A THEORETICAL AND EXPERIMENTAL INVESTIGATION FOR DEVELOPING A METHODOLOGY FOR CO/POLY-GENERATION SYSTEMS; WITH SPECIAL EMPHASIS ON TESTING, ENERGY AND EXERGY RATING

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ABSTRACT

A THEORETICAL AND EXPERIMENTAL INVESTIGATION FOR DEVELOPING A METHODOLOGY FOR CO/POLY-GENERATION SYSTEMS; WITH SPECIAL EMPHASIS ON TESTING, ENERGY AND EXERGY RATING

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A poly-generation system can be defined as the simultaneous and collocated generation of two or more energy supply types, aimed to maximize the utilization of the thermodynamic potential (efficiency) of the consumed energy resources. A Poly-generation system may involve co-generation (power and heat) or tri-generation (power, heat, and cold) processes and may also be connected to a district energy system. A poly-generation plant reclaims heat in a useful form that would be wasted otherwise in separate electricity and heat (and chilled water in some cases) generating systems. By this way a poly-generation plant provides a variety of benefits including improved efficiency and fuel savings, reduction of the primary energy demand total cost of utility service and unit fuel cost, independency for energy and protection of environment. With the overall efficiencies in the range of 70-90%, poly-generation systems are gaining popularity all around the world, including Turkey. In spite of all their potential benefits and increasing interest for poly-generation systems, there is not yet any rating, testing, metrication and classification guidelines and standards. It is indeed very important to rate the performance and energy savings potential, determine the heat and power outputs, estimate the system efficiency and the ratio of the split of the power produced between thermal and electric. These are the information which are hard to determine since there are not enough common test standards, rating standards and nor consensus-based terminology for combined heat and power systems in the world literature. Even the classification of the cogeneration systems is hardly globalized. Aim of this study is to develop a common procedure with respect to the above shortcomings for testing and rating poly-generation systems under realistic operating conditions with accurate formulae which will help to contribute energy and exergy economy by establishing a robust metrication standard based on new evaluation parameters. This study aims to find a procedure to evaluate a poly-generation system by establishing standard test methods and evaluation tools in terms of parameters like energy and exergy characteristics of thermal and electric loads, temperature demand and power split for determining operational characteristics of the system. This may be achieved by revising and expanding DIRECTIVE 2004/8/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL. A case study is expected to be based on a trigeneration power plant to be received within the framework of the EU FP6 HEGEL Poly-generation project, to be tested at METU, which has a capacity of 145 kW electric and 160 kW useful heat.

Key Words: Poly-generation, CHP, energy analysis, exergy analysis, REMM, Internal Combustion Engine

ÖZ

İKİLİ/ ÇOKLU BİRLİKTE ÜRETİM SİSTEMLERİ İÇİN TEST, ENERJİ VE EKSERJİ SINIFLANDIRILMASINA ODAKLI BİR METODOLOJİNİN GELİŞTİRİLMESİNE YÖNELİK TEORİK VE DENEYSEL ARAŞTIRMA

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Çoklu birlikte üretim, enerji kaynağından en yüksek termodinamik verimin sağlanabilmesi amacı ile, faydalı ısıdan buhar, soğutulmuş su, sıcak su ve elektrik gibi farklı ekonomik taleplerin aynı anda ve bir arada karşılanabilmesi olarak tanımlanabilir. Bu taleplerin ayrı ayrı üretimi sırasında kullanılamayıp harcanan ısının, çoklu birlikte üretim sayesinde kullanılabilmesi, termodinamik verimin %70-90 gibi yüksek değerlere çıkarılabilmesinin yanı sıra, ana enerji ihtiyacının ve toplam kurulum maliyetinin azaltılabilmesini, enerji üretiminde bağımsızlığı sağlamakta ve çevrenin korumasına katkıda bulunmaktadır. Tüm bu olumlu özellikleri ile dünyada olduğu gibi çoklu üretim sistemleri Türkiye'de de oldukça talep görmeye başlamıştır. Çoklu birlikte üretim sistemleri; kojenerasyon-çoklu üretim ve üçlü birlikte üretim olarak sınıflandırılabilir. Çoklu birlikte üretim sistemlerinin tüm bu potansiyel yararlarına ve oldukça artan taleplerine ramen sınıflandırılabilmeleri ve test edilebilmeleri için gerekli ilke ve standartlar oluşturulamamıştır. Çoklu birlikte üretim sistemleri için performans ve enerji tasarrufu potansiyelinin

değerlendirilebilmesi, faydalı ısı ve enerji üretimlerinin saptanması, sistem verimliliği ve faydalı ısı ve elektrik üretimleri oranının belirlenebilmesi oldukça önemlidir. Bu bilgiler, dünya literatüründe genel test ve sınıflandırma standartlarının olmaması ve çoklu birlikte üretim ile ilgili kavram karmaşaları sebebi ile genelde elde edilebilmesi oldukça zor bilgilerdir. Bu çalışmanın amacı, çoklu birlikte üretim sistemlerinin gerçekçi işletme koşullarında sınıflandırılabilmesi ve test edilebilmesi için standart test yöntemleri ve değerlendirme araçları oluşturarak genel bir yöntem geliştirmektir. Bu yöntem geliştirilirken, enerji ve ekserji ekonomisine de katkıda bulunabilecek eksiksiz denklemler oluşturulması hedeflenmiştir. Tez çalışmasının sonucunda, çoklu birlikte üretim sistemlerini giriş sıcaklığı, yük, enerji üretiminin oranı gibi temel parametreler cinsinden değerlendirebilen ve işletme niteliklerini tespit eden bir yöntem bulmak amaçlanmıştır. Bu amaca Avrupa Parlementosu ve Konseyinin yayımladığı 2004/8/EC yönergesi incelenip, geliştirilerek ulaşılabilir. Deneysel çalışmaların ODTÜ' de kurulacak olan 145 kW elektrik ve 160 kW ısı kapasiteli EU FP6 HEGEL Çoklu Birlikte Üretim projesi üzerinde yapılaması planlanmaktadır.

Anahtar Kelimler: Çoklu Birlikte Üretim, Birlikte Isı-Güç Üretimi, Enerji Analizi, Ekserji Analizi, REMM, İçten Yanmalı Motor

To All My Loved Ones, Especially My Family

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AEC	Alternatifve energy ratio, dimensionless
AF	Availability factor , dimensionless
BLF	Base load factor, dimensionless
BMEP	Break mean effective pressure, (kPa)
ci	Carbon content of the fuel, (kg CO_2/kWh)
C	Power to heat ratio, dimensionless
Cmin	Minimum acceptable power to heat ratio for CHP status, dimensionless
CHPEη	Electrical efficiency of CHP defined as annual electricity from CHP divided by the fuel input used to produce the sum of useful heat output and electricity from CHP, dimensionless
CHPHη	Thermal efficiency of CHP defined as annual useful heat output divided by the fuel input used to produce the sum of useful heat output and electricity from CHP, dimensionless
COP	Heat pump coefficient of performance, dimensionless
C_{pw}	Constant pressure spesific heat, (kJ/kg)
e	Specific flow exergy (kJ/kg)
e_{fuel}	Specific flow exergy of fuel (kJ/kg)
e_{in}	Spesific inflow exergy (kJ/kg)
e_{out}	Spesific outflow exergy (kJ/kg)
E	Electrical energy, or power (taken unity), kW-h (Btu), or kW (Btu·h^{-1})
$\dot{E}_{heat}$	Heat exergy rate, (kW)
$\dot{E}_{dest}$	Exergy destruction rate, (kW)
$\dot{E}_{exhaust}$	Exergy rate of exhaust gases, (kW)
E_{CHP}	Useful electric output of CHP system to the consumer, kW-h (Btu), or kW (Btu·h^{-1})
EL	Engine life expectancy factor, dimensionles

EUF	Energy utilization ratio/factor, dimensionles
f_c	Calorific value of fuel, kg·(kW-h)$^{-1}$ (lb·Btu^{-1})
H	Thermal energy (heat), or thermal power, kW-h (Btu), or kW (Btu·h^{-1})
IC	initial cost of, (TL or $)
FC	Fuel cost, (TL or $)
FC_{CHP}	Fuel cost of operating a poly-generation unit, (TL or $)
FS	Fuel savings ratio, dimensionless
H_{CHP}	Heat output of CHP to the consumer, kW-h (Btu), or kW (Btu·h^{-1})
he	Inlet enthalpy, (kJ/kg)
hi	Outlet enthalpy, (kJ/kg)
h_{sat_l}	Saturated liquid enthalpy, (kJ/kg)
h_{sup}	Superheater enthalpy, (kJ/kg)
h_{w_in}	Inlet water enthalpy, (kJ/kg)
HF	Heat following factor, dimensionless
$\dot{m}_i$	Inlet mass flow rate, (kg/s)
$\dot{m}_e$	Exit mass flow rate, (kg/s)
$\dot{m}_{fuel} \cdot H_i$	Fuel mass flow rate, (kg/s)
$\dot{m}_{wc}$	Mass flow rate of cooling water, (kg/s)
MS	Money saving,(TL or $)
n	Rotational speed, (rev/s)
NL	Noice level factor, dimensionless
P_{aux}	Auxilaries power output, (kW)
$\mathbb{P}_b$	Engine Break power, (kW)
$P_{generator}$	Generator power, (kW)
P_{gross}	Gross power output, (kW)
$\mathbb{P}_i$	Engine indicated power, (kW)
$\mathbb{P}_{ng\_comp}$	NG compressor power, (kW)
P_{net}	Net power of the system, (kW)
P_{pumps}	Pump power, (kW)
PES	Percent primary energy savings (according to Directive 2004/8/EC, dimensionless

PES_{RCHP}	Percent primary energy-exergy savings of CHP, dimensionless
PESR	Primary energy-exergy savings, dimensionless
Q	Quality of heat and power , dimensionless
$\dot{Q}$	Heat Power, (kW)
Q_a	The heat generated in the absorption process, (kJ)
Q_c	The heat rejected in condenser , (kJ)
Q_{ev}	The evaporation heat release, (kJ)
$\dot{Q}_{ICE}$	Heat Power of ICE, (kW)
$\dot{Q}_{EXCH}$	Heat Power of HEX, (kW)
$\dot{Q}_{COND}$	Heat Power of condenser, (kW)
R	Reliability , dimensionless
$RefH\eta$	Efficiency reference value for separate heat production, dimensionless
$RefE\eta$	Efficiency reference value for separate electricity production, dimensionless.
$Ref\psi_R$	Reference rational exergy efficiency, dimensionless
$Ref\psi_{RCHP}$	Reference rational exergy efficiency of CHP, dimensionless
s	Entropy, kJ/kg-K
s_0	Dead state entropy, kJ/kg-K
T	Temperature, K (°C)
T_0	Dead state surroundings temperature, K (°C)
T_a	Indoor dry-bulb air temperature, K (°C)
T_{app}	Application inlet temperature, K (°C)
T_E	Minimum source temperature that electricity can be generated, K (°C)
T_f	Flame temperature of the fuel spent in System (i), K (°C)
T_{ref}	Reference temperature of the environment that any process tends to become in equilibrium, K (°C)
T_{sat}	Saturation temperature at a given pressure, K (°C)
TIT	Turbine inlet temperature K (°C)
$\dot{W}$	Power rate, (kW)
$\dot{W}_{net}$	Net electrical power rate, (kW)

