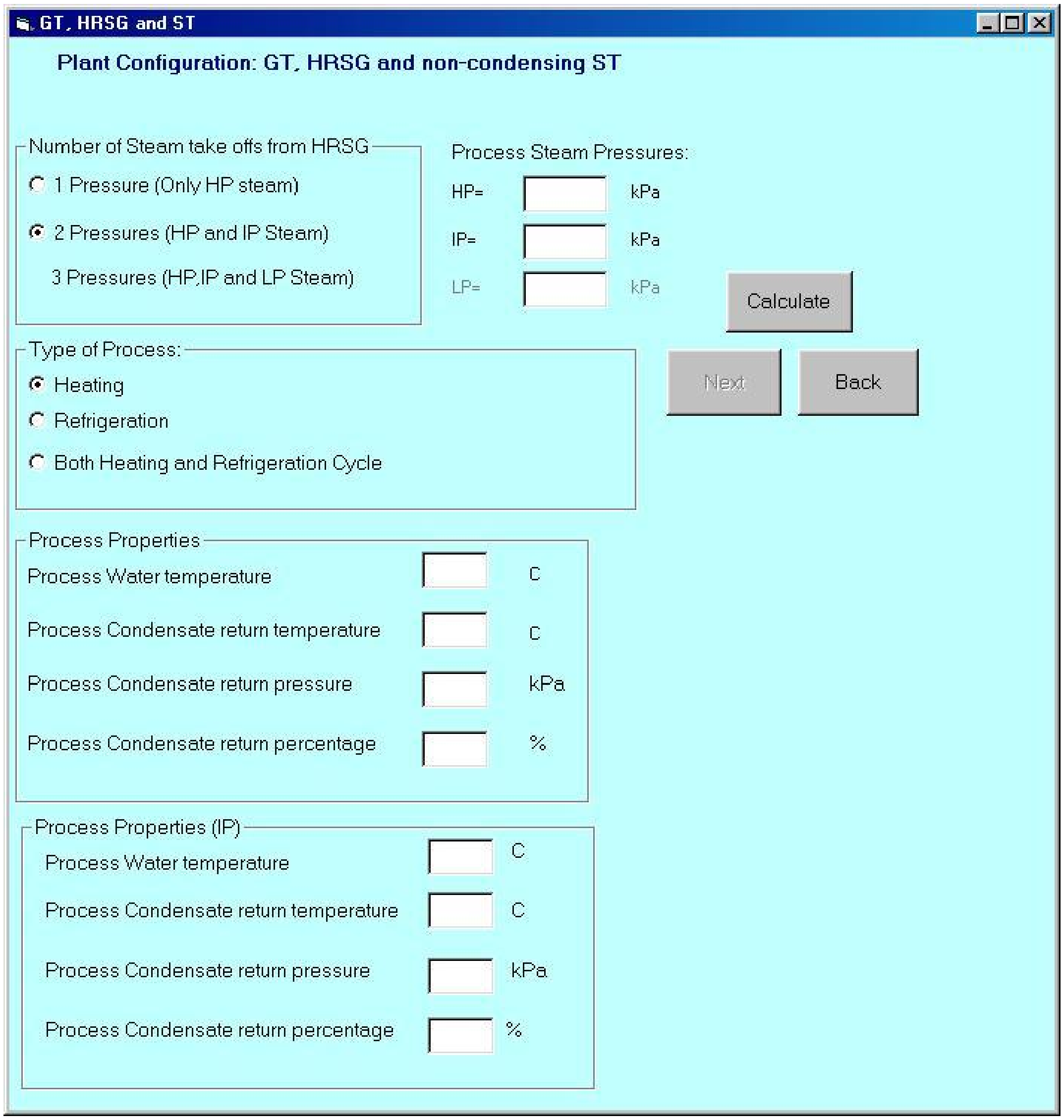


Figure 5.6 Cogeneration Design Program Form-3

Among these, process condensate return pressure is calculated by including 1% pressure loss in the HRSG as a default, but user may redefine this percentage. For condensing steam turbine case, cooling system type is to be determined, upon he given configurations. When all necessary inputs are given, user will proceed by pressing first “Calculate”, then “Next” buttons.

5.5. Choice of Design for the Cogeneration Power Plant

After all the inputs are determined, the design parameters are to be chosen. Again there are two forms, if “No Steam Turbine Case” is the choice, Form 4 in Figure 5.7, if steam turbine case iss chosen, Form 7 in Figure 5.8 will come up.

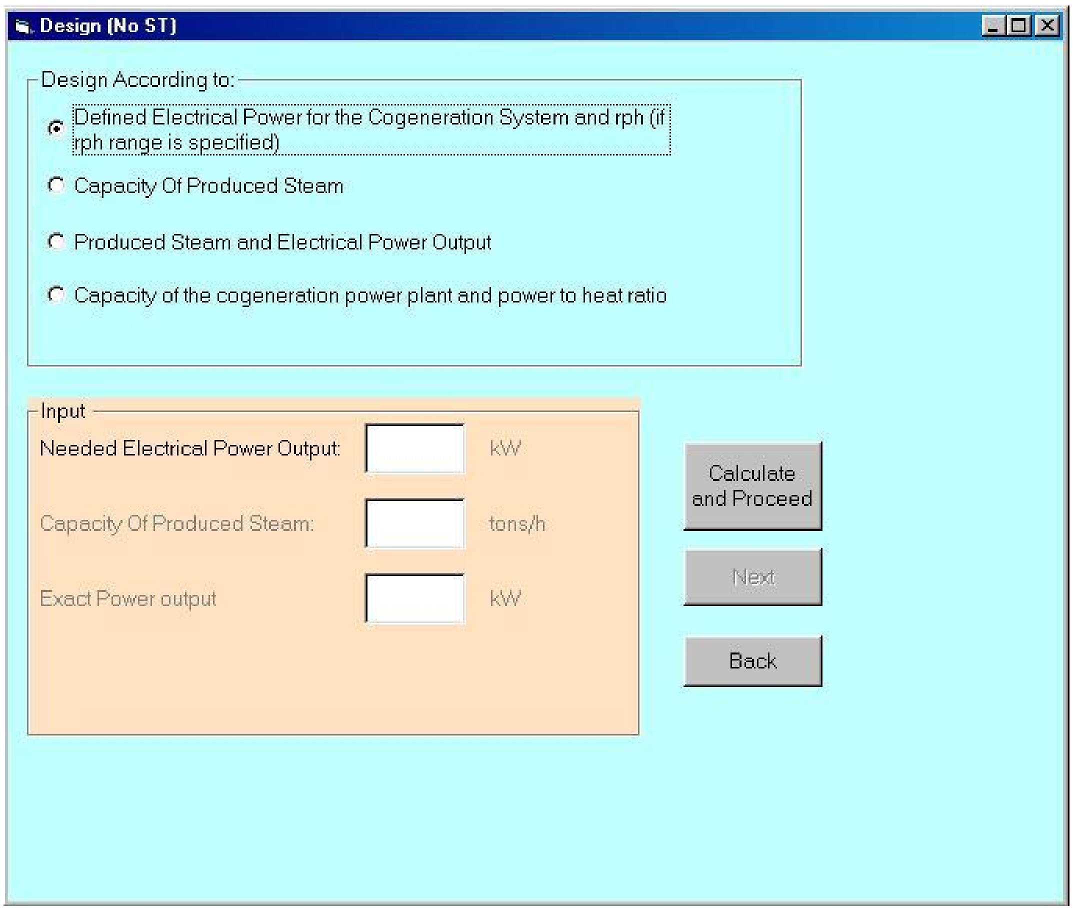


Figure 5.7 Cogeneration Design Program Form-4

The user may choose to define the electrical power for the cogeneration system, capacity of the produced steam, both the capacity of steam and electrical power output, or capacity of the cogeneration plant and power to heat ratio. Program will always help the user to specify the correct inputs by an interactive form and error pop ups.

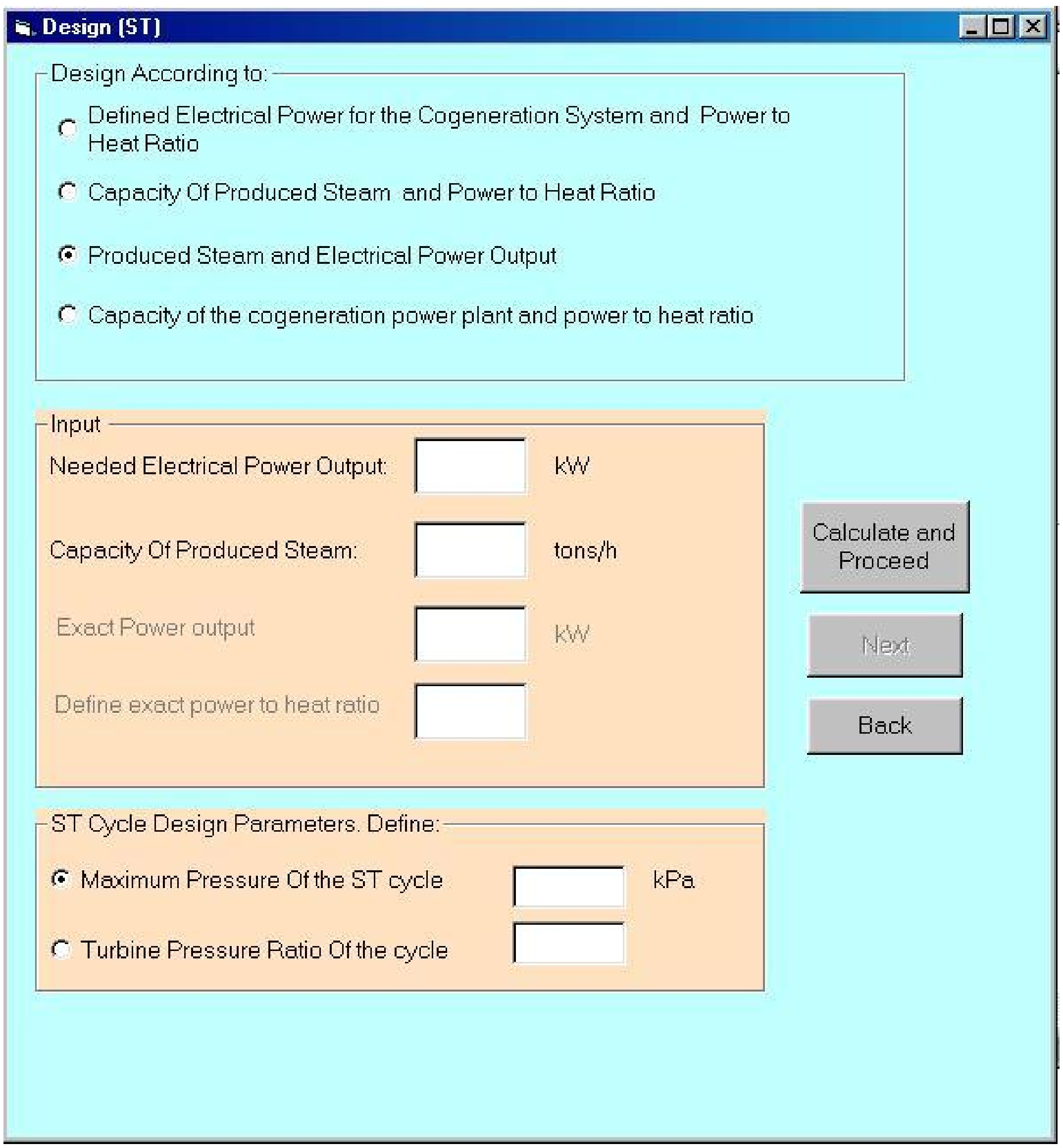


Figure 5.8 Cogeneration Design Program Form-7

If the user choose the refrigeration or both heating and refrigeration cycle (trigeneration) in forms 2 or 3, up on clicking calculate and then next, he will come up with Form 4 for “No Steam Turbine Case”, and Form 7 for “Steam Turbine Case” again, but this time forms will look different, as can be seen in Figures 5.9 and 5.10 respectively. Now the user is to input the refrigeration temperature, and either refrigeration power input, or mass flow rate. Total electrical power output of the system will be displayed on the same forms.

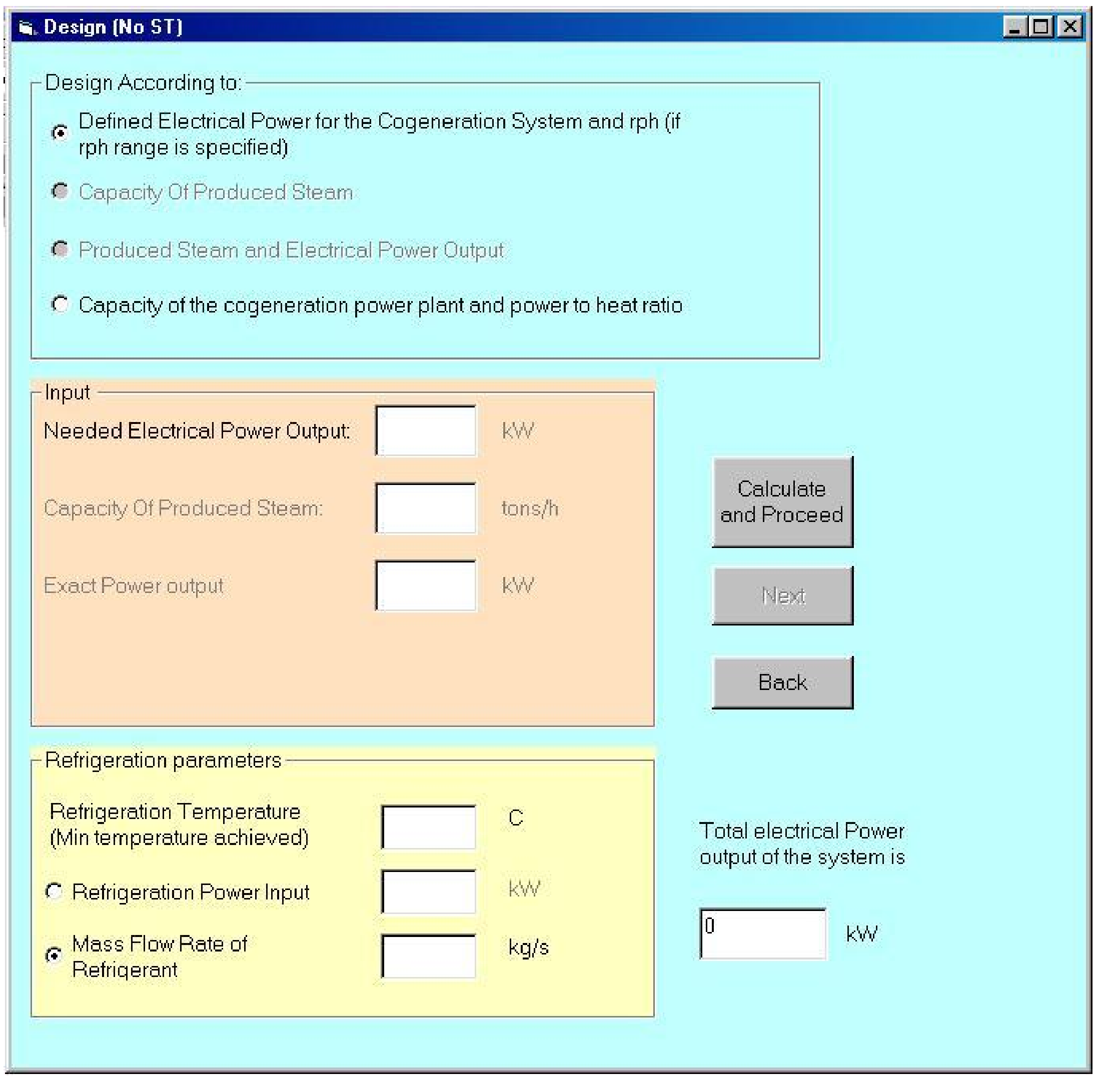


Figure 5.9 Cogeneration Design Program Form-7 with Refrigeration for no Steam Turbine

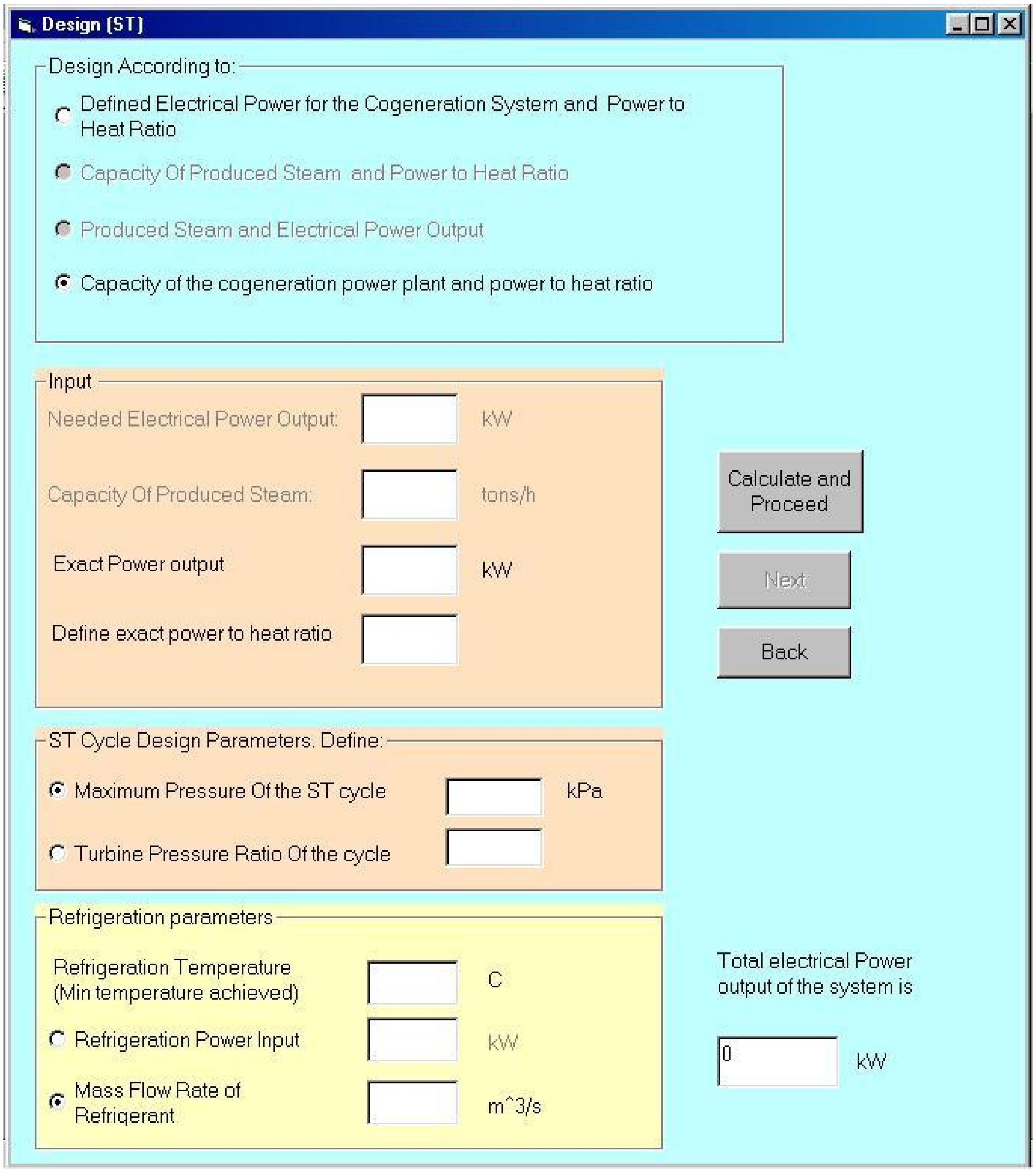


Figure 5.10 Cogeneration Design Program Form-7 with Refrigeration for Steam Turbine

5.6. Property Watch Form

During the design process while the program is working and while the calculations proceed, user may see the calculated values any time he/ she wants by simply calling the property watch form. Thus, the properties like turbine inlet temperature and pressure, compressor compression ratio, exhaust temperatures and etc. may easily be seen whenever user wants. A sample property watch form is given in Figure 5.11 below, for a design process without refrigeration.

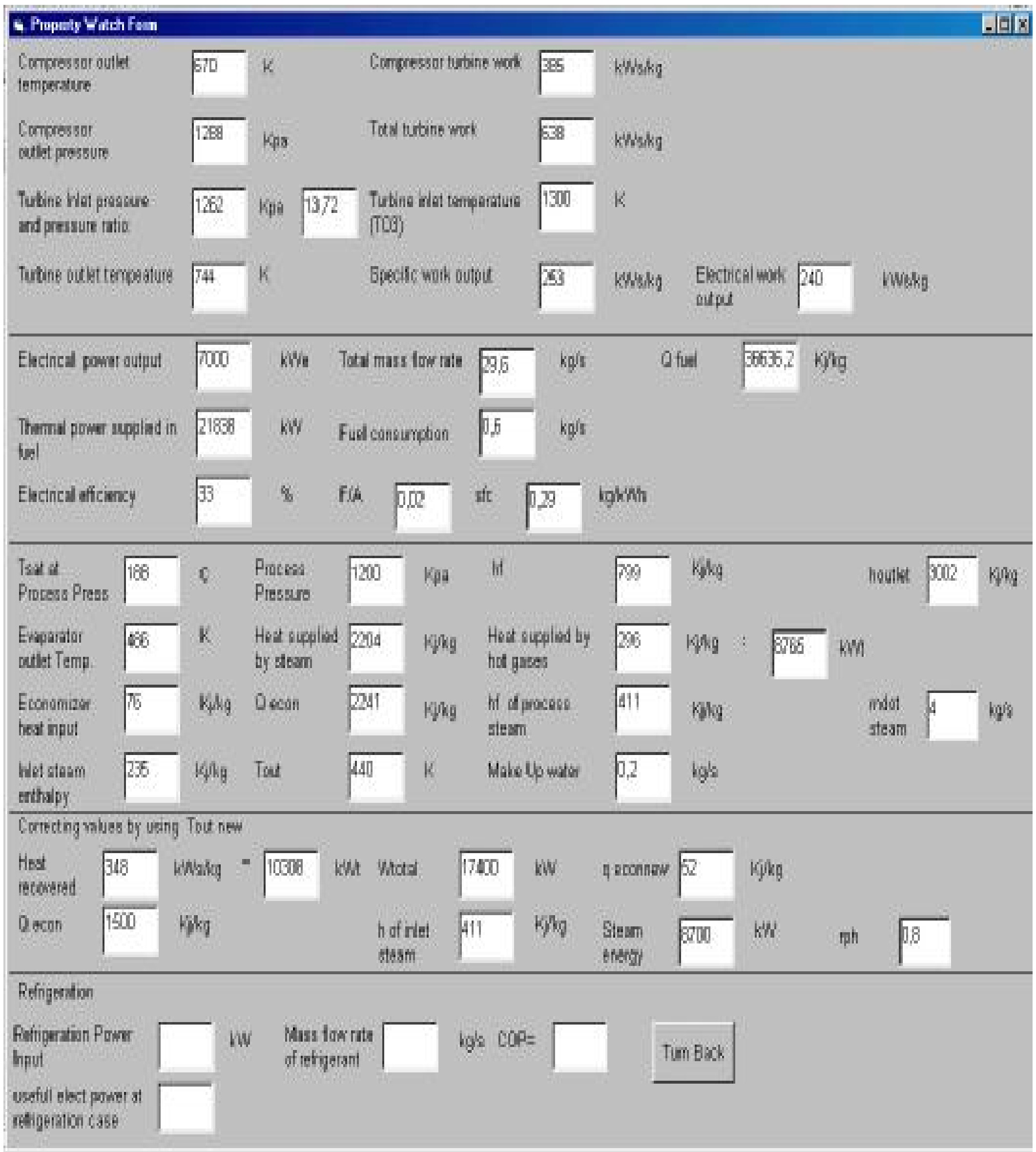


Figure 5.11 Cogeneration Design Program Property Watch Form

5.7. Steam Properties Calculation in Modules

Steam properties like saturation temperature, pressure, saturated fluid or gas internal energy, entropy, or internal energies and entropies of the water/steam at any temperature or pressure can be calculated by the help of the modules of the program.

5.8. Optimization (Correction) Forms

There are some self-corrected mistakes in the program. For instance, if the calculated overall capacity of the power plant exceeds the maximum value the user previously defines, the program, running an iterative loop, tries to recover the error. The form, seen in Figure 5.12 appears, and if the user chooses the correct option, program tries to put the plant capacity between the limits. If this would not lead to a solution, then the user should redefine some values, or choose another range for plant capacity.

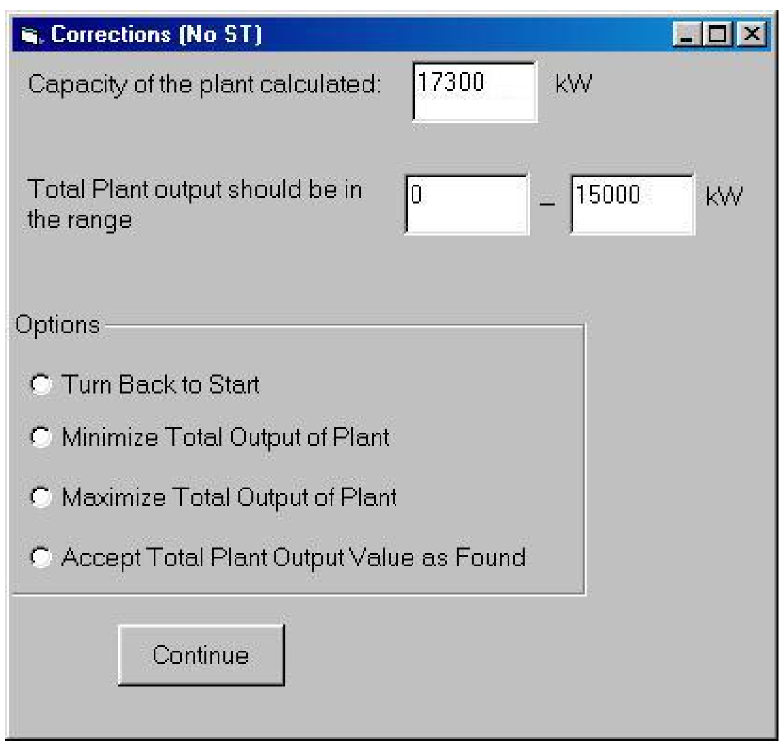


Figure 5.12 Correction Form for No Steam Turbine Case

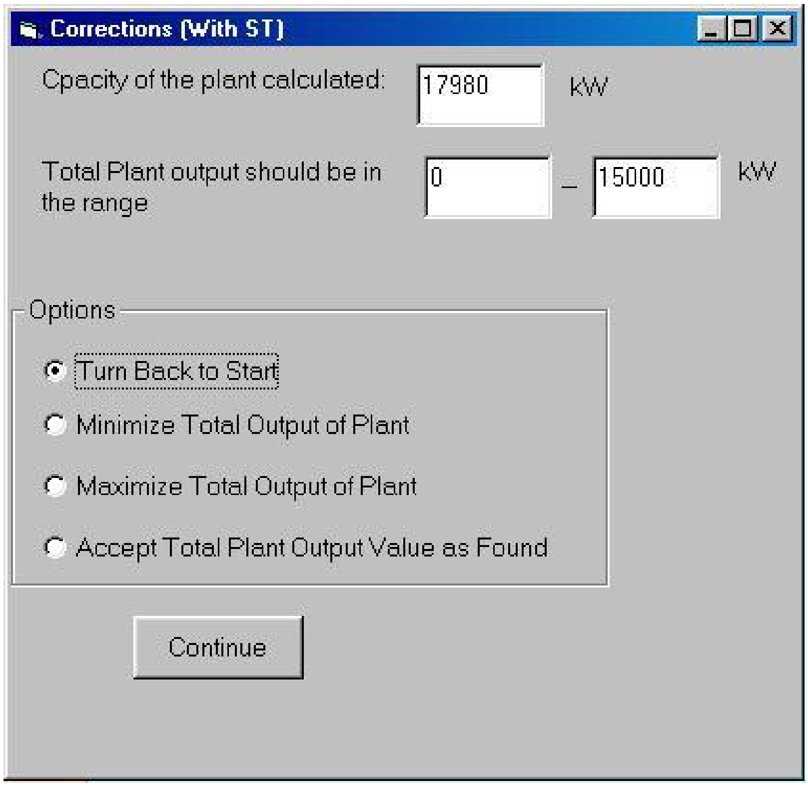


Figure 5.13. Correction Form for Steam Turbine Case

5.9. Output Form

As soon as the user presses “Calculate” button, due to the specified parameters and design criteria, program will start calculation and iterations to find the optimal design for the cogeneration power plant. All the data the user needs will then be presented on the output form of the program. User may always have the freedom to turn back, redefine some parameters and do the calculations for other design parameter.

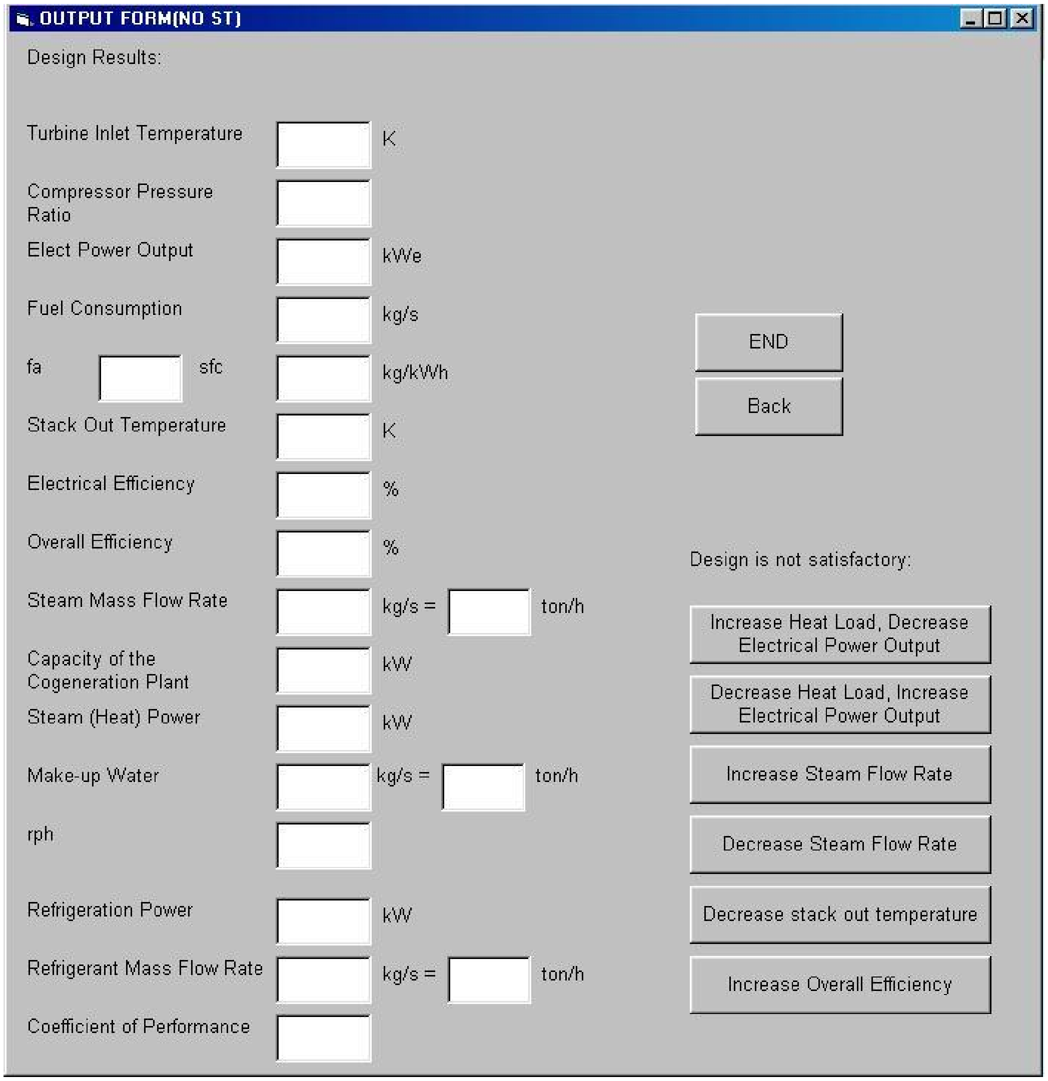


Figure 5.14 Cogeneration Design Program Output Form for no Steam Turbine Case

As seen in Figure 5.14, there are some option buttons for the user if he thinks the design is some how not satisfactory. By pressing any of them, which is suitable for the user, a more economical or efficient re-design can be done within the limits of the program.

In Appendix E, there is a sample cogeneration power plant design study for a better understanding of the program.

CHAPTER 6

COGENERATION ON A CAMPUS ENVIRONMENT-CASE STUDIES-

As mentioned before in section 1.1.3, university campuses are places where cogeneration would be the most cost-effective means of producing heat and electrical energy as well as the most realistic mechanism for controlling electrical energy costs. This is mostly because, in universities, heat and electrical demands differ a considerable amount throughout the year, and the ratio of heat demand to electrical demand is relatively high. With a cogeneration system, the necessary energy of any type may be produced anytime, in any quantity, in other words, independency for energy is gained.

Cogeneration facility gives the university an opportunity to control and reduce energy costs by investing in an on-site power plant.

With a cogeneration facility, the university may benefit from the reduced CO_2 emissions arising globally from the independent generation of power as well as lessened water pollution and may help conservation of fuel resources.

There are quite a lot of universities all around the world, use the opportunity of cogeneration, with the installed capacity ranging from 50 kW to 300 MW. The detailed list for these universities is given in Appendix A.

In the study, cogeneration power plants are conceptually designed to meet requirements of the METU Campus. The conceptual designs are developed by using "Cogeneration Design" Program which is capable of designing a wide range of cogeneration power plants, based on different inputs and cycle configurations.

In this chapter, there are eight different design scenarios /case studies developed for METU Campus. These include only gas turbine and HRSG design in different gas turbine outputs; gas turbine, HRSG and steam turbine design and a compression chiller added design (trigeneration case).

6.1. Cogeneration in METU Campus

For the first case study, cogeneration facilities in METU Campus is examined. For a better understanding, additional information for METU Campus is given below.

6.1.1. METU Campus Data and Description

The campus area is 4500 hectares and the forest area is 3043 hectares, including Lake Eymir, about 20 kilometres from the Centrum of Ankara. METU campus is located on the Ankara-Eskişehir highway and has been forested entirely through the efforts of the University employees and students since the early 1960's. All faculties and departments of the University are in the same campus area, except for the "Graduate School of Marine Sciences" which is located at İçel-Erdemli on the southern coast of Turkey.

On the campus, there is a natural gas fired heat plant, which is supplying university's hot water and is responsible from the district heating on the campus. There are five boilers (steam generators) in the heat and water plant, supplying the campus' heat, with capacities of 10,10,10,35 and 55 tons/h steam. There is a newly installed boiler, with a capacity of 65 ton/h, which is planned to be commisioned by April 2004, and to replace all the previous 5 boilers.

The heat plant produce steam at 1200 kPa, 280°C, rejects condensate(saturated) water at 98°C, with about a loss of 4% in the system because of blow downs in the steam generators, and in the heat exchangers. Considering a heat balance between the natural gas and steam, with %20 excess air, it is found that, with 1 Nm^3 natural gas, about 13 kg of steam at 280°C can be produced

The heat plant supply energy for about 430,000 m^2 of closed area in METU campus. This area includes most of the faculties, guesthouses, cafetaria, administative buildings, sport centers and dormitories. However, there are still some buildings with their own central heating systems like research assistants' residences and ODTÜKent, suming up to an area of 131,000 m^2. Lastly, there are some buildings, still in construction, with net area of 22,000 m^2. This means, the current heat plant supplies steam for 74% of the campus. If the whole campus is the target, the capacity of heat supply could be increased by about 26%.

The heat plant supply the steam by different sized pipelines. Diameters for different parts of the pipeline are 250, 180, 125 and 90 mm from the largest to the smallest, as can be found in Apendix F. Layout of utility infrastructure i.e. for natural gas pipelines, water pipelines and electricity distribution lines are also given in Appendix F.

University buys electricity directly from TEDAŞ.

6.1.2. METU Heat and Electric Demand

Ten year's data of campus for natural gas and electricity consumption is studied

The curves representing the University's electric demand in kWh and MW; natural gas demand and heat demand in MW with and without 26 % increased capacity, can be seen respectively in Figures 6.1, 6.2, 6.3, and 6.4.

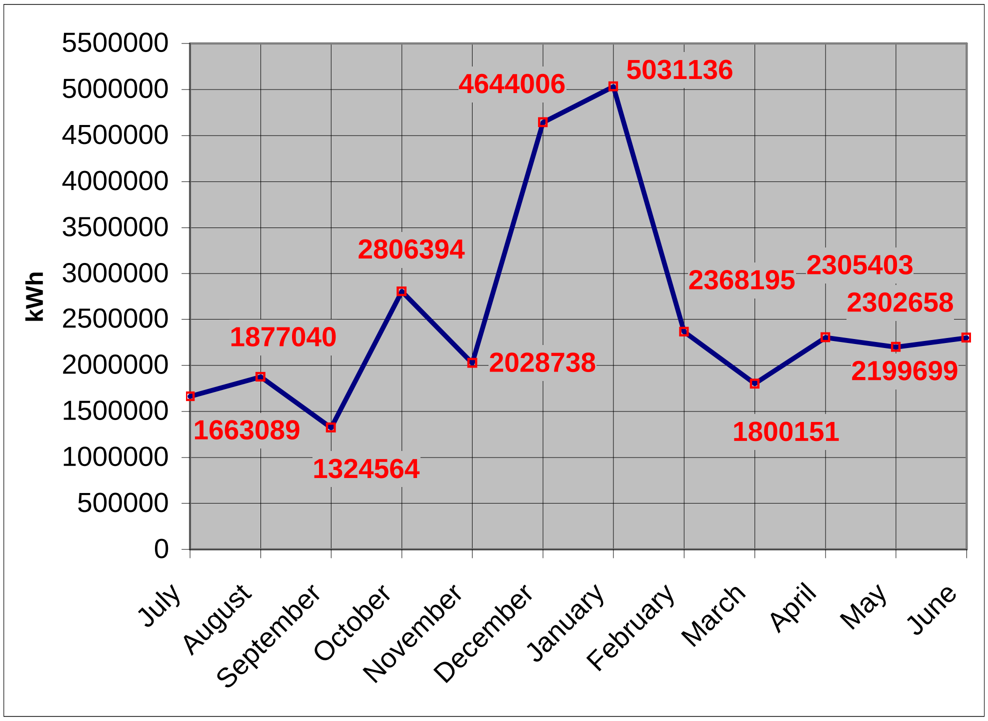


Figure 6.1 Annual Electric Consumption Trend based on 8 Years

More detailed data about the last eight year's electric and natural gas consumptions are given in Appendix G.1 and G.2.

It is clear that, in the Figures 6.1 and 6.2, electrical consumption differs a lot from month to month, even in the same season. The semester beginning and end dates even effect the daily consumptions in the University. This is quite same in the natural gas case, and corresponding heat demands for the University, are given in Figures 6.3 and 6.4. When all the campus area is considered, the increased capacity can be found.