Greek Letters

α	Stoichiometric combustion proportion, dimensionless
η	Energy efficiency, dimensionless
η_{cogen}	Co-generation energy utilization (first law) efficiency, dimensionless
$\eta_{cogen,II}$	Co-generation second law efficiency, dimensionless
$\eta_{CONDENSER,II}$	Condenser second law efficiency, dimensionless
$\eta_{heat\ exh,II}$	Second law efficiency of HEX, dimensionless
η_{HEX}	HEX efficiency, dimensionless
$\eta_{ice,II}$	Second law efficiency of ICE, dimensionless
η_{inv}	Inversion efficiency of ICE, dimensionless
η_{m}	Mechanical efficiency, dimensionless
$\eta_{PUMP,II}$	Pump second law efficiency, dimensionless
η_{SE}	SE first law efficiency, dimensionless
$\eta_{SE,II}$	SE second law efficiency, dimensionless
$\eta_{thermal,ice}$	Thermal efficiency, dimensionless
ψ_{R}	Rational exergy efficiency, dimensionless
ψ_{RE}	Rational exergy efficiency of electrical energy generation of CHP, dimensionless
ψ_{RH}	Rational exergy efficiency of thermal energy generation of CHP, dimensionless
ψ_{RCHP}	Overall rational exergy efficiency of the CHP system, dimensionless
ε	Useful work (exergy) that a unit thermal energy flow (fuel) can accomplish, dimensionless
ε_{max}	Unit exergy of the fuel input ($\varepsilon_{max}= \varepsilon_{Emax} +\varepsilon_{Hmax}$) to CHP, dimensionless
ε_{min}	Minimum exergy that satisfies a given task (application) that requires unit energy, dimensionless
ε_{Emax}	Unit exergy spent in providing electricity from CHP, dimensionless
ε_{Hmax}	Unit exergy spent in providing heat from CHP, dimensionless
ε_{Emin}	Minimum unit exergy that could provide the same electricity, dimensionless

ε_{Hmin}	Minimum unit exergy that could provide the same heat for a given application, dimensionless
ε_{sup}	Exergy supply, dimensionless
ε_{dst}	Destroyed exergy, dimensionless
v	Specific volume of steam (water), m^3/kg

Abbrivations

CCHP	Combined cooling, heating and power
CEN	European Committee for Standardization
CHP	Combined Heat and Power
CRF	FIAT Centre of Research
DEPC	Diesel Engine Powered Cogeneration
CI	Compression Ignition
EC	The Council of the European Union
EU	European Union
FP6	Sixth EU Framework Programme (For research and technology development)
FWP	Feed Water Pump
GSHP	Ground source Heat Pump
HEGEL	High Efficiency Combined-Cycle Gas Poly-generator for Ecological Local Generation,
HEX	Heat exchanger
HP	High pressure
HRSG	Heat Recovery Steam Generator
ICE	Internal Combustion Engine
IP	Intermediate pressure
ISO	International Organization for Standardization
SE	Steam Engine
SI	Spark Ignition
ST	Steam Turbine

LHV	Lover heating value
LP	Low pressure
NG	Natural gas
ORC	Organic Rankine cycle
PBP	Payback Period
PP	Pinch Point
PPTD	Pinch Point Temperature Difference
RC	Rankine Cycle
REMM	Rational Exergy Management Model
RICBED	Research and Implementation Centre for Built Environment and Design (MATPUM)

CHAPTER 1

THE NEED FOR THE RESEARCH

The rising cost of energy and power, depleting nature of natural resources and the recent global warming issues have led to the urgent need for developing advanced energy conversion systems. In this respect, high efficiency poly-generation systems are gaining more attention lately, due to their added advantages in terms of increased overall thermal and exergy efficiency and reducing harmful emissions.[1]

The generation of electricity and the recovery of heat in poly-generation plants achieve overall thermal efficiencies in between 70-90% and above, corresponding to efficiencies of the poly-generation equipment. The efficiency range when compared to separate production of electrical power (35 % power efficiency), heat (80 % efficiency) and cold water [2], is obviously high. This provides a great variety of benefits including improved efficiency for production, reduction of the site's total outside purchased energy requirements and unit fuel cost. This high efficiency of the power plant also leads to protection of natural resources and protection of the environment by lowering emissions of carbon dioxide, the main greenhouse gas. By making use of waste heat and waste products; besides air pollution water pollution is also lessened by such systems. The development of poly-generation will as well combat climate change.

Considering the above mentioned advantages, it can be said that increasing and wise application of poly-generation systems in our country will contribute to stimulating the economy, saving the environment, protection of primary energy resources, ensuring reliability of energy supplies and also lessening transmission and distribution losses for electrical energy.

Gathering statistics and monitoring developments in the combined heat and power sector is difficult and can contain a considerable number of uncertainties. For cogeneration, or more generally poly-generation systems, it is important to rate the performance and energy savings potential of commercial CHP systems, estimate the system efficiency, energy rating and the ratio of the split of the power produced between heat and electricity for the power plant. These are the information which are hard to determine since there are not enough common test standards, rating standards and terminology for combined heat and power systems in the world literature. There is insufficient guidance, and no general optimization method with any corrected proof either. Even the classification of the cogeneration systems is hardly globalized.

Aim of this study is to develop a common procedure with respect to the above shortcomings for testing and rating poly-generation systems under realistic operating conditions with accurate formulae which will help to contribute energy and exergy economy.

With the mentioned procedure, since it will be easier to standardize the definitions and some terminology regarding poly-generation, it will be possible to evaluate the cogeneration system in terms of basic parameters like inlet temperature, load, energy split of the power plant and determine operational characteristics of the system, determine the performance and see if the system will be acceptable as per EU Directive.

Case studies are developed, based on FP 6 HEGEL [3] co-generation system which will provide heat and power to MATPUM building of the Department of Architecture, METU. It consists of an ICE co-generator and two heat exchangers within it, which will be described briefly in the following chapters.

Academicians, PhD. and M.Sc. students, industrial staff, R&D engineers who are studying or involved with energy, exergy, heating, cooling, electric power production, green/sustainable buildings, co-generation etc. can benefit from the this study.

CHAPTER 2

INTRODUCTION

2.1. Poly-generation

A poly-generation system can be defined as the simultaneous and collocated generation of two or more energy supply types, aimed to maximize the utilization of the thermodynamic potential (efficiency) of the consumed energy resources. [1]

A Poly-generation system, representing the supply side may involve co-generation (power and heat) or tri-generation (power, heat, and cold) plant and may also be connected to a district energy system, which may represent the demand side. Natural Gas, fuel oil, coal or preferably renewable energy resources like biogas may be used as the primary energy resource to be inputted to the poly-generation system. Such poly-generation systems should be designed and controlled with an objective of optimizing all relevant interactions between supply and demand, such as energy balance, exergy balance, temperature compatibility, and optimum split of energy supply, while green-house emissions are minimized. [4]

Poly-generation when combined with an efficient district energy system and especially with low-exergy buildings and green buildings may provide a great variety of benefits including improved energy and exergy efficiency in the built environment, quality and continuity of energy supply, independence from interconnected power grid, local power generation for production, reduced cost, and better protection of natural resources and the environment. Below, cogeneration and tri-generation processes are defined in detail.

2.1.1. Co-generation

Co-generation system, aka combined heat and power system (CHP) is an on-site power plant, which simultaneously generates both heat and power from a single entry of the primary energy resource in a more efficient way when compared to separate generation of heat and power systems having separate and/or different types of primary energy resource inputs. A co-generation system recovers heat that would normally be wasted in separate production of heat and power. The recovered heat in a cogeneration system is utilized by the customer in order to satisfy some or all of the heat demand on the customer side. By this way, a cogenerating system reduces the primary energy requirements and harmful emissions. [5,6]

Co-generation has substantial contributions to economy, environment, energy conservation and social welfare with the responsible use of primary energy resources-whether fossil or alternative- and by establishing local security of heat and power supply. It is quite usual to obtain fuel savings in the range of 15% to 45%. By making use of waste heat and waste products; besides air pollution water pollution is also reduced by eliminating or downsizing cooling towers in large power plants. Considering these contributions of cogeneration power plants, TUBİTAK and other partners have very recently completed the TSAD (Utilization of Waste Heat of Thermal Power Plants) research project and investigated efficiencies of all thermal power plants in Turkey, and prepared a project to utilize the waste heat of Yatağan Lignite power plant for heating applications in the district buildings. [7]

Co-generation is not only an option for big industry and district heating; Small enterprises, public authorities, even the owners of single-family houses can use this principle and realize the same benefits. Many colleges and municipalities abroad, which have extensive district heating and cooling systems, have cogeneration facilities.

The wide range of applications where cogeneration may be used can be classified in four groups;

1. Residential and commercial buildings including but not limited to office buildings, hospitals, schools, universities, community heating schemes, government buildings, banks etc,
2. Services sector involving hotels, swimming pools and leisure centers, stores and supermarkets, airports, shopping centers, restaurants, laundries, car washes,
3. Agriculture and horticulture sector, involving production and use of biogas and biomass, bio-diesel, textile, drying crops or wood etc
4. Industry, especially glass, ceramics and similar energy intensive industrial plants. food processing, textile production, brewing, distilling and malting, timber processing, motor industry, industrial zones or parks. Industrial parks are another good candidate for successful cogeneration applications with a district energy system. [8]

Considering all these benefits of cogeneration, thus the importance of encouragement of the existence of co-generation plants, European Parliament and The Council of The European Union published the DIRECTIVE 2004/8/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL [9] to standardize the definitions and some terminology regarding CHP. The directive defines a highly efficient co-generation system according to the primary energy savings value. (PES). PES for a feasible cogeneration system should be at least 10% compared with the references for separate production of heat and electricity, in terms of the first law of thermodynamics, which will later be explained in more detail in part 1.5.

2.1.2. Tri-generation

A tri-generation plant is a poly-generation plant that has an added service to the customer in the form of cold- mostly generated by absorption chillers that rely upon part of the heat supply to the customer. However, if the cold generation takes place on the customer side rather than the poly-generation plant, the plant remains as a cogeneration plant. [10]

A well-designed tri-generation plant produces and can supply simultaneously three different forms of energy to the customer. These are namely; heat (hot fluid and or steam), cold (chilled water) and power (electrical or mechanical). Tri-generation can be used in many industrial processes where there is a simultaneous need for power, heat and refrigeration. Best examples where tri-generation can be used are the food and beverage industry, chemical industry, buildings and industrial parks. [11, 12]

2.2. Poly-generation Technology

Especially in large power plants that only generate electrical power whether single-cycle or combined cycle, large clouds pouring out of massive cooling towers is customary, as can be seen in Figure 2.1.

Figure 2.1 Exhaust Gases Pouring out from Conventional Plants

Actually the question why all this heat irreversibly is being wasted should be asked. Poly-generation may substantially reduce this waste of heat in a typical power plant and also preserves precious water resources. With a poly-generation 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. [13]

The figure below, (Figure 2.2) illustrates this principle; in the upper half, it is shown how much electricity and heat a typical small-scale cogeneration system produces with a given number of energy input, compared to separate heat and power production. In this example, the cogeneration unit has an efficiency of 89% and in the case of separate production of heat and electricity the efficiency is much lower. Far more fuel is needed, because of the high losses in the power station, and additional losses in the electricity network and in the boiler. [8]

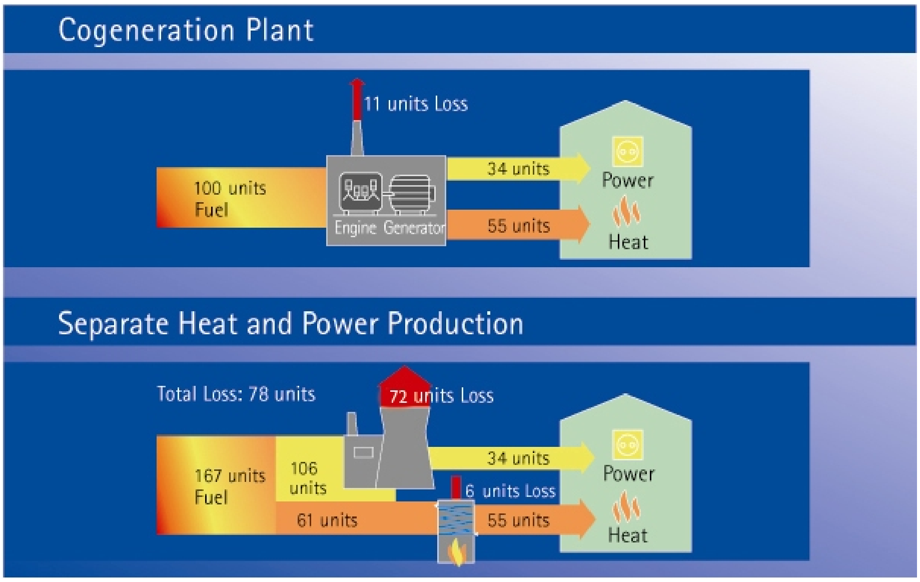


Figure 2.2 Comparison of Cogeneration Plant and Separate Heat and Power Production

Poly-generation power plants utilize a fuel source to produce electricity, hot water/steam and /or cold water/ ice simultaneously. For these processes, a typical plant consists of a prime mover where fuel is converted to mechanical power and heat and drives an electrical generator, a waste heat exchanger (heat recovery system) that recovers waste heat from the prime mover and/or exhaust gases to produce hot water or steam, a bottoming Rankine cycle including a steam turbine or a steam engine with all necessary auxiliary equipment (condenser, pump etc), a heat rejection system, electrical and mechanical interconnections between the poly-generator and the energy user, and a control system. Among all these equipment, prime mover has the greatest importance since it is the heart of the poly-generation system. The choice of the prime mover depends on the sites' heating and operating requirements, equipment availability and fuel availability. Below, different prime movers and their technological, economical and environmental features are discussed in detail.

2.3. Prime Movers

There are four principal types prime movers; gas turbine, steam turbine, steam engine and reciprocating engines (internal combustion engines-ICE). These prime movers which are the main components of poly-generation systems, transform chemical and/or thermal energy to electrical and heat energy

2.3.1. Gas Turbine

The gas turbine engine is known to have a number of attractive features, principally: low capital cost, compact size, short delivery, high flexibility and reliability, fast starting and loading [14]

In gas turbine systems, 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, or may be used directly in some process applications. Gas turbines range from 25kWe 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 poly-generation 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.

In the study, micro-turbines are of attention. They are scaled down versions of combustion turbines that provide reasonable electrical efficiency of about 30%, multi-fuel capability, low emission levels, and heat recovery potential, and need minimal maintenance. For cogeneration applications, an overall efficiency of 80% and above can be achieved. Existing micro-turbine systems range in size from 25 to 80 kW. In addition, research is ongoing for systems with capacities less than 25 kW, which will be suitable for the single-family residential buildings.

Micro-turbines offer a number of advantages including compact size, low weight, small number of moving parts and lower noise. In addition, micro-turbine based cogeneration systems have high-grade waste heat, low maintenance requirements (but require skilled personnel), low vibration and short delivery time. However, in the lower power ranges, reciprocating ICE's have higher efficiency. Besides the use of natural gas, other fuels like diesel, landfill gas, ethanol, industrial off-gases and other bio based liquids and gases can be used. [15]

2.3.1.1. Technological Aspects

The thermodynamic process of a micro-turbine involves the pressurization of intake air by the compressor. The compressed air and fuel are mixed and ignited in a combustion chamber. The resulting hot combustion gas expands turning the turbine, which drives the compressor and provides power by rotating the compressor turbine shaft. With a recuperator, the hot exhaust gas helps preheat the air as it passes from the compressor to the combustion chamber.

As shown in Figure 2.3, the basic components of micro-turbine systems are the compressor, turbine generator and the recuperator

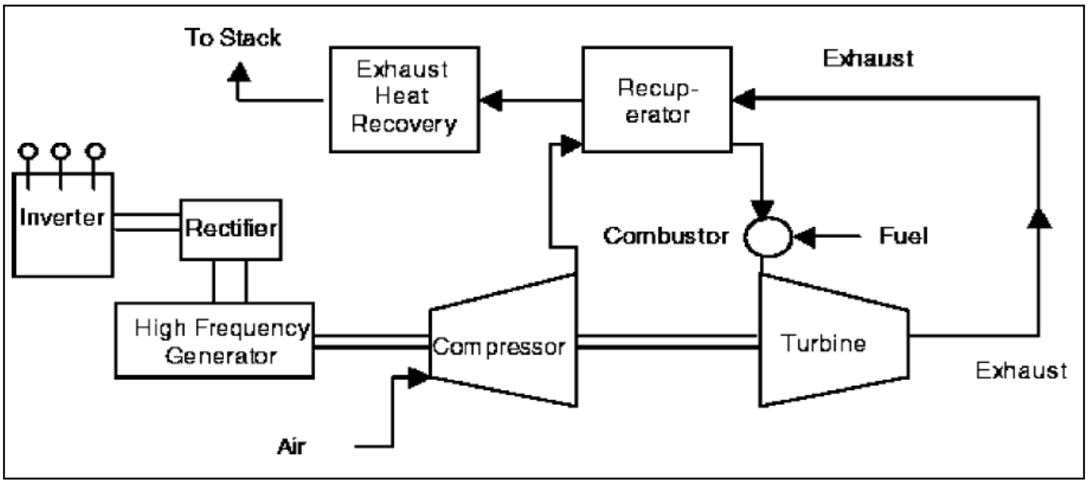


Figure 2.3 Schematic of a recuperated Micro-Turbine Based Cogeneration Unit

In micro-turbines, the turbo-compressor shaft generally turns at high rotational speeds of about 80,000–120,000 rpm. The physical size of components and rotational speed of micro-turbine systems are strongly influenced by the specific turbine and compressor design characteristics.

Most micro-turbines are based on single-stage radial flow compressors. This is attributed to the size range of micro-turbines (0.23–2.3 kg/s of air/gas flow). Radial flow turbine-driven compressors are similar to small reciprocating engine turbochargers in terms of design and volumetric flow.

The exhaust gas of a micro turbine based cogeneration system is the source of heat recovery. Commonly, an integrated heat exchanger is used to extract heat from the exhaust gas before releasing the gas to the atmosphere. Depending on the application, hot water or steam may be produced. Use of a recuperator increases the electrical efficiency of a micro turbine cogeneration system, but reduces the recoverable heat from the exhaust gas. This may or may not be desirable depending on the application. [15]

2.3.1.2. Performance Characteristics and Efficiency

There is scant performance information on micro-turbine systems that is obtained from a limited number of demonstration projects. Manufacturers claim availability to be in the range of 90–95%

Micro-turbine designs are more complex than conventional simple cycle gas turbines because of the addition of a recuperator to reduce fuel consumption, thereby substantially increasing efficiency.

When the performance characteristics of several commercially available micro-turbine based cogeneration systems are examined, it can be seen that electrical efficiency increases w.r.t. electricity capacity of micro-turbines. Each manufacturer uses a different recuperator design, and tradeoff is often made between cost and performance. The pressure ratio is also an important factor; however, it is generally limited by material selection since the maximum temperature in the cycle increases with pressure. Electrical and the overall efficiencies of micro-turbine based cogeneration systems are lower than those of reciprocating engines and fuel cells.

The efficiency of micro-turbine based cogeneration systems can be increased by increasing the peak pressure and temperature in the cycle, requiring the development of high-temperature materials suitable for this purpose. However, higher temperatures can lead to higher NO_x emissions, necessitating the use of sophisticated combustor design to reduce NO_x emissions.

The output of a micro-turbine system can be reduced by a combination of mass flow rate reduction (i.e. decreasing the compressor speed) and turbine inlet temperature reduction. Consequently, along with the output, the efficiency of a micro-turbine operating at part load is reduced. The variation of efficiency of a 30 kW micro turbine is given in Figure 2.4.

Ambient conditions affect the power output and the efficiency of micro-turbine systems. Both power and efficiency decreases at elevated inlet temperatures. The power decrease is attributed to the decreased air mass flow rate and the efficiency decrease is due to the higher power requirement by the compressor to compress air of higher temperature. Power output and efficiency also decrease with increasing altitude and thus decreasing pressure. [15]

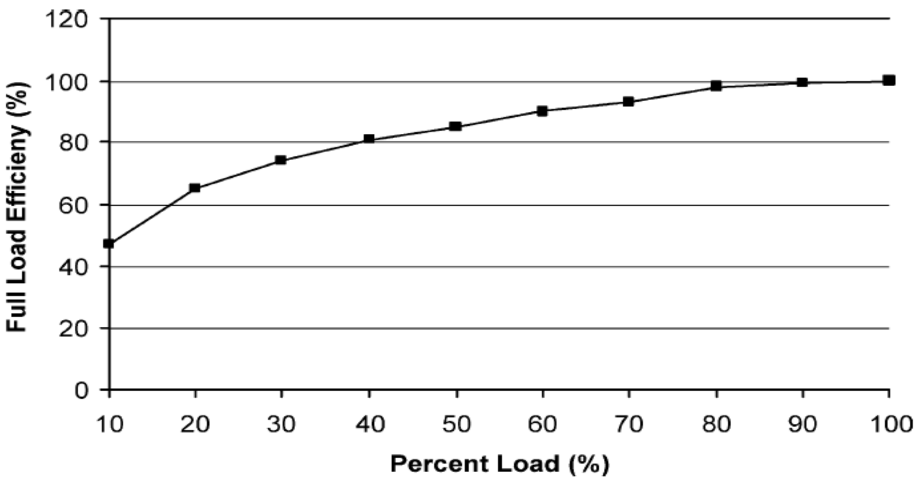


Figure 2.4 Micro-turbine Part-Load Power Performance for a 30 kW Micro-turbine (Single Shaft, High Speed Alternating System) [15]

2.3.1.3. Maintenance

Due to their simple construction and few moving parts, micro-turbine systems have the potential for lower maintenance costs than that of reciprocating internal combustion engines. For example, since the lubricating oil is isolated from the Normally, scheduled maintenance is carried out once annually, with maintenance costs in the 0.006–0.01 $/kW h range. An overhaul is required every 20,000–40,000 h depending on the product developers, design, and service. [15]

2.3.1.4. Emissions

Micro-turbines have the potential for producing low emissions. They are designed to achieve low emissions at full load, however, emissions are higher when operating under reduced load. The main pollutants from the use of micro-turbine systems are NOx, CO and unburnt hydrocarbons, and negligible amount of SO_2.

NOx is a mixture of (mostly) nitric oxide (NO) and nitrogen dioxide (NO_2) in variable composition. NOx, is formed when nitrogen and oxygen in the combustion air combine in the flame during the combustion of gases and light oils. The factors influencing the level of NOx emissions from a boiler are the flame temperature, the amount of nitrogen in the fuel being used, excess air level and combustion air temperature.

CO forms during combustion when carbon in the fuel oxidizes incompletely, ending up as CO instead of CO_2. Poor burner design or firing conditions can be responsible for high levels of CO emissions. Proper burner maintenance or equipment upgrades, or using an oxygen control package, can control CO emissions successfully

Emissions of sulfur are related directly to the sulfur content of the fuel, and are not dependent on boiler size or burner design. About 95 % of the sulfur content of the fuel is emitted as sulfur dioxide (SO_2) with about 5 percent as sulfur trioxide (SO_3). SOx are classified as a pollutant because they react with water vapor in the air and in flue gas to form sulfuric acid mist, which is extremely corrosive and damaging in its air-, water- and soil-borne forms. Boiler fuels containing sulfur are primarily coal, oil and some types of waste.