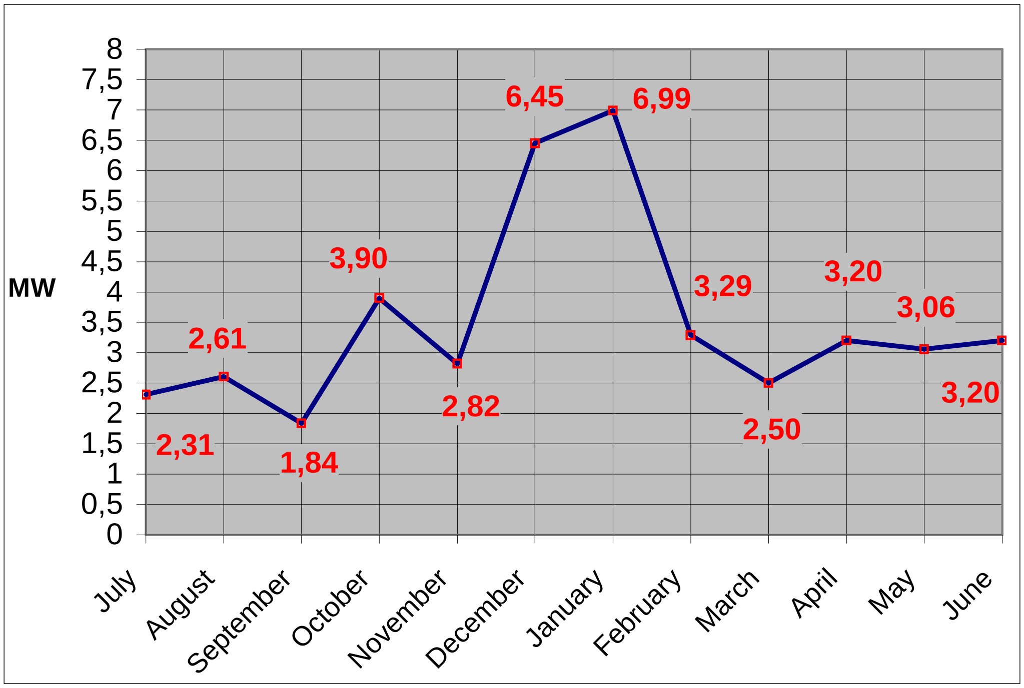


Figure 6.2 Average Electric Demand in MW

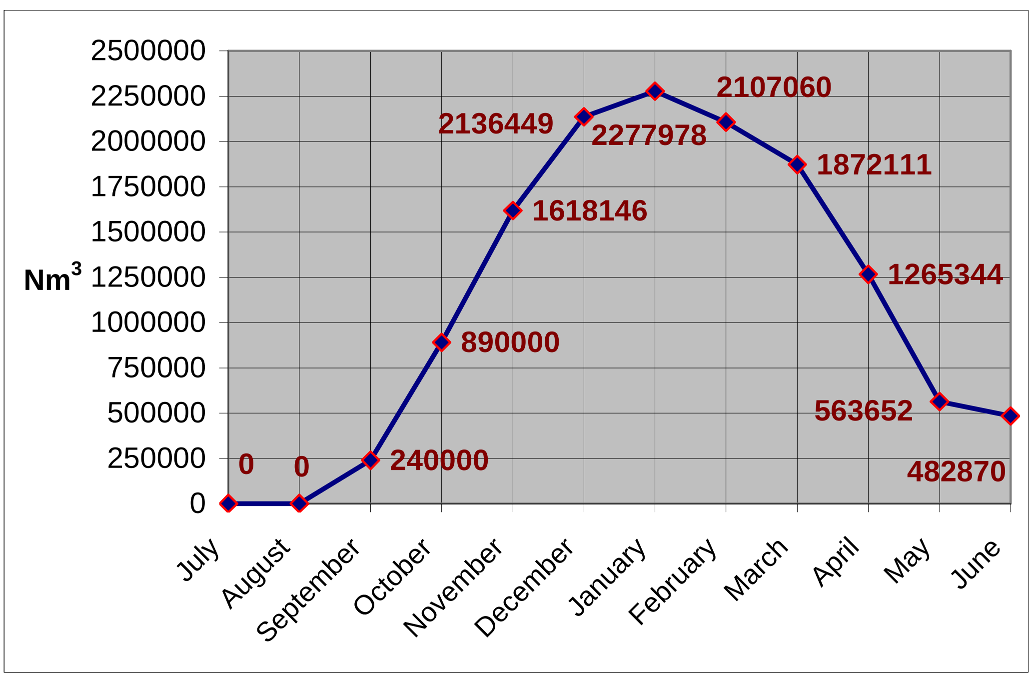


Figure 6.3 Maximum Natural Gas Consumption

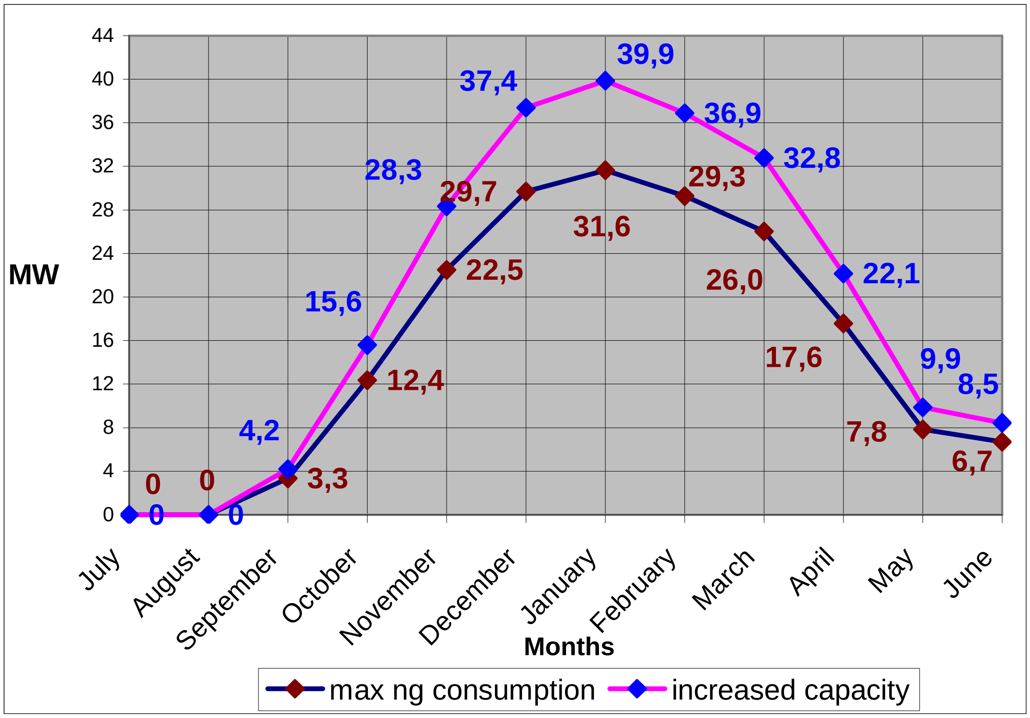


Figure 6.4 Heat Demand in MW

6.1.3. METU Campus Input Data

Table 6.1 Common Input Data for METU Campus

Average Outside Temperature	190 K
Altitude	800 m
Outside Pressure	91.9 kPa
Average Relative Hummidity	60%
Fuel Type	Natural Gas

On the first step, the data which is to be input by the user include the site properties, fuel type, and plant configuration. Table 6.1, shows the common inputs for the case studies at METU campus.

6.1.4. Cases Regarding Cogeneration in METU Campus (Heat and Power Cycle)

Primarily, cogeneration facilities for METU campus are examined, using first a gas turbine and a heat recovery steam generator; second, a gas turbine, a steam turbine and a heat recovery steam generator.

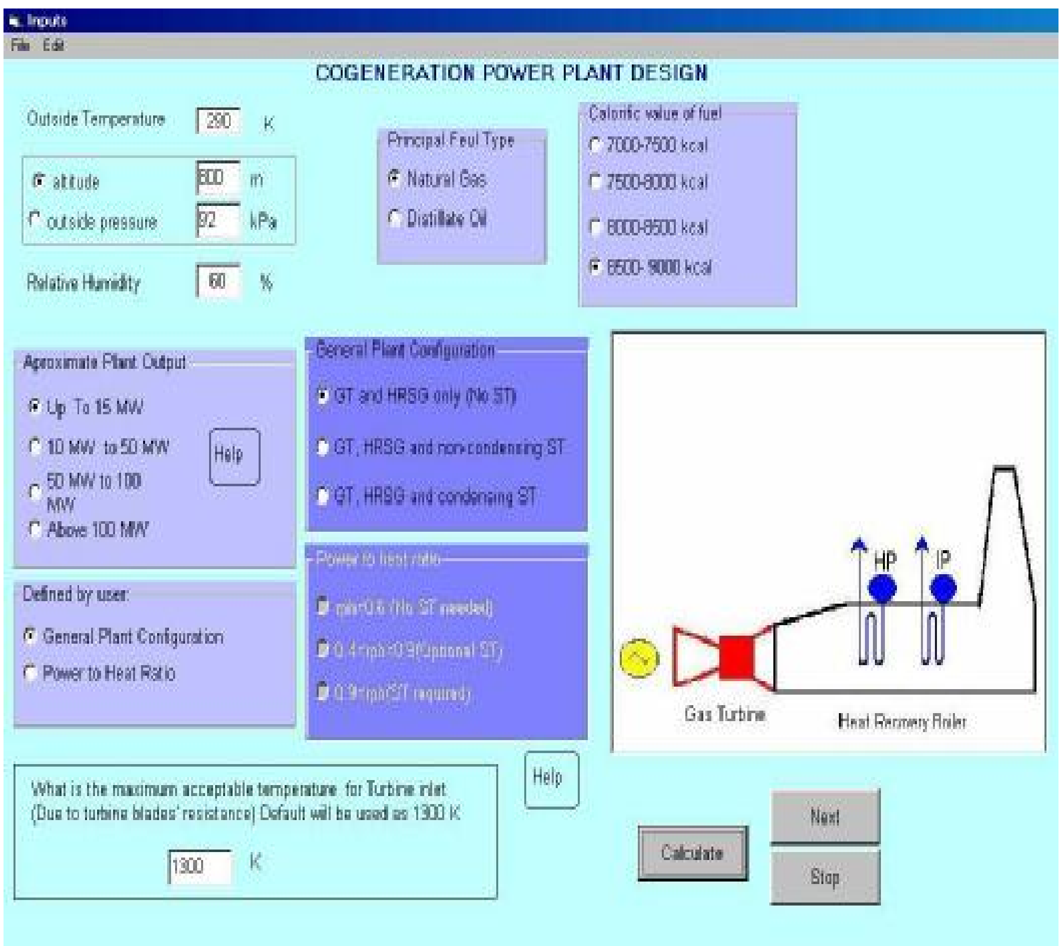


Figure 6.5 Input Form for Cogeneration Plant Design

The inputs for the first form are as shown in the Figure 6.5. Natural gas is chosen as the principal fuel type, with a quite high calorific value corresponding to 35600-37600 kJ/kg. Maximum acceptable turbine inlet temperature is chosen as the default value, (1300 K) which is a good value for a small cogeneration unit.

For determining the approximate plant output, the first help form, shown in Figure 6.6 should be used by inputting data such as the annual electric consumption for METU campus, and defining the operating availability and generation factor. This way, the help form will give the necessary capacity for electric consumption, using average annual demand values. For the heat consumption, it is necessary to define the annual natural gas consumption for the campus, and the average calorific value of the natural gas used in the campus. Then the program will calculate the capacity corresponding to heat consumption.

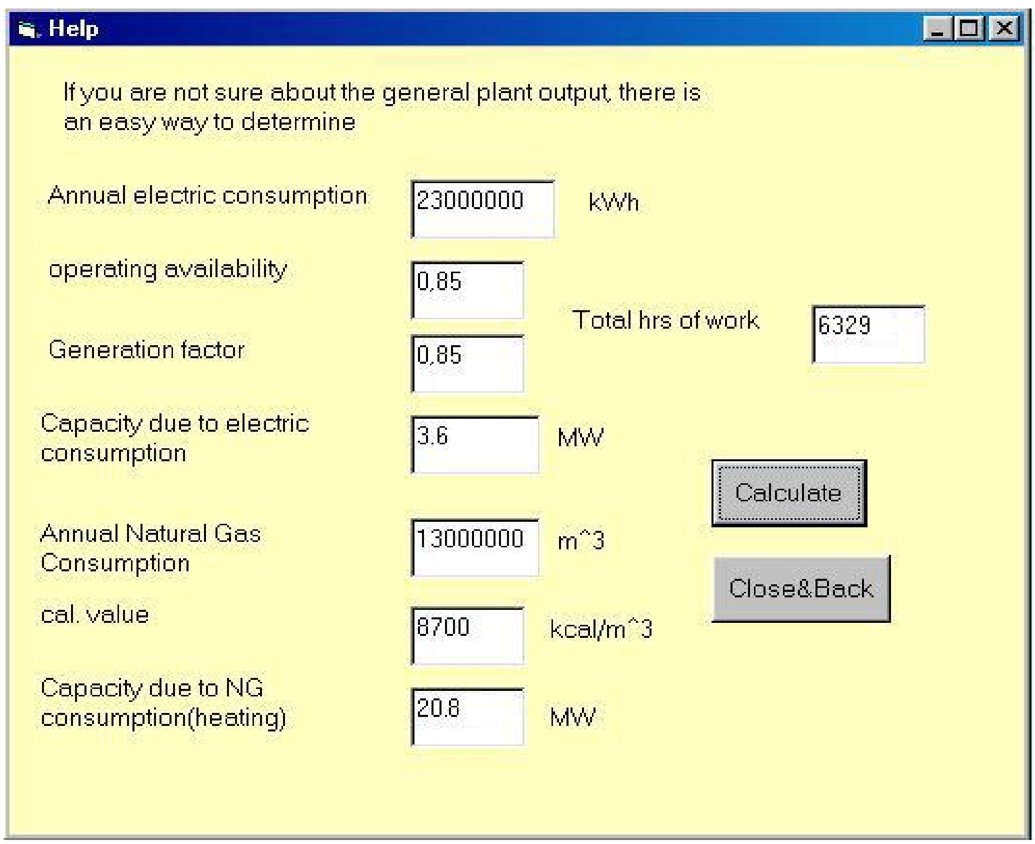


Figure 6.6 Help Form 1

As shown above, the annual average electric consumption of 23 million kWh corresponds to a power capacity of 3.65 MW while the natural gas consumption corresponds to a capacity of 20.8 MW. On the other hand, since the demand differs so much during the year for summer and winter, the annual distribution should be considered. Thus, examining the heat and electric demand curves, it can be seen that, maximum electric need is in January, about 3.7 million kWh during the month, corresponding to a 5.1 MW established power plant; and minimum need is seen on July and August, the summer months, as about 1.2 million kWh, again corresponding to a value of 2 MW of electric power. Also, the natural gas consumptions, and corresponding necessary heat power can be seen on the same figure as maximum heat demand is about 32 MW, corresponds to 20 million kWh during the month of January. There is very little consumption during the summer months.

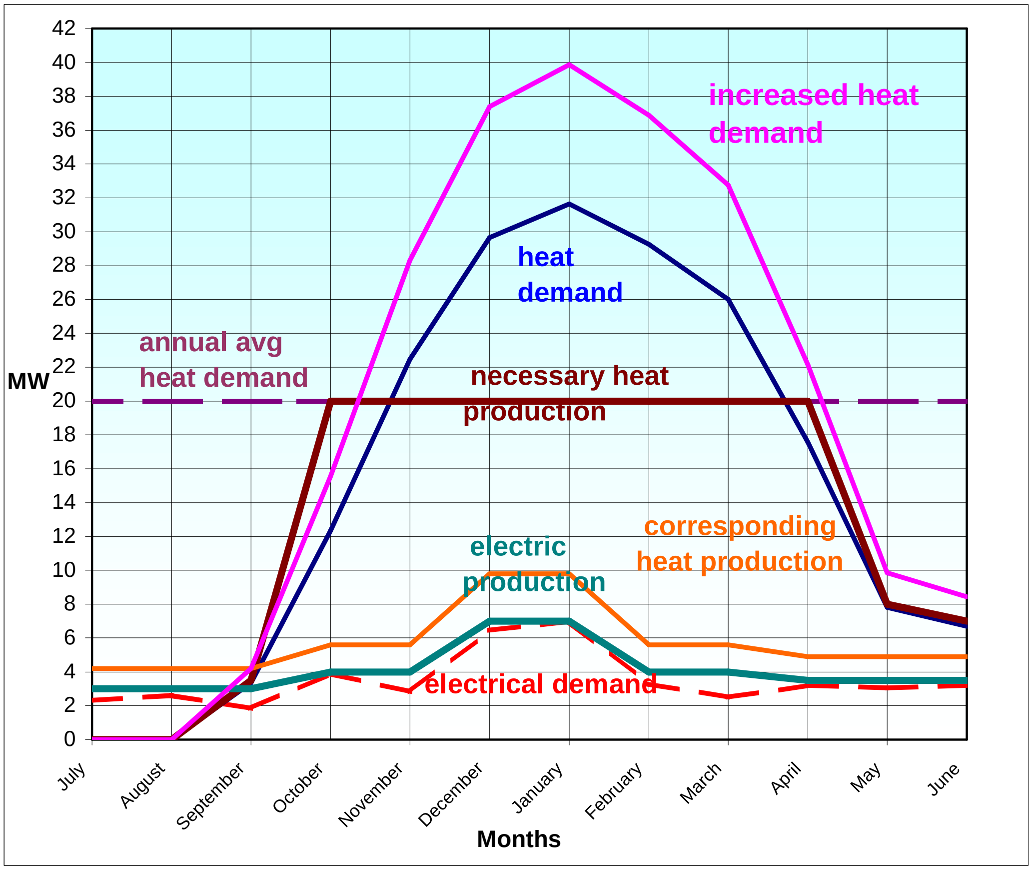


Figure 6.7 Sample Demand and Supply Curves for METU Campus

The data found is examined and sample supply curves are prepared, which can be seen on Figure 6.7. This scenario consists of supplying just the necessary electricity for the campus throughout the year. But on winter months, since heat demand is so much more than which can be supplied, supplementary heating would be necessary; which means burning natural gas directly in the boilers.

6.1.4.1. Cogeneration in METU Campus Without a Steam Turbine

6.1.4.1.1. Introduction, Important Parameters and Design Principles

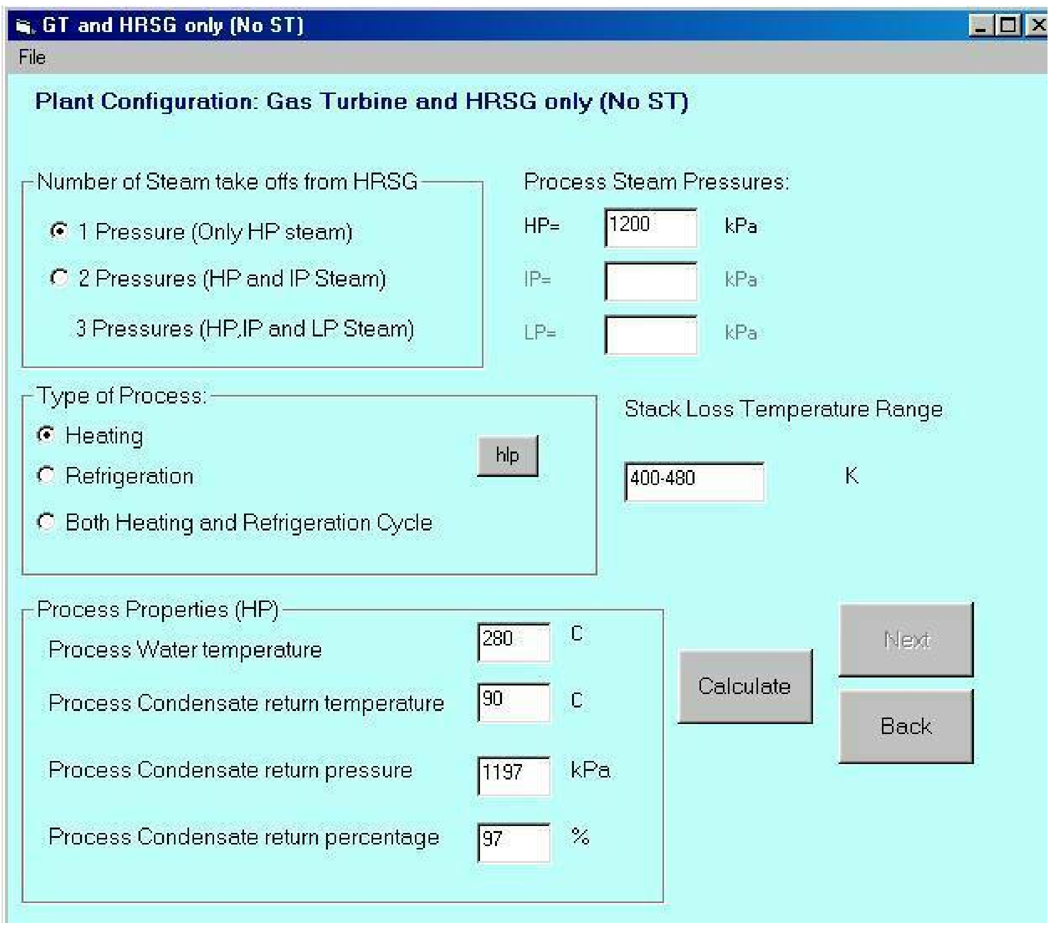


Figure 6.8 Cogeneration Design Program Form-2

First of all, a system with a gas turbine and a HRSG will be chosen for accomplishing this task. After finishing the calculations, “Next” button on Form 1 seen in Figure 6.5 will be clicked, so that the second form (Figure 6.8) will appear. For campus heating, steam at 280°C and 1200 kPa is needed. The program calculates the condensate return pressure as 1198 kPa, and the return percentage is assumed to be %97. It should be noted that, “Up to 15 MW of plant capacity” is chosen on the first form, which is quite below the capacity needed.

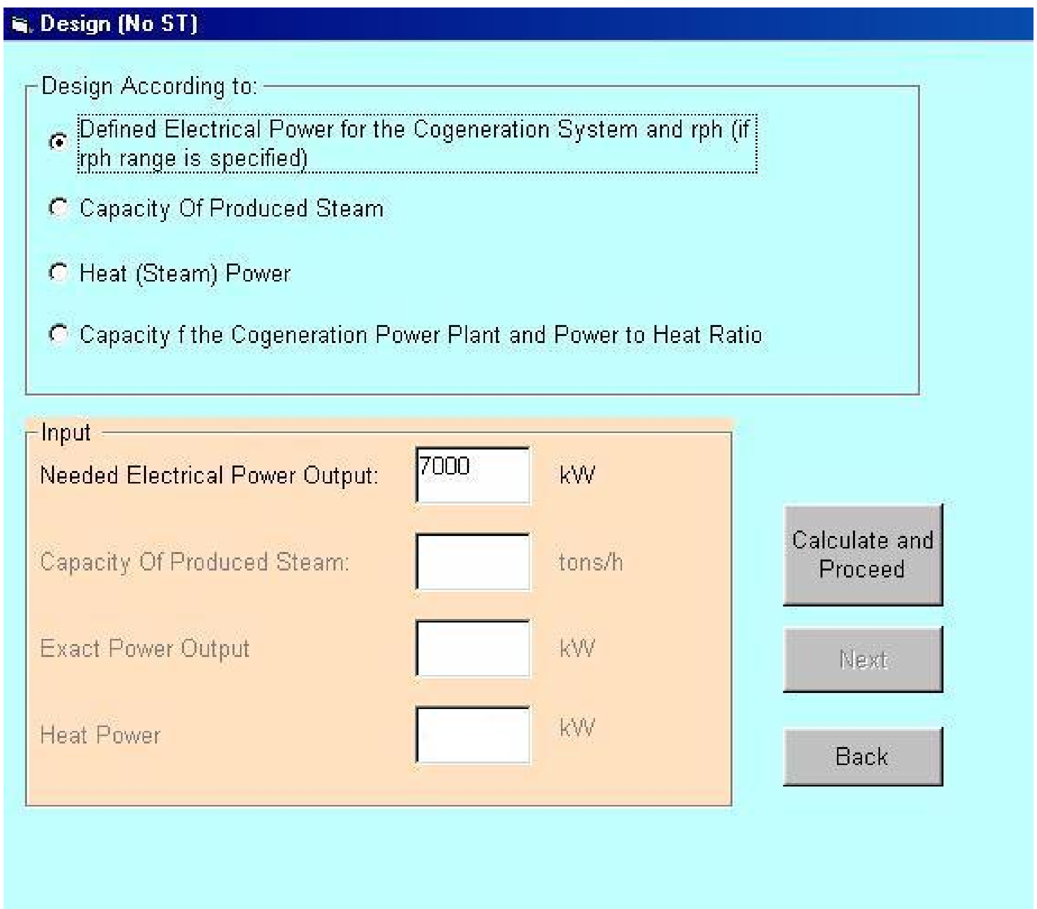


Figure 6.9 Cogeneration Design Program Form-4

After “Calculate” and “Next” buttons are clicked, the design form- Form 4 appears which is given in Figure 6.9. When the first option is chosen, the required electrical power is input as 7 MW according to the supply curve given in Figure 6.2.This corresponds to the highest value on the month of January.

Figure 6.10 Error Box

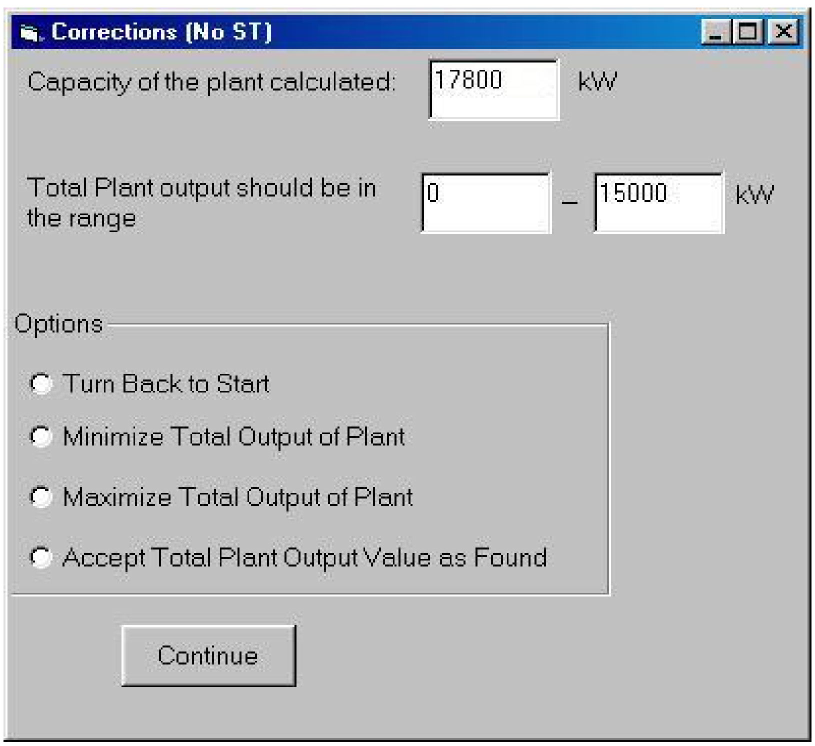


Figure 6.11 Correction and Optimization Form

When “Calculate and Proceed” button is clicked, the error box given in Figure 6.10 will appear, leading to another form- Form 10, which is seen in Figure 6,11. This form is for optimizing a power plant design, based on the user’s inputs, but if the user had inconsistent values and if design of such a plant is impossible, program will give a warning.

As seen from the outputs, to have an electric output of 6.5 MW, a 17.3 MW power plant is required. Now the program asks if the user wishes to return back to start to change the values- to minimize/maximize the total output of the plant, or accept the capacity of the plant as calculated. It is sensible to choose the second option, trying to minimize the total output of the plant, keeping the electrical power as 6.5 MW.

This time, the following error message in Figure 6.12 appears. The following table and the corresponding figure shows the optimization process for the plant design, but no solution can be reached for this case. Thus, choosing “Return Back to Start” option, a higher range for the cogeneration plant capacity should be input.

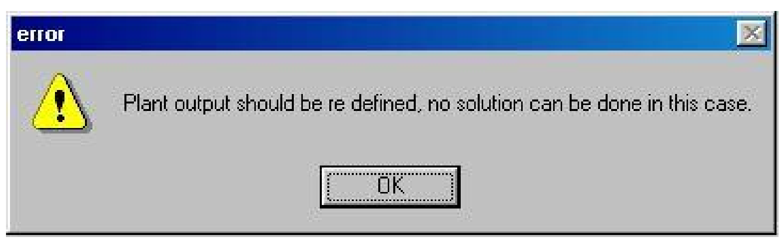


Figure 6.12 Error Form

As can be seen in Table 6.2, and the graphs given in Figures 6.13 and 6.14, increasing pressure ratio and decreasing TIT accomplish a fall in the plant output till the value of 16.2 MW; but further proceeding causes the output to rise again. This means that no solution can be obtained for capacities below 15 MW.

Table 6.2 Important Parameters in Design Calculations For Cogeneration Power Plants

rp (comp. ratio)	TO3 (TIT) K	Capacity of Plant(kW)	Heat recovered (kj/kg)	mtot (air flow rate) m3/s	Ep (kJ/kg) elect.Work
15	1285	17716	323,7	32,8	217,1
15.5	1280	17577	313,5	33,4	213,0
16	1275	17447	303,5	34,1	208,9
16.5	1270	17325	293,9	34,8	204,7
17	1265	17212	284,5	35,5	200,4
17.5	1260	17105	275,4	36,3	196,0
18	1255	17006	266,5	37,1	191,6
18.5	1250	16913	257,8	38,0	187,1
19	1245	16826	249,3	39,0	182,5
19.5	1240	16745	241,1	40,0	178,0
20	1235	16670	233,0	41,0	173,3
20.5	1230	16600	225,0	42,2	168,7
21	1225	16535	217,3	43,4	164,0
21.5	1220	16476	209,7	44,7	159,3
22	1215	16421	202,2	46,0	154,5
22.5	1210	16372	194,9	47,5	149,7
23	1205	16327	187,7	49,1	144,9
23.5	1200	16287	180,7	50,8	140,1
24	1195	16253	173,8	52,6	135,2
24.5	1190	16223	167,0	54,6	130,3
25	1185	16199	160,3	56,7	125,4
25.5	1180	16181	153,7	59,0	120,5
26	1175	16169	147,2	61,5	115,6
26.5	1170	16163	140,8	64,3	110,6
27	1165	16164	134,5	67,3	105,7
27.5	1160	16173	128,3	70,6	100,7
28	1155	16191	122,2	74,3	95,7
28.5	1150	16218	116,1	78,4	90,7
29	1145	16256	110,2	83,0	85,7
29.5	1140	16306	104,3	88,2	80,7
30	1135	16372	98,5	94,0	75,7

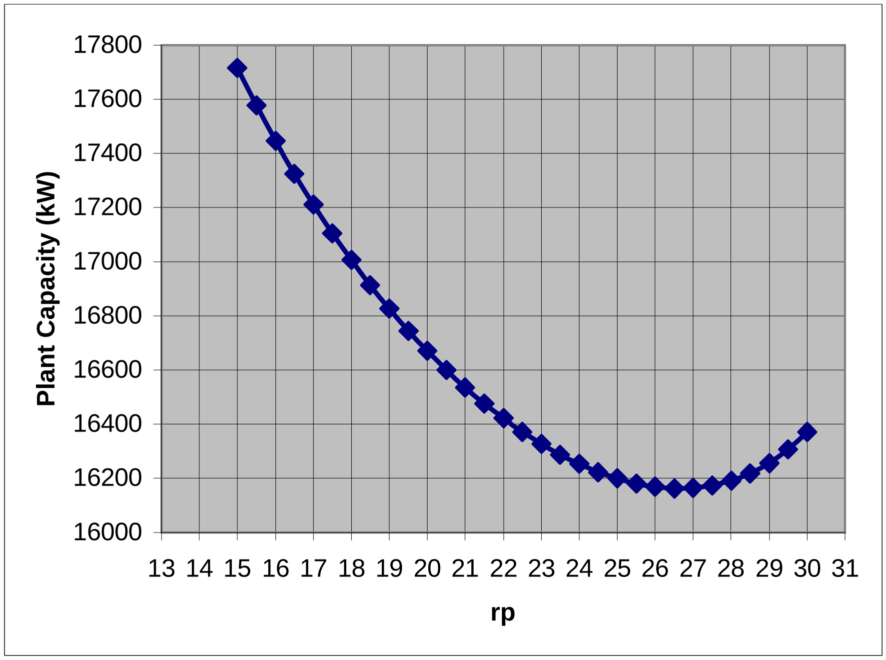


Figure 6.13 Plant Capacity vs Pressure Ratio

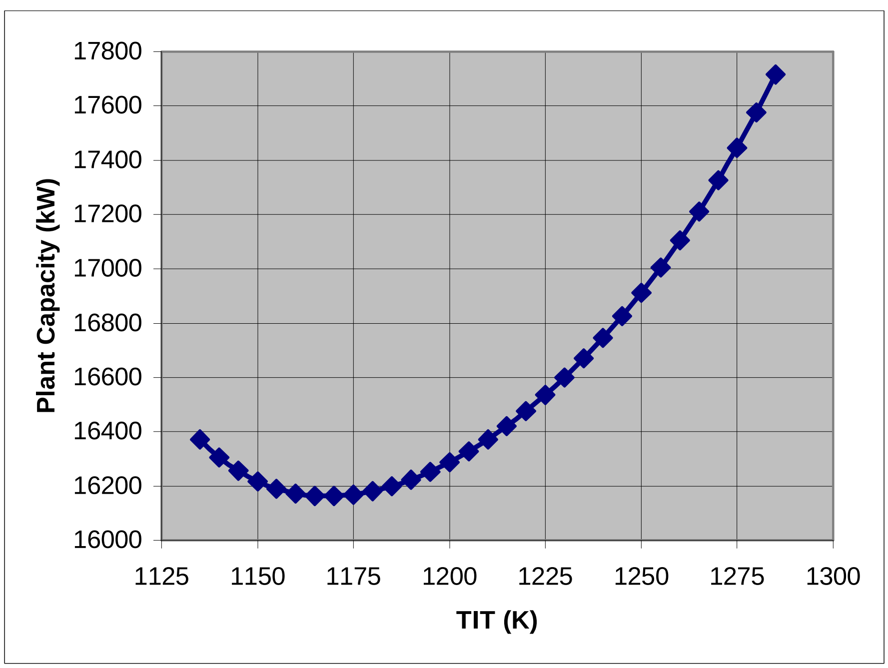


Figure 6.14 Plant Capacity vs TIT

The computer program works in the manner described above with the help of the figures. How the program response to the users actions and how it guides the user by the error and help forms is clearly seen.

Starting over again with the input form, aproximate output between 10 and 50 MWs is chosen. On the design form, when 7000 kW is input for eletrical work output, the following results are obtained:

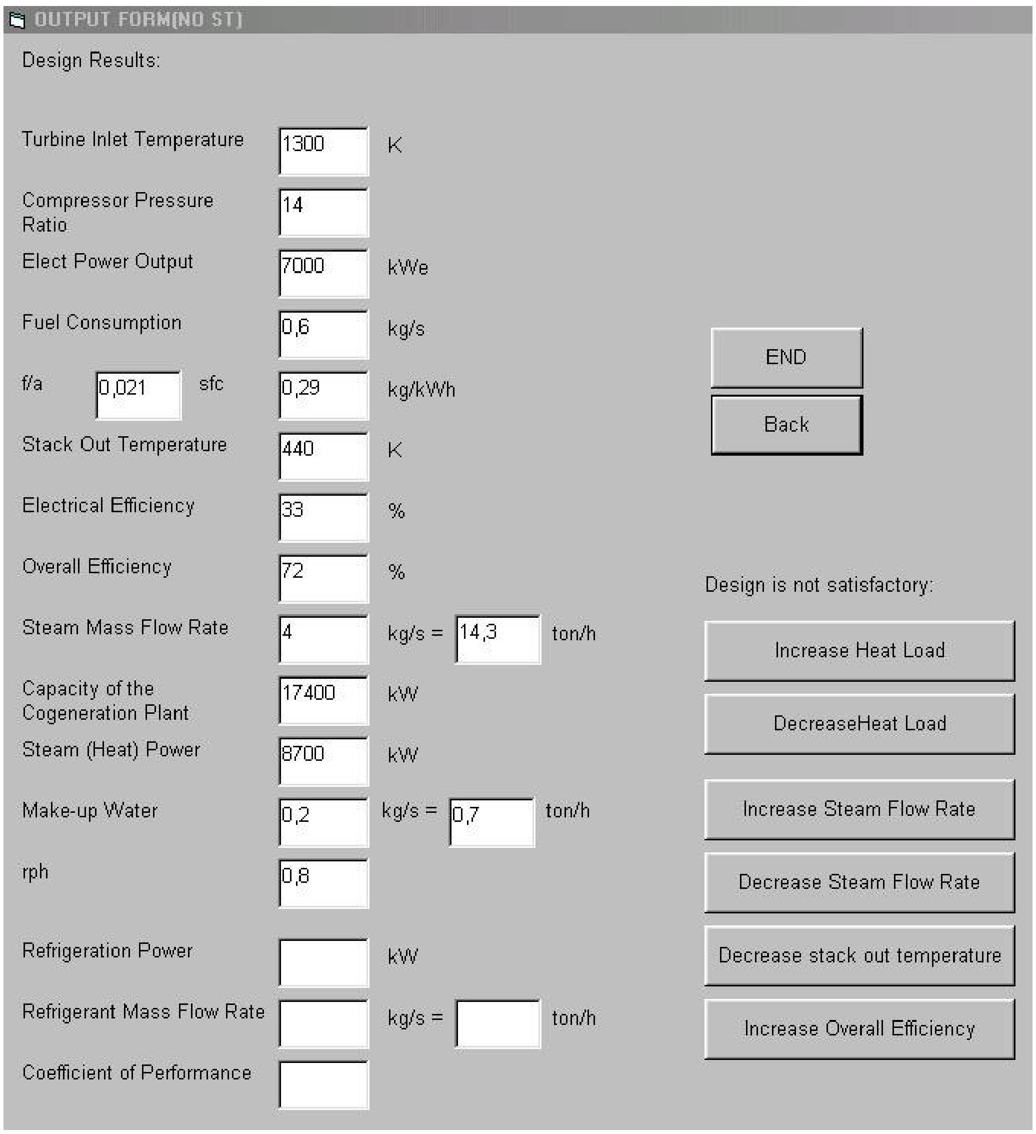


Figure 6.15 Output for 7 MW Gas Turbine

As calculated, not even half of the heat demand is satisfied with 7 MW Gas Turbine. Heat power is found to be 8780 kW_h, as seen in the Figure 6.15.

6.1.4.1.2 Case Study: 1

First case will be two 4MW gas turbines since the average electric demand is 4 MW. This means that, heat production is so much below the existing heat demand of the campus.

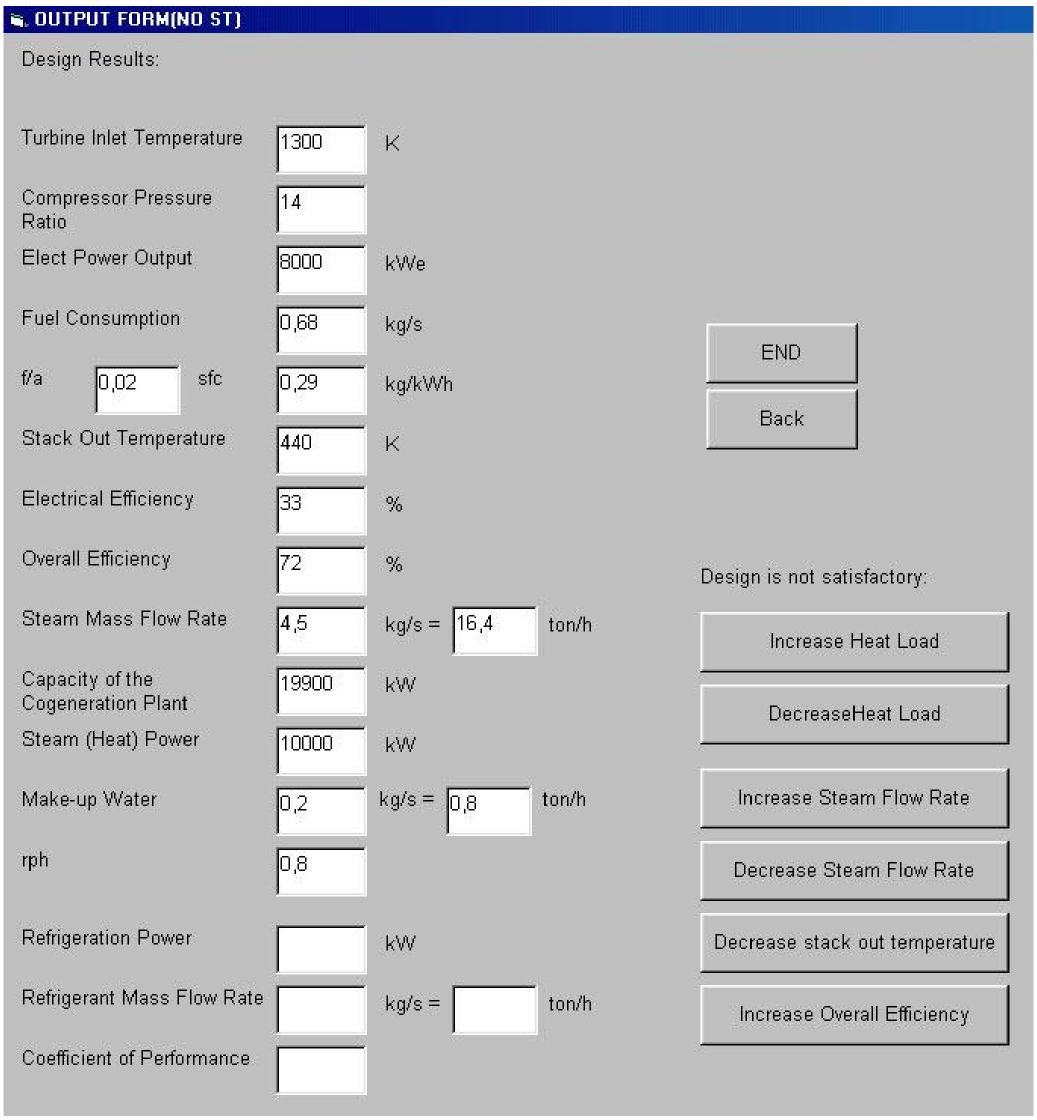


Figure 6.16 Output Form for Case Study:1

In this case, maximum possible heat power output with this system is 10000 kW_h as it is shown in Figure 6.16. Again this is so much below the current heat demand, which is seen in the Figure 6.17.