Emission characteristics of micro-turbine systems based on manufacturers' guaranteed levels are given in Table 2.1. [15]

CO emissions in micro-turbines occur as a result of incomplete combustion. At low loads, micro-turbines tend to have incomplete combustion. CO_2 emissions are of concern because of their potential contribution to the greenhouse effect. CO_2

emission from micro-turbines is a function of both fuel carbon content and system efficiency. Use of natural gas and the high overall efficiency results in low CO_2 emissions. [15]

Table 2.1 Micro-Turbine Emission Characteristics

	Capstone model 330 micro-turbine	IR energy systems 70LM (two shaft)	Turbec T100
Nominal electricity cap.(kW)	30	70	100
Electrical efficiency (%) HHV	23	25	27
NO_x, ppmv	9	9	15
NO_x, lb/MW h[a]	0.54	0.50	0.80
CO, ppmv	40	9	15
CO, lb/MW h	1.46	0.30	0.49
THC, ppmv	<9	<9	<10
THC, lb/MW h	<0.19	<0.17	<0.19
CO_2, lb/MW h	1928	1774	1706
Carbon, lb/MW h	526	484	465

a Conversion from volumetric emission rate (ppmv at 15% O_2) to output based rate lbs/MW h) for both NOx and CO based on conversion multipliers provided by Capstone Turbine Corporation and corrected for differences in efficiency.

2.3.1.5. Cogeneration Systems and Their Costs

Micro-turbine based cogeneration systems are being introduced into the market by manufacturers to meet the electrical and thermal demands of both building and industrial applications. Existing micro-turbine system sizes vary from 25 to 80 kW, but research is being carried out to develop systems in sizes less than 25 kW, for example 1 and 10 kW Today, several U.S manufacturers have micro-turbine units suitable for multi-family residential, commercial and institutional cogeneration applications.

Installed costs of micro-turbines can vary depending on the scope of the plant equipment, geographical area, competitive market conditions, special site requirements, emission control requirements, current labor rates. Comprehensive cost estimates for micro-turbine cogeneration systems supplying electricity and hot water are given in Table 2.2, assuming that the cogeneration system produces hot water.

Table 2.2 Estimated Capital Costs for Micro-Turbine Based Cogeneration [15]

	Capstone model 330 micro-turbine	IR energy systems 70LM (two shaft)	Turbec T100
Nominal electricity cap (kW)	30	70	100
Equipment costs ($):			
Micro-turbine	1000	1030	800
Gas booster compressor	Included	Included	Included
Heat recovery	225	Included	Included
Controls/monitoring	179	143	120
Total equipment	1403	1173	920
Labor/material costs ($):			
Project and construction Mgt	418	336	226
Engineering and fees	154	146	112
Project contingency	72	58	45
Project financial (Interest during construction)	40	32	25
Total plant cost ($/kW)	2516	2031	1561

2.3.2. 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. Steam turbines 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. Steam turbine is often used in CHP applications; 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. These turbines typically range in size from 100 kW_e to over 1000 MW_e.

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. 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. [16]

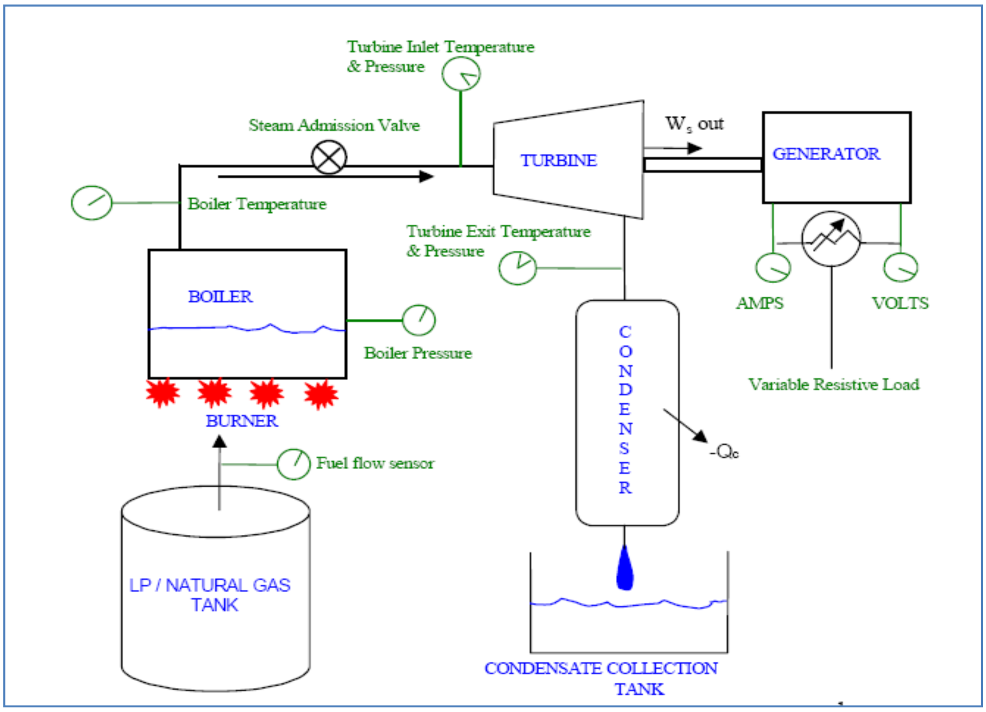


Figure 2.5 Schematical Representation of Steam Turbine System [17]

2.3.2.1. Technological Aspects

The typical steam turbine based system consists of several major components; a heat source, such as a boiler which is capable of burning fuels and delivers high pressure steam, a steam power turbine and a heat sink, which can be a thermal process, or just the environment. In most cases, the power source is a heat recovery steam generator (HRSG), which is fired with exhaust gases of another prime mover; (ICE of gas turbine). The system operates in a Rankine Cycle and is equipped with a condenser, as shown in the Figure 2.5.

The modern steam turbine may have three stages, like shown in Figure 2.6. The high-pressure (HP) section has small blades because the incoming steam has very high energy at very high temperature (about 1200 ºK). After the steam passes through the HP section, it is sent back to the boiler to be reheated to about 1,000 ºC. The steam is then sent to the next section of the turbine, called the intermediate pressure (IP) section where the blades are larger. After that, the steam is sent to the low-pressure (LP) section of the turbine. [16]

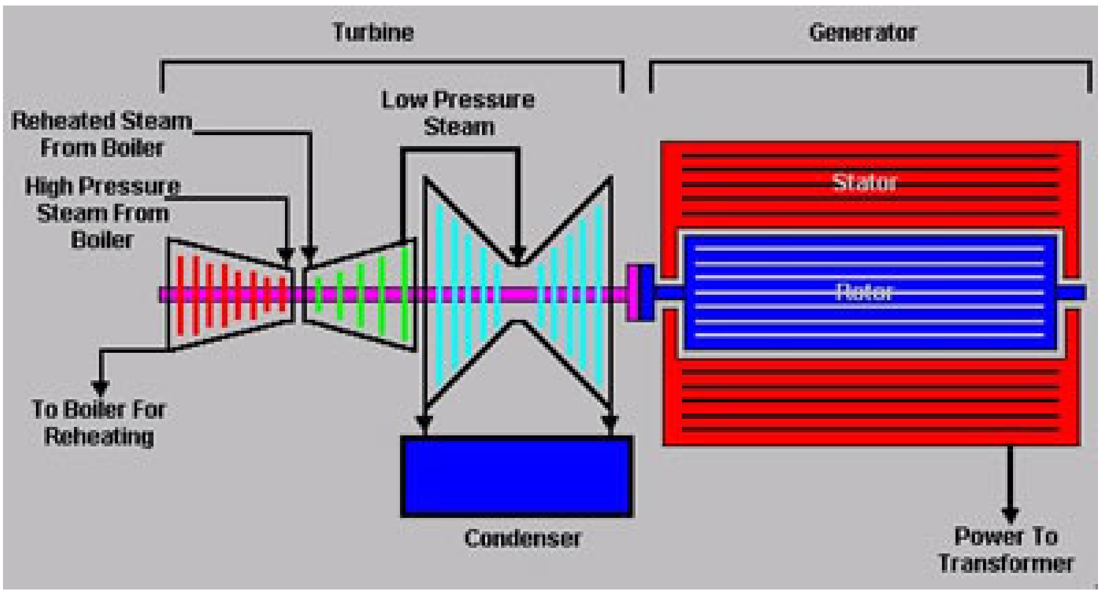


Figure 2.6. Steam Turbine Overview [21]

Because most of the energy has already been removed from the steam, the blades here are the largest. 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.

An ideal steam turbine is considered to be an isentropic process, or constant entropy process, in which the entropy of the steam entering the turbine is equal to the entropy of the steam leaving the turbine. No steam turbine is truly “isentropic”, however, with typical isentropic efficiencies ranging from 20-90% based on the application of the turbine. The interior of a turbine comprises several sets of blades, or “buckets” as they are more commonly referred to. One set of stationary blades is connected to the casing and one set of rotating blades is connected to the shaft. The sets intermesh with certain minimum clearances.

These stages are characterized by how the energy is extracted from them and are known as either impulse or reaction turbines. Most steam turbines use a mixture of the reaction and impulse designs: each stage behaves as either one or the other, but the overall turbine uses both. Typically, higher pressure sections are impulse type and lower pressure stages are reaction type.

An **impulse turbine** has fixed nozzles that orient the steam flow into high speed jets. These jets contain significant kinetic energy, which the rotor blades, shaped like buckets, convert into shaft rotation as the steam jet changes direction. A pressure drop occurs across only the stationary blades, with a net increase in steam velocity across the stage.

In the **reaction turbine**, the rotor blades themselves are arranged to form convergent nozzles. This type of turbine makes use of the reaction force produced as the steam accelerates through the nozzles formed by the rotor. Steam is directed onto the rotor by the fixed vanes of the stator. It leaves the stator as a jet that fills the entire circumference of the rotor. The steam then changes direction and increases its speed relative to the speed of the blades. A pressure drop occurs across both the stator and the rotor, with steam accelerating through the stator and decelerating through the rotor, with no net change in steam velocity across the stage but with a decrease in both pressure and temperature, reflecting the work performed in the driving of the rotor.

Turbines are also classified w.r.t. steam supply and exhaust conditions as; condensing, noncondensing, reheat, extraction and induction turbines**;**

Noncondensing or backpressure turbines are most widely used for process steam applications. The exhaust pressure is controlled by a regulating valve to suit the needs of the process steam pressure. These are commonly found at refineries, district heating units, pulp and paper plants, and desalination facilities.

Condensing turbines are most commonly found in electrical power plants. These turbines exhaust steam in a partially condensed state, typically of a quality near 90%, at a pressure well below atmospheric to a condenser.

Reheat turbines are also used almost exclusively in electrical power plants. In a reheat turbine, steam flow exits from a high pressure section of the turbine and is returned to the boiler where additional superheat is added. The steam then goes back into an intermediate pressure section of the turbine and continues its expansion.

Extracting type turbines are common in all applications. In an extracting type turbine, steam is released from various stages of the turbine, and used for industrial process needs or sent to boiler feed water heaters to improve overall cycle efficiency. Extraction flows may be controlled with a valve, or left uncontrolled.

Induction turbines introduce low pressure steam at an intermediate stage to produce additional power. [16]

2.3.2.2. Performance characteristics and Efficiency

Steam turbine systems are high efficiency systems, efficiencies ranging between 65-80%. At part loads, efficiency decreases. Electrical production efficiencies range between %25-42. Turbine inlet pressure and temperature values for a typical turbine system range between 1,033 kPa and 20,670 kPa and saturated temperature values to 583 °C respectively.

Steam turbines are usually rated at the maximum load the turbine can carry, and this point is usually greater than the load at which the turbine achieves at maximum efficiency. Optimum performance typically occurs at 95% of the rated load.

Steam turbines provide the greatest flexibility in meeting a site’s thermal requirements. Multiple extraction ports allow the turbine to satisfy differing steam requirements, while multiple induction ports allow the use of process by-product steam for power generation. Combination extraction/condensing turbines provide further flexibility, allowing the maximum production of power from steam that is not required for process.

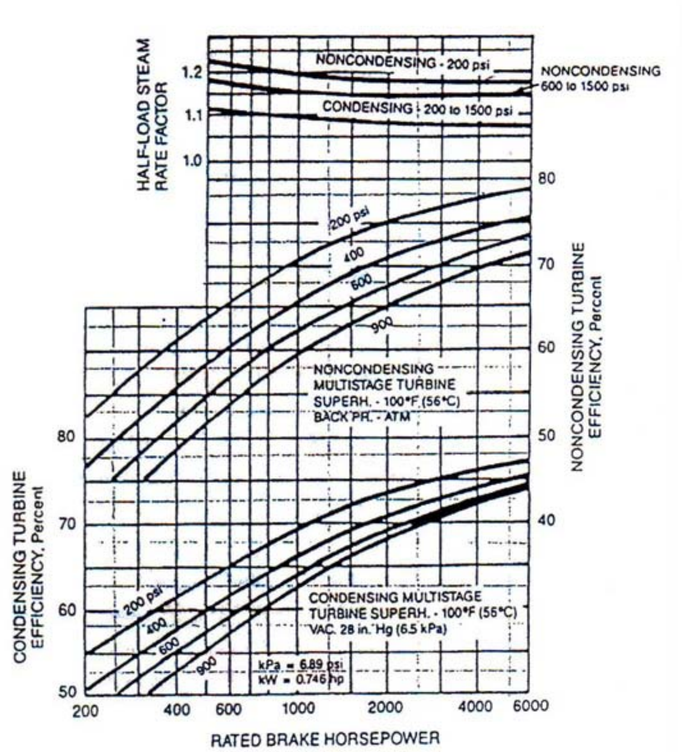


Figure 2.7 Steam Turbine Efficiency [16]

Unlike combustion turbines, steam-turbine based cogeneration systems can require a rather lengthy start-up period, which includes the warm-up of the fired boilers. Start-up can require several hours and sometimes, where more rapid availability is important, it may be cost-effective to keep a boiler warm to reduce start-up time.

The steam turbines maximum efficiency is limited to the Carnot efficiency, which is a function of the temperatures of the cycle's heat source and the heat sink. That efficiency is reduced by mechanical inefficiencies, steam losses, and imperfections in the flow path. An obvious opportunity for increasing the turbine efficiency is to increase the difference in the energy content of the turbines throttle and exit steam conditions. Increasing the throttle pressure and/ or temperature and decreasing the condensing pressure and/ or temperature will increase the Carnot efficiency.

Other measures such as increasing turbine speed or increasing the number of turbine stages will also increase the efficiency. Mechanical efficiency will typically range between 50% and 80%. As exit steam quality decreases, the recoverable heat becomes less useful, and the cogeneration system's total energy efficiency may decrease. Additionally, as condenser temperatures decrease, the steam's moisture content can increase, resulting in added maintenance costs. Figure 2.7 illustrate typical efficiency data for low-pressure backpressure and condensing turbines. [16]

2.3.2.3. Maintenance

Steam turbines are capable of extremely high availabilities (95%) with forced outage rates of less than 2 %. Most turbine outages are the result of blade corrosion, and therefore water quality is the most significant factor impacting availability and maintenance costs. Since steam is raised in external boilers, the type of fuel has no effect on turbine maintenance and availability, although fuel handling and availability problems may limit overall system availability. Finally, auxiliary equipment such as pumps and valves may limit system availability, and redundant ancillary components are usually cost-effective for critical systems.

The interval between steam turbine overhauls will depend primarily on water quality and turbine duty cycle. A complete internal inspection of a steam turbine can require between 150 and 350 hours; however, problems encountered during the inspection can significantly increase the length of the outage. Continued performance monitoring, valve inspections, vibration monitoring, and water quality testing can aid in minimizing outage time.

It is generally believed that more than 50% of steam turbine plant outages are attributable to the corrosion; therefore, water and steam chemistry are critical components of an effective steam turbine maintenance program. As throttle pressures and temperatures increase, water quality must be increased. [16]

2.3.2.4. Emissions

Steam turbines produce two forms of emissions: combustion gas emissions, which result from the boiler, and thermal emissions which result from the condensing operation. Combustion emissions are discusses below; however, thermal emissions can be reduced to the extent that recoverable heat is used in process applications. [16]

Boilers emissions include nitrogen oxide (NOx), sulfur oxides (SOx), particulate matter (PM), carbon monoxide (CO), and carbon dioxide (CO_2) and UHC.

In industrial boilers, the predominant NOx formation mechanisms are thermal and fuel-bound. Thermal NOx, formed when nitrogen and oxygen in the combustion air combine in the flame during the combustion of gases and light oils. Fuel-bound NOx is associated with oil fuels and is formed when nitrogen in the fuel and oxygen in the combustion air react.

Emissions of sulfur are mostly sulfur dioxide (SO_2) with about 5 percent as sulfur trioxide (SO_3).

PM emissions are largely dependent on the grade of boiler fuel, and consist of many different compounds, including nitrates, sulfates, carbons, oxides and other uncombusted fuel elements. PM levels from natural gas are significantly lower than those of oils, and distillate oils much lower than residual oils. For industrial and commercial boilers, the most effective method of PM control is use of higher-grade fuel, and ensuring proper burner setup, adjustment and maintenance.

CO forms during combustion when carbon in the fuel oxidizes incompletely. Older boilers generally have higher levels of CO than new equipment because older burners were not designed with CO control as a design parameter.

While not considered a regulated pollutant in the ordinary sense of directly affecting public health, emissions of carbon dioxide (CO_2) are of concern due to its contribution to global warming. Atmospheric warming occurs because solar radiation readily penetrates to the surface of the planet but infrared (thermal) radiation from the surface is absorbed by the CO_2 (and other polyatomic gases such as methane, unburned hydrocarbons, refrigerants and volatile chemicals) in the atmosphere, with resultant increase in temperature of the atmosphere. The amount of CO_2 emitted is a function of both fuel carbon content and system efficiency

Table 2.3 below illustrates typical emissions of NOx, PM and CO for boilers by size of steam turbine system and by fuel type.

Table 2.3 Typical Emissions of NOx, PM and CO for Boilers [17]

Boiler Fuel	System 1 500 kW			Systems 2 and 3 3 MW / 15 MW		
	NO_x	CO	PM	NO_x	CO	PM
Coal (lbs/MMBtu)	N/A	N/A	N/A	0.20-1.24	0.0.02-0.7	
Wood (lbs/MMBtu)	0.22-0.49	0.6	0.33-0.56	0.22-0.49	0.06	0.33-0.56
Fuel Oil (lbs/MMBtu)	0.15-0.37	0.03	0.01-0.08	0.07-0.31	0.03	0.01-0.08
Natural Gas (lbs/MMBtu)	0.03-0.1	0.08	-	0.1 – 0.28	0.08	-

2.3.2.5. Cogeneration Systems and Their Costs

Modern large steam turbine plants (over 500 MW) have efficiencies approaching 40-45%. These plants have installed costs between $800 and$2000/kW, depending on environmental permitting requirements. [16]

Table 2.4 Boiler/Steam Turbine CHP System Cost and Performance Characteristics [16]

Cost & Performance Characteristics[5] Steam Turbine Parameters	System 1	System 2	System 3
Nominal Electricity Capacity (kW)	500	3,000	15,000
Turbine Type	Back Pressure	Back Pressure	Back Pressure
Typical Application	Chemicals plant	Paper mill	Paper mill
Equipment Cost ($/kW) [6]	$657	$278	$252
Total Installed Cost ($/kW)[7]	$1,117	$475	$429
Turbine Isentropic Efficiency (percent)[8]	50%	70%	80%
Generator/Gearbox Efficiency (percent)	94%	94%	97%
Steam Flow (lbs/hr)	21,500	126,000	450,000
Inlet Pressure (psig)	500	600	700
Inlet Temperature (° Fahrenheit)	550	575	650
Outlet Pressure (psig)	50	150	150
Outlet Temperature (° Fahrenheit)	298	366	366
CHP System Parameters			
Boiler Efficiency (percent), HHV	80%	80%	80%
CHP Electric Efficiency (percent), HHV[9]	6.4%	6.9%	9.3%
Fuel Input (MMBtu/hr)[10]	26.7	147.4	549.0
Steam to Process (MMBtu/hr)	19.6	107.0	386.6
Steam to Process (kW)	5,740	31,352	113,291
Total CHP Efficiency (percent), HHV[11]	79.6%	79.5%	79.7%
Power/Heat Ratio[12]	0.09	0.10	0.13
Net Heat Rate (Btu/kWh)[13]	4,515	4,568	4,388
Effective Electrical Efficiency (percent), HHV[14]	75.6%	75.1%	77.8%
Heat/Fuel Ratio	0.73	0.72	0.70
Electricity/Fuel Ratio	0.06	0.07	0.09

5 Characteristics for "typical" commercially available ST generator systems, data.from: TurboSteam, Inc.
6 Equipment cost includes turbine, gearbox, generator, controls and switchgear.
7 Installed costs vary greatly based on site-specific conditions; Installed costs of a "typical" simple installation were estimated to be 70% of the equipment costs.
8 The Isentropic efficiency of a turbine is a comparison of the actual power output compared to the ideal, or isentropic, output. It is a measure of the effectiveness of extracting work from the expansion process and is used to determine the outlet conditions of the steam from the turbine.
9 CHP electrical efficiency= Net electricity generated/Total fuel into boiler; (boiler fuel converted to electricity)
10 Fuel input based on condensate return at steam outlet pressure and saturation temperature
11 Total CHP efficiency = (Net electricity generated+ Net steam to process)/Total fuel into boiler
12 Power/Heat Ratio = CHP electrical power output (Btu)/ useful heat output (Btu)
13 Net Heat Rate = (total fuel input to the boiler - the fuel that would required to generate the steam to process assuming the same boiler efficiency/steam turbine electric output (kW).
14 Effective Electrical Efficiency = (Steam turbine electric power output)/(Total fuel into boiler (steam to process/boiler efficiency)). Equivalent to 3,412 Btu/kWh/Net Heat Rate.

2.3.3 Steam Engine

The oldest type of power cycle is the Rankine cycle or steam power. Steam power is today mainly used in large power plant generating electricity by burning fossil fuel or splitting atoms in nuclear power plant. When it comes to smaller steam power plant, Rankine power cycles has several inherent unique qualities that makes it very attractive as a future propulsion system in mobile applications as well as small scale CHP applications.

The novel high-tech micro Rankine cycle embodied as steam engine is not only a choice of replacing conventional prime movers but may also be an attractive complement to an ICE, as turbine and fuel cell where waste heat is with high efficiency, flexibility and reliability. However, in many applications a self-contained steam engine will offer the best possibility to use different energy sources including solar energy. [18]

Among different types of power cycles, Rankine cycle has largest possibility to use various primary energy resources. As long as there is heat source above 400 ºC, it is possible to boil water and generate electricity. Steam power is unique when it comes to external combustion and low temperature. Combustion in a steam engine starts at ambient temperature whereas a Stirling engine for instance uses a preheating process in order to get acceptable efficiency. That means that the highest flame temperature in a steam power system is low and hence very little nitrogen in the air is oxidized. The ability to harness low temperature makes the steam power also be able to recover waste heat from other sources and act as a bottoming cycle.

The choice of steam engine over conventional organic cycle (steam turbine) is due to its high electrical power generation efficiency, high flexibility and reliability and low flame temperatures in the system, causing very little nitrogen oxide, thus protecting the environment.