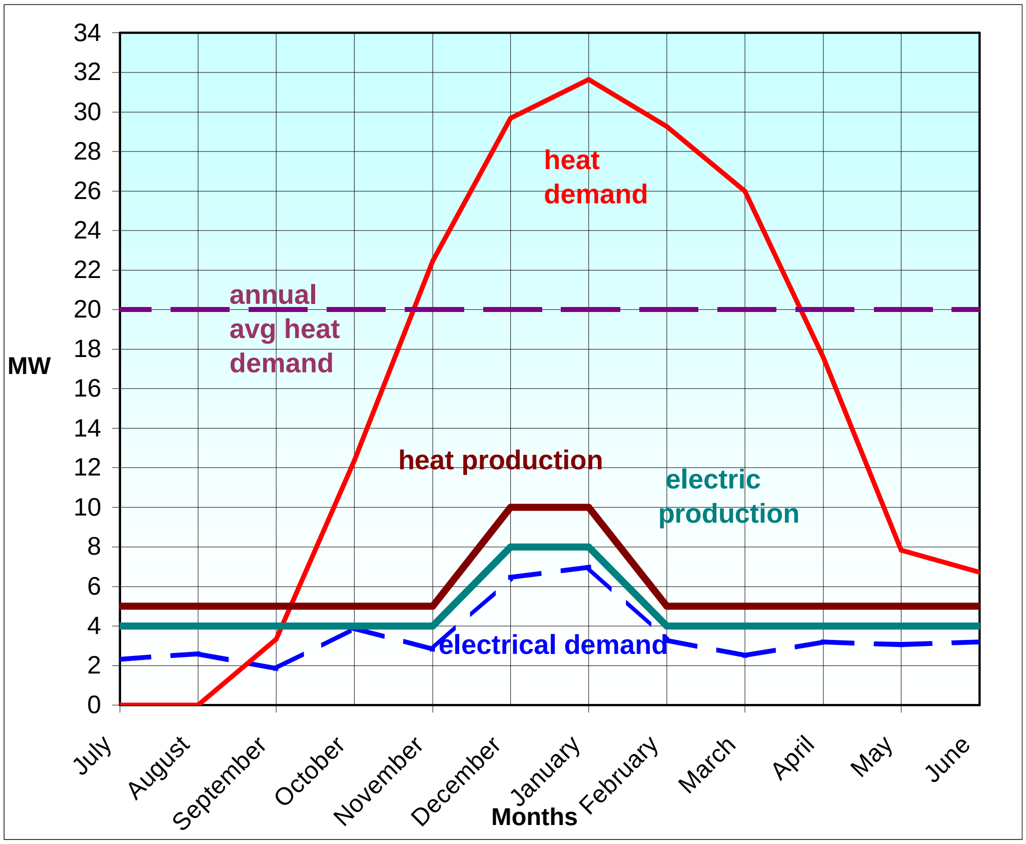


Figure 6.17 Demand and Supply Curves for Case Study: 1

Corresponding gas turbine selection, details of the system parameters, economical analysis and cost summary of design is given in Appendix H, Part 1.

6.1.4.1.3. Finding Electrical Capacity for Maximum Heat Power

For choosing the capacity, electrical power needed for meeting the overall heat load of the campus should be calculated. Thus, the program is ran for calculating this maximum plant capacity, corresponding to 32 MW of heat load, which is input on the design form in Figure 6.18.

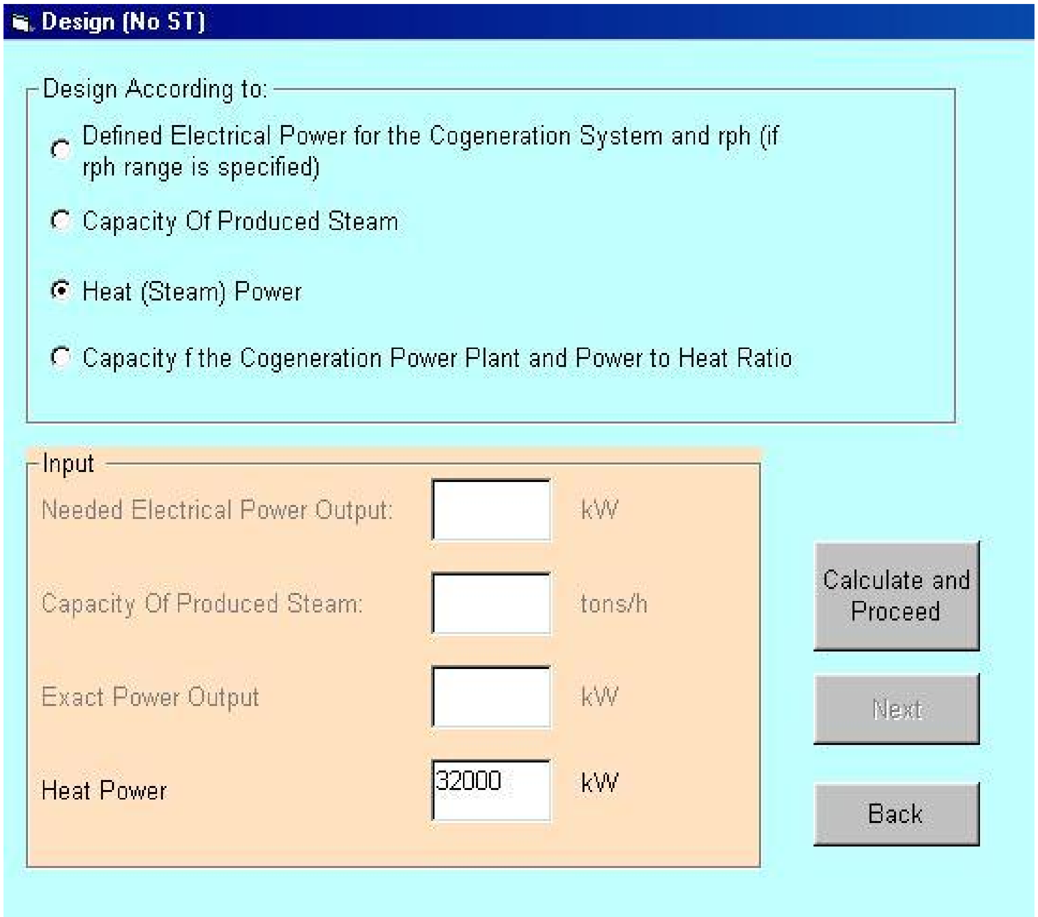


Figure 6.18 Design Form for Finding Electrical Power for Maximum Heat Power Capacity

As can be seen in Figure 6.19, this number is about 23.5 MW. This should be the minimum electrical output of the gas turbine, if additional firing will not be used for further heating on the campus.

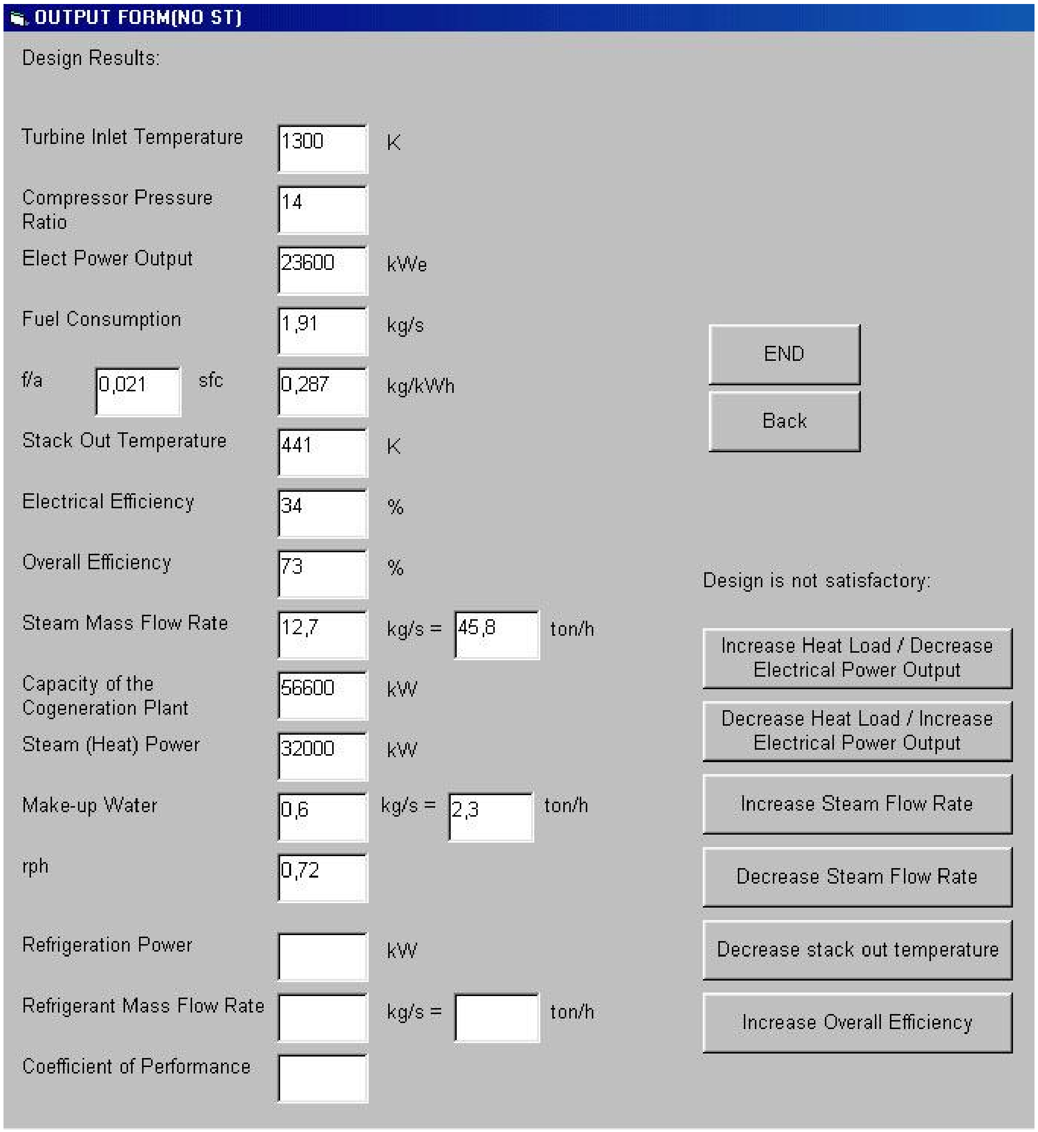


Figure 6.19 Output Form for Finding Electrical Power for Maximum Heat Power Capacity

6.1.4.1.4. Finding Electrical Power for Maximum Steam Flow Rate

There is another way to find the plant output and corresponding electrical capacity. In the design form (Figure 6.18), capacity of produced steam is chosen to be 55 ton/h. After this is input the “Calculate” and “Next” buttons are clicked. The output form appearing is given in Figure 6.20 below.

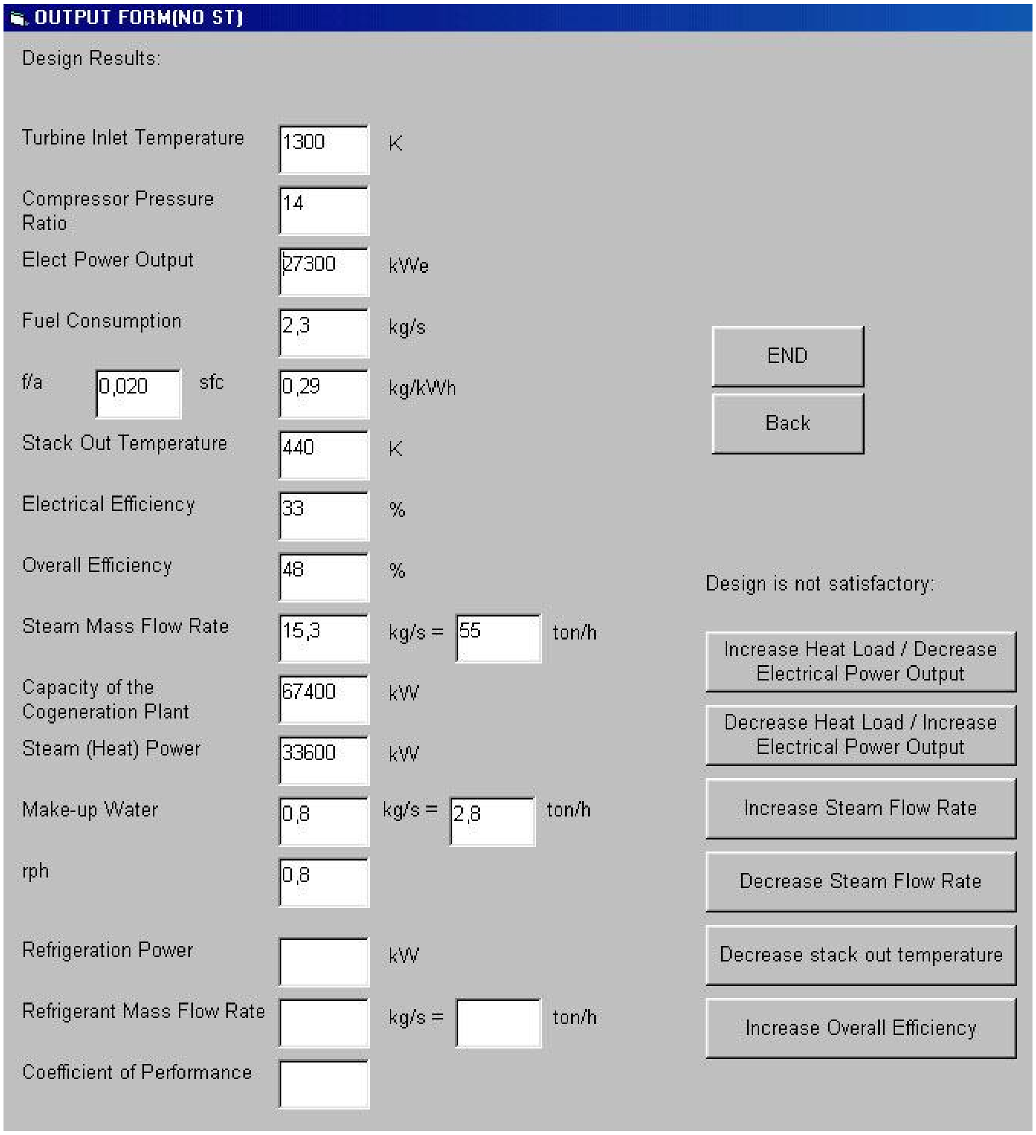


Figure 6.20 Output Form for Finding Electrical Power for Maximum Steam Flow Rate

6.1.4.1.5 Case Study: 2

In the second case two of 12 MW gas turbines are used, making up to 24 MW totally. When full capacity is used, it supplies all the heat necessary, with 16 MW excess electricity, and in summer and spring months, only one gas turbine is to be operated which will supply enough heat for the campus. The corresponding outputs for the program can be seen in Figure 6.21, and the demand-supply relationship is as given in Figure 6.22.

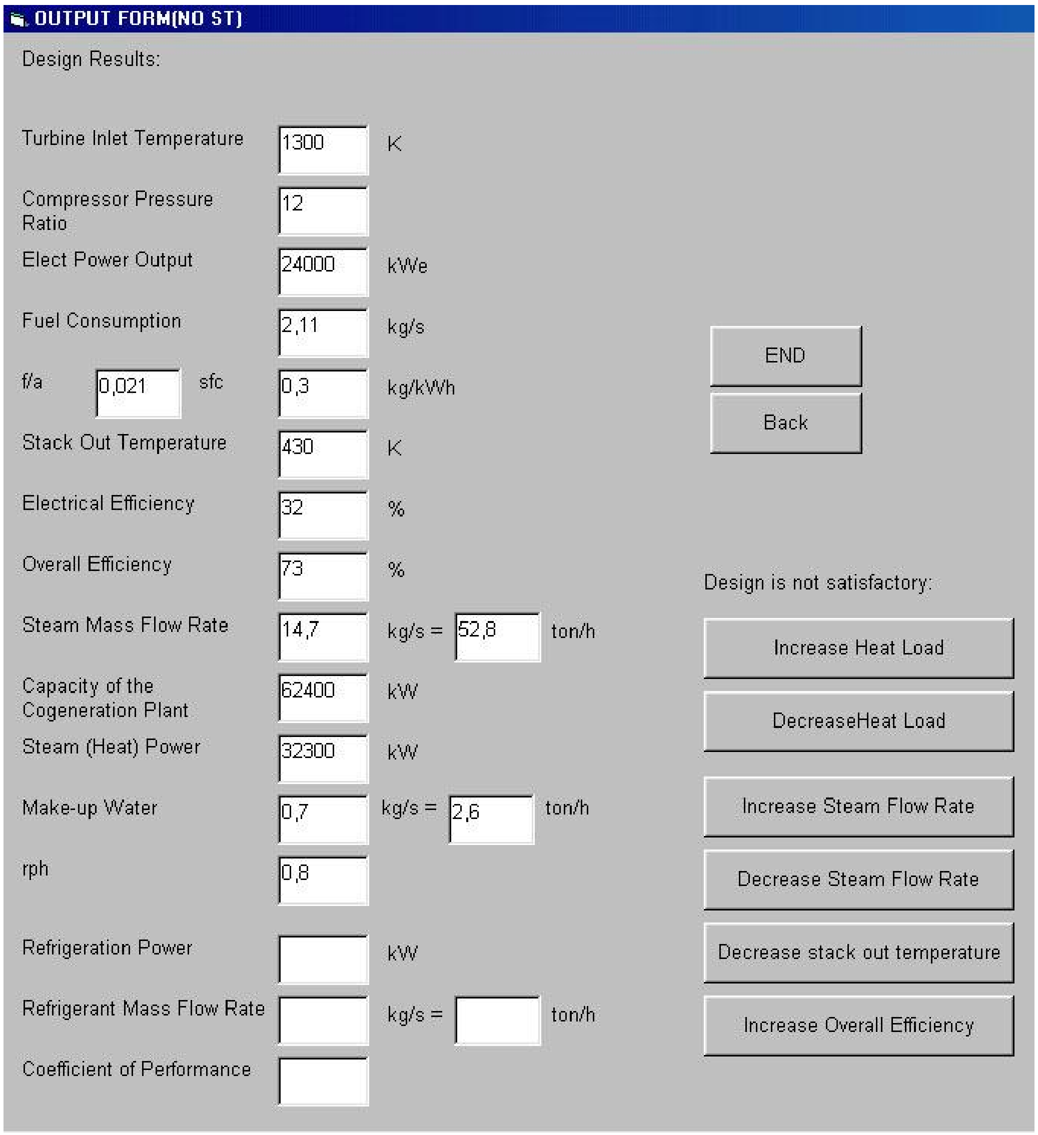


Figure 6.21 Output Form for Case Study: 2

Corresponding gas turbine selection, details of the system parameters, economical analysis and cost summary of design is given in Appendix H, Part2.

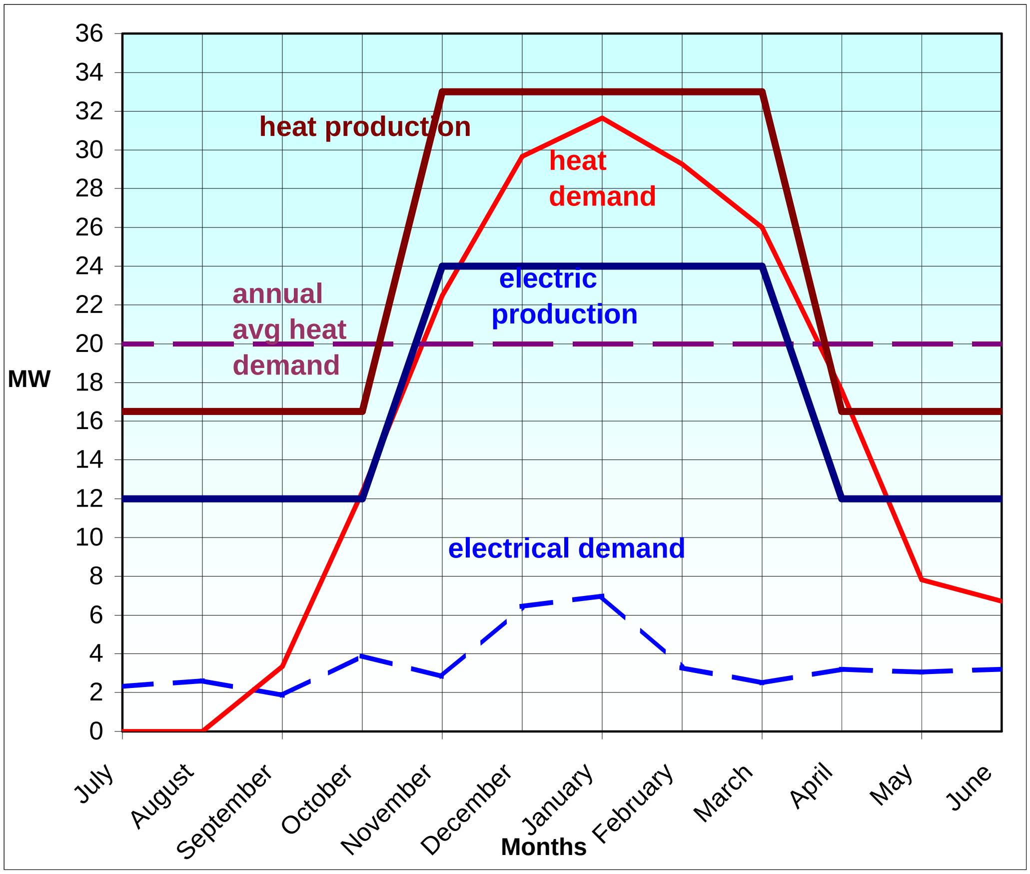


Figure 6.22 Demand and Supply Curves for Case Study: 2

6.1.4.1.6 Case Study: 3

For the third case, one 12 MW gas turbine is to be used with extra firing of natural gas for heating during coldest months. Since heat power is slightly above 15 MW, extra firing may vary from 10 to 30 tons/h. There is excess elecricity of about 5 MW minimum. Output form for this configuration is in Figure 6.23, and the demand-supply curves can be found in Figure 6.24.

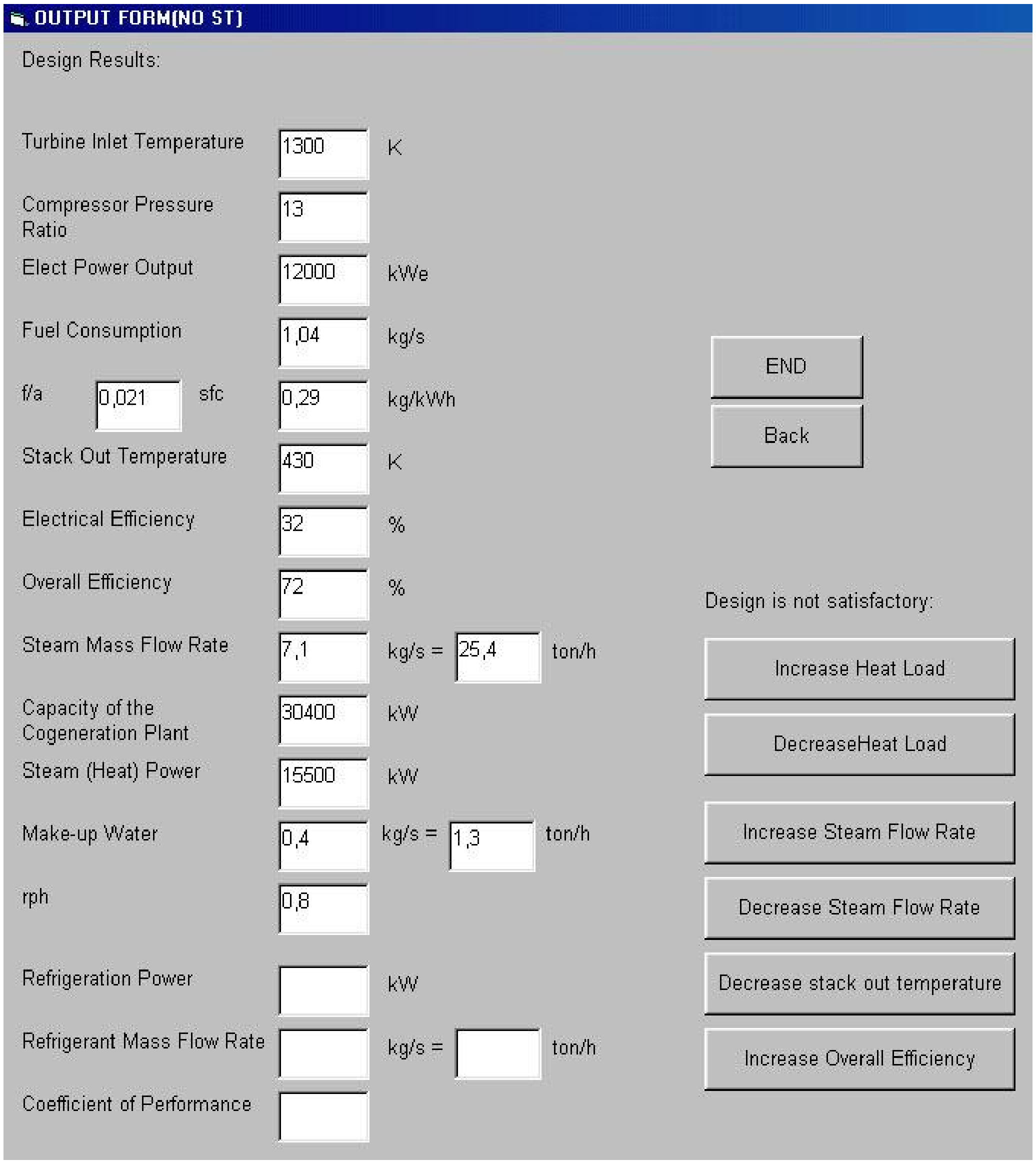


Figure 6.23 Output Form for Case Study: 3

Corresponding gas turbine selection, details of the system parameters, economical analysis and cost summary of design for Case Study:3 is given in Appendix H, Part 3.

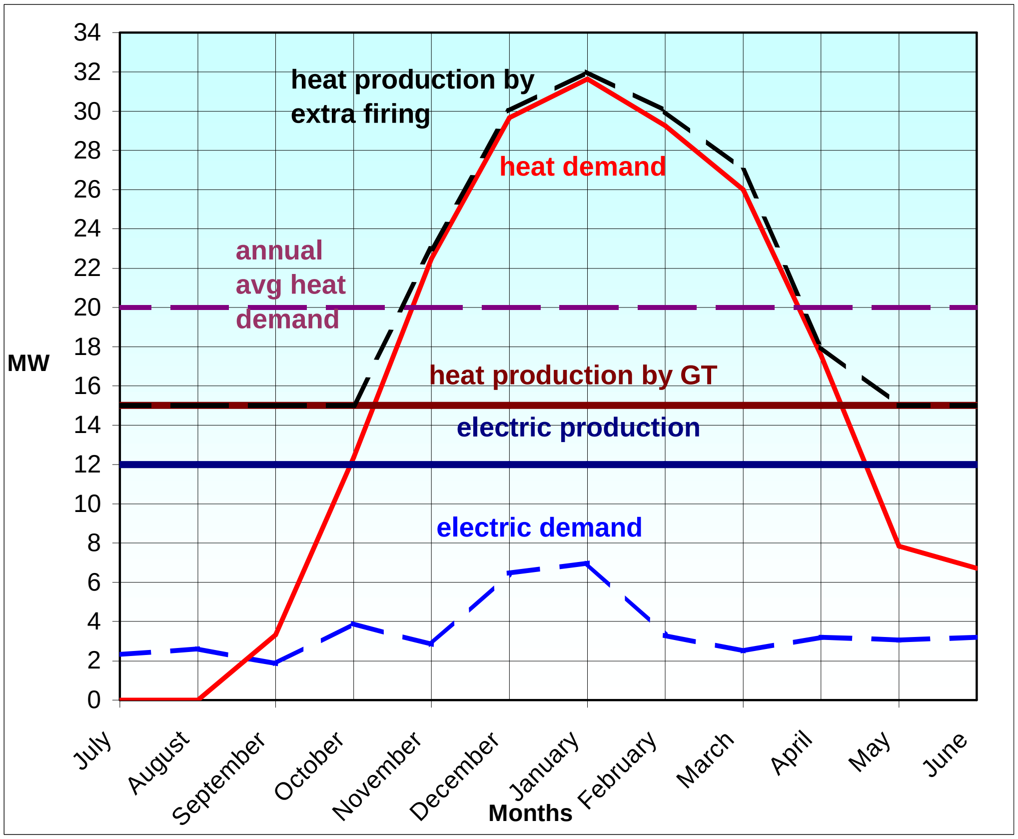


Figure 6.24 Demand and Supply Curves for Case Study: 3

6.1.4.1.7 Case Study: 4

For the last case with heat and power cycle applications, increased capacity to 39 MW_h will be studied. Total gas turbine power about 29 MW, which is in fact higher than most of the cases, will be produced by two 14.5 MW gas turbines for Case: 4_1 and three 10 MW gas turbines for Case: 4_2. Outputs for both configurations can be found in Figure 6.25. Corresponding demand and supply curves are given in Figure 6.26 for case:4_1 and in Figure 6.27 for Case: 4_2.

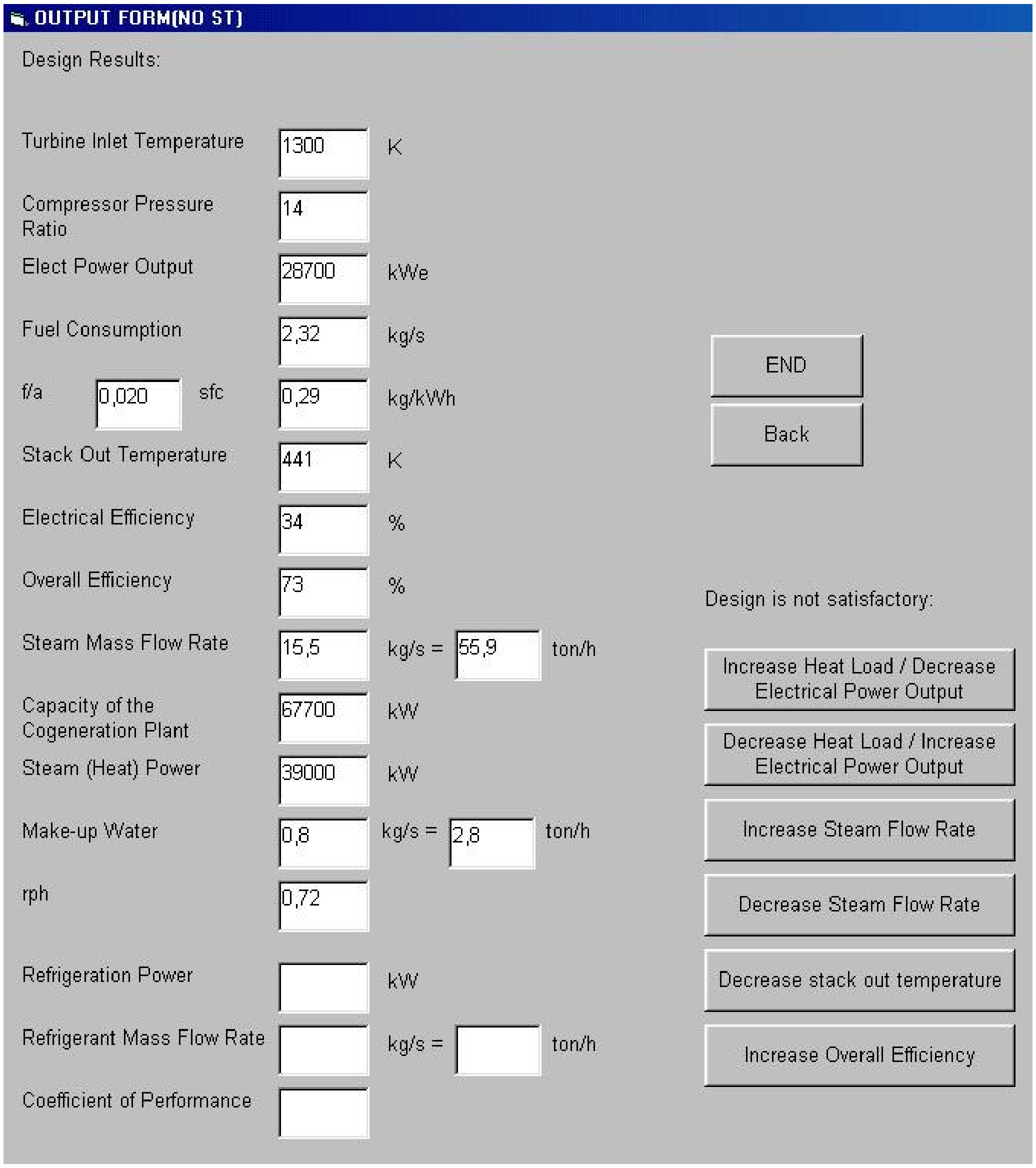


Figure 6.25 Output Form for Case Study: 4

Gas turbine selection, details of the system parameters, economical analysis and cost summary of the design is given in Appendix H, Part 4.

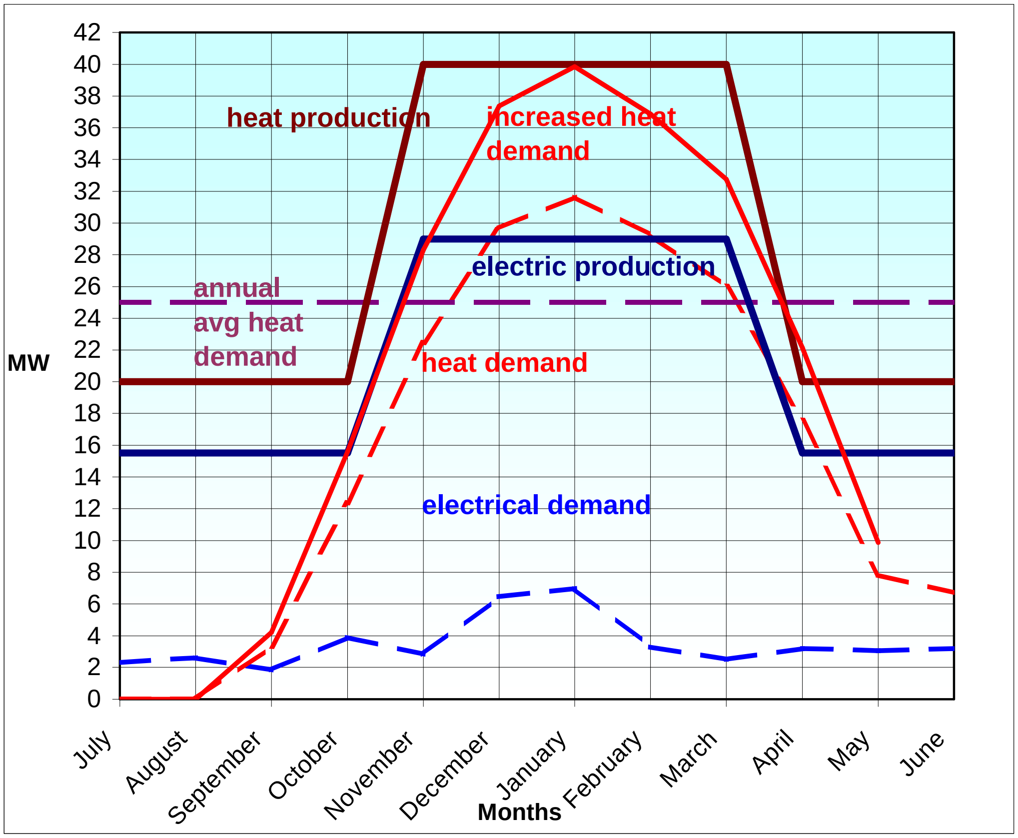


Figure 6.26 Demand and Supply Curves for Case Study: 4_1

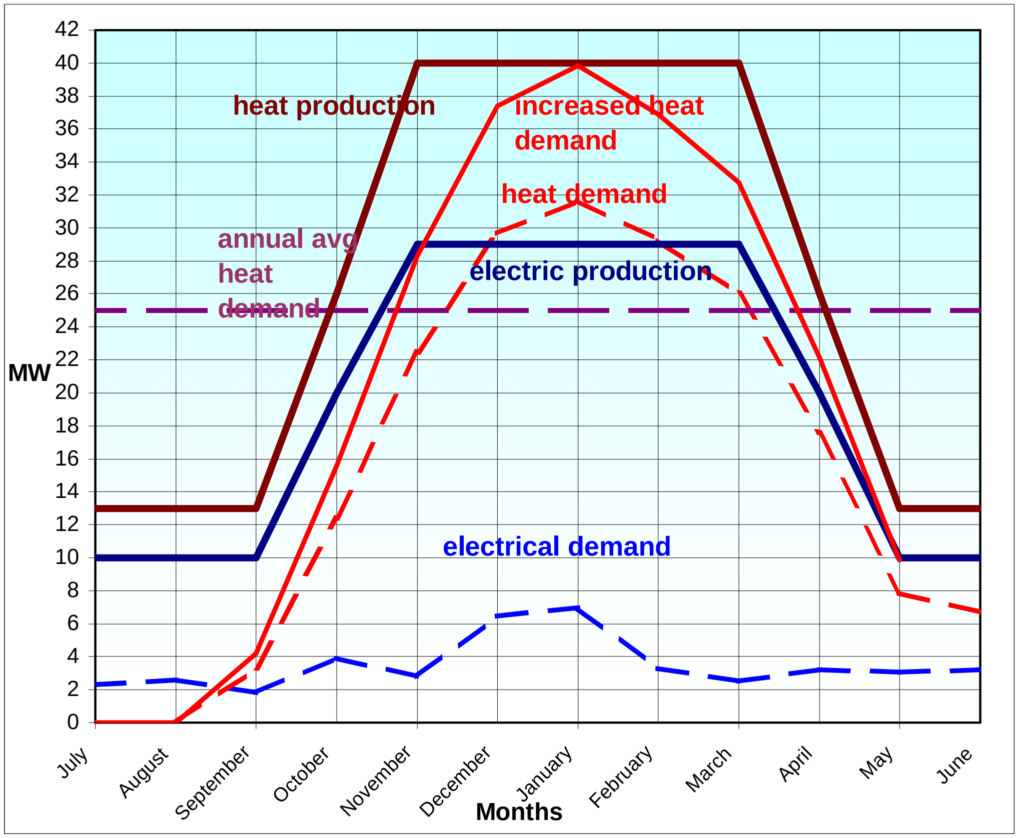


Figure 6.27 Demand and Supply Curves for Case Study: 4_2

6.1.4.2. Cogeneration in METU Campus With Steam Turbine

6.1.4.2.1. Case Study: 5

For the first case of combined cycle application for cogeneration with steam turbine, 12 MW total power may be used with extra firing and burning of natural gas directly for heating during coldest months, to supply maximum 35 ton/h additional steam. The corresponding design and output forms of the program can be seen in Figures 6.28 and 6.29 respectively.

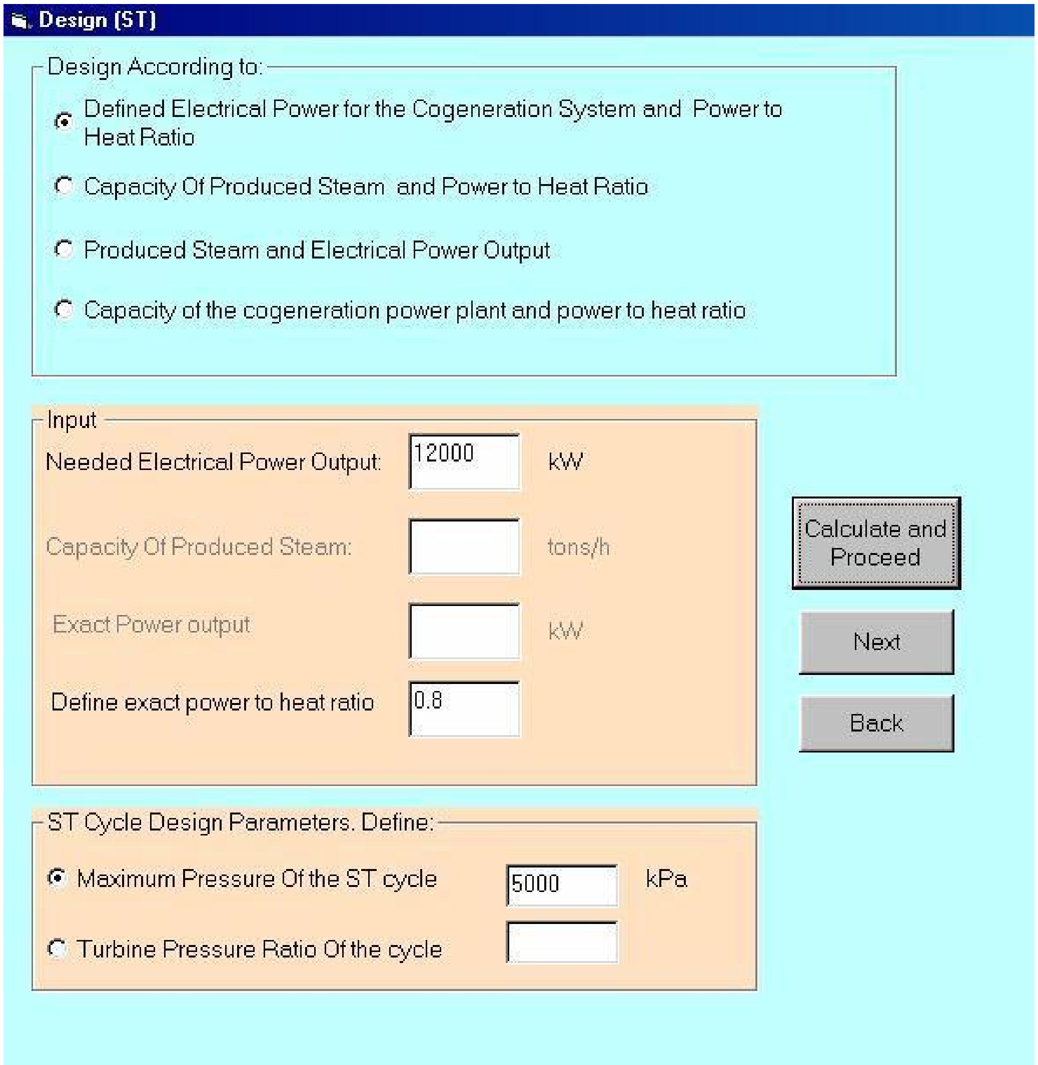


Figure 6.28 Design Form for Case:5

For this case, to decrease the electric output of steam turbine, a higher value for r_{ph} can be input, or simply clicking the "Increase Gas Turbine/ Decrease Steam Turbine Output" button, this can be accomplished. Then, the following outputs given in Figure 6.30 will appear.

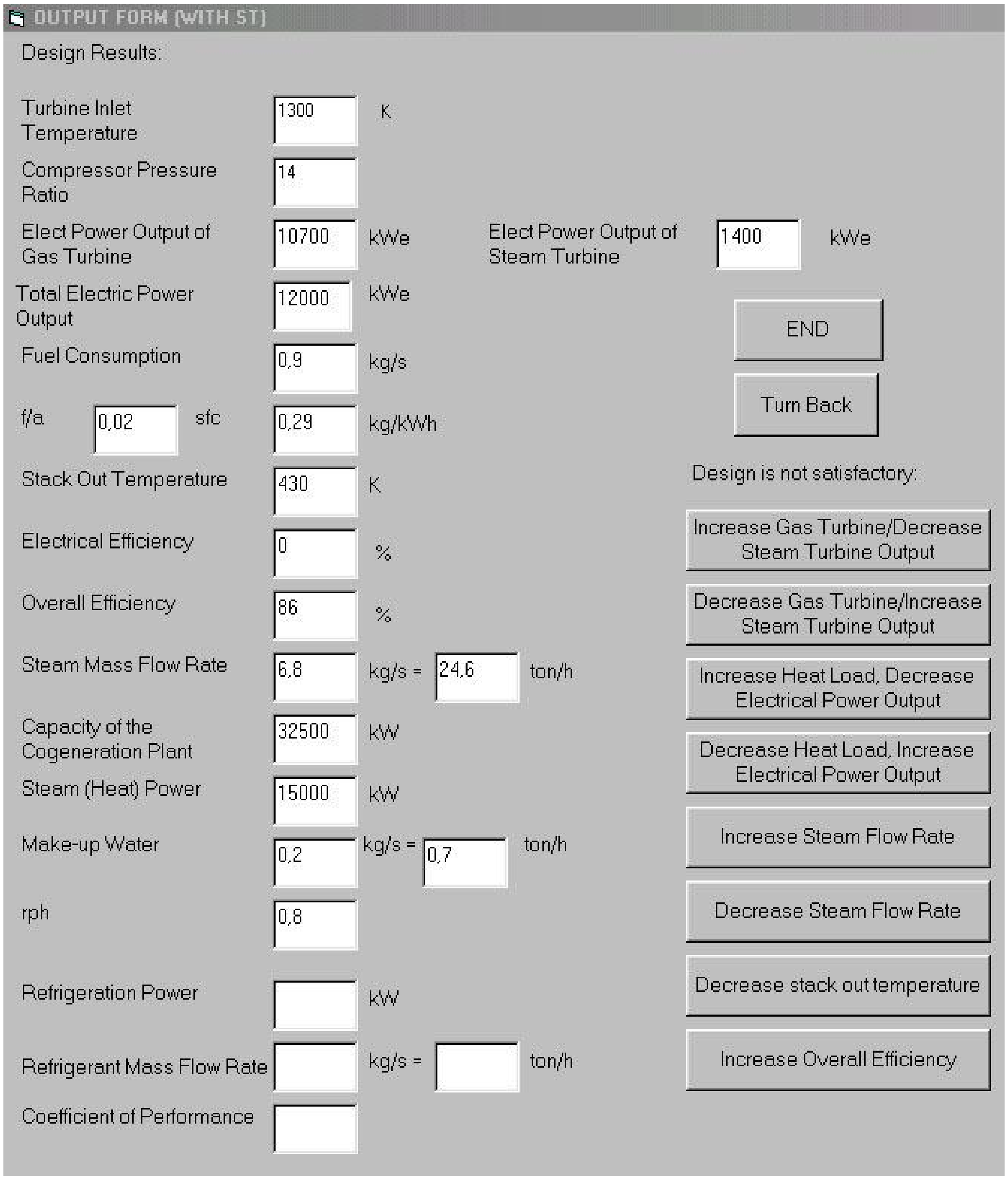


Figure 6.29 Output Form for Case Study: 5

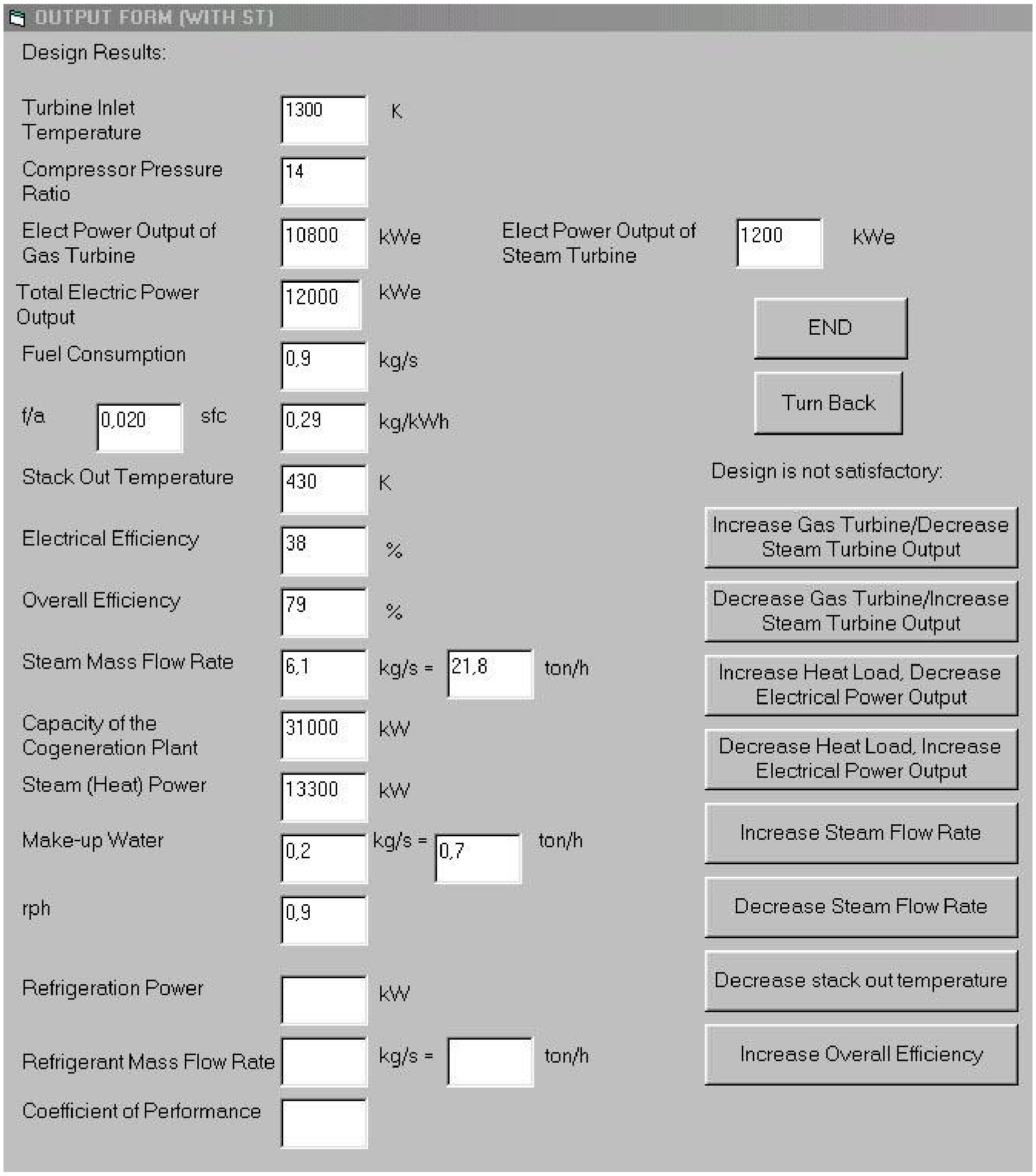


Figure 6.30 Output Form for Case Study: 5 After Re-design

Demand and supply curves are shown in Figure 6.31. Corresponding gas turbine and steam turbine selection, details of the system parameters, economical analysis and cost summary of the design is given in Appendix H, Part 5.

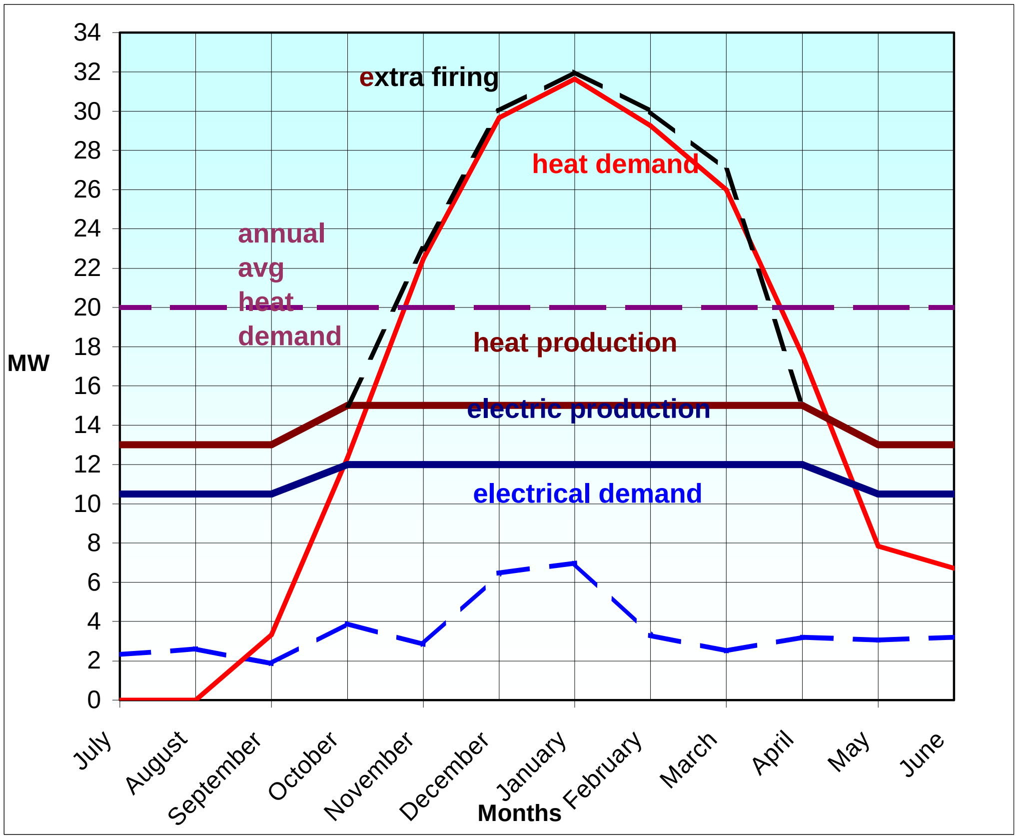


Figure 6.31 Demand and Supply Curves for Case Study: 5

6.1.4.2.2. Case Study: 6

Two 9 MW gas turbines and 2 MW steam turbine are utilized in the 6th case study, which means totally about 20 MW electric power. The steam turbine run when more electrical power is required, and can be stopped when more steam is required for heating. Only one gas turbine may be operated in summer and spring months. There will be some extra firing necessary.

The output is in Figure 6.32. Corresponding demand and supply curves can be found in Figure 6.33.

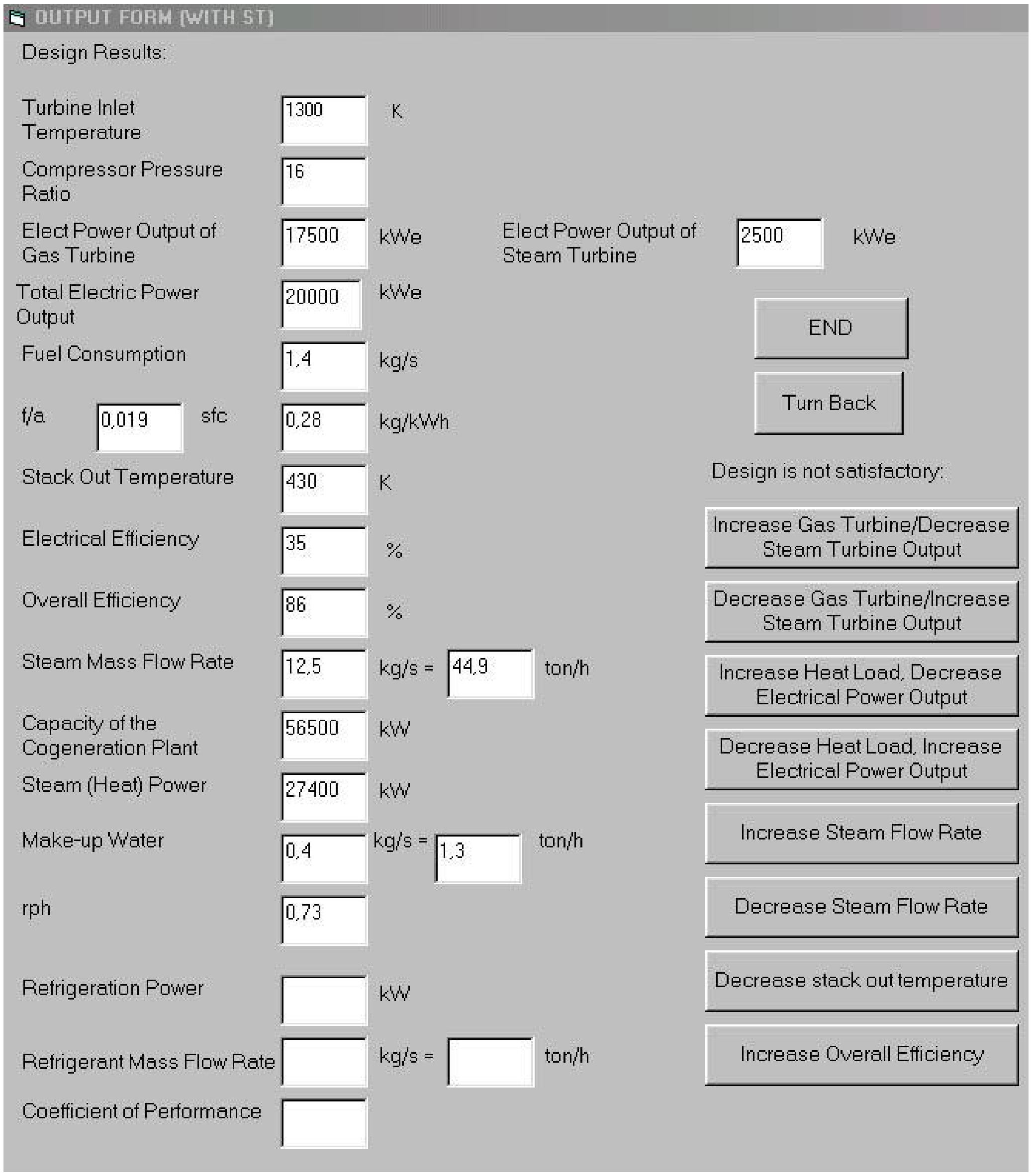


Figure 6.32 Output Form for Case Study: 6

Gas turbine and steam turbine selection, details of the system parameters, economical analysis and cost summary of design for Case Study: 6 are given in Appendix H, Part 6.

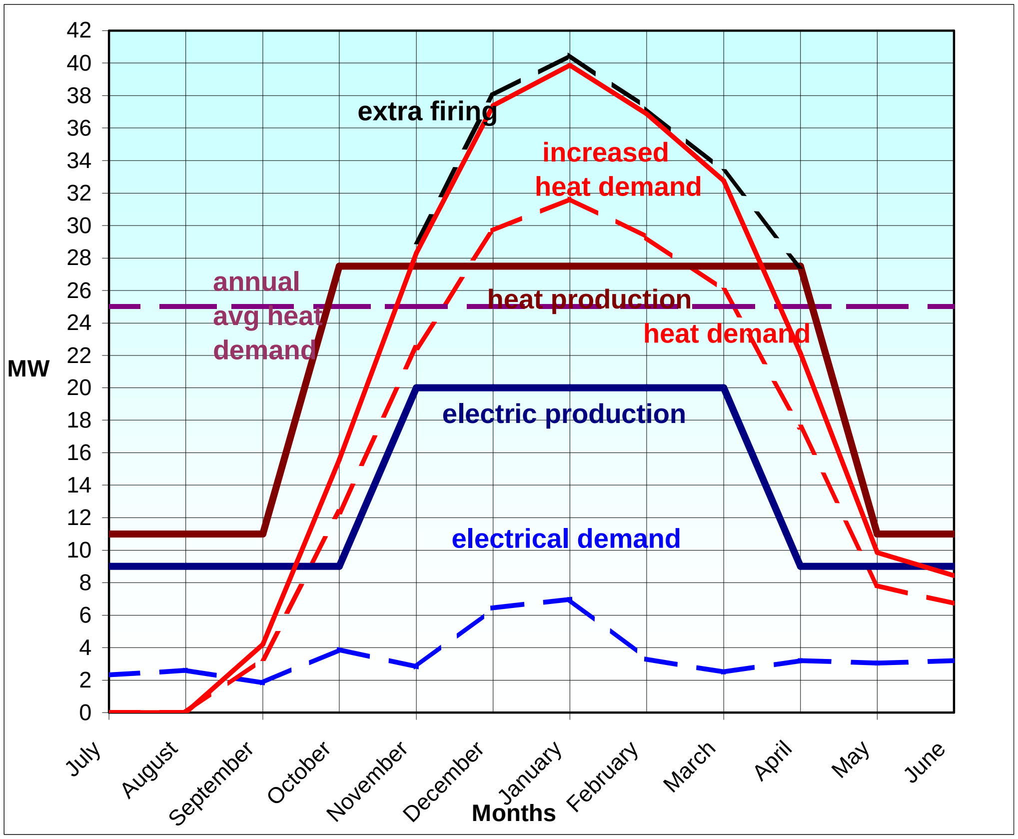


Figure 6.33 Demand and Supply Curves for Case Study: 6

6.2. Cases Regarding Trigeneration in METU Campus (Heat, Power and Refrigeration Cycle)

6.2.1. Case Study: 7

This case is of 12 MW gas turbine total power with extra firing and burning of NG directly for heating during coldest months. This time, a refrigeration unit will be used for cooling or ice making. During cold months, refrigeration capacity will be decreased, while during summer months, 8 MW refrigeration power can be produced for obtaining low temperatures.

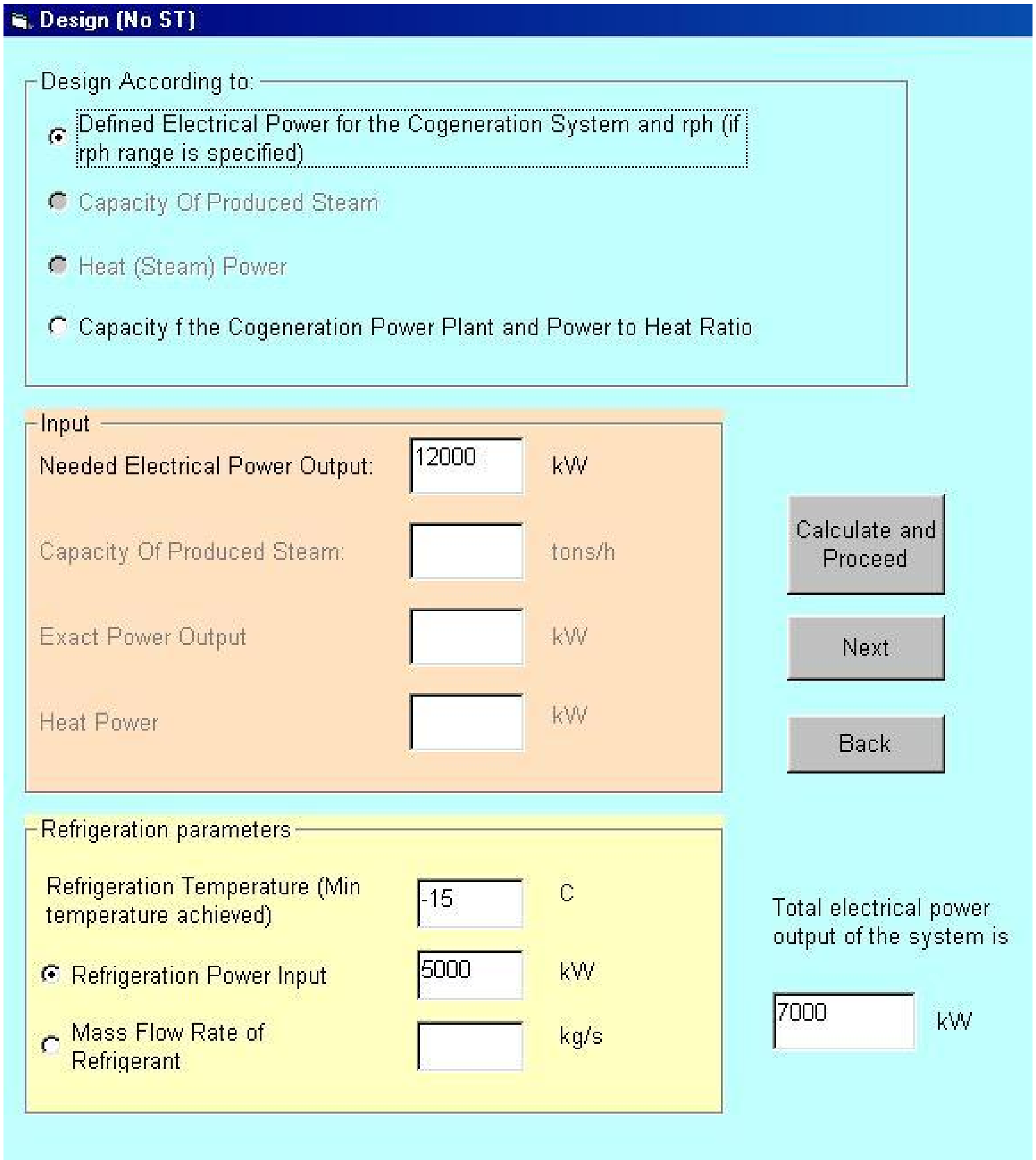


Figure 6.34 Design Form for Trigeneration Without Steam Turbine, Case Study: 7

Design form for this case is given in Figure 6.34 above. Useful electrical output for the system is 7 MW as can be seen.

The corresponding outputs of the program, for the above case are in Figure 6.35. Demand and supply curves can be found in Figure 6.36. Gas turbine selection, details of the system parameters, economical analysis and cost summary of design is given in Appendix H, Part 7.

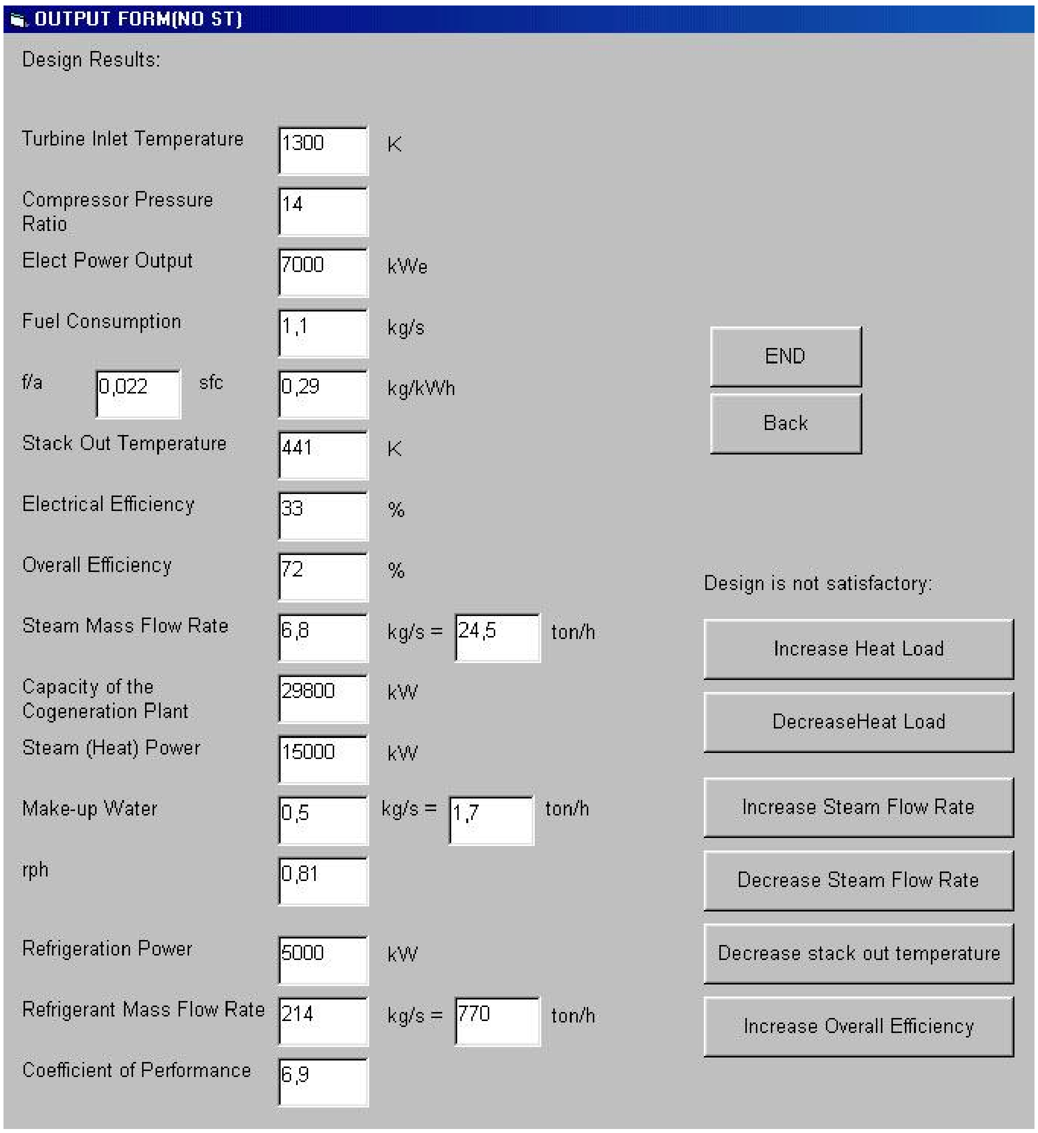


Figure 6.35 Output Form for Case Study: 7

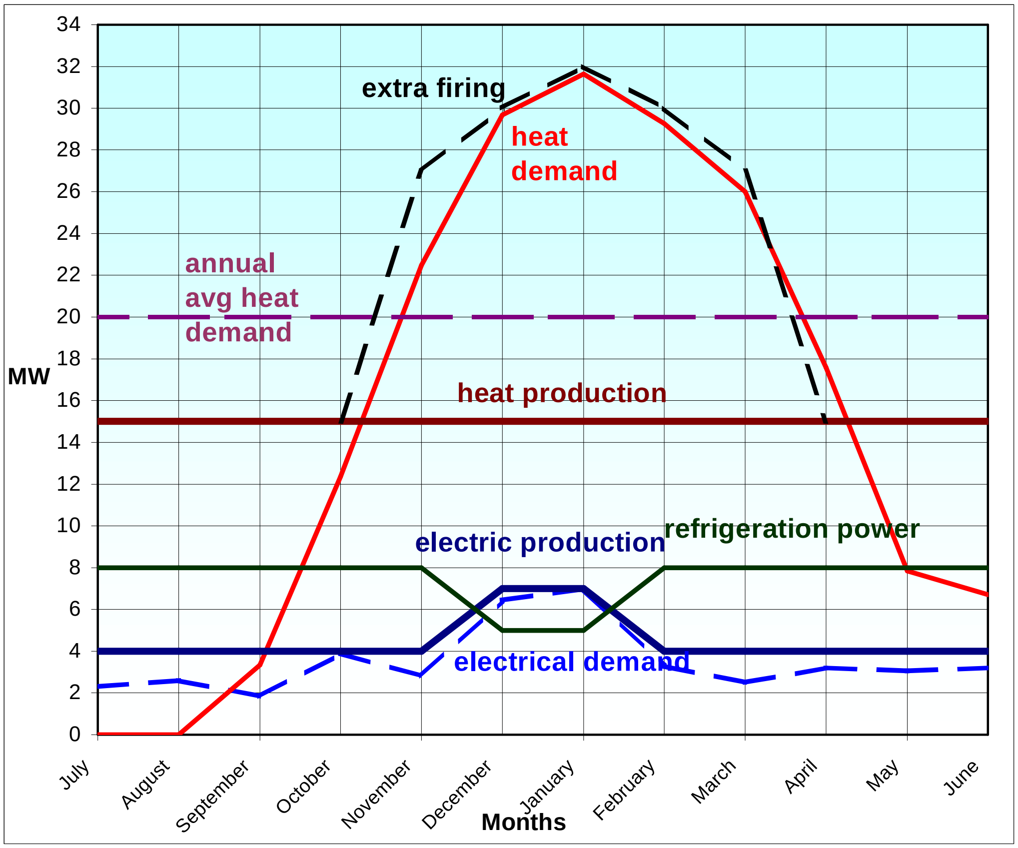
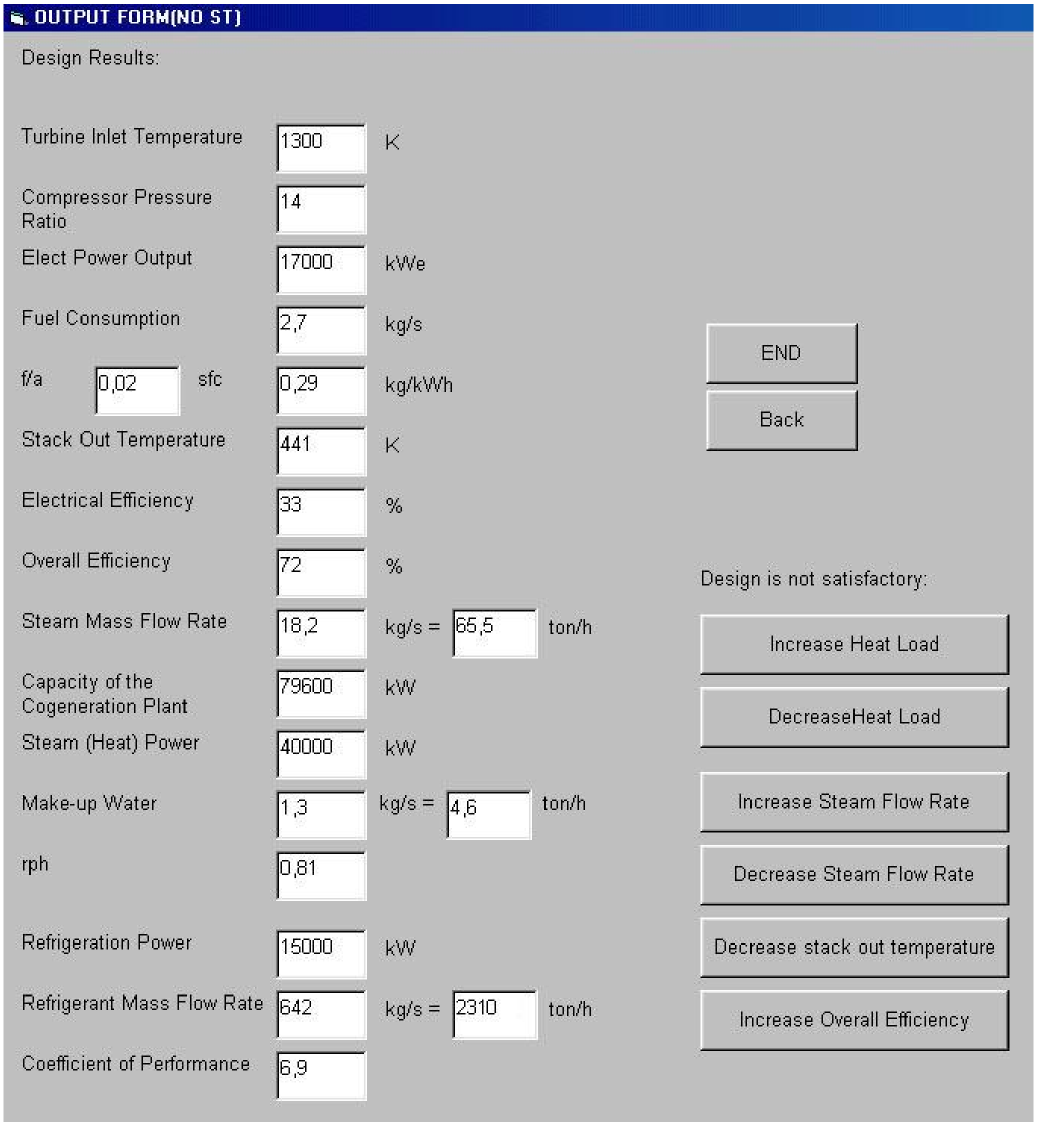


Figure 6.36 Demand and Supply Curves for Case Study: 7

6.2.2. Case Study: 8

The last study is for producing totally 32 MW of electrical power with two different configurations. First one, Case: 8_1 is two 8 MW gas turbines, and a 15 MW capacity refrigeration unit. When only one gas turbine is operated during summer months, 20 MW heat power, which is more than enough, will be produced together with a refrigeration capacity of 7.5 MW. During coldest months, refrigeration unit may be operated on lower capacity. Output form for this case can be seen in Figure 6.37 below.


OUTPUT FORM(NO ST)

Design Results:

Turbine Inlet Temperature	1300	K		
Compressor Pressure Ratio	14			
Elect Power Output	17000	kWe		
Fuel Consumption	2,7	kg/s		
f/a 0,02 sfc	0,29	kg/kWh		
Stack Out Temperature	441	K		
Electrical Efficiency	33	%		
Overall Efficiency	72	%		
Steam Mass Flow Rate	18,2	kg/s =	65,5	ton/h
Capacity of the Cogeneration Plant	79600	kW		
Steam (Heat) Power	40000	kW		
Make-up Water	1,3	kg/s =	4,6	ton/h
rph	0,81			
Refrigeration Power	15000	kW		
Refrigerant Mass Flow Rate	642	kg/s =	2310	ton/h
Coefficient of Performance	6,9			

END

Back

Design is not satisfactory:

Increase Heat Load

DecreaseHeat Load

Increase Steam Flow Rate

Decrease Steam Flow Rate

Decrease stack out temperature

Increase Overall Efficiency

Figure 6.37 Output Form for Case Study: 8_1

Supply and demand curves for the first configuration (2x8 MW gas turbines) are in Figure 6.38.

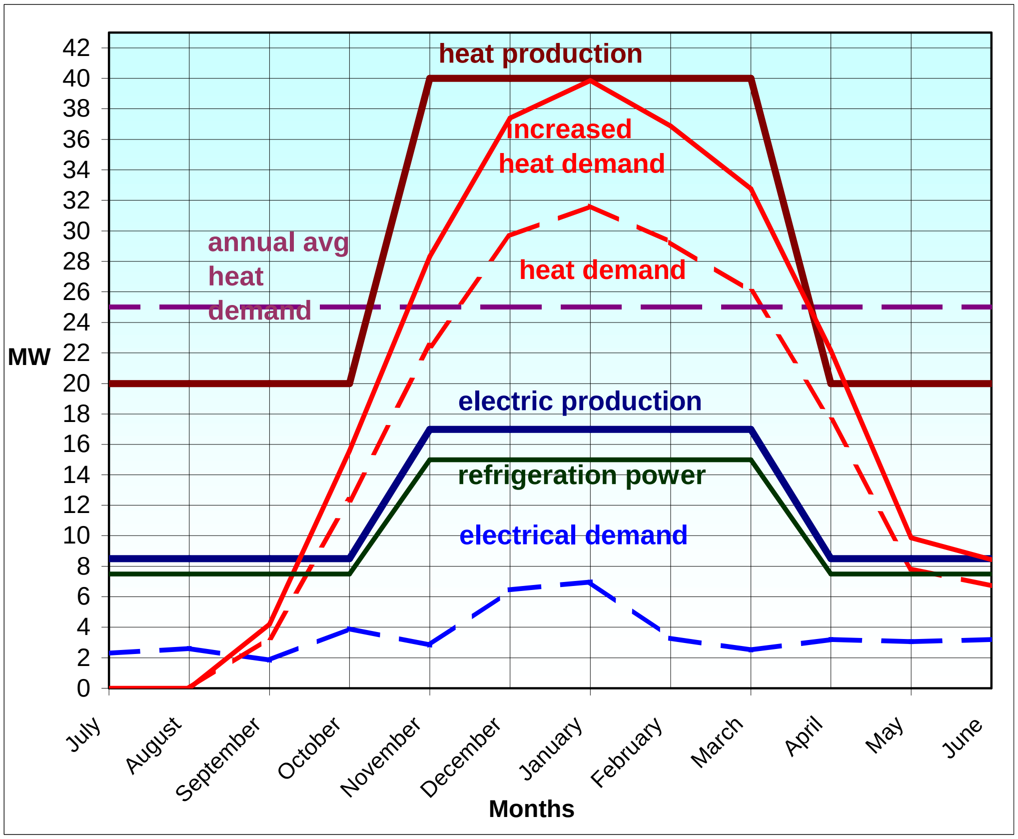


Figure 6.38 Demand and Supply Curves for Case Study: 8_1

For Case: 8_2, 3x10 MW gas turbines are used and refrigeration capacity is increased. The outputs for this configuration do not differ except the refrigeration power and the corresponding demand and supply curves are given in Figure 6.38 can be build.