2.3.3.1. Technological Aspects

All steam engines are designed to extract energy from high pressure steam and convert it into motion by allowing the steam to expand. The components in a novel steam engine system are a burner, a steam generator (HRSG), an expander, a condenser and a feed pump. The steam engine cycle can be seen in Figure 2.8. [18]

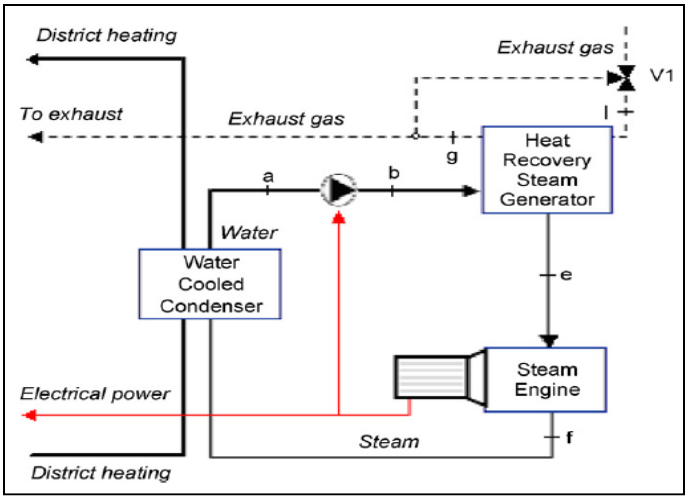


Figure 2.8 Schematical Representation of SE Cycle

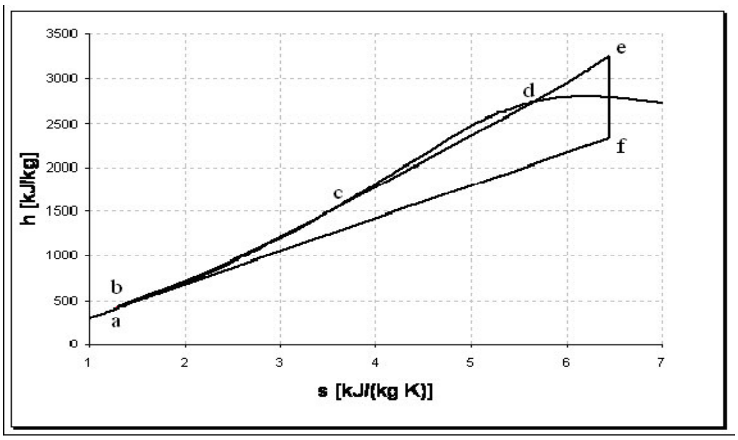


Figure 2.9. Mollier Diagram for the Ideal Rankine Cycle [3]

Figure 2.9 shows the Molier Diagram of the ideal Rankine cycle showing the relation between enthalpy (h) and entropy (s) of steam.

The ability to use of low temperature sources makes it possible to harness solar energy by evaporating water or other working fluids in parabolic trough. Stirling engine requires about + 800 ºC to operate which requires high concentrating solar collectors, which in turn means complex and expensive solar collector technology, whereas steam power can accept lower temperature and thus simpler and cheaper solar collectors. Below is the qualitative cycle for a steam engine as shown in Figure 2.10.

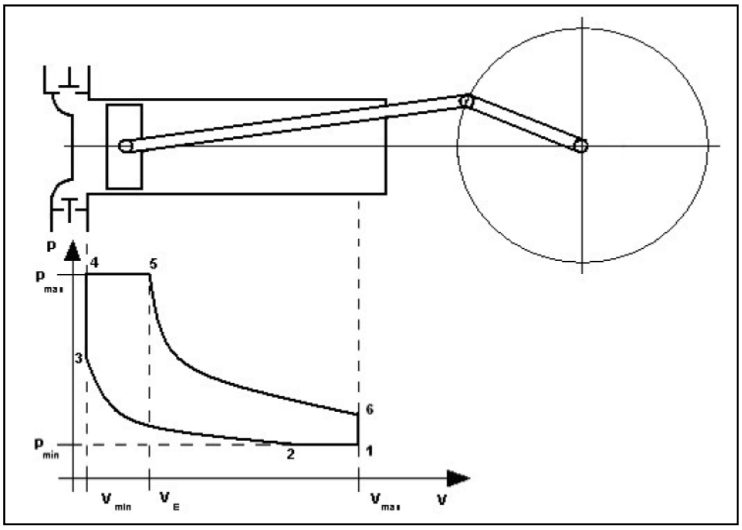


FIGURE 2.10 Qualitative cycle for a Steam Engine [18]

2.3.3.2. Performance Characteristics and Efficiency

In Steam engines, since water is the working fluid it will allow to deal with high admission pressure and large expansion ratio for high efficiency. Typical expansion ratios for internal combustion engine are 1:10-1:20 whereas the steam engine should have an expansion ratio of 1:100. High Pressure means a high-density working fluid, which in turn means a high-speed nature of the engine. When realizing a high-speed steam engine several advantages are possible to achieve. High speed will give reduced internal leakage (Blow-by) and hence, improved efficiency. Furthermore, high-speed will imply low heat losses and last but not least, high specific power (kW/kg). Steam engine has an inherent quality to be very compact and having much higher power density than ICEs. This is because it's possible to fill almost all the displacement volume with working fluid before expanding. Also, mean effective pressure and power density are high compared to a typical ICE. [19]

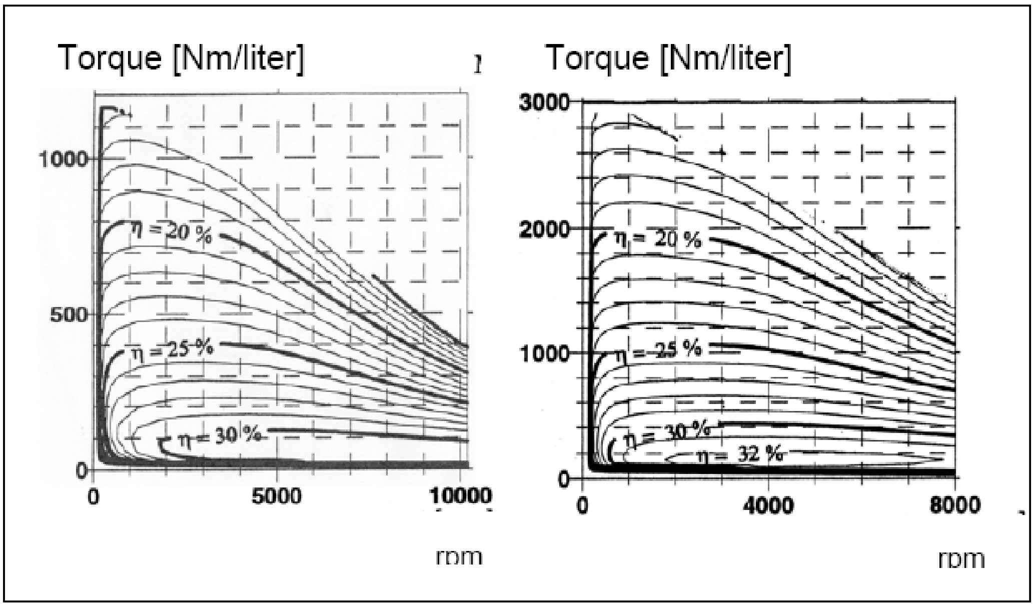


Figure 2.11 Performance Maps for a 100 bar and a 250 bar Steam Engine [19]

For steam engines, system efficiency is insensitive to load and can respond to rapid changes in steam conditions. So steam engine efficiency is not affected very much by shaft speed. Engines performance can be presented in so-called performance maps. In Figure 2.11, such a performance map based on computer simulation is illustrated. When Figure 2.11 is examined, it can be seen that the highest efficiency is obtained at low loads, that is, low torque. This is the opposite to conventional ICE which have the highest efficiency at full or close to full load. The efficiency characteristic of a steam engine in part load operation is very advantages in most applications.

If the prime mover is 30% efficient half of the waste heat is captured, a typical CHP efficiency is calculated as 65%.

No heat engine can be more efficient than the Carnot cycle, in which heat is moved from a high temperature reservoir to one at a low temperature, and the efficiency depends on the temperature difference. For the greatest efficiency, steam engines should be operated at the highest steam temperature possible (superheated steam). [19]

2.3.3.3. Maintenance

Steam engine startup is fast and steam engines can be out-of-service for long periods of time and can be started up easily after a long layoff. Like in steam turbines, auxiliary equipment such as pumps and valves may limit system availability and continued performance monitoring, valve inspections, vibration monitoring, and water quality testing can aid in minimizing outage time. [19]

2.3.3.4. Emissions

The highest flame temperature in a steam power system is low and hence very little nitrogen in the air is oxidized. Low pressure combustion – combined with no preheating allows for modest combustion temperatures and very little problem with NOx. Some new designs include coating the steam generator surfaces with an oxidation catalytic layer so that other emissions such as CO and unburned hydrocarbons can be reduced in a cost effective way. [18]

2.3.3.5. Cogeneration Systems and Their Costs

While the low capital cost of steam engine power is a critical design consideration, long out-of-service times and the ability to start up easily after a long layoff make the decision obvious.

There are not so many practical applications of SE cogeneration systems. SE is more often used as a bottoming cycle with ICE or a gas turbine.

2.3.4. Reciprocating Internal Combustion Engine

Reciprocating internal combustion engines (ICE) are mostly employed in low and medium power systems ranging from 50 kW_e to around 10 MW_e, and are found in applications where production of hot water (rather than steam) is the main requirement. 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 jacket cooling water. [15]

Heat from the engine jacket cooling water often between 85 and 90 ºC, accounts for up to 30% of the energy input. The heat recovered from the engine exhaust represents 30–50%, as hot water or low-pressure steam is from 100 to120 ºC. Thus, by recovering heat from the cooling systems and exhaust, approximately 70–80% of the energy derived from the fuel is utilized to produce both electricity and useful heat. The recovered heat can therefore be used to generate hot water or low-pressure steam for space heating, domestic hot water heating, or absorption cooling.

Reciprocating ICE's are suitable for small-scale cogeneration applications because of their robust and well-proven technology; however they do need regular maintenance and servicing to ensure availability. They can be fired on a broad variety of fuels with excellent availability making them suitable for numerous cogeneration applications in residential, commercial, institutional and small-scale industrial loads. [56]

The advantages of ICE technology over other cogeneration technologies are low capital cost, reliable onsite energy, low operating cost, ease of maintenance, and wide service infrastructure.

2.3.4.1. Technological Aspects

Reciprocating ICE's are classified by their method of ignition; compression ignition (CI-Diesel) engines and spark ignition (SI-Otto) engines. Diesel engines can be used for large-scale and small-scale cogeneration. These engines are mainly four-stroke direct injection engines fitted with a turbo-charger and intercooler. Diesel engines run on diesel fuel or heavy oil, or they can be set up to operate on a dual fuel mode that burns primarily natural gas with a small amount of diesel pilot fuel. between 500 and 1500 rpm.

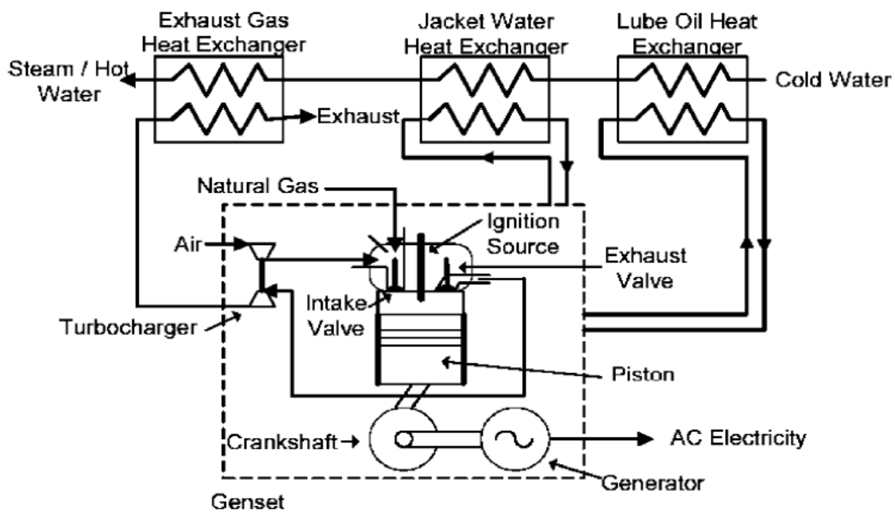


Figure 2.12 Typical packaged SI- ICE Based Cogeneration System [15]

Cooling systems for diesel engines are more complex in comparison to the cooling systems of SI engines and temperature are often lower, usually 85 ºC maximum, thus limiting the heat recovery potential.