Gas turbine selections, details of the system parameters, economical analysis and cost summary of the two designs are given in Appendix H, Part 8.

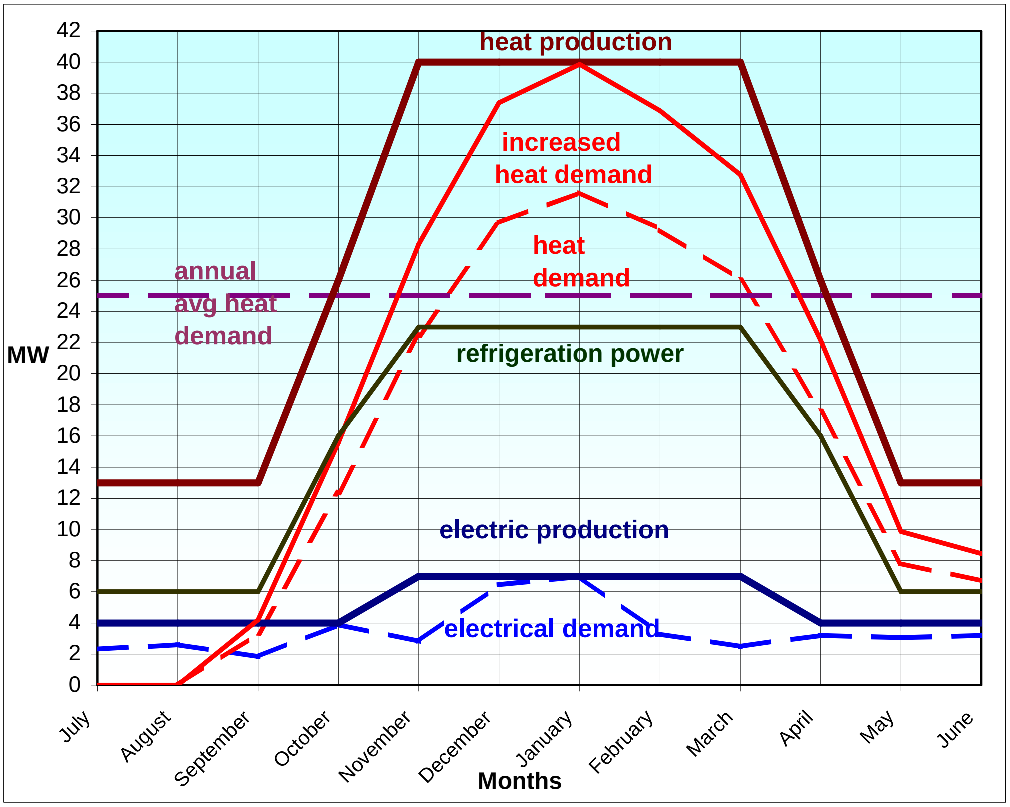


Figure 6.39 Demand and Supply Curves for Case Study: 8_2

6.3. Results and Conclusions

A comparison for showing satisfaction of requirements and economic evaluations for the designed cogeneration power plants for all the cases are given in Table 6.3 and Table 6.4.

First of all, in the campus environment, condensing steam turbine can not be used, since it is not possible to supply large amounts of water all the time, and besides it appreciably increases the construction and maintenence cost a lot.That is why in non of the cases condensing turbine is used.

When a gas turbine just big enough to supply all campus' electricity is chosen, it is found that, heat demand is so much above the heat supplied by the gas turbine.

Table 6.3. Technical and Economical Information About Case Studies

CASE	POWER	COST (US$)	SYSTEM DESCRIPTION
Case 1	7.7 MW	8.750.000	1XGT+ HRSG+Add. Firing(35 ton/h)
Case 2	24.5 MW	23.500.000	1XGT+ HRSG
Case 3	12.2 MW	13.600.000	1XGT+ HRSG+Add. Firing(30 ton/h)
Case 4_1	37.2 MW	37.900.000	2XGT+ HRSG
Case 4_2	32.8 MW	33.700.000	3XGT+ HRSG
Case 5	12 MW	15.400.000	1XGT+ HRSG+ST+Add Firing(35 ton/h)
Case 6	17.6 MW	21.000.000	2XGT+ HRSG+ST+Add Firing(25 ton/h)
Case 7	12.2 MW	13.600.000+ref system	1XGT+ HRSG+Refrigeration +Add. Firing(30 ton/h)
Case 8_1	37.2 MW	37.900.000+ref system	2XGT+ HRSG+Refrigeration
Case 8_2	32.8 MW	33.700.000+ref system	3XGT+ HRSG+Refrigeration

Table 6.4. Comments and Payback Periods For Case Studies

CASE	COMMENTS	YEARS FOR PAY BACK OF EQUITY
Case 1	Considerably below heat demand (Add firing needed)	4
Case 2	Above electrical demand	3
Case 3	Higher electrical production, much below heat demand. (Add firing needed)	4.2
Case 4_1	No gas turbine in the electrical output range	2.7
Case 4_2	Above electrical demand	2.7
Case 5	Considerably below heat demand (Add firing needed), expensive system	5
Case 6	Below heat demand (Add firing needed), expensive system	6
Case 7	Below heat demand (Add firing needed)	4.3
Case 8_1	No gas turbine in the electrical output range	2.8
Case 8_2	Above electrical demand	2.9

This time additional firing (burning of natural gas directly in the auxilary boilers for steam production) takes importance, since there are natural gas fired boilers present on the campus' heat plant. Burning of natural gas decreases overall system efficiency and also increases fuel cost, but since this will be necessary only during 4-5 months, and no additional construction will be done, it may be considered. The existing boilers on the campus' heat plant may be used together with the HRSG, supplying steam to the same line.

When a gas turbine capacity is chosen to supply all necessary heat demand, it is seen that there is considerably excess electricity produced. This excess elecrticity can be sold, or may be used in a refrigeration system to produce chilled water or ice. This way efficiency of the system is increased, and payback period can be shortened. But since METU is a Government University, there are some regulations that make it difficult and rather disadvantageous to sell this excess electricity generated.

When a steam turbine is constructed, more heat energy is converted into electicity, which does not seem to be sensible for a campus environment, since heat demand is always much more than the electric demand. So the cases with a steam turbine come out not to be feasible, not only because of this fact, but also because construction cost for steam turbines is much higher than for gas turbines, thus the system expenditure increases.

Systems of two or more gas turbines are more convinient for a campus, since day and nigth demands differ so much thus some units may be stoped when demand is low. This way, a more economical operation can be done.

The economical summaries and cost reports in Appendix H are based on the assumption all the excess electricity and hot water are sold. So when using hot water for own demand is considered, in all cases, pay back period will increase.

For the cogeneration power plant to be feasable in the campus, agreements should be done for selling this excess electricity and even excess steam, if there will be any and if refrigeration cases will be used, the customers should be found and preliminary agreements should be signed before starting the construction. All the heat power should be primarily supplied to the campus as hot water, if necessary, additional firing should be used. When these requirements are met, the university will receive maximum benefit from the cogeneration power plant with respect to following points:

First of all, university will be able to produce its own electricity, heating and hot water considering its requirements, independent from other firms or companies. This way, university will not be affected from electricity shortage occurred for any reason among the grid. Also the customers buying electricity will not be affected.

Secondly, university may be able to sell the excess electricity, and earn money out of it. Steam production costs with a cogeneration system compared to natural gas fired boiles decrease a considerable amount. Also, electricity production cost will be lower than buying electricity from the government. Even though calculations of the monitary expressions are not aimed in the thesis it is clear that, university will profit within a time period of 6 to 8 years.

Thirdly, if a compression refrigeration unit is installed, selling or using the benefits of this facility, university may gain further profit. Since vapour absorbtion systems are less efficient, have a high capital cost, and consume water vapour which is a problem because heat demand is so much more than electrical demand, it is turns out to be unfeasable to install such a system in the campus of METU.

By the way, when all the above statements are considered more convenient cases come up to be; Case: 3, Case: 4_2 , Case: 7 and Case: 8_2.

Case: 3 (1x12.2 MW GT+HRSG+Additional Firing of max 30 tons/h) is economical, since the construction cost is low and pay back of equity period is not so long. Also, capacity of the power plant is smaller compared to most of the other cases, which will decrease the size of the equipment and maintenence costs. During cold months, additional firing is to be used to supply heating and hot water upon the demand of the campus, but burning natural gas decreases overall system efficiency and also increases fuel cost, as mantioned before, but no additional construction will be necessary, since the boilers are available on site.

Case: 4_2 (3x11 MW GT+HRSG) requires a higher construction cost compared to most of the cases, but has the lowest pay back period if the excess electricity can be sold. This system is flexible since there are 3 gas turbines working all together, so one or two of them may be used according to the electric and heat demand of the campus. Capacity of the power plant is quite high compared to most of the other cases, but since no additional firing is necessary even with the further increased heat demand, system efficiency will be higher than the other cases, where additional firing is used.

Case: 7 (1x12.2 MW GT+HRSG+Refrigeration +Additional Firing of max 30 tons/h) is less economical than Case:3 because of the construction cost of the refrigeration plant, but cold storage facility services instead of electricity will increase the feasibility and the gain of profit. The pay back period is not so long. During cold months, additional firing is to be used to supply heating and hot water, again, burning natural gas decreases overall system efficiency and also increases fuel cost, as mantioned before.

Case: 8_2 (3x11MW GT+HRSG+Refrigeration), requires a higher construction cost compared to most of the cases, but has a lower pay back period. This system is flexible since there are 3 gas turbines working all together, so one or two of them may be used according to the electric and heat demand of the campus. Capacity of the power plant is quite high compared to most of the other cases, but since no additional firing is necessary, even with the further increased heat demand, system efficiency will be higher than the other cases, where additional firing is used. Also, using cold storage facilities instead of electricity will increase the gain of profit for the power plant.

CHAPTER 7

DISCUSSION AND CONCLUSIONS

7.1. Discussion and Conclusions

“Cogeneration Design” program is developed using Microsoft Visual Basic 6.0 programming language for conceptually designing cogeneration power plants. Program is developed based on the formulation and assumptions given in Chapter 3. Design is focused on power plants to be built in Middle East Technical University Campus, where there is mainly heating, hot water, electricity and sometimes cooling demands.

8 different cases (scenarios) are studied, and compared with each other and discussed in Section 6.3. More convenient ones are chosen among them and further discussed giving the advantageous and disadvantageous aspects of the designs.

When the results obtained from the “Cogeneration Design” program, given in Chapter 6 are compared to the results of “Thermoflow Software” which is one of the widely used power plant design programs in the world, it is found that the cycle parameters and output parameters are convenient. The cycle schematics found with “Thermoflow Software” are given in Appendix H.

Detailed comparison between these 8 studied cases is done, general concluding remarks are developed and feasibility of the cases are discussed briefly in Chapter 6, Part 6.3.

The conclusions came out to be:

- Condensing steam turbine cannot be used in the METU Campus.

- Power plant capacity chosen to meet the heat demand for the campus provides considerable excess electricity. This electricity should be sold or used in a refrigeration unit for cold storage facility services.

- Using a steam turbine on the campus should be carefully considered since it may not be feasable due to its expense and steam energy which is necessary for heating is converted to electricity although electrical demand is lower than heat demand.

- Flexible systems composed of two or more gas turbines each having its own HRSG are more convenient for a campus.

- In most of the cases, additional firing would be necessary (to supply more hot water and heat) and this can be done by the boilers already present in the heat plant on METU campus.

Considering these, more convenient cases come up to be; Case: 3, Case: 4_2 , Case: 7 and Case: 8_2. Detailed comparsion between these cases are given in Section 6.3.

University will recieve maximum benefit from the cogeneration plant designs studied in any of these four cases with respect to following points:

- University will be able to produce its own electricity heating and hot water considering its requirements, independent from other firms or companies.

- University will not be affected from electricity shortage occurred for any reason among the grid.

- University may be able to sell the excess electricity to profit, produce steam more cheaply.

- If a compression refrigeration unit is installed, university may gain further profit by sellling or using the benefits of cold storage facility.

7.2.Recommendations for future work

In the "Cogeneration Design" program, HRSG design is done using one steam pressure level. Multiple steam take offs were beyond the scope of the master thesis, but for a future work, HRSGs with two or three steam take offs may be modelled for designing a cogeneration power plant.

In the program, for trigeration cases, accepted inputs are limited, and design process does not allow so many different cases. A more detailed trigeneration design may be recomended as a future work , since it was again beyond the scope of this thesis.

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APPENDIX A

Table A.1 Universities Having Cogeneration Facilities In US And Canada

NAME AND LOCATION OF THE UNIVERSITY	CAPACITY
Albion College, Albion, Michigan	360 kW
Atlantic Union College, Lancaster MA	NA
Alvin Community College, Alvin, TX,	1,000 kW
Baylor University, Waco, Texas,	3,300 kW
Biola University, California,	1,200 kW
Brown University, Providence, Rhode Island	3,250 kW
Bucknell University, Lewisburg, Pennsylvania	600 kW
California Institute of Technology	11,000 kW
California Polytechnic State University, San Luis Obisbo	350 kW
California State University-Long Beach, Long Beach, California	350 kW
Central Michigan University, Mount Pleasant, Michigan	950 kW
Cerritos Community College, Norwalk, California	150 kW
City of San Diego, San Diego, California	27,000 kW
Claremont Colleges, Claremont, California	50 kW
Clark University, Worcester, Massachusetts	1,800 kW
Colby College, Waterville, Maine	100 kW
College of Wooster, Wooster, Ohio	375 kW
Cornell University, Ithaca, New York	8,500 kW
Dartmouth College, Hanover, New Hampshire	4,000 kW
Dundee University, Dundee, Scotland	3,000 kW
Duquense University, Pittsburgh, Pennsylvania(under Construction)	5,000 kW
Eastern Michigan University, Ypsilanti, Michigan	4,000 kW
Elgin Community Collage, Elgin, Illinois, Waukesha engines	4 x 800 kW
Foothill-De Anza Community College, Los Altos Hills, California	65 kW
Georgetown University, Washington, (D.C turbine)	2,800 kW

NAME AND LOCATION OF THE UNIVERSITY	CAPACITY
Gordon-Conwell Theological Seminary South Hamilton, Massachusetts	NA
Harding University, Harding, Arkansas	5,200 kW
Henry Ford Community College, Dearborn, Michigan. (Gas/Diesel)	70 kW
Highland Community College, Freeport, Illinois	60 kW
Hofstra University, Long Island, New York	NA
Illinois Central College, East Peoria, Illinois	650 kW
Illinois Institute of Technology, Chicago, Illinois	8,000 kW
Iowa State University, Ames, Iowa(Coal-fired steam turbine)	36,000 kW
Loma Linda University, Loma Linda,California,2xAllison501KH	10,600 kW
Kansas State University, Manhattan, Kansas	3,000 kW
Massachusetts Institute of Technology, ABB GT-10 Gas Turbine	23,000 kW
Stanford University, Stanford, California	39,000 kW
State University of New York, Stony Brook,Long Island,NY (LM6000)	NA
Syracuse University, Syracuse, New York (2 x LM5000)	NA
Texas A&M University, College Station, TX	36,500 kW
Texas Tech, Lubbock, Texas	NA
The College of New Jersey, Trenton, New Jersey (Solar Turbine)	3,200 kW
The Hotchkiss School, Lakeville, Connecticut	135 kW
The Rockefeller University, New York	NA
The University of Medicine and Dentistry of New Jersey, Piscataway	NA
Trent University, Peterborough, Ontario	2,500 kW
Turabo University, Gurabo, Puerto Rico	38,000 kW
University of Alaska, Fairbanks, Alaska (coal- and oil-fired)	13,000 W
University of British Columbia,	NA
University of California, Berkeley, California	NA
University of California, Davis, California	7,000 kW
University of California, Los Angeles, California (2 x LM1600)	NA
University of California, San Francisco, California	NA
University of California, Santa Cruz, California	2,600 kW

NAME AND LOCATION OF THE UNIVERSITY	CAPACITY
University of Colorado, Boulder, Colorado, 2 x MF111	33,000 kW
University of Evansville	1,100 kW
University of Florida LM6000 gas turbine.	42,000 kW
University of Illinois at Urbana-Champaign, Champaign, Illinois	30,000 kW
University of Iowa, Iowa City, Iowa	21,000 kW
University of Lethbridge, Lethbridge, Alberta	NA
University of Maryland, Baltimore, Maryland	NA
University of Massachusetts, Amherst, Massachusetts	3,600 kW
University of Michigan, Ann Arbor, Michigan	39,000 kW
University of Michigan, Dearborn, Michigan	350 kW
University of Missouri - Columbia, Columbia, Missouri	52,000 kW
University of Nebraska, Lincoln, Nebraska (1,5 MW steam & 3 MW gas)	4,500 kW
University of New Mexico, Albuquerque, New Mexico	3,500 kW
University of North Carolina, Chapel Hill, North Carolina	28,000 kW
University of Northern Colorado, Greeley, Colorado2 x LM 5000	NA
University of Northern Iowa, Cedar Falls, Iowa	7,500 kW
University of Notre Dame, Notre Dame, Indiana	32,000 kW
University of Oklahoma, Norman, Oklahoma	12,500 kW
University of Oregon, Eugene, Oregon	5,500 kW
University of San Diego, San Diego, California	1,050 kW
University of San Francisco	1,500 kW
University of South Florida, Tampa, Florida	1,775 kW
University of Texas, Austin, Texas	100,000 kW
University of Texas, South West Medical Center, Dallas, Texas	NA
University of Toronto, Toronto, Ontario	8,000 kW
University of Washington, Seattle, Washington	5,000 kW
University of Western Ontario, London, Ontario	1,600 kW
University of Wisconsin, Madison, Wisconsin	3,000 kW
University of Wisconsin, Whitewater	285,000 kW
Vanderbilt University, Nashville,	11,000 kW

NAME AND LOCATION OF THE UNIVERSITY	CAPACITY
Virginia Polytechnic Institute and State University, Blacksburg	24,000 kW
Wellesley College, Wellesley, Massachusetts (3 x Jenbacher	4,500 kW
Wentworth Institute of Technology, Boston, Massachusetts	660 kW
Williams College, Williamstown, Massachusetts	500 kW

APPENDIX B

HRSG DESCRIPTION

B.1. Types and Configurations of HRSG According to Evaporator Layouts

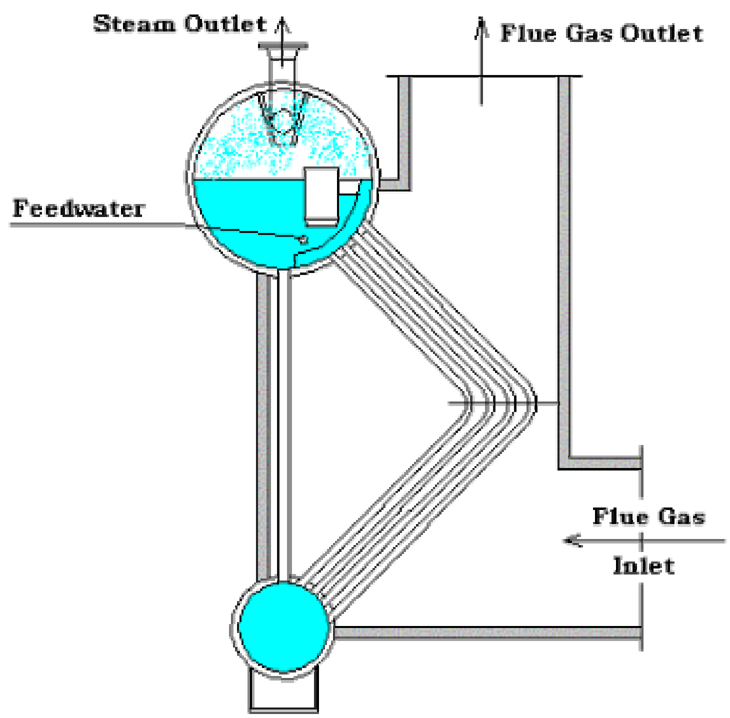

Figure B.1 D-Frame Evaporator Layout

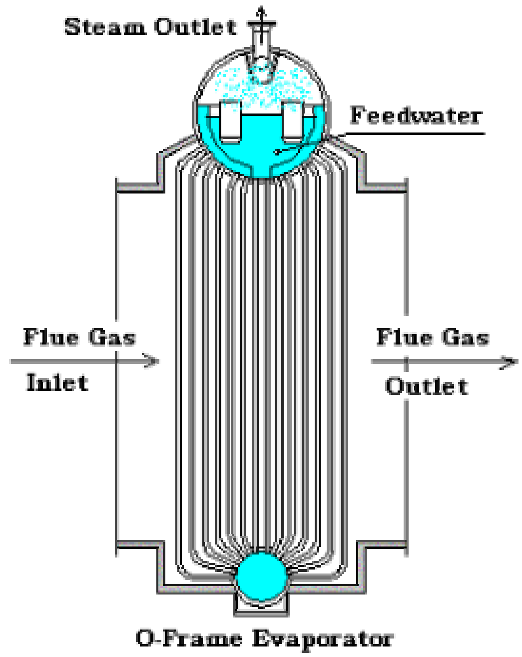

Figure B.2 O-frame evaporator layout

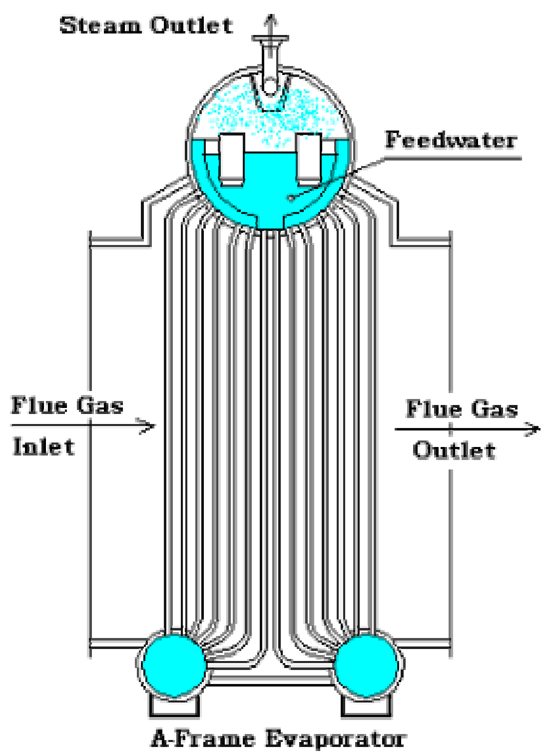

Figure B.3 A-Frame Evaporator Layout

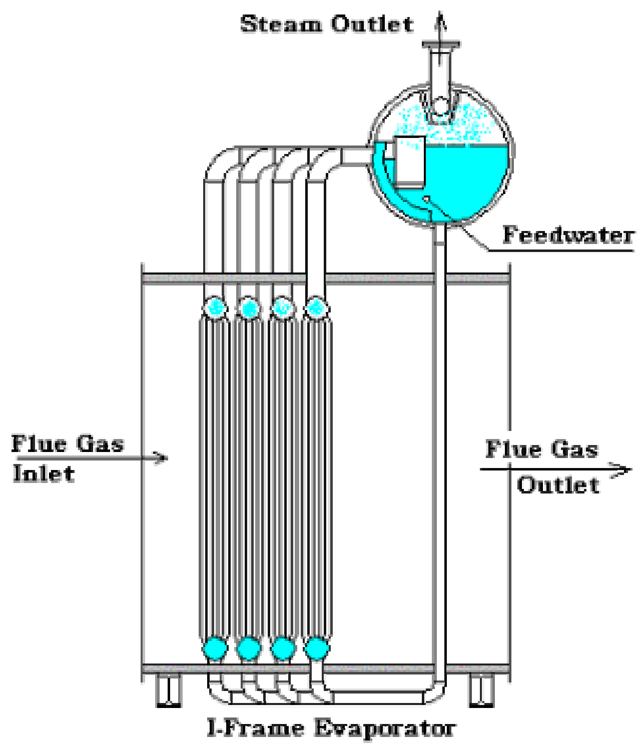

Figure B. 4 I-Frame Evaporator Layout

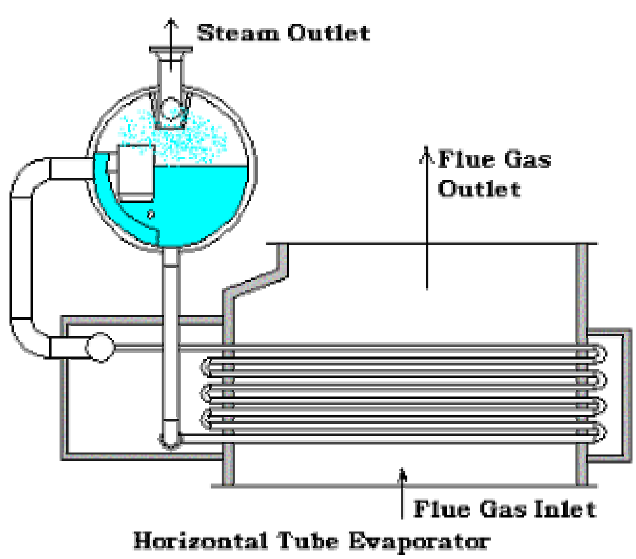

Figure B.5 Horizontal Tube Evaporator Layout.

2. Types and Configurations of HRSG According to Superheater Layouts

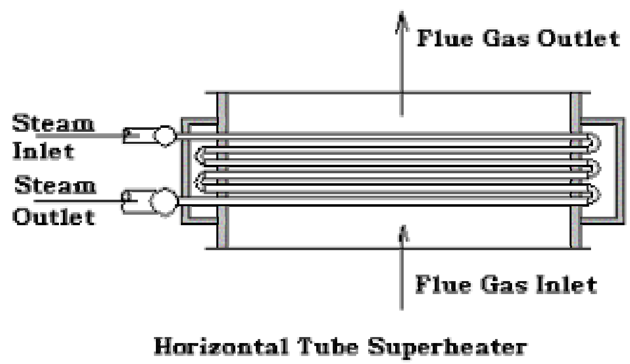

Figure B.1. Horizontal Tube Type Superheater Layout

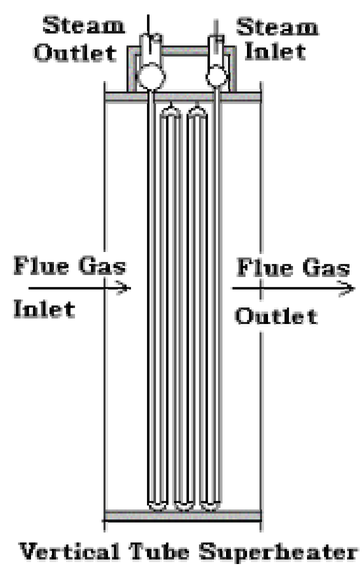

Figure B.2. Vertical Tube Type Superheater Layout

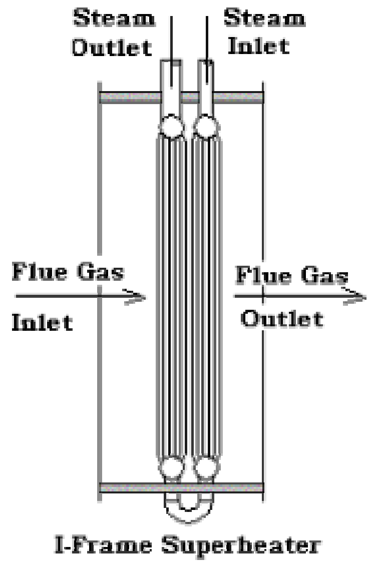

Figure B.3 I-Frame Type Superheater Layout

[9]

APPENDIX C

SCHEMATICS OF POWER CYCLES

C.1. COGENERATION CYCLE WITHOUT STEAM TURBINE

C.2. COGENERATION CYCLE WITH NON CONDENSING STEAM TURBİNE

C.3. COGENERATION CYCLE WITH CONDENSING STEAM TURBINE

APPENDIX C.1

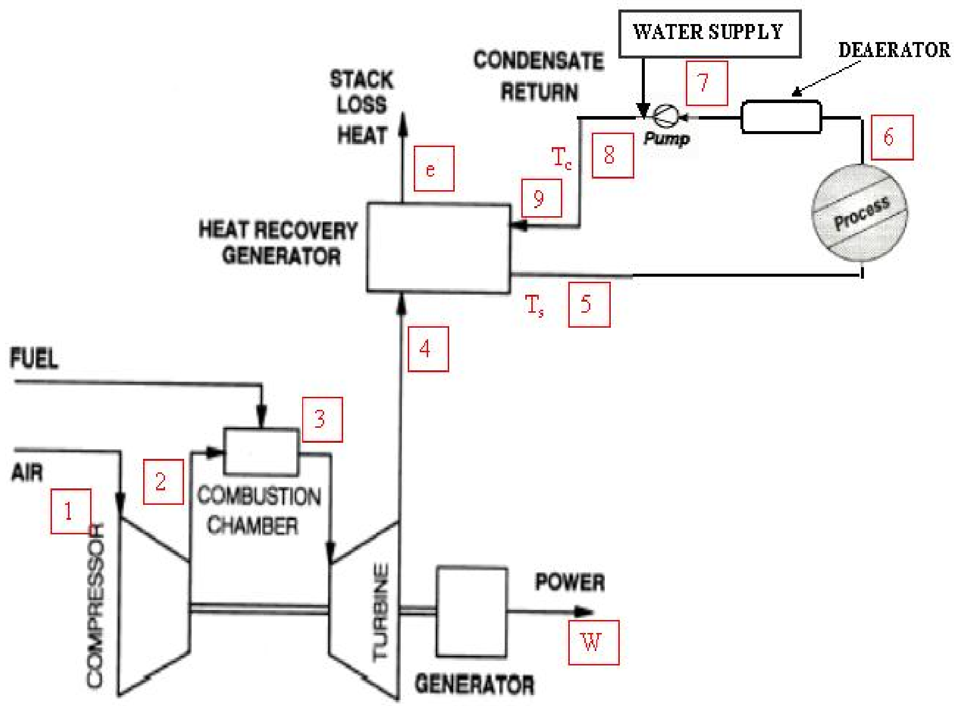

Figure C.1. Cogeneration Cycle Without Steam Turbine

APPENDIX C.2

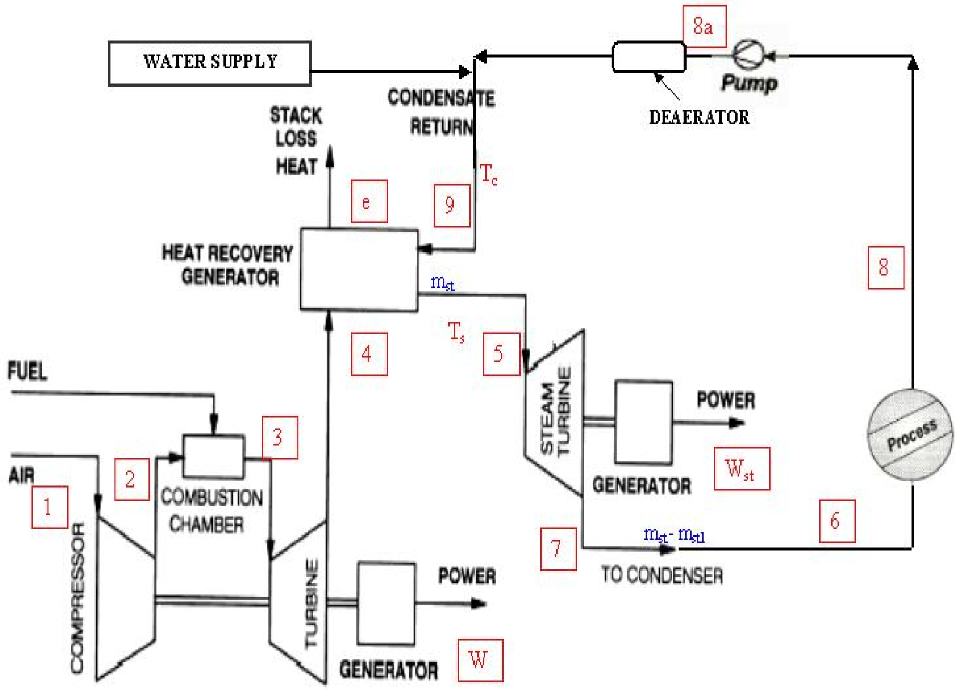

Figure C.2. Cogeneration Cycle With Non Condensing Steam Turbine

APPENDIX C.3

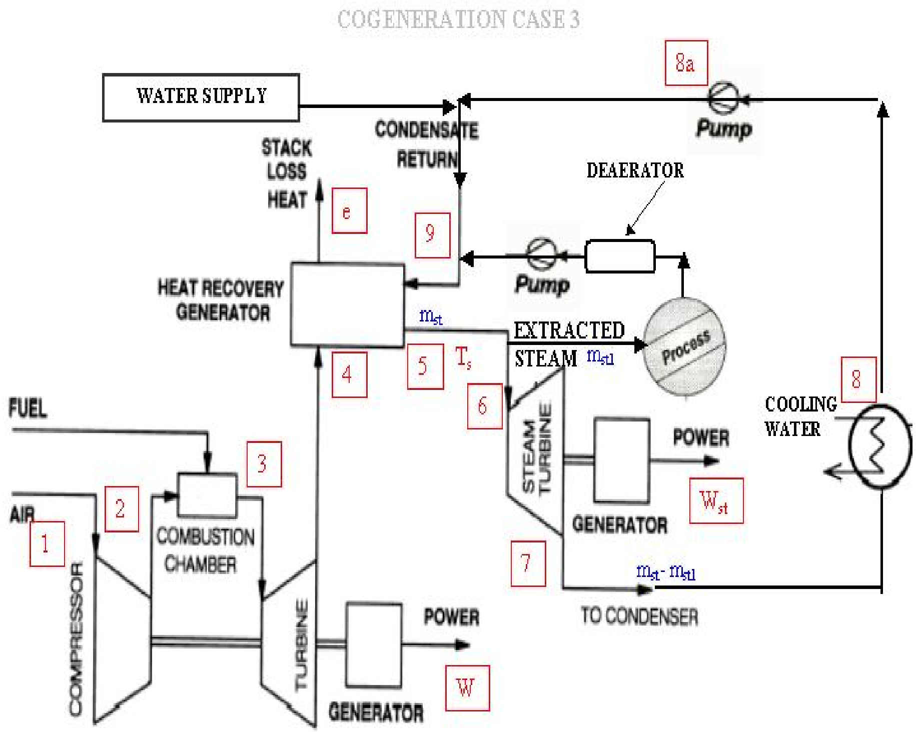

Figure C.3. Cogeneration Cycle With Condensing Steam Turbine

APPENDIX D

D.1. INDUSTRY PRICING FACTORS FOR SIMPLE CYCLE AND COMBINED CYCLE POWER PLANTS (TAKEN FROM GAS TURBINE WORLD 2001-2002)

D.2. INDUSTRY PRICE LEVELS FOR SIMPLE CYCLE AND COMBINED CYCLE POWER PLANTS (TAKEN FROM GAS TURBINE WORLD 2001-2002)

D.1. INDUSTRY PRICING FACTORS FOR SIMPLE CYCLE AND COMBINED CYCLE POWER PLANTS

Industry Pricing Factors

Simple Cycle Power Plants

Budget price levels reported in the Gas Turbine World Handbook are derived from a number of different sources including owner-operators, consulting firms, packagers, and gas turbine builders.

The individual prices sometimes vary considerably. We adjust the results of our field research to arrive at a consensus price that the majority of industry contacts consider reasonable for a single unit purchase.

In the case of simple cycle gensets and packaged plants, the budget prices quoted are F.O.B. the factory in year 2001 U.S. dollars for basic gas turbine packages without the bells and whistles or accessory systems.

Based on 'bare bones' gas turbine and electric generator package equipped with basic systems and controls needed for an operational installation:

Genset package. Skid mounted single-fuel gas turbine and driven electric generator. Includes regular gas turbine start, fuel forwarding and lube oil systems, standard controls.

Electric generator. Primarily air-cooled designs (TEWAC) for generators below 150 MW output and hydrogen-cooled designs over 150 MW. Even for the larger units, however, air cooling is becoming more popular as a lower priced alternative.

Balance of plant. Air inlet filter and intake silencer, stack with exhaust silencing, vibration monitoring system and controls. Packaged gensets are normally housed outdoors in acoustic enclosures with ventilation and fire protection systems. Fuel gas compressor not included.

Emissions control. Dry low NOx combustion is included when it is a standard design feature of the specified gas turbine model. But the budget prices do not include NOx water or steam injection, nor post-firing treatment such as NOx or CO catalytic reduction.

$ per kW pricing

It is important to evaluate and compare gas turbine genset and power plant prices on the same basis.

Industry practice is to relate the total price to base load output on natural gas fuel at 59F (15C) ambient sea level site conditions and 60% relative humidity, without water or steam injection for NOx or power augmentation unless otherwise specified, and without duct losses.

Gas turbine models identified as steam injected designs are an exception. In this case, the output rating is quoted for base load operation with steam injection—but without inlet or exhaust duct losses.

For electric gensets, the quoted nominal ISO rating represents the gross power output measured across the electric generator terminals. As such it includes electric generator efficiency and any reduction gearing losses.

Installation extra

The prices shown in the GTW Handbook do not reflect associated plant costs such as site engineering and installation services that typically can more than double equipment-only acquisition costs.

Complete turnkey plant outlay such as transportation and taxes, engineering procurement and construction services, legal and financial fees, start-up, commissioning, spares and operator training can add tremendously to project costs.

For steam and water injected gas turbine units, the quoted price includes all of the on-engine components and hardware necessary to run steam or water through the machine.

But it does not include the off-engine steam production equipment such as heat recovery boiler or once-through steam generator, nor any water treatment hardware and supplies.

Depending on the number of units ordered, scope of equipment supply, site specific requirements, geographic location, and competitive market conditions, prices vary considerably.

As the past two years have shown, market demand and supply inevitably are the most important factors in determining price levels. Under all scenarios, however, big buys for multiple unit installations can reduce the unit cost substantially.

Changes in currency valuations also play an important role, sometimes dramatically, since competitive suppliers must take into account their impact on profit margins and costs.

Fuel efficiency

In areas of premium fuel prices, the better thermal efficiency designs almost always command higher first cost than lower-efficiency models in the same output range.

This reflects the increased engineering, manufacturing and materials costs that suppliers of advanced gas turbine designs must recover through higher equipment prices.

In the long run, however, the level of natural gas pricing (primary fuel for most of these machines) determines the value of thermal efficiency relative to fuel costs and number of hours the plant will operate.

For mid-range to base load service, higher efficiency plants can produce equipment payback periods many years faster than lower-efficiency competitors, justifying their increased first cost.

By contrast, fuel efficiency is relatively unimportant for peaking machines that run less than a few hundred hours a year. Availability, reliability and start time take precedence over thermal efficiency.

This is especially true in grid areas where daily, hourly or seasonal price swings are high.

Low emissions

Increasingly stringent emissions regulations are spreading beyond industrialized countries to developing nations and even remote offshore platform operations.

Depending on fuel type and allowable emission levels, the cost of gas turbine emissions controls and post combustion treatment systems can add substantially to the base price of a plant.

In general, the tighter the air quality emission regulations, the more you will have to spend on gas turbine and plant equipment.

Basic emission control systems include water or steam injection for NOx reduction on natural gas or distillate fuels.

Some installations also add post-firing treatment with NOx and CO catalytic reduction, adding substantially to balance-of-plant and operating costs.

One extreme example is an LM6000 peaking station project in the U.S. northeast that is budgeting an additional $5 million per unit for NOx and CO reduction.

This covers the costs of an engine water injection system, a downstream selective catalytic reduction system to reduce NOx, and a separate CO removal system.

Today, several gas turbines are being equipped with dry low-NOx/CO combustors for operation on natural gas fuel. However, a few systems are coming out with dry low emissions on distillate as well.

On the larger frame engines, dry NOx systems are often provided as standard equipment, and the $ per kW levels do not increase that much due to the production volume of the fuel system equipment.

This is generally true of DLE system design that are relatively simple to engineer and install. (There is more room in the combustion section.)

In other cases, especially on the aeroderivative machines, complex dry low emissions systems can add up to 10% or more to the engine cost.

In most cases, these DLE units are completely new replacements of the original aircraft propulsion designs that are fully annular units designed for liquid fuel only.

Extras cost more

Not covered in the prices quoted are the electrical substation, switchyard, pipeline connec tions, fuel gas compressor skid.

Nor are fuel storage and treatment systems for liquid fuel included. No black start generator sets. Administrative offices, separate modular control room, workshops, storage buildings, spares and consumables are not included.

Also not covered: water or steam injection systems for NOx control; complex multi-level inlet filtration, inlet chillers or anti-ice systems; tall exhaust stacks or chimneys; electrical distribution or main step-up transformers and switchgear and motor control centers; poured concrete foundations and foundation bolting.

Pricing factors

Bulk purchases can create significant volume discounts, affecting unit price levels. How badly an OEM wants to enter or succeed in a particular market makes a difference.

Trade tariffs set up to tax imported equipment can significantly add to the packaged price at rates of 3% and higher.

Similarly, attractive financing packages and low-interest (or no interest) loan availability can affect the budget price of the gas turbine generator set.

Age of the gas turbine design can also be a factor in setting prices. New machines are often heavily discounted to get production prototypes out into the field.

Some are downright 'giveaways' to accumulate operating hours (field experience) and provide a showcase for prospective customers.

Later, as the design is accepted into the marketplace, prices are increased to normal levels.

Older machines, besides being less efficient, can often be steeply discounted since the original costs of engineering design, product development and production tooling and facilities have long since been repaid.

Unit upratings also tend to reduce $ per kW levels. An uprated machine, for example, can carry exactly the same equipment price as its predecessor. But, because of its higher output, it will show up with lower $ per kW cost in comparative evaluations.

Gross versus net plant ratings will also have an impact on $ per kW pricing. In the GTW Handbook, we try as much as possible to quote net plant ratings where available.

Scoping studies

Budgetary $ per kW prices are intended for preliminary project assessment and evaluation of power generating equipment.

Installed and complete turnkey plant costs can conservatively add between 60 and 100% to the equipment-only prices shown here.

While equipment prices listed reflect an average level, there is a fairly wide range—upward and downward—resulting from OEM or packager competitive position, geographical area, marketing strategies and production capabilities.

Most important price factor is supply and demand. In a seller's market for the 180-MW-class 'F' technology units, price sometimes takes a back seat to product availability.

Industry analysts note that, despite major price increases, buyers are paying full retail for these machines, and are queuing up for 'delivery slots' some of which stretch out through 2005.

As a project developer, owner or operator, it is up to you and your engineering consultants to evaluate all these factors during the bid process when shopping for new gas turbine generating capacity.

Industry Pricing Factors

Combined Cycle Power Plants

In the last year or so, turnkey prices for large utility-scale and merchant combined cycle power plant installations have increased by around 10% to 13% overall.

Understandably, turnkey prices vary widely from one project to another depending on the need for access roads, fuel gas pipeline extensions, training centers, repair facilities, site location, and the like.

For budgetary pricing purposes we have focused on projects with comparable requirements and scope of supply. We have excluded cost items such as overseas shipment, catalytic NOx and CO reduction systems, water treatment, power augmentation, etc.

Turnkey projects include supply and installation of gas turbine, heat recovery steam generator, steam turbine, and electric generator equipment; associated balance of plant components; plant engineering, design; and construction services; plant startup and commissioning.

We are quoting combined cycle plant prices in year 2001 U.S. dollars for turnkey supply and construction of standard 1x1 and 2x1 modular designs equipped with basic balance of plant equipment and controls needed for an operational installation:

Gas turbine. Skid mounted, with minimal enclosure, generally for indoors installation. Standard starting and controls. No steam or water injection for NOx and no inlet air heating or chilling. Includes reduction gearing for smaller engines.

Steam turbine. Condensing, subcritical, single or dual-pressure level; some units triple pressure with reheat. Axial or radial exhaust and water cooled heat rejection.

Unfired HRSG. Outdoors mounted heat recovery steam generator with ductwork, but no bypass damper or catalytic section. Dual-pressure level, some units triple pressure with reheat. Short exhaust stack.

Electric generators. Generally air-cooled on smaller machines, hydrogen cooling on larger units. Main step-up transformer, neutral grounding cubicle, and non-segregated bus included.

Balance of plant. Standard controls (not DCS) and auxiliaries. Does not include substation, pipeline, fuel gas compressor. Includes minimal tank storage for liquid fuels but no treatment system. Office and workshop buildings, special tools, operational spares, consumables, black start generator not included.

$ per kW pricing

Industry practice is to evaluate and compare combined cycle plant prices on the basis of net plant output and efficiency at 59°F (15C) seal level and 60% relative humidity on natural gas fuel with system losses.

Calculated $ per kW cost figures are based on net plant power output measured across the electric generator terminals.

These dollar figures are designed for scoping studies and preliminary project assessment. They do not include indirect costs that add considerably to project budgets.

Prices can vary considerably depending on the scope of equipment supply, site specifics, geographic location, currency valuations, and competitive market conditions.

Construction costs also can vary dramatically as a function of labor rates and specific construction requirements at different site locations worldwide.

Fuel cost is also a factor. There is a first-cost premium for high efficiency gas turbines and steam turbines. For example, a more efficient (and more complex) steam cycle will increase the overall plant cost.

Triple pressure heat recovery boilers cost more. So do units with reheat, and the multi-casing steam turbines that match these boilers also are more expensive.

Efficiency effects

In mid-range to base load service up to 8000 hours per year, typically how these plants operate, the higher efficiency units produce equipment payback years faster than lower-efficiency counterparts, justifying their greater initial cost.

For a typical plant installation that is expected to be in service 20 to 30 years, fuel costs are still the biggest single cost of running a power plant.

Figure that over the roughly 25-year life of a base load combined cycle, up to 70 percent of total plant costs—including acquisition, owning and operating costs and debt service—are for fuel alone. It's easy to see why efficiency is so important.

One OEM notes that "an increase of even a single percentage point in efficiency can reduce operating costs by $15-20 million over the life of a typical gas-fired combined cycle plant in the 400-500 MW range."

The relative value of thermal efficiency is specific to each site. It depends on equipment, price of fuel, size of plant, and operational profile, among others.

Pricing roller coaster

Prices for large IPP and utility-scale combined cycles plummeted over a five to six year period to hit historic low levels around 1997-98.

Compared to the early 1990s, these plants were selling for 45-50% less than earlier models installed at the beginning of the decade. But, around mid-1998, prices started turning around.

Industry analysts point to the North

American gas turbine 'buying spree' starting mid-summer 1998 as the key to the rising price trend.

The release of pent-up demand, particularly for large 60-Hz 'F' technology gas turbines, resulted in the production pipeline for most OEMs currently being full.

This has resulted in customers queuing up for delivery slots and putting up with longer lead times before equipment is delivered. More recently, economic downturn has caused a number of power project postponements and cancellations.

Multi-unit buys and delivery slots are changing hands, and OEM vendors are reportedly looking at speeding up deliveries on certain models. Prices are falling, fast.

Pre-packaged designs

EPC firms and OEMs have dramatically cut plant building schedules, sometime in half, by developing standardized, pre-engineered and easily replicated package modules to simplify plant design.

These 'reference plants' are designed for and built with maximized factory production and packaging for minimum on-site work.

Computerized overall plant design cuts materials and labor costs and jobsite problems and delays by allowing a complete plant, down to the piping and wiring, to be designed and reviewed before any earth is moved.

Today's pre-engineered complete combined cycle power plants can routinely be installed in well under two years from contract signing to commissioning.

Dollars per kilowatt

Standardized combined cycle plant $ per kW prices are a function of size (output) of the gas and steam turbines.

Prices vary according to multiples of units that make up the plant—as well as the design configuration of both the plant and its components.

Multi-shaft plants, where each gas turbine and steam turbine drives its own electric generator, are generally more costly than single shaft designs.

The single-shaft combined cycle—with the gas and steam turbine together driving opposite ends of an electric generator in a single power train—eliminates one complete electric generator and its attendant auxiliaries.

Any reduction in power equipment usually reduces price. However, some single-shaft plant designs fit an overrunning clutch between the steam turbine and generator.

This allows running the gas turbine simple cycle on its own without the steam cycle. However, it increases cost so that the single-shaft configuration ends up only about 3 to 5% cheaper than a multi-shaft unit.

Fudge factors

As noted, turnkey prices for large plants have increased substantially. Part of the reason is these plants have grown through higher technology designs which can include air cooled condensers and tighter emissions control.

One power plant developer notes that the steam turbines have also increased in price over the past 12 months. He indicates that now they are also paying premiums for shipment slots for steam turbines, just as they are for gas turbines.

An industry analyst claims that the increase in combined cycle prices is primarily due to non-OEM components that make up the balance of the plant and which are not actually manufactured by the OEMs.

In order to maximize production the OEM have farmed out much of the combined cycl equipment as modules, he claims.

"If you look at a combined cycle plant lay out it is really made up of about 10 modules gas turbine, electric generator(s), steam turbine, heat recovery boiler, condenser, cooling, lube oil, fuel, controls, fire suppression, water treatment, and probably one or two more," he says.

"All these modules are supplied by an out side source and put together by a contractor Modules and contracting costs make up approximately 50 to 55 percent of the total plant's cost."

In addition, there has been a shift to go to dry cooling to conserve water (makes permitting easier) and putting the combined cycle power modules inside a building on raised pedestals.

"This would be called a dry indoor plant and adds anywhere from 3 to 5 percent to tl total cost over a comparable wet, outdoor plant."

There is another factor which he calls 'hidden costs'. "This is an advance payment required in order to secure a production space in the current schedule.

"If the down payment (not recoverable) is say, three percent, that gives you at least a 1% price increase based on the interest cost of the down payment."

Advanced technology

Manufacturers have improved both gas and steam turbine technologies and performance. Increased power density reduces costs on the gas turbine and steam turbine portions.

For a given amount of labor and materials, the advanced technology designs produce many more kW of power than their predecessors of only a decade ago.

Optimizing the gas turbine-to-steam turbine size and design has produced economies and boosted overall plant performance.

Precise matching of the gas turbine design performance parameters to the HRSG and steam turbine design greatly improves overall plant performance.

Scoping studies

As noted, the $ figures quoted in the GTW Handbook are designed for scoping studies and preliminary project assessment. They are not 'sticker prices'.

While equipment prices listed reflect an average level, there is a fairly wide range—usually upward, rarely downward.

This results from OEM or packager competitive position, geographical area, marketing strategies, currency valuations and production capabilities.

As noted, currently many order books are filled, boosting prices for the foreseeable future unless market conditions change dramatically.

Excluded costs

These turnkey price levels are for no-frills plants with minimal equipment and services.

Extended site work such as cogenerated process steam or utility plant tie-ins are not covered, nor are extensive buildings, workshops, substations.

Special tools and operational spares such as combustor baskets, blades and vanes, etc., are also not inluded.

Additional costs must be added to these prices for emission controls which can include water or steam injection for NOx treatment in the combustor.

If post-firing treatment with selective catalytic reduction is applied to meet tight regulatory levels, this will add substantially to the plant initial capital costs (and operational expenses).

For example, one project developer in the U.S. is budgeting an additional $5 million per 44-MW unit to pay for combustion water injection and downstream catalytic reduction to remove post-combustion NOx and a CO emissions.

Also not included are the indirect costs for items such as interest during construction, financing and legal fees, licensing and permitting, insurance and bonding, workman's compensation, sales tax, extensive inland freight, owner's cost and overhead, and project contingency funds.

It is up to you and your engineering consultants to review and evaluate all these factors during the bidding process when shopping for new gas turbine combined cycle generating capacity.

D.2. INDUSTRY PRICE LEVELS FOR SIMPLE CYCLE AND COMBINED CYCLE POWER PLANTS

Industry Price Levels

Simple Cycle Power Plants

Budget prices in year 2001 U.S. dollars for basic electric power generator packages including a single-fuel gas turbine, air cooled electric generator (some larger units hydrogen cooled), skid and enclosure, inlet and exhaust ducts with silencers, standard control and starting systems, conventional combustion system unless otherwise designated as dry low emissions (DLE) models.

Plant Model	Base Load Output	Heat Rate Btu/kWh	Efficiency	Budget Price	$ per kW
VPS1	496 kW	16,570 Btu	20.6%	$435,000	$877
ST6L-813	848 kW	13,125 Btu	26.0%	$677,500	$799
Makila TI	1050 kW	12,580 Btu	27.1%	$880,000	$838
Saturn 20	1210 kW	14,025 Btu	24.3%	$675,000	$558
KG2-3C	1450 kW	21,620 Btu	15.8%	$1,070,000	$738
M1A13D	1473 kW	14,230 Btu	24.0%	$940,000	$638
KG2-3E	1830 kW	21,070 Btu	16.2%	$1,200,000	$656
ST18A	1960 kW	11,300 Btu	30.2%	$1,200,000	$611
OGT2500	2730 kW	12,515 Btu	27.3%	$1,435,000	$526
UGT-2500	2850 kW	12,430 Btu	27.5%	$1,390,000	$488
M1T13D	2900 kW	14,460 Btu	23.6%	$1,625,000	$560
VPS3	3105 kW	12,775 Btu	26.7%	$1,520,000	$490
ST30	3340 kW	10,660 Btu	32.0%	$1,600,000	$479
Centaur 40	3515 kW	12,240 Btu	27.9%	$1,400,000	$398
VPS4	3570 kW	11,800 Btu	28.9%	$1,601,000	$449
501-KB5S	3950 kW	11,765 Btu	29.0%	$1,600,000	$405
GTES-4	4100 kW	14,130 Btu	24.1%	$1,230,000	$300
ST40	4040 kW	10,310 Btu	33.1%	$1,800,000	$446

Plant Model	Base Load Output	Heat Rate Btu/kWh	Efficiency	Budget Price	$ per kW
Centaur 50	4600 kW	11,630 Btu	29.3%	$1,600,000	$348
GTES-5	5200 kW	13,050 Btu	26.1%	$1,534,000	$295
Taurus 60	5200 kW	11,225 Btu	30.4%	$1,800,000	$346
PGT5	5220 kW	12,720 Btu	26.8%	$1,900,000	$364
Typhoon 5.25	5250 kW	11,200 Btu	30.5%	$1,850,000	$352
501-KB7	5275 kW	11,200 Btu	30.5%	$1,750,000	$332
M7A-01	5840 kW	11,230 Btu	30.4%	$2,310,000	$396
PGT5B	5900 kW	10,700 Btu	31.9%	$2,050,000	$347
GTES-6	6200 kW	12,780 Btu	26.7%	$1,705,000	$275
501-KH5 (steam injection)	6420 kW	8560 Btu	39.9%	$2,300,000	$358
601-KB9	6450 kW	10,615 Btu	32.1%	$2,450,000	$380
GT6001	6700 kW	10,840 Btu	31.5%	$2,700,000	$403
UGT-6000	6700 kW	11,270 Btu	30.3%	$2,100,000	$313
Tornado	6750 kW	10,820 Btu	31.5%	$2,650,000	$393
M7A-02	6960 kW	11,050 Btu	30.9%	$2,700,000	$388
Taurus 70	7520 kW	10,100 Btu	33.8%	$2,670,000	$355
Tempest	7910 kW	10,940 Btu	31.2%	$2,750,000	$348
601-KB11	7920 kW	10,350 Btu	33.0%	$3,200,000	$404
UGT-6000+	8300 kW	10,650 Btu	32.0%	$2,350,000	$283
THM1304-10	9320 kW	12,170 Btu	28.0%	$3,520,000	$378
UGT-10000	10,000 kW	10,220 Btu	34.2%	$3,350,000	$335
G3142J	10,450 kW	13,320 Btu	25.6%	$3,750,000	$359
Mars 100	10,690 kW	10,520 Btu	32.4%	$4,000,000	$374
THM1304-11	10,760 kW	11,460 Btu	29.8%	$3,730,000	$347
PGT10B	11,700 kW	10,660 Btu	32.0%	$4,700,000	$402

Plant Model	Base Load Output	Heat Rate Btu/kWh	Efficiency	Budget Price	$ per kW
GTES-12	12,000 kW	10,240 Btu	33.3%	$3,000,000	$250
Cyclone DLE	12,875 kW	9820 Btu	34.8%	$4,650,000	$361
Titan 130	13,500 kW	10,250 Btu	33.3%	$4,500,000	$335
SB60-1	13,570 kW	11,490 Btu	29.7%	$5,930,000	$437
PGT16	13,750 kW	9670 Btu	35.3%	$6,750,000	$491
LM1600PA	13,750 kW	9865 Btu	34.6%	$8,000,000	$582
LM1600DLE	13,750 kW	9865 Btu	34.6%	$8,500,000	$618
H-15	13,800 kW	11,010 Btu	31.0%	$6,300,000	$456
MF111B	14,570 kW	11,020 Btu	31.0%	$6,200,000	$425
Avon	14,580 kW	12,100 Btu	28.2%	$5,200,000	$357
GTES-16	16,000 kW	9790 Btu	34.9%	$4,000,000	$250
UGT-10000 STIG. (steam injection)	16,000 kW	7950 Btu	43.0%	$4,500,000	$281
UGT-16000	16,300 kW	11,230 Btu	30.4%	$3,950,000	$242
LM1600-PB STIG (steam injection)	16,900 kW	8605 Btu	39.7%	$8,280,000	$490
GT35	17,000 kW	10,600 Btu	32.2%	$5,914,000	$348
L20A	17,000 kW	10,040 Btu	34.0%	$6,665,000	$392
UGT-15000.	17,500 kW	9750 Btu	35.0%	$6,275,000	$359
LM2000	18,000 kW	9615 Btu	35.5%	$7,950,000	$440
UGT-15000+.	20,000 kW	9480 Btu	36.0%	$6,500,000	$325
PGT25	22,450 kW	9395 Btu	36.3%	$9,900,000	$441
LM2500PE	22,800 kW	9280 Btu	36.8%	$9,575,000	$420
GT10B	24,770 kW	9985 Btu	34.2%	$7,495,000	$303
UGT-15000 STIG (steam injection)	25,000 kW	8100 Btu	42.1%	$7,250,000	$290
RB211-6556	25,360 kW	9745 Btu	35.0%	$7,900,000	$311

Plant Model	Base Load Output	Heat Rate Btu/kWh	Efficiency	Budget Price	$ per kW
FT8.	25,490 kW	8950 Btu	38.1%	$9,725,000	$382
UGT-25000.	26,200 kW	9550 Btu	35.7%	$6,800,000	$260
PG5371PA	26,300 kW	11,990 Btu	28.5%	$7,680,000	$292
H-25	26,900 kW	10,280 Btu	33.2%	$8,300,000	$309
LM2500PH (steam injection)	28,280 kW	8325 Btu	41.0%	$11,500,000	$407
LM2500+PK	28,600 kW	8860 Btu	38.5%	$10,500,000	$367
RB211-6562	28,775 kW	9225 Btu	37.0%	$8,900,000	$309
GT10C	29,060 kW	9480 Btu	36.0%	$8,495,000	$292
RB211-6762DLE.	29,430 kW	9030 Btu	37.8%	$9,600,000	$326
MF-221	30,000 kW	10,670 Btu	32.0%	$10,000,000	$333
RB211-6761DLE	31,750 kW	8735 Btu	39.1%	$10,300,000	$324
IM5000	33,550 kW	9210 Btu	37.1%	$12,900,000	$384
PG6561B	39,620 kW	10,710 Btu	31.9%	$13,100,000	$331
UGT-25000 STIG. (steam injection)	40,100 kW	7990 Btu	42.7%	$8,200,000	$20
PG6581B.	42,100 kW	10,640 Btu	32.1%	$14,600,000	$34
GTX100	43,000 kW	9215 Btu	37.0%	$11,828,000	$27
LM6000PD	42,330 kW	8310 Btu	41.1%	$14,600,000	$34
LM6000PD(DLE).	42,400 kW	8200 Btu	41.6%	$15,400,000	$36
LM6000PC	43,700 kW	8105 Btu	42.1%	$14,200,000	$32
LM6000PC Sprint (water injection)	48,060 kW	8430 Btu	40.5%	$16,100,000	$33
W251B11/12	49,500 kW	10,450 Btu	32.6%	$14,000,000	$28
IM5000-STIG(steam injection)	50,100 kW	7950 Btu	42.9%	$15,150,000	$30
Trent DLE	51,190 kW	8210 Btu	41.6%	$16,000,000	$31
FT8 Twin	51,350 kW	8890 Btu	38.4%	$16,500,000	$32
GT8C2	57,000 kW	10,100 Btu	33.8%	$16,100,000	$28

Plant Model	Base Load Output	Heat Rate Btu/kWh	Efficiency	Budget Price	$ per kW
Trent	58,000 kW	8528 Btu	40.0%	$17,350,000	$299
V64.3	63,000 kW	9640 Btu	35.4%	$17,700,000	$281
V64.3A	67,100 kW	9810 Btu	34.8%	$20,600,000	$308
PG6101FA.	70,140 kW	9980 Btu	34.2%	$22,200,000	$317
PG7121EA	85,400 kW	10,420 Btu	32.8%	$21,200,000	$248
UGT-110000	114,500 kW	9480 Btu	35.0%	$14,000,000	$122
GT11N2	116,500 kW	10,050 Btu	33.9%	$24,100,000	$207
W501D5A	120,500 kW	9840 Btu	34.7%	$25,800,000	$214
PG9171E	123,400 kW	10,100 Btu	33.8%	$25,900,000	$210
M701DA.	144,100 kW	9810 Btu	34.8%	$29,400,000	$204
V94.2	157,000 kW	9920 Btu	34.4%	$30,500,000	$194
GT13E2	165,100 kW	9560 Btu	35.7%	$35,200,000	$213
PG9231EC	169,200 kW	9770 Btu	34.9%	$35,200,000	$208
PG7241FA.	171,700 kW	9420 Btu	36.2%	$40,500,000	$236
GT24	179,000 kW	9098 Btu	37.5%	$39,300,000	$219
V84.3A	180,000 kW	8980 Btu	38.0%	$39,700,000	$220
W501F	186,500 kW	9130 Btu	37.4%	$40,400,000	$217
V94.2A	190,700 kW	9660 Btu	35.3%	$37,500,000	$197
PG9311FA	243,000 kW	9360 Btu	36.4%	$47,100,000	$194
W501G	253,000 kW	8760 Btu	38.5%	$49,700,000	$196
PG9351FA.	255,600 kW	9250 Btu	36.9%	$51,000,000	$199
GT26	262,000 kW	8930 Btu	38.2%	$51,900,000	$198
V94.3A.	265,900 kW	8840 Btu	38.6%	$50,400,000	$190
M701F.	270,300 kW	8930 Btu	38.2%	$51,000,000	$189
M701G	334,000 kW	8630 Btu	39.5%	$60,700,000	$182

Industry Price Levels

Combined Cycle Power Plants

Budget prices in year 2001 U.S. dollars for turnkey equipment supply and installation of modular plants powered by natural gas-fired gas turbine, unfired multi-pressure HRSG without a bypass stack, condensing multi-pressure steam turbine, electric generators, associated balance of plant equipment, engineering procurement construction services, plant startup and commissioning.

Plant Model	Net Plant Output	Heat Rate Btu/kWh	Net Effic	Gas Turbines	Steam Turbines	Budget Price	$ per kW
STAC 60	7.3 MW	8620 Btu	39.6%	1xTaurus 60	1x1.8 MW, 1P	$5,475,000	$750
GPCS 80	7.9 MW	8470 Btu	40.3%	1xM7A-01	1x2.4 MW, 1P	$7,900,000	$1000
STAC 70.	9.5 MW	8180 Btu	41.7%	1xTaurus 70	1x2.0 MW, 1P	$7,125,000	$750
STAC 100.	13.8 MW	8380 Btu	40.7%	1xMars 100	1x3.0 MW, 1P	$10,350,000	$750
S-LM1600PA	17.4 MW	7280 Btu	46.8%	1xLM1600	1x4.6 MW, 2P	$15,900,000	$912
STAC 130	17.7 MW	8000 Btu	42.7%	1xTitan 130	1x3.7 MW, 1P	$12,900,000	$730
KA35-1	22.8 MW	7880 Btu	43.3%	1xGT35	1x6.2 MW, 2P	$19,100,000	$840
CC-201	28.3 MW	7670 Btu	44.5%	2xPGT10	1x10 MW, 2P	$24,100,000	$852
CC1-2500	31.7 MW	6850 Btu	49.8%	1xLM2500	1x8.4 MW, 2P	$25,600,000	$808
THM1304-11	32.9 MW	7497 Btu	45.5%	2x1304-11	1x11 MW, 2P	$26,000,000	$790
FT8	32.9 MW	6865 Btu	49.7%	1xFT8	1x8.8 MW, 2P	$25,800,000	$740
KA10B-1	36.1 MW	6760 Btu	50.5%	1xGT10B	1x12 MW, 2P	$28,340,000	$785
1 x RB211-6556	36.7 MW	6725 Btu	50.7%	1xRB211	1x11 MW, 2P	$24,400,000	$665
CC1-2500+	38.4 MW	6570 Btu	51.9%	1xLM2500+	1x12 MW, 2P	$27,300,000	$710
CC105P	38.5 MW	8180 Btu	41.7%	1xFr.5PA	1x18 MW, 2P	$24,260,000	$630
1 x RB211-6562	40.6 MW	6535 Btu	52.2%	1xRB211	1x12 MW, 2P	$27,000,000	$665
1 x RB211-6762	41.5 MW	6435 Btu	53.0%	1xRB211	1x12 MW, 2P	$27,200,000	$657
1 x RB211-6761DLE	44.2 MW	6275 Btu	54.4%	1xRB211	1x12 MW, 2P	$28,700,000	$650
CC1-6000	55.6 MW	6620 Btu	52.5%	1xLM6000PC	1x13 MW, 2P	$36,975,000	$665

Plant Model	Net Plant Output	Heat Rate Btu/kWh	Net Effic	Gas Turbines	Steam Turbines	Budget Price	$ p k
KAX100-1	62.0 MW	6320 Btu	54.0%	1xGTX100	1x21 MW, 2P	$41,000,000	$6
S-106B	64.3 MW	6970 Btu	49.0%	1xFr. 6B	1x24 MW, 2P	$41,800,000	$6
1 x Trent-DLE	66.0 MW	6285 Btu	54.3%	1xTrent	1x16 MW, 2P	$42,900,000	$6
FT8 Twin	66.7 MW	6770 Btu	50.4%	2xFT8	1x18 MW, 2P	$42,200,000	$6
1.W251B	71.5 MW	7140 Btu	47.8%	1xW251B11/12	1x25 MW, 2P	$55,600,000	$7
KA10B-2	73.2 MW	6730 Btu	50.7%	2xGT10B	1x25 MW, 2P	$48,500,000	$[illegible]
2 x RB211-6556 ...	73.5 MW	6725 Btu	50.7%	2xRB211	1x23 MW, 2P	$46,300,000	$[illegible]
1 x Trent	74.2 MW	6470 Btu	52.7%	1xTrent	1x16 MW, 2P	$44,500,000	$[illegible]
KA8C-1	77.4 MW	6740 Btu	50.6%	1xGT8C	1x25 MW, 2P	$52,300,000	$[illegible]
CC205P	77.8 MW	8110 Btu	42.1%	2xFr.5PA	1x27 MW, 2P	$47,850,000	$[illegible]
2025	80.5 MW	6890 Btu	49.5%	2xH-25	1x28 MW, 2P	$49,500,000	$[illegible]
2 x RB211-6562 ...	81.3 MW	6530 Btu	52.2%	2xRB211	1x24 MW, 2P	$51,200,000	$[illegible]
KA8C2-1S	82.0 MW	6825 Btu	50.0%	1xGT8C2	1x26 MW, 2P	$52,000,000	$[illegible]
2 x RB211-6762. ..	82.8 MW	6355 Btu	53.7%	2xRB211	1x24 MW, 2P	$52,200,000	$[illegible]
KA10C-2	83.6 MW	6590 Btu	51.8%	2xGT10C	1x27 MW, 2P	$53,000,000	$[illegible]
2 x RB211-6761DLE	88.4 MW	6270 Btu	54.4%	2xRB211	1x25 MW, 2P	$55,700,000	$[illegible]
1S.64.3A	99.8 MW	6540 Btu	52.2%	1xV64.3A	1x31 MW, 2P	$83,300,000	$[illegible]
1xP200-PFBC	100.0 MW	8030 Btu	42.5%	1xGT35P	1x83 MW, Cond.	$110,000,000	$1
CC2-6000	106.5 MW	6610 Btu	51.6%	2xLM6000PC	1x22 MW, 2P	$69,760,000	$[illegible]
S-106FA	107.1 MW	6440 Btu	53.0%	1xFr.6FA	1x40 MW, 3P, RH	$88,400,000	$[illegible]
KAX100-2	124.5 MW	6285 Btu	54.3%	2xGTX100	1x42 MW, 2P	$75,000,000	$[illegible]
S-107EA	130.2 MW	6800 Btu	50.2%	1xFr. 7EA	1x48 MW, 3P	$79,100,000	$[illegible]
S-206B	130.7 MW	6850 Btu	49.8%	2xFr. 6B	1x49 MW, 2P	$84,800,000	$[illegible]
2 x Trent-DLE	132.0 MW	6285 Btu	54.3%	2xTrent	1x32 MW, 2P	$85,000,000	$[illegible]
2.W251B	143.5 MW	7110 Btu	48.0%	2x251B11/12	1x51 MW, 2P	$96,000,000	$[illegible]
KA13D-1	147.1 MW	6920 Btu	48.6%	1xGT13D	1x53 MW, 1P	$83,100,000	$[illegible]
2 x Trent	148.2 MW	6470 Btu	52,7%	2xTrent	1x32 MW, 2P	$85,000,000	$[illegible]

Plant Model	Net Plant Output	Heat Rate Btu/kWh	Net Effic	Gas Turbines	Steam Turbines	Budget Price	$ pe kW
1.84.2	163.0 MW	6630 Btu	51.5%	1xV84.2	1x60 MW, 2P	$89,900,000	$55
KA11N2-1	168.0 MW	6860 Btu	49.7%	1xGT11N2	1x56 MW, 2P	$91,600,000	$54
1.W501D5A	173.0 MW	6760 Btu	50.5%	1x501D5A	1x59 MW, 2P	$95,900,000	$55
Cobra 264.3	183.4 MW	6595 Btu	51.7%	2xV64.3	1x64 MW, 2P	$96,600,000	$52
S-109E	189.2 MW	6570 Btu	52.0%	1xFr. 9E	1x70 MW, 2P	$101,695,000	$53
2.64.3A.	201.1 MW	6490 Btu	52.6%	2xV64.3A	1x75 MW, 2P	$108,270,000	$53
MPCP1-M701D ...	212.5 MW	6635 Btu	51.4%	1xM701D	1x70 MW, 2P	$110,860,000	$52
S-206FA	218.7 MW	6305 Btu	54.1%	2xFr. 6FA	1x84 MW, 3P, R	$119,500,000	$54
1.V94.2	232.9 MW	6600 Btu	51.7%	1xV94.2	1x86 MW, 2P	$118,100,000	$50
1S84.3A	260.0 MW	5980 Btu	57.1%	1xV84.3A	1x84 MW, 3P, R	$126,440,000	$48
KA24-1 ICS	260.0 MW	6040 Btu	56.5%	1xGT24	1x102 MW, 2P	$126,300,000	$48
S-107FA	262.6 MW	6090 Btu	56.0%	1xFr. 7FA	1x95 MW, 3P, R	$130,960,000	$49
S-207EA	263.6 MW	6700 Btu	50.9%	2xFr. 7EA	1x101 MW, 3P	$130,800,000	$49
1.W501F	283.3 MW	6090 Btu	56.0%	1xW501F	1x103 MW, 3P, R	$133,150,000	$47
1S.94.2A	294.3 MW	6190 Btu	55.1%	1xV94.2A	1x95 MW, 3P, R	$128,900,000	$43
2.W501D5A	346.9 MW	6740 Btu	50.6%	2x501D5A	1x118 MW, 2P	$157,800,000	$45
1S.W501G	365.0 MW	5880 Btu	58.0%	1xW501G	1x125 MW, 3P, R	$158,400,000	$43
KA26-1	378.0 MW	5985 Btu	57.0%	1xGT26	1x140 MW, 3P, R	$157,360,000	$41
S-109FA	390.8 MW	6020 Btu	56.7%	1xFr. 9FA	1x142 MW, 3P, R	$157,200,000	$40
1S.V94.3A	392.2 MW	5945 Btu	57.4%	1xV94.3A	1x120 MW, 3P, R	$155,230,000	$39
MPCP1-M701F.. .	397.7 MW	5988 Btu	57.0%	1xM701F	1x132 MW, 3P, R	$157,300,000	$39
S107H	400.0 MW	5690 Btu	60.0%	1xMS7001H	1x140 MW, 3P, R	$200,000,000	$50
MPCP2-M701D.. .	426.6 MW	6610 Btu	51.6%	2xM701D	1x142 MW, 2P	$182,200,000	$42
2.V94.2	466.6 MW	6590 Btu	51.8%	2xV94.2	1x173 MW, 2P	$181,500,000	$38
Cobra 294.2	477.9 MW	6506 Btu	52.4%	2xV94.2	1x178 MW, 2P	$183,000,000	$38
KA13E2-2	480.0 MW	6450 Btu	52.9%	2xGT13E2	1x167 MW, 2P	$185,900,000	$38
KA11N2-3	517.0 MW	6550 Btu	52.1%	3xGT11N2	1x172 MW, 2P	$198,000,000	$38

New Gas Turbine Models

Introduced Between 1998 and 2002

Close to 40 new gas turbine designs for electric power generation and prime mover applications were released for production from 1998 up to 2002. Except for the TF50A, which is a marine propulsion engine, the base load design ratings listed apply to electric power output at the generator terminals.