Compared to Diesel engines, SI engines are more suitable for smaller cogeneration applications, with their heat recovery system producing up to 160 ºC hot water or 20 bar steam output. In cogeneration applications, SI engines are mostly run on natural gas. Many SI engines derived from Diesel engines (i.e. they use the same engine block, crankshaft, main bearings, camshaft, and connecting rods as the diesel engine) operate at lower brake mean effective pressure (BMEP) and peak pressure levels to prevent knock. SI engines have compression ratios of 8 to 11, while CI engines have compression ratios in the range 12 to 24.

In the current study, natural gas fired OTTO (SI) engine will be examined. The emission profile such engines has improved significantly through better design and control of the combustion process and the use of exhaust catalysts. In addition, they offer low first cost, fast start up, and significant heat recovery potential. [15]

The basic elements of a reciprocating ICE based cogeneration system are the engine, generator, heat recovery system, exhaust system, controls and acoustic enclosure. The generator is driven by the engine, and the useful heat is recovered from the engine exhaust and cooling systems. The architecture of a typical packaged ICE based cogeneration system is shown in Figure 2.12.

The engines used in cogeneration systems are lean/ stoichiometric mixture engines since they have lower emission levels, and the excess oxygen in the exhaust gases can be used for supplementary firing.

2.3.4.2. Performance Characteristics and Efficiency

In general, diesel engines are more efficient than SI engines because of their higher compression ratios. However, the efficiency of large SI engines approaches that of diesel engines of the same size.

Reciprocating ICEs are generally rated at ISO conditions of 25 ºC and 1 bar pressure [36]. Both output and efficiency of a reciprocating ICE degrades by approximately 4% per 333 m of altitude above 333 m, and about 1% for every 5.6ºC above 25 ºC.

Overall efficiency for reciprocating ICE based cogeneration systems is in the range of 85–90% with little variation due to size. The electrical efficiency was shown to be in the range of 28–39%, and this increases as engine size becomes larger.

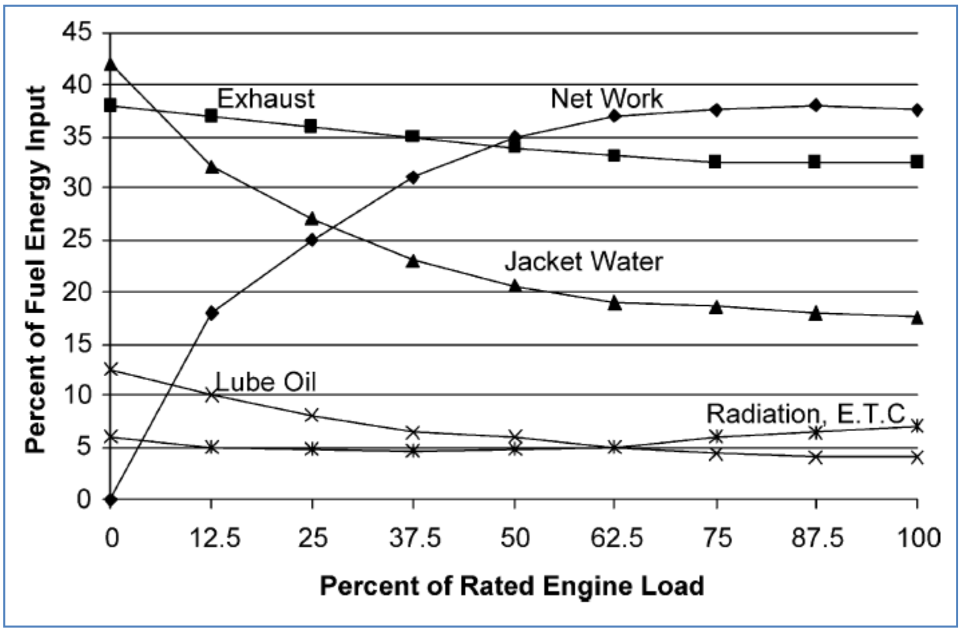


Figure 2.13 Heat Balance of Reciprocating Internal Combustion Engine

For cogeneration applications, the heat to power ratio of the engine is critical. As can be seen in Figure 2.13, the percentage of fuel energy input used in producing mechanical work, which results in electrical generation, remains fairly constant until 75% of full load, and thereafter starts decreasing. This means that more fuel is required per kWh of electricity produced at lower partial loadings, thereby leading to decreased efficiency. Also the amount of heat generated from the jacket coolant water and exhaust gases increases as electrical efficiency of the engine decreases. [15]

2.3.4.3. Maintenance

Routine inspections, adjustments and periodic maintenance are required with reciprocating ICE's. These involve changing of engine oil, coolant and spark plugs, often carried out for every 500–2000 h. Manufacturers often recommend a time interval for overhaul, from 12,000 to 15,000 h of operation for a top-end overhaul and 24,000–30,000 h of operation for a major overhaul. A typical maintenance cost for reciprocating ICE that include overhaul is from 0.01 to 0.015 $/kW h. With proper maintenance, modern ICE based cogeneration systems operate at high levels of availability. [15]

2.3.4.4. Emissions

The primary pollutants associated with ICEs are oxides of nitrogen (NOx), carbon monoxide (CO), and volatile organic compounds (VOCs—unburned, non-methane hydrocarbons). Other pollutants like oxides of sulphur (SOx) are related to large, slow speed diesel engines fuelled by heavy oils [36]. Particulate matter is an issue for Diesels operated with liquid fuels.

NOx emissions are critical with ICEs. They are produced by burning fossil fuels in the presence of oxygen. NOx production is dependent on temperature, pressure, combustion chamber geometry and air–fuel mixture of the engine. Lean burn natural gas fired engines produce the lowest while diesel engines produce the highest NOx emissions. A three-way catalytic converter can be used to treat the exhaust gases and reduce concentrations of NOx, CO and unburned hydrocarbons.

Carbon monoxide is caused by the incomplete combustion of fossil fuels due to inadequate oxygen or insufficient residence time at high temperature. In addition, CO emissions can occur at the combustion chamber walls as a result of cooling and due to reaction quenching in the exhaust process. Also, too lean conditions can lead to incomplete and unstable combustion and increasing the CO emission levels. Unburned hydrocarbons are caused by incomplete oxidation during combustion of long chain hydrocarbons.

Use of oxidation catalysts can reduce CO and unburned hydrocarbon emissions.

Particulates are the product from poorly adjusted combustion processes, i.e. incomplete combustion of fuel hydrocarbon. They are solid particles and appear as exhaust coloration or smoke. Emissions characteristics provided by manufacturers for a range of reciprocating internal combustion engines are given in Table 2.5. [15]

Table 2.5 Emission Characteristics of Reciprocating ICEs Used in Cogeneration

	Cummins						Coastintelligen	
Electrical output (kW)	7.5	16	16	20	35	50	55	80
Engine/fuel type	Diesel/ diesel	SI/NG	Diesel/ diesel	SI/NG	Diesel/ diesel	Diesel/ diesel	SI/NG	SI/NG
Emission control device	None	None	None	None	None	Turbo-charger	Advanced catalytic converter	Advanced catalytic converter
Air–fuel ratio		16.8		16.6				
CR	18.5:1	9.4:1	18.5:1	9.4:1	17.3:1	16.5:1		
NO_x, (g/bhph)	12.6	7.8	12.6	8.2	6.99	7.97	<0.15[a]	<0.15[a]
CO, (g/bhph)	3.13	36.8	3.13	38.6	1.26	0.75	<0.60[a]	<0.60[a]
Unburned hydrocarbon (g/bhph)	1.64	1.3	1.64	1.2	0.50	0.4	<0.15[a]	<0.15[a]
SO_2, (g/bhph)					0.62	0.6		
Particulates (g/bhph)	0.66	Negligible	0.66	Negligible	N/A	0.13		

2.3.4.5. Cogeneration Systems and Their Costs

A number of reciprocating internal combustion engine based cogeneration systems suitable for the residential sector are currently available in the market, range over 1–100 kW size. The basic cost of a reciprocating ICE based cogeneration system depends on its rated output. Smaller packaged reciprocating internal combustion engines typically run at a higher RPM than larger systems and they are often modified from automotive or truck engines. These two factors combined make smaller packaged engines cost less than larger, slow speed engines. The smaller reciprocating internal combustion engines are skid mounted, and the package includes the necessary radiators, fans, starting, control and fuel systems, and piping connections

Generally, reciprocating internal combustion based cogeneration systems less than 500 kW in size cost between 800 and 1300 $/kW, with higher cost for smaller cogeneration systems [16]. Estimated capital costs of various sizes of reciprocating ICE based cogeneration systems are given in Table 2.6.

Maintenance costs differ with the type, speed, size, and number of cylinders of an engine. These costs include maintenance labor, engine components and materials such as oil filter, air filters, spark plugs, gaskets, valves, piston rings, and oil. In addition, maintenance costs include minor and major overhauls. Data obtained from a manufacturer's survey [15] suggests that the maintenance costs for reciprocating internal combustion engine based cogeneration systems lie in the cost band of 0.008–0.015 $/kW h.

Table 2.6 Capital Costs of Reciprocating ICE Based Cogeneration Systems

Cost Component	Senertec[a]	North American cogeneration systems [9]			MAN [19]
Electrical capacity (kW)	5.5	7.1–10.7	20.1–23.3	30.3–35.0	100.0
Electrical efficiency (%)	27.5	28.1	37.4	33.1	30.6
Thermal efficiency (%)	62.5	56.5	50.0	51.2	50.4
Installed cost ($/kW)	2720	2800	1600	1300	1080

2.4. Heat Recovery Steam Generator

The heat recovery steam generator, (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 2.14 and 2.15.

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 and 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.

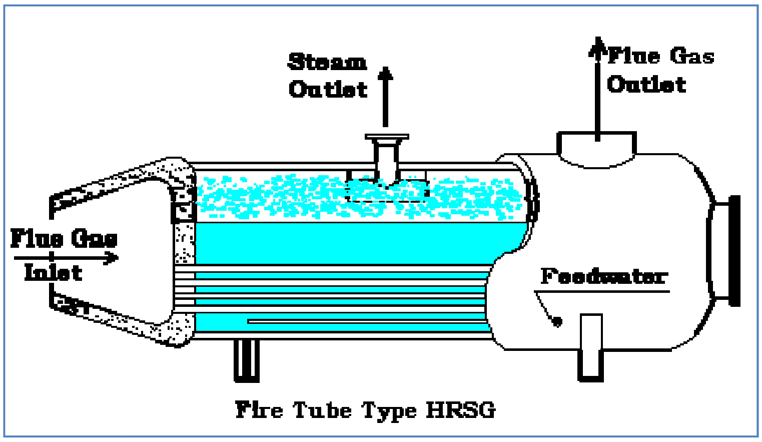


Figure 2.14 Fire Tube Type HRSG [20]

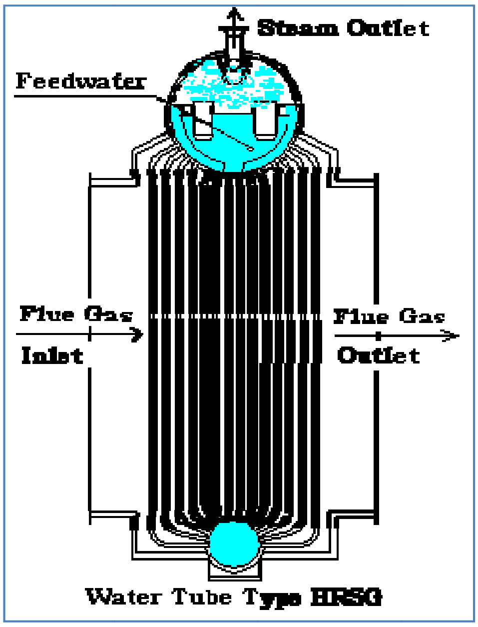


Figure 2.15 Water Tube Type HRSG [20]

2.4.1. Components of HRSG

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.