Model	ISO Base Load	LHV Heat Rate	Efficiency	Firing Temp	Intro	Gas Turbine Design
ST5R	395 kW	10,435 Btu/kWh	32.7%	---	2002	Pratt & Whitney
ST5	457 kW	14,520 Btu/kWh	23.5%	---	2002	Pratt & Whitney
TB3-137	1100 kW	14,630 Btu/kWh	23.3%	1520°F	1999	Motor Sich-Progress
VT2600	2250 kW	11,270 Btu/kWh	30.3%	1810°F	2001	Volvo Aero
Eurodyn	2370 kW	11,515 Btu/kWh	29.6%	---	1998	Turbomeca
ST30	3340 kW	10,660 Btu/kWh	32.0%	---	1999	Pratt & Whitney
ASE50	3770 kW	11,170 Btu/kWh	30.6%	2020°F	1999	Vericor
GTU-4P	4000 kW	14,355 Btu/kWh	23.8%	1435°F	1998	Aviadvigatel
ST40	4040 kW	10,310 Btu/kWh	33.1%	---	1999	Pratt & Whitney
Mercury 50	4200 kW	8980 Btu/kWh	38.0%	2125°F	1998	Solar Turbines
VT4400	4220 kW	12,180 Btu/kWh	28.0%	1930°F	1999	Volvo Aero
TF50A	5365 hp	0.456 lbs/hp-hr	---	1940°F	1993	Vericor
THM1203-A	5760 kW	15,158 Btu/kWh	22.5%	1725°F	1999	MAN Turbo
PGT5B	5900 kW	10,695 Btu/kWh	31.9%	1975°F	1999	Nuovo Pignone
GTU-6P	6150 kW	13,075 Btu/kWh	26.1%	1710°F	2000	Aviadvigatel

Model	ISO Base Load	LHV Heat Rate	Firing Efficiency	Temp	Intro	Gas Turbine Design
601-KB9	6450 kW	10,615 Btu/kWh	32.2%	2020°F	1998	Rolls-Royce
601-KB11	7920 kW	10,350 Btu/kWh	33.0%	2030°F	1999	Rolls-Royce
ASE 120	9780 kW	9765 Btu/kWh	34.9%	2320°F	2000	Vericor
THM1304-11	10,760 kW	11,460 Btu/kWh	29.8%	1823°F	2000	MAN Turbo
UGT-10000	10,700 kW	9440 Btu/kWh	36.2%	2160°F	1998	Mashproekt
GTE-70	10,780 kW	9480 Btu/kWh	36.0%	2165°F	1999	Zorya
GTU-12PER	12,360 kW	10,375 Btu/kWh	32.9%	2060°F	2001	Aviadvigatel
Titan 130	12,800 kW	10,250 Btu/kWh	33.3%	2050°F	1998	Solar Turbines
Cyclone	12,880 kW	9820 Btu/kWh	34.8%	2280°F	1998	Alstom
GTU-16PER	16,400 kW	9810 Btu/kWh	34.8%	2200°F	2001	Aviadvigatel
L20A	17,000 kW	10,040 Btu/kWh	34.0%	2280°F (nozzle)	2001	Kawasaki
GTU-25PER	24,650 kW	9030 Btu/kWh	37.8%	2325°F	2002	Aviadvigatel
FT8 Plus	27,970 kW	8900 Btu/kWh	38.3%	2160°F	2001	Pratt & Whitney
DR61P Vectra	30,380 kW	8690 Btu/kWh	39.3%	2270°F	1998	Dresser-Rand
RB211-6761. DLE	31,750 kW	8735 Btu/kWh	39.1%	2260°F	1999	Rolls-Royce
GTX100	43,000 kW	9215 Btu/kWh	37.0%	2190°F	1998	Alstom
LM6000 Sprint	47,300 kW	8250 Btu/kWh	41.4%	2265°F	1998	General Electric
Twin FT8 Plus	56,340 kW	8840 Btu/kWh	38.6%	2160°F	2001	Pratt & Whitney
GT8C2	57,200 kW	9835 Btu/kWh	34.7%	2012°F	1998	Alstom
UGT-110000	114,500 kW	9480 Btu/kWh	36.0%	2210°F	1999	Mashproekt
PG7001B	183,150 kW	9200 Btu/kWh	36.9%	2500°F	2000	General Electric
PG7001H	260,000 kW	8640 Btu/kWh	39.5%	2600°F	2001	General Electric
PG9001H	292,000 kW	8640 Btu/kWh	39.5%	2600°F	2000	General Electric
M701G	334,000 kW	8630 Btu/kWh	39.5%	2580°F	1999	Mitsubishi

APPENDIX E

SAMPLE COGENERATION POWER PLANT DESIGN USING "COGENERATION DESIGN" PROGRAM

Table E.1 General Inputs for the Cogeneration System

Avarage Ambiant Temperature	280 K
Outside Pressure	101.1 kPa
Relative Humidity	% 70
Fuel Type	Natural Gas
System Configuration	GT, HRSG, ST+ Refrigeration
Max TIT	1250 K
Approximate Plant Output	50MW-100MW

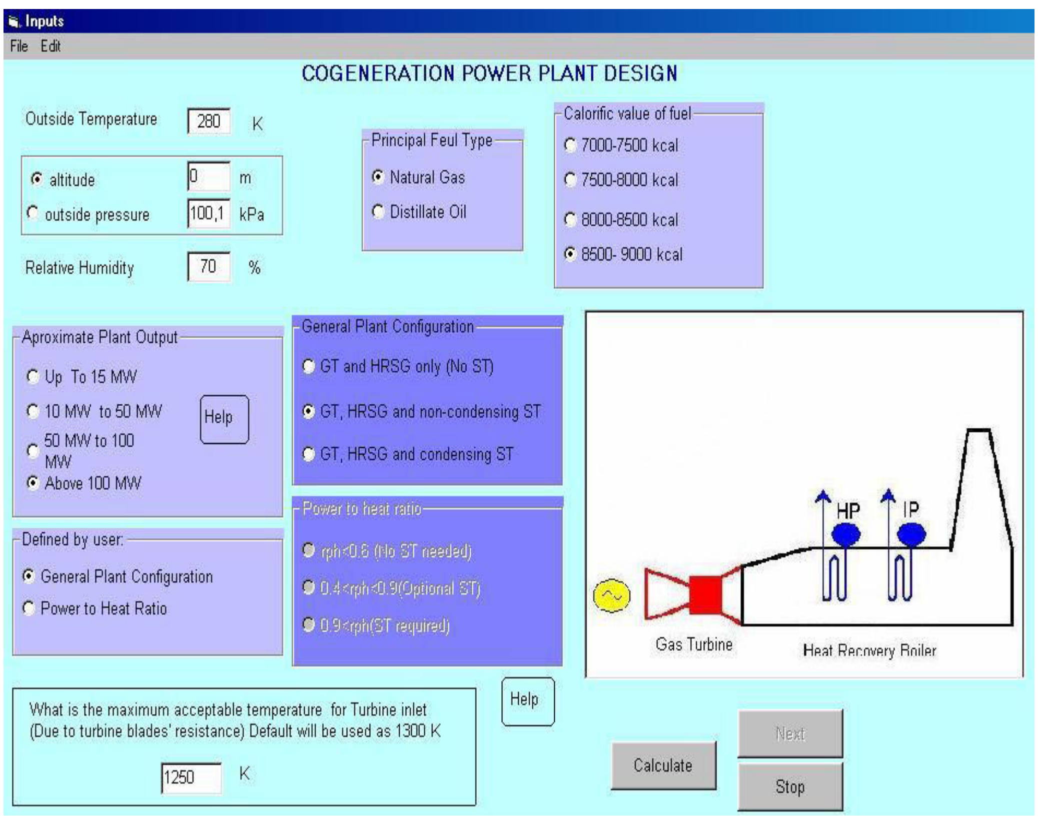

Figure E.1. Input Form

Table E.2. Inputs for the System Design

Number of Steam Take offs	1
Process Steam Pressure	800 kPa
Type Of Process	Heating+Refrigeration
Process Water Temperature	250 C
Condensate Return Temperature	98 C
Condensate Return Percentage	%97

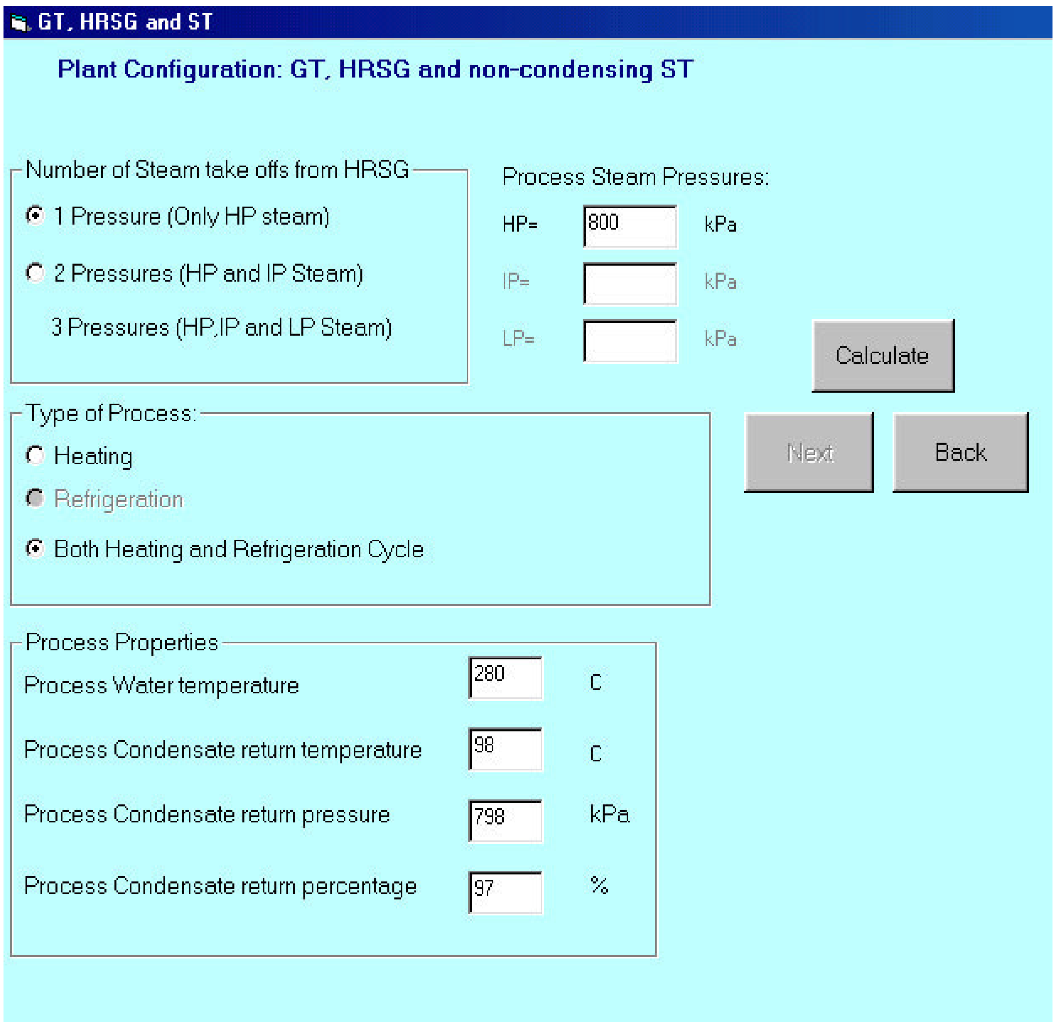

Figure E.2. System Design Form

Table E.3. Required Power and Process Needs

Design According To	Electrical Power and rph
Needed Electrical Power Output	80000 kW
Power to Heat Ratio	0.8
ST Cycle Design: Max Cycle Pressure	8000
Refrigeration Temperature	-15
Refrigeration Power Input	10000 kW

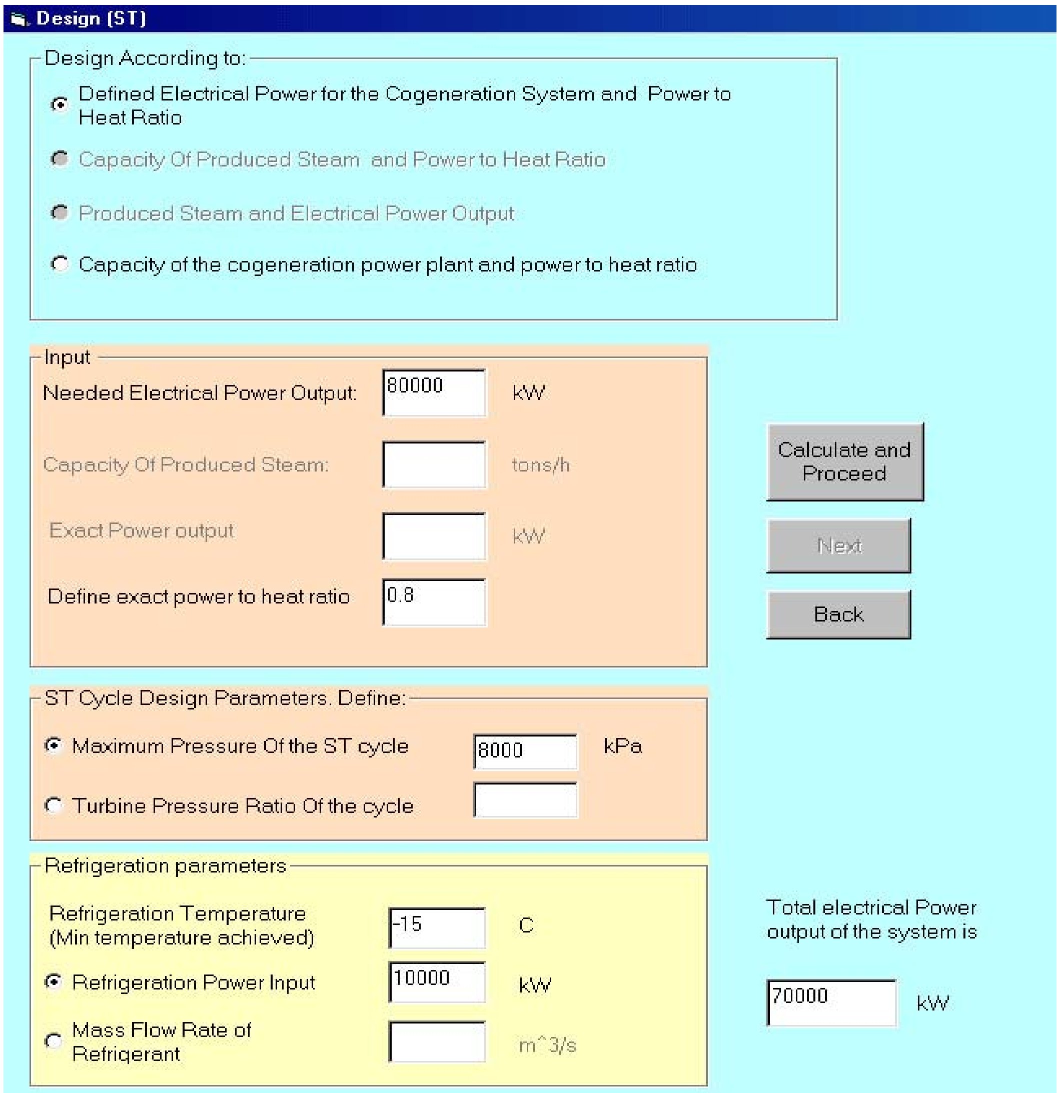

Figure E.3. Design Form-2

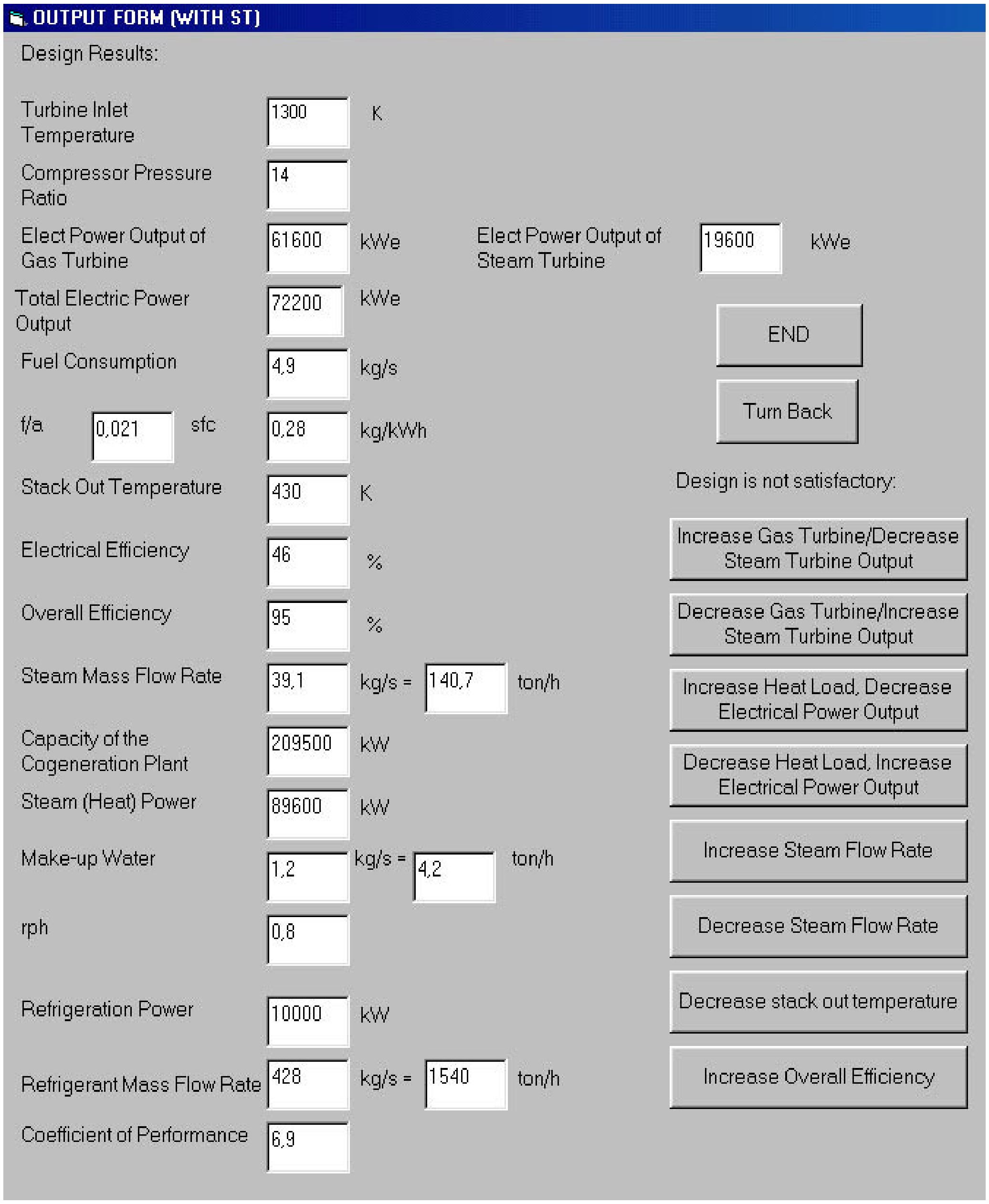

Figure E.4. Output Form

APPENDIX F

APPENDIX F

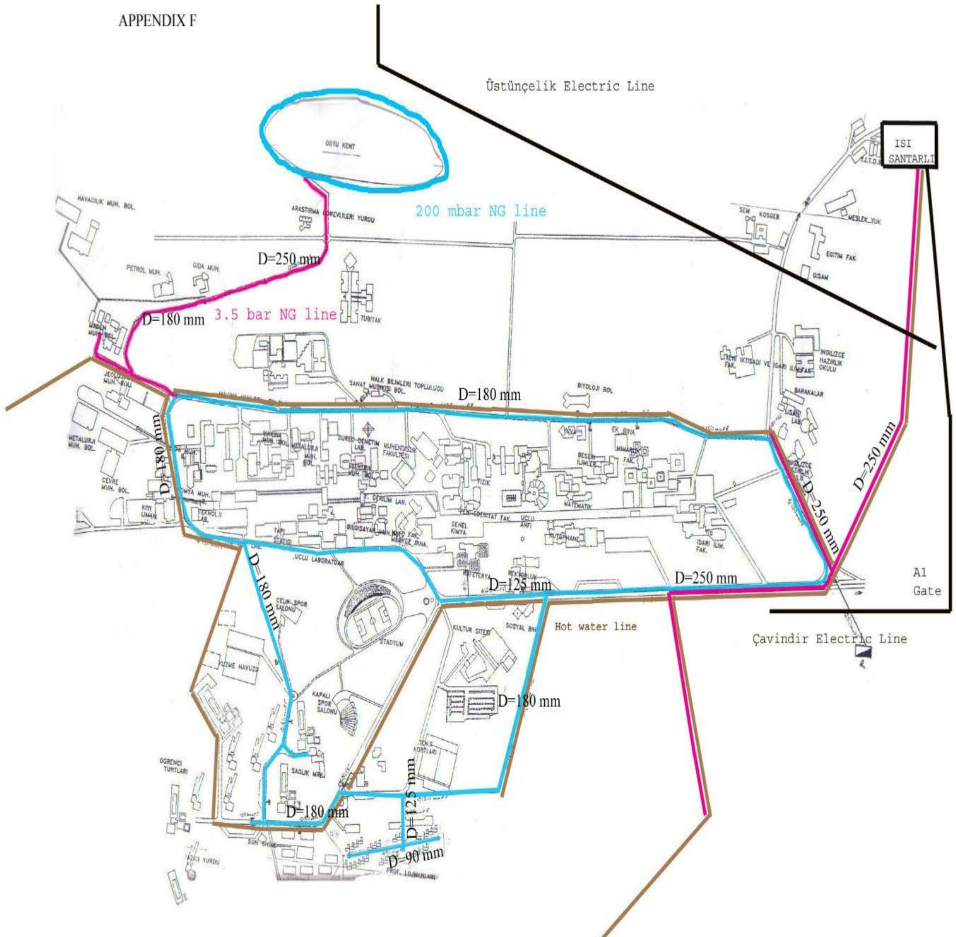

Figure F.1. METU Campus Layout

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APPENDIX G

G.1. METU 8 YEARS ELECTRICAL CONSUMPTIONS

G.2 METU 8 YEARS NATURAL GAS CONSUMPTIONS

	January	February	March	April	May	June	July	August	September	October	November	December
Year	kwh	kwh	kwh	kwh	kwh	kwh	kwh	kwh	kwh	kwh	kwh	kwh
1996	2294170	1381300	1512745	1422840	790815	164140	581040	566755	777895	1314715	13544830	7919500
1997	3210320	918310	1074535	1800685	2295035	1506765	754010	909020	999825	1428570	1917063	3210320
1998	2601300	1780200	875610	1894050	1710338	1526625	863190	1055700	941850	1032048	3256110	2928705
1999	3061530	2924048	2786565	607890	2793180	1417950	1367580	890100	934950	1203360	1411050	2236290
2000	4592640	1314450	1935450	1504200	1407600	1425540	1380000	1117800	1112280	1649100	1902330	3100000
2001	3726000	1718790	1336530	1741215	1726380	1920960	1107450	1223370	1461420	1389660	2274240	3000120
2002	5031826	2368195	1800151	2305403	2199699	2302658	1663089	1877040	1324564	2806394	2028738	4644006
2003	4640250	2160369	1960614	2668304	1775076	1984980	1763121	1665062	1665213	3615725	1402846	3021548

Figure G.1 METU Electrical Consumption

	January	February	March	April	May	June	July	August	September	October	November	December	TOTAL(NM^3
1995	1637840	1366747	1356068	1099996	474188	182806	70859	0	121470	763980	1618146	1802238	**10494338**
1996	1777871	1433051	1694067	1159433	427620	341245	96406	0	181874	730704	1294059	1428485	**10564815**
1997	1785140	1732618	1744937	1328371	514480	351876	94424	0	239570	785196	1439199	1900839	**11916650**
1998	1914735	1695695	1719779	789656	427171	330872	0	0	239569	470052	1343520	1867536	**10798585**
1999	1770715	1713996	1554354	1217772	462516	382216	69720	0	164184	709608	1497468	1827236	**11369785**
2000	2215742	2107060	964680	995820	563652	427884	46536	0	354468	887064	1383252	1773208	**11719366**
2001	1861791	1648375	1188120	709440	529668	365700	0	0	191080	507526	1485140	1906042	**10392882**
2002	2277978	1531876	1125897	1187618	532841	482870	0	0	203078	427631	1482186	2136449	**11388424,2**
2003	1809624	1836751	1872111	1265344	466135	324629	0	0	212824	596643	1396435	1980630	**11761126**

Table G.1 Natural Gas Consumption

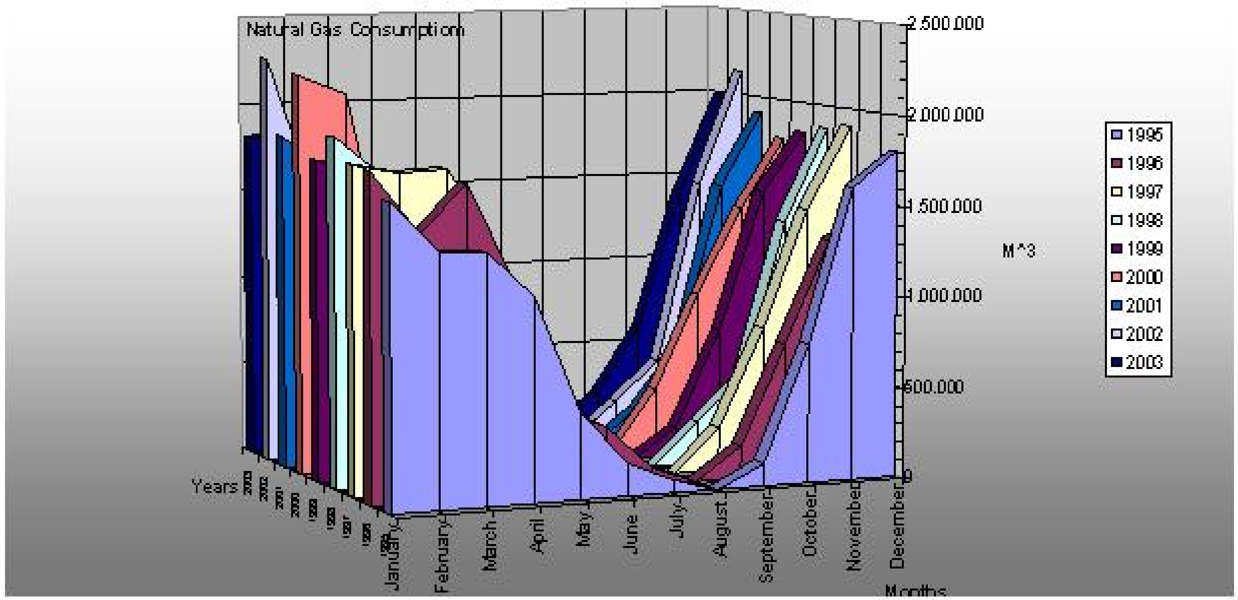

Figure G.2 Natural Gas Consumption

APPENDIX H

VERIFICATION OF 8 CASES DISCUSSED IN CHAPTER 6, BY USING THERMOFLOW SOFTWARE;

- **GAS TURBINE SELECTIONS WITH DETAILS OF THE SYSTEM PARAMETERS**
- **ECONOMICAL ANALYSIS**
- **COST SUMMARIES**

APPENDIX H_PART 1

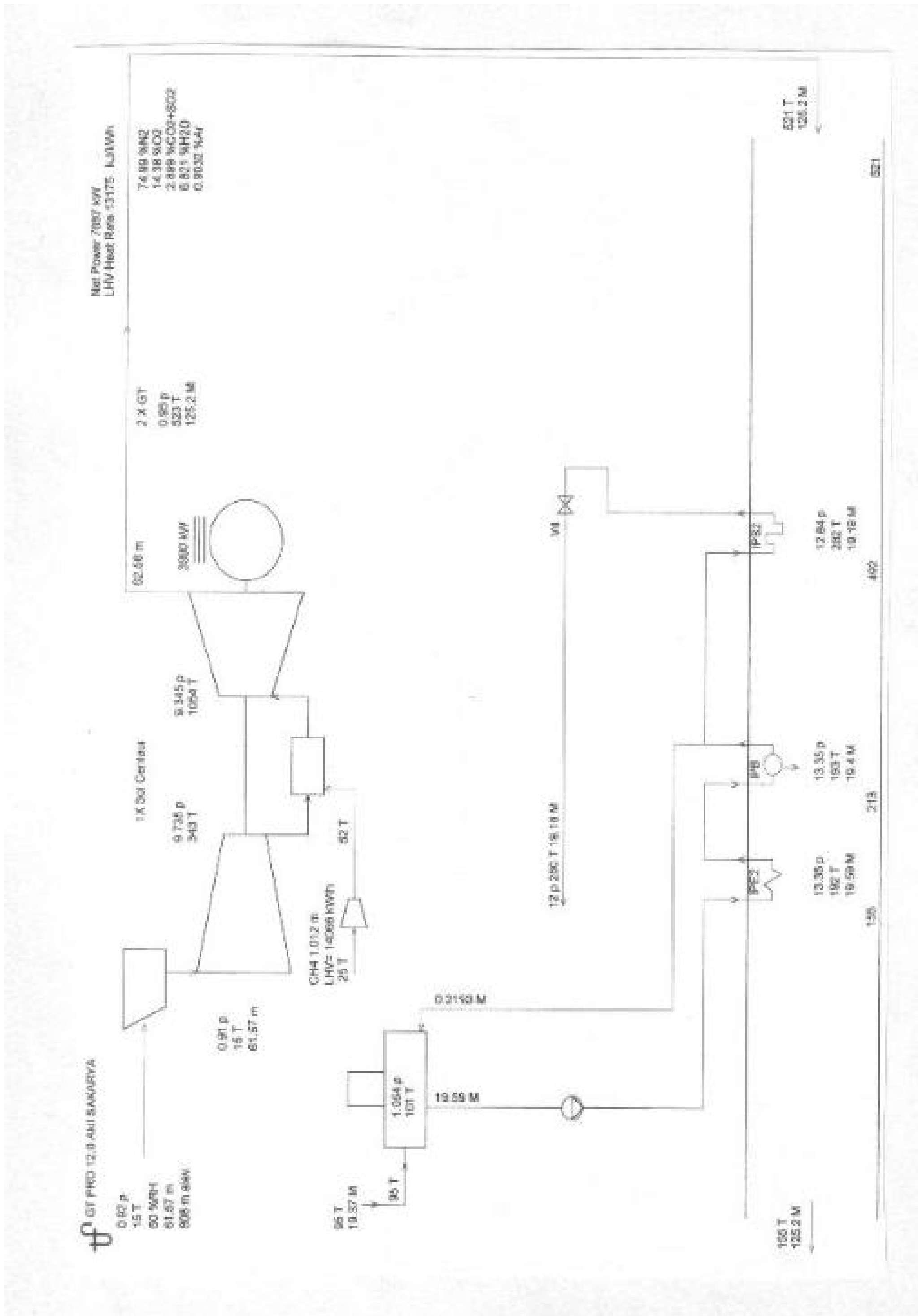

Figure H.1.1. Gas Turbine Schematics for Case:1

Table H.1.1. Project Cost Summary for Case:1

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	4.281.870	4.710.057	USD
II Other Equipment	461.231	507.355	USD
III Civil	323.842	199.059	USD
IV Mechanical	837.737	542.174	USD
V Electrical	499.422	294.432	USD
VI Buildings & Structures	126.533	80.507	USD
VII Engineering & Plant Startup	804.867	802.437	USD
Subtotal - Contractor's Internal Cost	**7.335.503**	**7.136.020**	**USD**
VIII Contractor's Soft & Miscellaneous Costs	1.065.591	883.212	USD
Contractor's Price	**8.401.094**	**8.019.232**	**USD**
IX Owner's Soft & Miscellaneous Costs	756.098	721.731	USD
Total - Owner's Cost	**9.157.192**	**8.740.963**	**USD**
Net Plant Output	**7,7**	**7,7**	**MW**
Cost per kW - Contractor's	**1.093**	**1.043**	**USD per kW**
Cost per kW - Owner's	**1.191**	**1.137**	**USD per kW**

Total Plant (Reference Basis):	Reference Cost	Hours
Commodities	662.152	
Labor	1.055.217	36.373

Effective Labor Rates:	Cost per Hour
Civil Account	25,04
Mechanical Account	29,00
Electrical Account	30,00

Buildings	% of Total Cost	Estimated Cost
Labor	50	63.267
Material	50	63.267
Labor Hours		

Table H.1.2. Financial Summary for Case:1

Financial Summary

Annual Electricity Exported	62.27	10^6 kWh
Annual Steam Exported	405	TJ
Annual Fuel Imported	820	TJ LHV
Total Investment	8,741,000	USD
Specific Investment	1137.1	USD per kW
Initial Equity	2,622,000	USD
Cumulative Net Cash Flow	26,698,000	USD
Internal Rate of Return on Investment (ROI)	16.773	%
Internal Rate of Return on Equity (ROE)	31.693	%
Years for Payback of Equity	3.544	years
Net Present Value	3,562,000	USD
Break-even Electricity Price @ Input Fuel Price	0.0392	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	4.614	USD/GJ

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APPENDIX H_PART 2

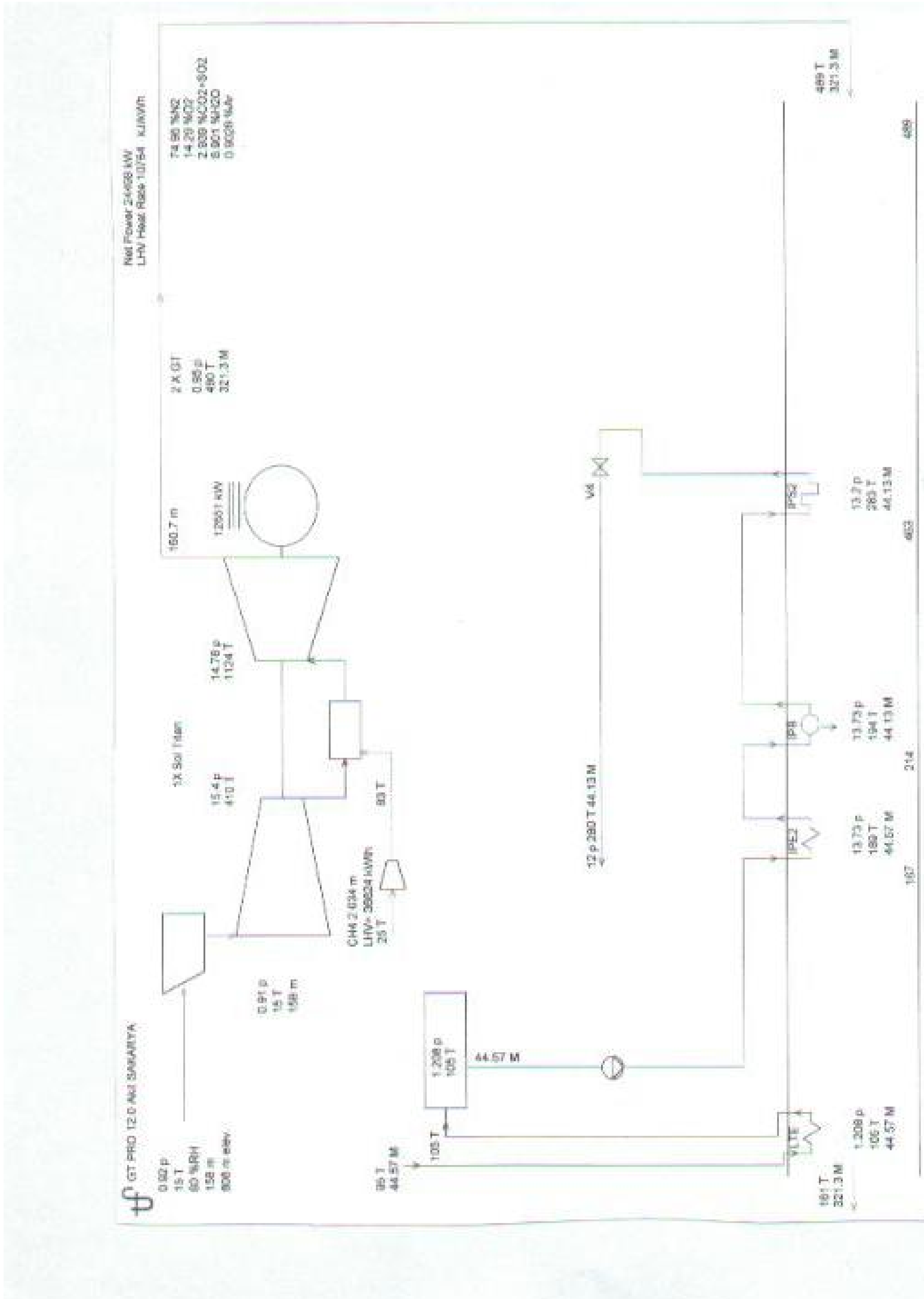

Figure H.2.1. Gas Turbine Schematics for Case:2

Table H.2.1. Project Cost Summary for Case:2

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	11.424.910	11.996.156	USD
II Other Equipment	1.100.075	1.155.078	USD
III Civil	640.003	743.719	USD
IV Mechanical	1.506.133	1.769.072	USD
V Electrical	738.423	871.167	USD
VI Buildings & Structures	803.570	924.106	USD
VII Engineering & Plant Startup	1.443.503	1.444.934	USD
Subtotal - Contractor's Internal Cost	**17.656.617**	**18.904.231**	**USD**
VIII Contractor's Soft & Miscellaneous Costs	2.354.438	2.615.193	USD
Contractor's Price	**20.011.055**	**21.519.424**	**USD**
IX Owner's Soft & Miscellaneous Costs	1.800.995	1.936.748	USD
Total - Owner's Cost	**21.812.050**	**23.456.172**	**USD**
Net Plant Output	**24,5**	**24,5**	**MW**
Cost per kW - Contractor's	**817**	**878**	**USD per kW**
Cost per kW - Owner's	**890**	**957**	**USD per kW**

Total Plant (Reference Basis):	Reference Cost	Hours
Commodities	1.182.184	
Labor	1.834.873	63.175

Effective Labor Rates:	Cost per Hour
Civil Account	25,02
Mechanical Account	29,00
Electrical Account	30,00

Buildings	% of Total Cost	Estimated Cost	Hours
Labor	50	401.785	
Material	50	401.785	
Labor Hours			15.211

Table H.2.2. Financial Summary for Case:2

Financial Summary

AnnualElectricityExported	198	10^6 kWh
AnnualSteamExported	931	TJ
AnnualFuelImported	2,140	TJ LHV
TotalInvestment	23,455,000	USD
SpecificInvestment	957.4	USD per kW
InitialEquity	7,036,000	USD
CumulativeNetCashFlow	90,839,000	USD
InternalRateofReturnonInvestment(ROI)	19.890	%
Internal Rate of Return on Equity (ROE)	40.230	%
Years forPayback ofEquity	2.732	years
NetPresentValue	14,529,000	USD
Break-even ElectricityPrice @ InputFuelPrice	0.0361	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	5.08	USD/GJ

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APPENDIX H_PART 3

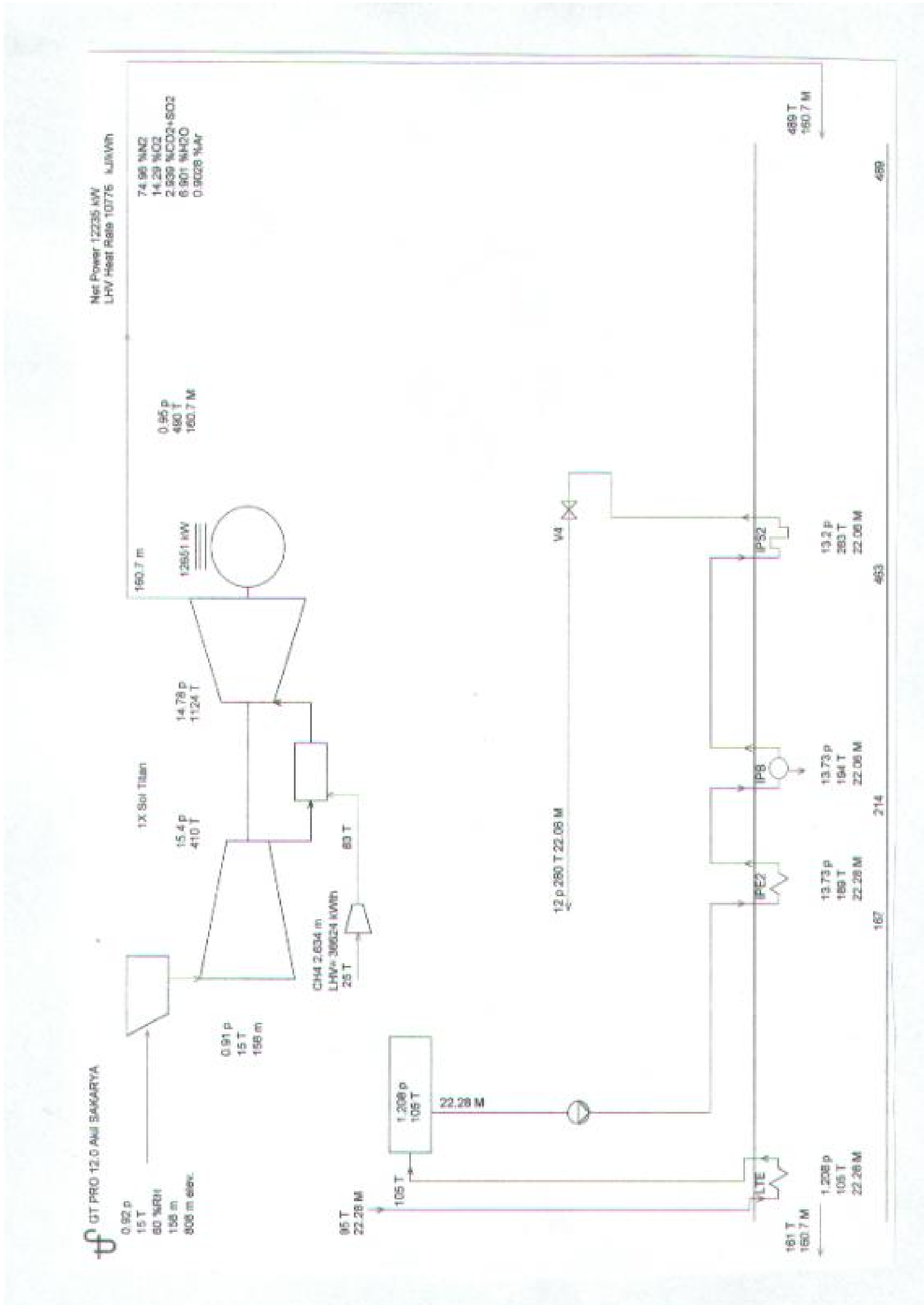

Figure H.3.1. Gas Turbine Schematics for Case:3

Table H.3.1. Project Cost Summary for Case:3

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	6.079.820	6.383.811	USD
II Other Equipment	672.418	706.039	USD
III Civil	418.743	485.630	USD
IV Mechanical	869.614	1.022.145	USD
V Electrical	593.273	698.022	USD
VI Buildings & Structures	535.677	616.029	USD
VII Engineering & Plant Startup	1.005.786	1.006.609	USD
Subtotal - Contractor's Internal Cost	10.175.331	10.918.285	USD
VIII Contractor's Soft & Miscellaneous Costs	1.378.708	1.540.040	USD
Contractor's Price	11.554.039	12.458.325	USD
IX Owner's Soft & Miscellaneous Costs	1.039.864	1.121.249	USD
Total - Owner's Cost	12.593.903	13.579.575	USD
Net Plant Output	12,2	12,2	MW
Cost per kW - Contractor's	944	1.018	USD per kW
Cost per kW - Owner's	1.029	1.109	USD per kW

Total Plant (Reference Basis):	Reference Cost	Hours	
Commodities	771.686		
Labor	1.186.131	40.855	
Effective Labor Rates:	Cost per Hour		
Civil Account	25,03		
Mechanical Account	29,00		
Electrical Account	30,00		
Buildings	% of Total Cost	Estimated Cost	
Labor	50	267.839	
Material	50	267.839	Hours
Labor Hours			
			10.136

Table H.3.2. Financial Summary for Case:3

Financial Summary

AnnualElectricityExported	99.11	10^6 kWh
AnnualSteamExported	465	TJ
AnnualFuelImported	1,070	TJ LHV
TotalInvestment	13,580,000	USD
SpecificInvestment	1109.9	USD per kW
InitialEquity	4,074,000	USD
CumulativeNetCashFlow	43,878,000	USD
InternalRate ofReturn onInvestment(ROI)	17.462	%
Internal Rate of Return on Equity (ROE)	33.537	%
Years forPayback ofEquity	3.331	years
NetPresentValue	6,156,000	USD
Break-even ElectricityPrice @ InputFuelPrice	0.0382	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	4.884	USD/GJ

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APPENDIX H_PART 4_1

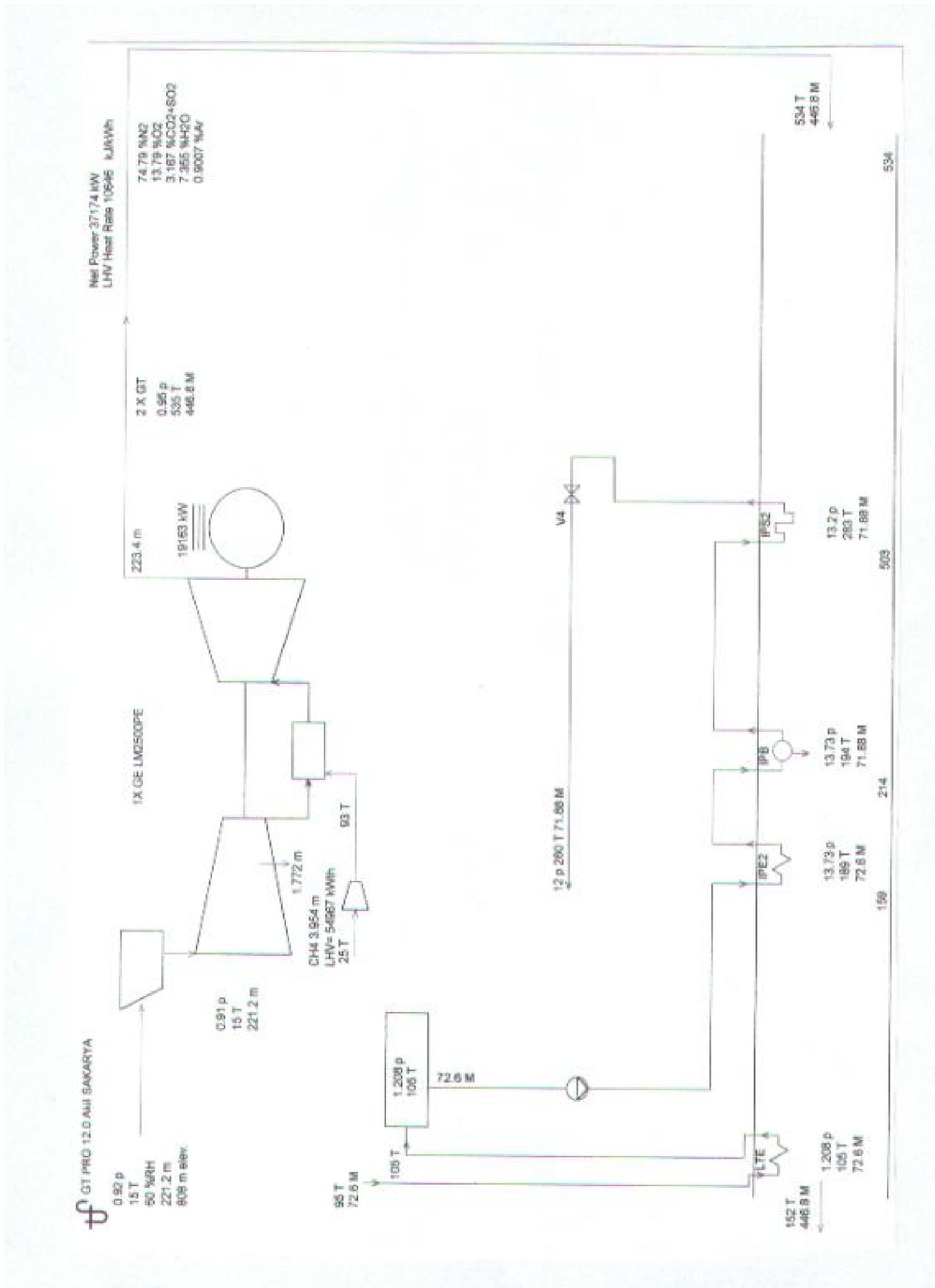

Figure H.4.1.1 Gas Turbine Schematics for Case: 4_1

Table H.4.1.1 Project Cost Summary for Case: 4_1

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	20.795.810	21.835.601	USD
II Other Equipment	1.288.161	1.352.569	USD
III Civil	912.196	1.064.427	USD
IV Mechanical	2.095.204	2.466.085	USD
V Electrical	892.645	1.053.572	USD
VI Buildings & Structures	988.117	1.136.335	USD
VII Engineering & Plant Startup	1.811.612	1.813.689	USD
Subtotal - Contractor's Internal Cost	**28.783.746**	**30.722.278**	**USD**
VIII Contractor's Soft & Miscellaneous Costs	3.743.390	4.125.258	USD
Contractor's Price	**32.527.136**	**34.847.536**	**USD**
IX Owner's Soft & Miscellaneous Costs	2.927.442	3.136.278	USD
Total - Owner's Cost	**35.454.578**	**37.983.814**	**USD**
Net Plant Output	**37,2**	**37,2**	**MW**
Cost per kW - Contractor's	**875**	**938**	**USD per kW**
Cost per kW - Owner's	**954**	**1.022**	**USD per kW**

Total Plant (Reference Basis):	Reference Cost	Hours
Commodities	1.596.596	
Labor	2.496.060	86.132

Effective Labor Rates:	Cost per Hour
Civil Account	25,01
Mechanical Account	29,00
Electrical Account	30,00

Buildings	% of Total Cost	Estimated Cost	Hours
Labor	50	494.059	
Material	50	494.059	
Labor Hours			18.707

Table H.4.1.2. Financial Summary for Case: 4_1

Financial Summary

AnnualElectricityExported	301	10^6 kWh
AnnualSteamExported	1,520	TJ
AnnualFuelImported	3,210	TJ LHV
TotalInvestment	37,987,000	USD
SpecificInvestment	1021.9	USD per kW
InitialEquity	11,396,000	USD
CumulativeNetCashFlow	148,755,000	USD
InternalRate of Return on Investment (ROI)	20.048	%
Internal Rate of Return on Equity (ROE)	40.677	%
Years for Payback of Equity	2.699	years
NetPresentValue	23,954,000	USD
Break-even Electricity Price @ Input Fuel Price	0.0349	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	5.207	USD/GJ

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APPENDIX H_PART 4_2

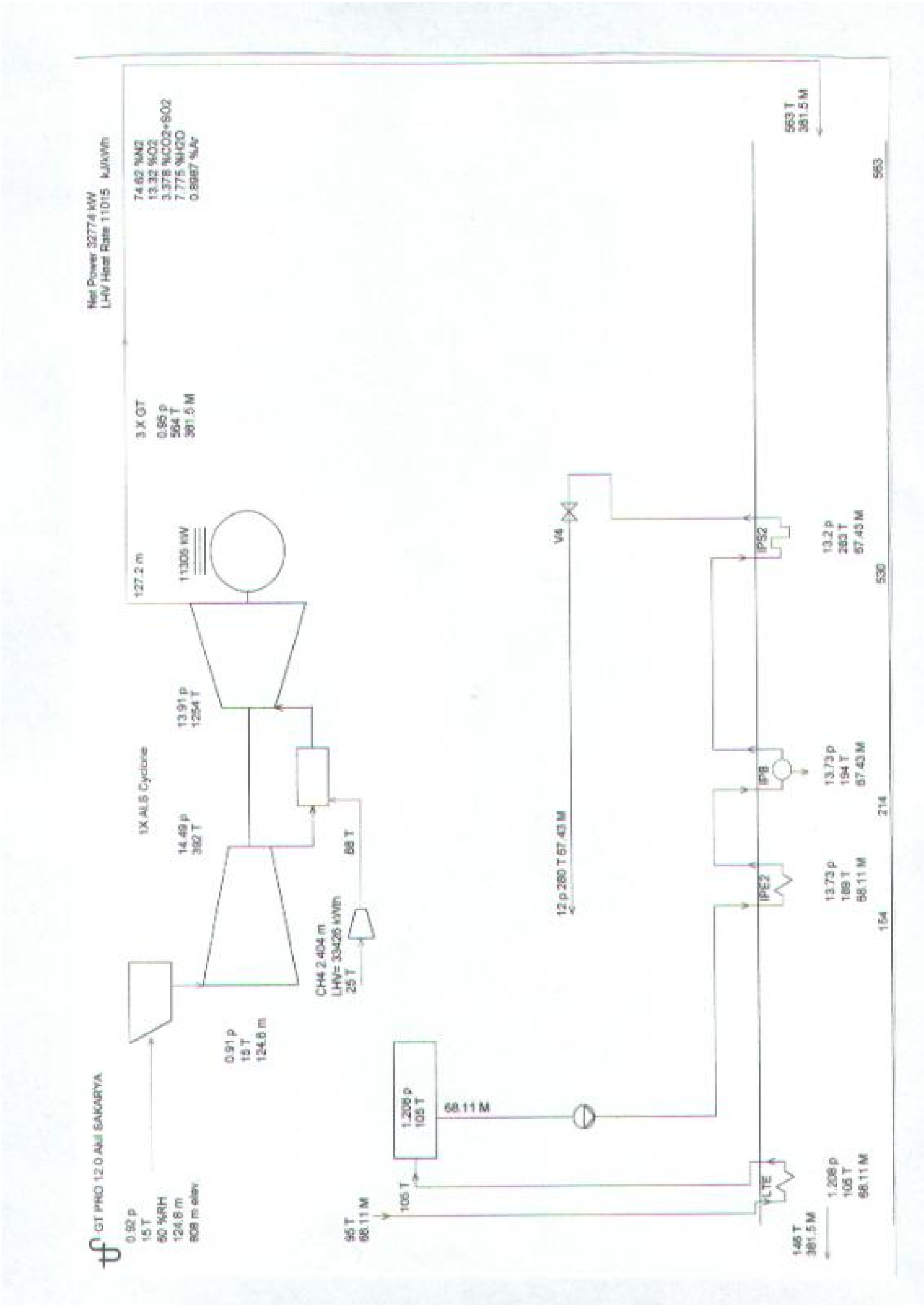

Figure H.4.2.1 Gas Turbine Schematics for Case: 4_2

Table H.4.2.1 Project Cost Summary for Case: 4_2

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	16.172.510	16.981.136	USD
II Other Equipment	1.411.326	1.481.892	USD
III Civil	960.126	1.123.462	USD
IV Mechanical	2.774.164	3.272.481	USD
V Electrical	1.027.830	1.213.571	USD
VI Buildings & Structures	1.102.402	1.267.762	USD
VII Engineering & Plant Startup	1.695.733	1.697.594	USD
Subtotal - Contractor's Internal Cost	25.144.091	27.037.898	USD
VIII Contractor's Soft & Miscellaneous Costs	3.482.192	3.888.395	USD
Contractor's Price	28.626.283	30.926.293	USD
IX Owner's Soft & Miscellaneous Costs	2.576.365	2.783.366	USD
Total - Owner's Cost	31.202.649	33.709.659	USD
Net Plant Output	32,8	32,8	MW
Cost per kW - Contractor's	874	944	USD per kW
Cost per kW - Owner's	952	1.029	USD per kW

Total Plant (Reference Basis):	Reference Cost	Hours
Commodities	1.983.017	
Labor	2.951.843	102.212

Effective Labor Rates:	Cost per Hour
Civil Account	25,01
Mechanical Account	29,00
Electrical Account	30,00

Buildings	% of Total Cost	Estimated Cost	Hours
Labor	50	551.201	
Material	50	551.201	
Labor Hours			20.872

Table H.4.2.2. Financial Summary for Case: 4_2

Financial Summary

AnnualElectricityExported	265	10^6 kWh
AnnualSteamExported	1,420	TJ
AnnualFuelImported	2,920	TJ LHV
TotalInvestment	33,708,000	USD
SpecificInvestment	1028.5	USD per kW
InitialEquity	10,113,000	USD
CumulativeNetCashFlow	131,683,000	USD
InternalRate of Return on Investment(ROI)	20.013	%
Internal Rate of Return on Equity (ROE)	40.580	%
Years for Payback of Equity	2.706	years
NetPresentValue	21,174,000	USD
Break-even Electricity Price @ Input Fuel Price	0.0349	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	5.163	USD/GJ

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APPENDIX H_PART 5

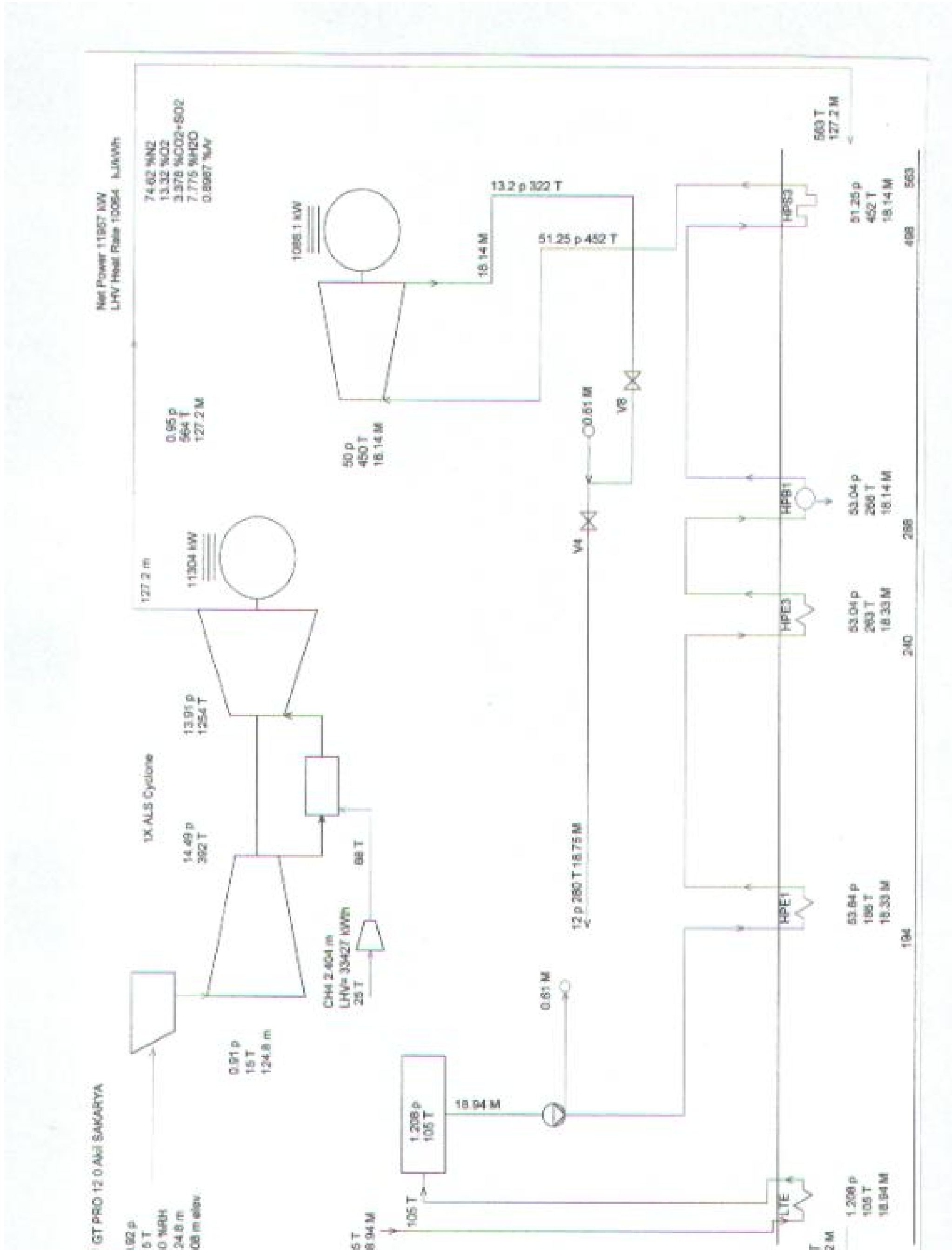

Figure H.5.1 Gas Turbine Schematics for Case: 5

Table H.5.1 Project Cost Summary for Case: 5

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	6.485.150	6.809.408	USD
II Other Equipment	823.710	864.896	USD
III Civil	481.023	560.490	USD
IV Mechanical	1.224.507	1.445.804	USD
V Electrical	741.783	872.659	USD
VI Buildings & Structures	544.951	626.694	USD
VII Engineering & Plant Startup	1.112.474	1.113.287	USD
Subtotal - Contractor's Internal Cost	11.413.599	12.293.237	USD
VIII Contractor's Soft & Miscellaneous Costs	1.604.040	1.801.701	USD
Contractor's Price	13.017.639	14.094.938	USD
IX Owner's Soft & Miscellaneous Costs	1.171.588	1.268.544	USD
Total - Owner's Cost	14.189.227	15.363.482	USD
Net Plant Output	12,0	12,0	MW
Cost per kW - Contractor's	1.088	1.179	USD per kW
Cost per kW - Owner's	1.186	1.285	USD per kW

Total Plant (Reference Basis):	Reference Cost	Hours	
Commodities	1.021.341		
Labor	1.501.442	51.889	
Effective Labor Rates:	Cost per Hour		
Civil Account	25,03		
Mechanical Account	29,00		
Electrical Account	30,00		
Buildings	% of Total Cost	Estimated Cost	Hours
Labor	50	272.476	
Material	50	272.476	
Labor Hours			10.314

Table H.5.2. Financial Summary for Case: 5

Financial Summary

AnnualElectricityExported	96.85	10^6 kWh
AnnualSteamExported	396	TJ
AnnualFuelImported	975	TJ LHV
TotalInvestment	15,364,000	USD
SpecificInvestment	1284.9	USD per kW
InitialEquity	4,609,000	USD
CumulativeNetCashFlow	40,861,000	USD
InternalRate of Return on Investment (ROI)	15.194	%
Internal Rate of Return on Equity (ROE)	27.576	%
Years for Payback of Equity	4.136	years
NetPresentValue	4,692,000	USD
Break-even Electricity Price @ Input Fuel Price	0.0408	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	4.703	USD/GJ

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APPENDIX H_PART 6

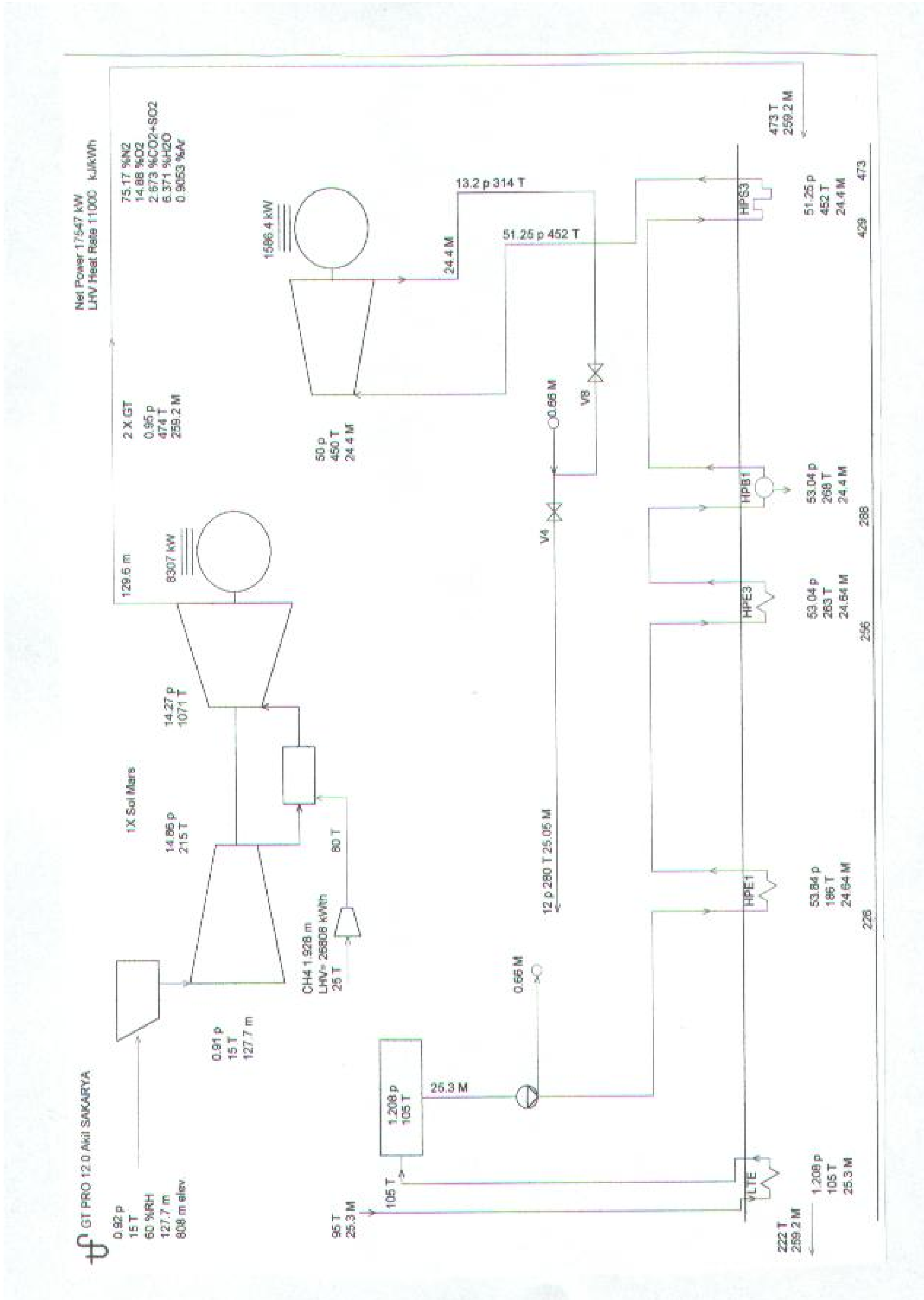

Figure H.6.1 Gas Turbine Schematics for Case: 6

Table H.6.1 Project Cost Summary for Case: 6

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	9.498.390	9.973.310	USD
II Other Equipment	1.239.462	1.301.435	USD
III Civil	578.658	673.755	USD
IV Mechanical	1.469.811	1.732.999	USD
V Electrical	857.887	1.008.847	USD
VI Buildings & Structures	715.963	823.357	USD
VII Engineering & Plant Startup	1.355.130	1.356.222	USD
Subtotal - Contractor's Internal Cost	15.715.300	16.869.925	USD
VIII Contractor's Soft & Miscellaneous Costs	2.148.854	2.398.577	USD
Contractor's Price	17.864.154	19.268.502	USD
IX Owner's Soft & Miscellaneous Costs	1.607.774	1.734.165	USD
Total - Owner's Cost	19.471.928	21.002.668	USD
Net Plant Output	17,6	17,6	MW
Cost per kW - Contractor's	1.018	1.098	USD per kW
Cost per kW - Owner's	1.110	1.197	USD per kW

Total Plant (Reference Basis):	Reference Cost	Hours
Commodities	1.185.681	
Labor	1.821.799	62.858

Effective Labor Rates:	Cost per Hour
Civil Account	25,02
Mechanical Account	29,00
Electrical Account	30,00

Buildings	% of Total Cost	Estimated Cost	Hours
Labor	50	357.981	
Material	50	357.981	
Labor Hours			13.552

Table H.6.2. Financial Summary for Case: 6

Financial Summary

AnnualElectricityExported	142	10^6 kWh
AnnualSteamExported	529	TJ
AnnualFuelImported	1,560	TJ LHV
TotalInvestment	21,003,000	USD
SpecificInvestment	1197	USD per kW
InitialEquity	6,301,000	USD
CumulativeNetCashFlow	45,843,000	USD
InternalRate of Return on Investment(ROI)	13.193	%
Internal Rate of Return on Equity (ROE)	22.570	%
Years for Payback of Equity	5.16	years
NetPresentValue	3,821,000	USD
Break-even Electricity Price @ Input Fuel Price	0.0449	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	4.255	USD/GJ

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APPENDIX H_PART 7

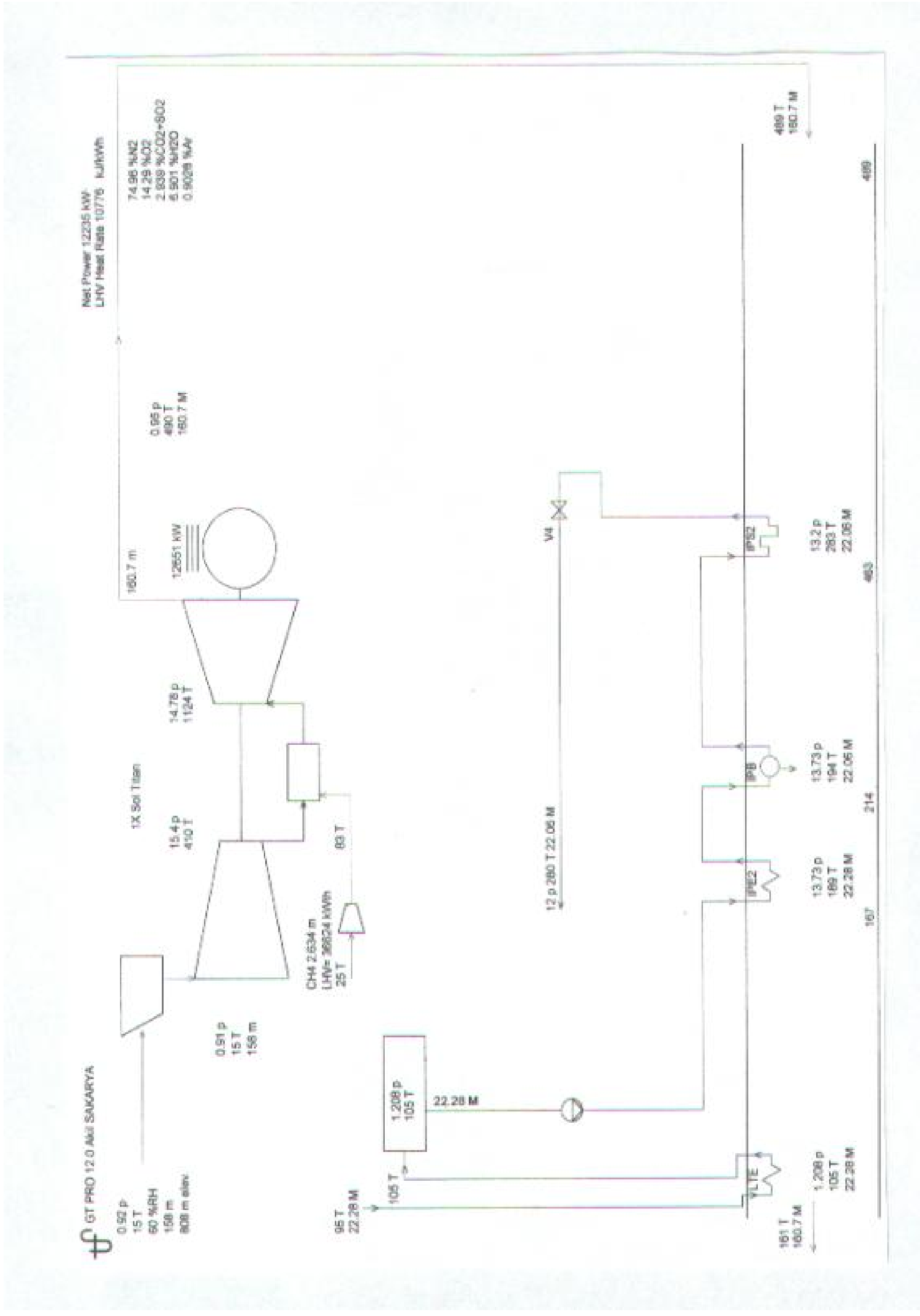

Figure H.7.1 Gas Turbine Schematics for Case: 7

Table H.7.1 Project Cost Summary for Case: 7

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	6.079.820	6.383.811	USD
II Other Equipment	672.418	706.039	USD
III Civil	418.743	485.630	USD
IV Mechanical	869.614	1.022.145	USD
V Electrical	593.273	698.022	USD
VI Buildings & Structures	535.677	616.029	USD
VII Engineering & Plant Startup	1.005.786	1.006.609	USD
Subtotal - Contractor's Internal Cost	10.175.331	10.918.285	USD
VIII Contractor's Soft & Miscellaneous Costs	1.378.708	1.540.040	USD
Contractor's Price	11.554.039	12.458.325	USD
IX Owner's Soft & Miscellaneous Costs	1.039.864	1.121.249	USD
Total - Owner's Cost	12.593.903	13.579.575	USD
Net Plant Output	12,2	12,2	MW
Cost per kW - Contractor's	944	1.018	USD per kW
Cost per kW - Owner's	1.029	1.109	USD per kW

Total Plant (Reference Basis):	Reference Cost	Hours	
Commodities	771.686		
Labor	1.186.131	40.855	
Effective Labor Rates:	Cost per Hour		
Civil Account	25,03		
Mechanical Account	29,00		
Electrical Account	30,00		
Buildings	% of Total Cost	Estimated Cost	Hours
Labor	50	267.839	
Material	50	267.839	
Labor Hours			10.136

Table H.7.2. Financial Summary for Case: 7

Financial Summary

AnnualElectricityExported	99.11	10^6 kWh
AnnualSteamExported	465	TJ
AnnualFuelImported	1,070	TJ LHV
TotalInvestment	13,580,000	USD
SpecificInvestment	1109.9	USD per kW
InitialEquity	4,074,000	USD
CumulativeNetCashFlow	43,878,000	USD
InternalRate of Return on Investment (ROI)	17.462	%
Internal Rate of Return on Equity (ROE)	33.537	%
Years for Payback of Equity	3.331	years
NetPresentValue	6,156,000	USD
Break-even Electricity Price @ Input Fuel Price	0.0382	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	4.884	USD/GJ

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APPENDIX H_PART 8_1

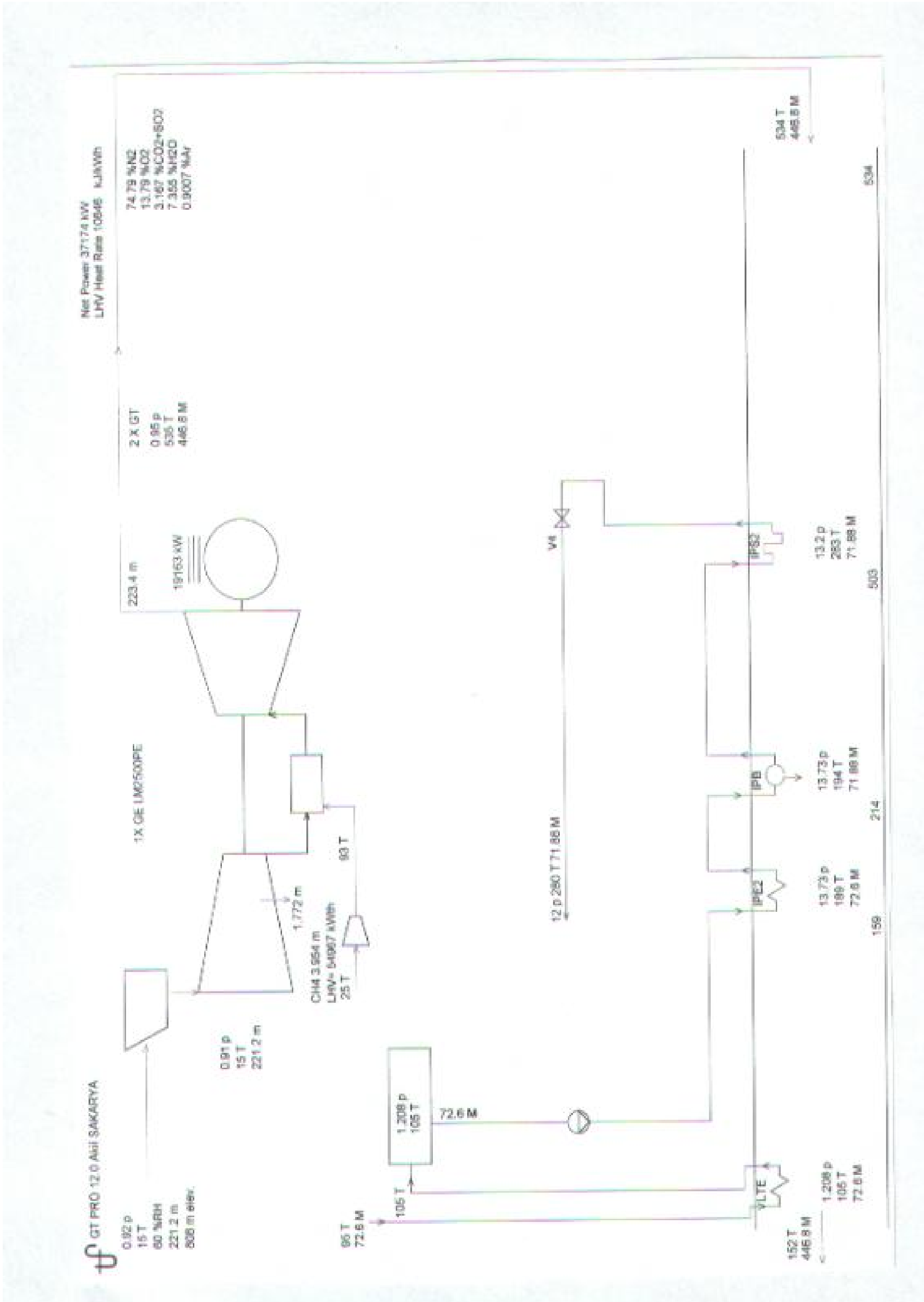

Figure H.8.1.1 Gas Turbine Schematics for Case: 8_1

Table H.8.1.1 Project Cost Summary for Case: 8_1

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	20.795.810	21.835.601	USD
II Other Equipment	1.288.161	1.352.569	USD
III Civil	912.196	1.064.427	USD
IV Mechanical	2.095.204	2.466.085	USD
V Electrical	892.645	1.053.572	USD
VI Buildings & Structures	988.117	1.136.335	USD
VII Engineering & Plant Startup	1.811.612	1.813.689	USD
Subtotal - Contractor's Internal Cost	28.783.746	30.722.278	USD
VIII Contractor's Soft & Miscellaneous Costs	3.743.390	4.125.258	USD
Contractor's Price	32.527.136	34.847.536	USD
IX Owner's Soft & Miscellaneous Costs	2.927.442	3.136.278	USD
Total - Owner's Cost	35.454.578	37.983.814	USD
Net Plant Output	37,2	37,2	MW
Cost per kW - Contractor's	875	938	USD per kW
Cost per kW - Owner's	954	1.022	USD per kW

Total Plant (Reference Basis):	Reference Cost	Hours	
Commodities	1.596.596		
Labor	2.496.060	86.132	
Effective Labor Rates:	Cost per Hour		
Civil Account	25,01		
Mechanical Account	29,00		
Electrical Account	30,00		
Buildings	% of Total Cost	Estimated Cost	Hours
Labor	50	494.059	
Material	50	494.059	
Labor Hours			18.707

Table H.8.1.2. Financial Summary for Case: 8_1

Financial Summary

AnnualElectricityExported	301	10^6 kWh
AnnualSteamExported	1,520	TJ
AnnualFuelImported	3,210	TJ LHV
TotalInvestment	37,987,000	USD
SpecificInvestment	1021.9	USD per kW
InitialEquity	11,396,000	USD
CumulativeNetCashFlow	148,755,000	USD
InternalRate of Return on Investment (ROI)	20.048	%
Internal Rate of Return on Equity (ROE)	40.677	%
Years for Payback of Equity	2.699	years
NetPresentValue	23,954,000	USD
Break-even Electricity Price @ Input Fuel Price	0.0349	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	5.207	USD/GJ

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APPENDIX H_PART 8_2

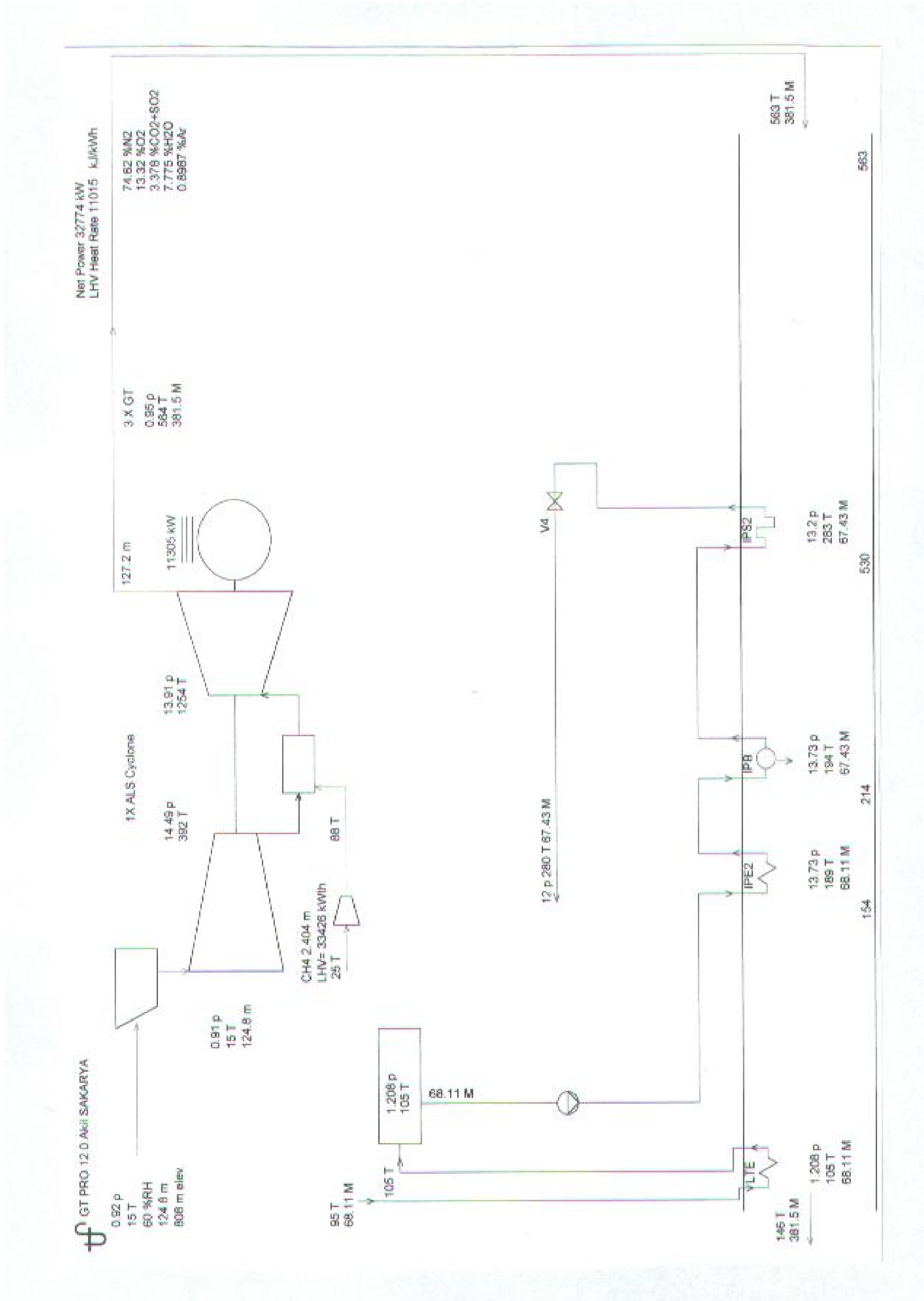

Figure H.8.2.1 Gas Turbine Schematics for Case: 8_2

Table H.8.2.1 Project Cost Summary for Case: 8_2

Project Cost Summary	Reference Cost	Estimated Cost	
I Specialized Equipment	16.172.510	16.981.136	USD
II Other Equipment	1.411.326	1.481.892	USD
III Civil	960.126	1.123.462	USD
IV Mechanical	2.774.164	3.272.481	USD
V Electrical	1.027.830	1.213.571	USD
VI Buildings & Structures	1.102.402	1.267.762	USD
VII Engineering & Plant Startup	1.695.733	1.697.594	USD
Subtotal - Contractor's Internal Cost	25.144.091	27.037.898	USD
VIII Contractor's Soft & Miscellaneous Costs	3.482.192	3.888.395	USD
Contractor's Price	28.626.283	30.926.293	USD
IX Owner's Soft & Miscellaneous Costs	2.576.365	2.783.366	USD
Total - Owner's Cost	31.202.649	33.709.659	USD
Net Plant Output	32,8	32,8	MW
Cost per kW - Contractor's	874	944	USD per kW
Cost per kW - Owner's	952	1.029	USD per kW

Total Plant (Reference Basis):	Reference Cost	Hours
Commodities	1.983.017	
Labor	2.951.843	102.212

Effective Labor Rates:	Cost per Hour
Civil Account	25,01
Mechanical Account	29,00
Electrical Account	30,00

Buildings	% of Total Cost	Estimated Cost	Hours
Labor	50	551.201	
Material	50	551.201	
Labor Hours			20.872

Table H.8.2.2. Financial Summary for Case: 8_2

Financial Summary

AnnualElectricityExported	265	10^6 kWh
AnnualSteamExported	1,420	TJ
AnnualFuelImported	2,920	TJ LHV
TotalInvestment	33,708,000	USD
SpecificInvestment	1028.5	USD per kW
InitialEquity	10,113,000	USD
CumulativeNetCashFlow	131,683,000	USD
InternalRateofReturnonInvestment(ROI)	20.013	%
Internal Rate of Return on Equity (ROE)	40.580	%
Years for Payback of Equity	2.706	years
NetPresentValue	21,174,000	USD
Break-even ElectricityPrice @ InputFuelPrice	0.0349	USD/kWhr
Break-even Fuel LHV Price @ Input Electricity Price	5.163	USD/GJ

PEACE/GT PRO 12.0 Thermoflow, Inc. *Note: Totals may not tally due to round-off.*
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