The next section; **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.

The last section; **economizer section**, sometimes called a preheater or preheat coil, is used to preheat the feed water 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.

Boiler Drums are the other important sections of the HRSG. The steam and water separation is achieved by means of separators installed in the upper part of the drum. Other purposes of the drums are; insuring a good mixing of feed water and boiler water and constituting a water reserve required for the controlled circulation system.

In Figure 2.16, a general view of a common HRSG can be found. Most HRSG designs function with more than one pressure stream. Below, in the figure mentioned, high pressure (HP), intermediate pressure (IP) and low pressure (LP) parts can be seen. In triple pressure systems, there is a substantial increase in the amount of heat recovered in the HRSG. Usually, the HP steam can be optimized at a higher pressure than the single pressure cycle; more of the energy is transferred into exergy. In addition, LP steam is produced, recovering the heat at the low temperature end of the HRSG and lowering the stack gas temperature by the addition of economizer and preheater. [55]

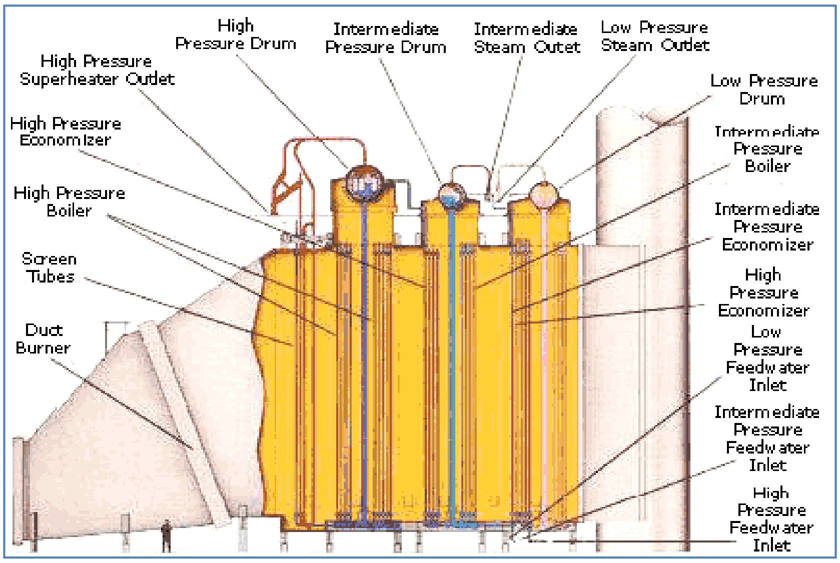


Figure 2.16 A General View of a HRSG [20]

2.4.2. 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 in Appendix A, Part A.1.

2.4.3. 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 A, Part A.2, Figures A.6, A.7 and A.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.

2.4.4. 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 A, Part A.2, for the superheaters.

2.4.5. 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 2.17, shows this relationship between the heat given up, and the three primary coils found in an HRSG.

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 at the economizer inlet, and going the other way, at a lower inlet temperature, this may occur at the superheater outlet. Pinch and Approach temperatures will be discussed in next chapters of the study.

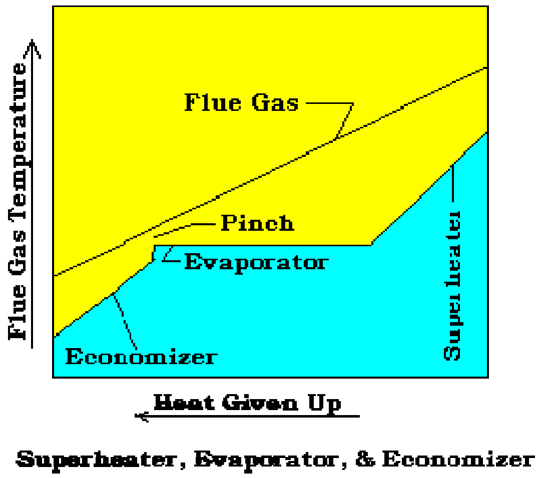


Figure 2.17 Relationship Between Heat Given Up and Three Primary Coils

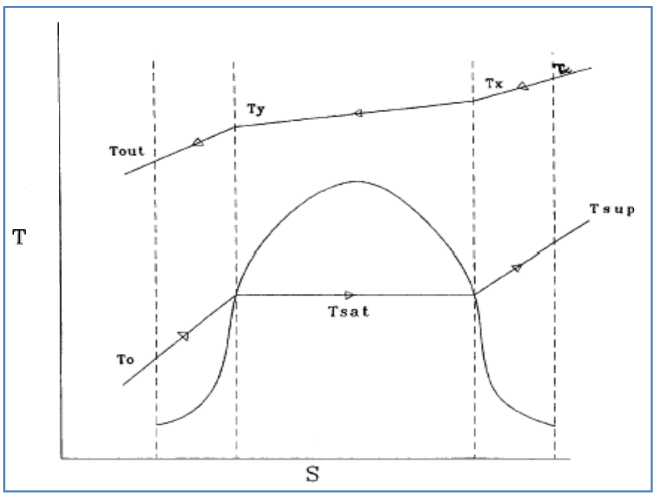


Figure 2.18 T-S Diagram of Waste Heat Recovery Boiler [21]

In Figure 2.18, T-S diagram of a simple HRSG unit can be found. 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. However, in the current study, a simple HRSG design will be considered, as can be seen in Figure 2.19.

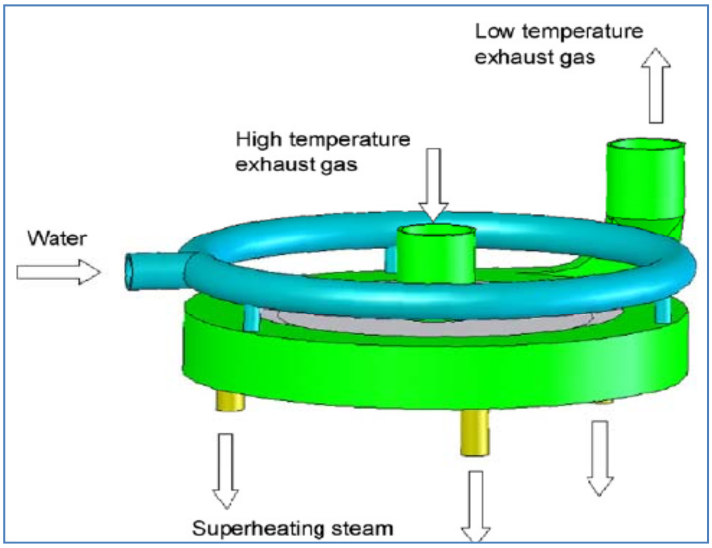


Figure 2.19 Single Pressure Flow Schematic for HRSG [11]

2.4.6. 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.

The pinch point temperature therefore affects the fuel utilization efficiency, power to heat ratio and second law efficiency, thus PES and PES_R values of the energy production system. [22]

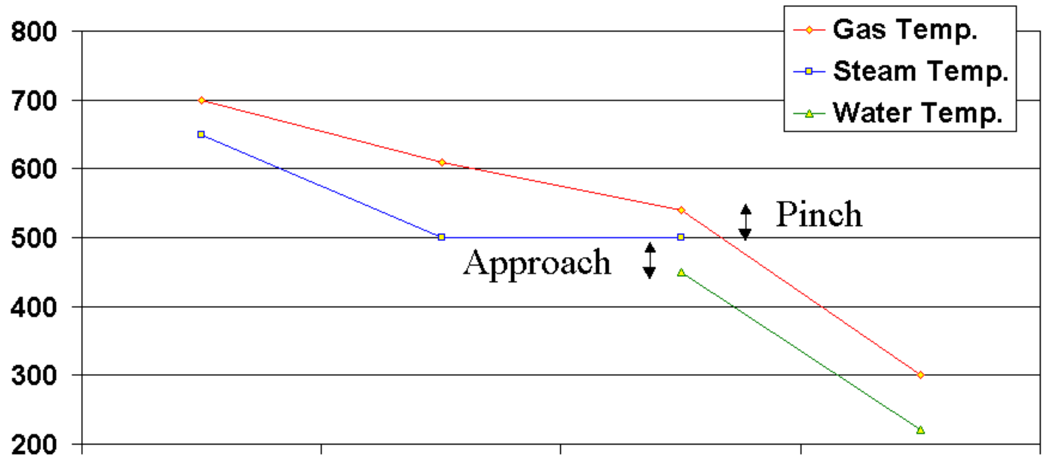


Figure 2.20 Approach and Pinch Point Illustrations

For many general purpose HRSGs 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 35 °C pinch design for these HRSG should be considered. [21]

2.5. Other Components of Poly-generation Plants

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.

2.6. Important Parameters in Poly-generation

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 energy utilization ratio/factor (EUF)** of the power generated or the electricity yield of a CHP process.

CHP fuel energy is the fuel energy based on lower heating value (LHV) needed in a CHP process to co-generate CHP electrical/mechanical energy and CHP useful heat energy in a reporting period.

CHP useful heat energy is the heat energy (thermal energy) supplied by a CHP process to a network or a production process in a reporting period. It is heat energy that would otherwise be supplied from other sources (see Article 3 (b) in [9]).

CHP electrical/mechanical energy is defined as the gross electrical/mechanical energy, which is generated in direct relation to the generation of CHP useful heat (see Article 3 (d) in [9]) in a reporting period.

The sum of the products **CHP electrical/mechanical energy** and **CHP useful heat energy** in relation to the **CHP fuel energy** provides the total **utilization ratio** of the fuel, or fuel utilization. This utilization ratio is an important quality criterion of CHP. The achievable utilization ratio for the use of solid fuels is slightly lower than for gaseous or liquid fuels, e.g. 80 to 85%.

The ratio of the gross electrical/mechanical CHP energy and CHP useful heat energy is known as the **power-to-heat ratio (C)**. 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. [23]

The 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 work-related power-to-heat ratios:

- Waste incineration: 0.25
- Internal Combustion Engine: 0.75
- Combined-cycle with heat recovery: 0.95
- Steam backpressure turbine: 0.45
- Steam condensing extraction turbine : 0.45
- Gas turbine with heat recovery: 0.55 [24, 8]

CHP overall efficiency is the ratio of CHP energy output to CHP energy inputs of the CHP plant in a reporting period.

PES (**Primary Energy Saving)** is the amount of primary energy savings provided by cogeneration production. Defined in to DIRECTIVE 2004/8/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL's current version, Annex III-b of [9].

CHPHη is the heat efficiency of the cogeneration production defined as annual useful heat output divided by the fuel input used to produce the sum of useful heat output and electricity from cogeneration.

CHPEη is the electrical efficiency of the cogeneration production defined as annual electricity from cogeneration divided by the fuel input used to produce the sum of useful heat output and electricity from cogeneration. Where a cogeneration unit generates mechanical energy, the annual electricity from cogeneration may be increased by an additional element representing the amount of electricity which is equivalent to that of mechanical energy.

C is the power to heat ratio as already mantioned, $\boldsymbol{\eta_{CHP}}$ **(overall efficiency)** is the ratio of CHP energy output to CHP energy inputs of the CHP plant, $\mathbf{RefH_{\eta}}$ is the efficiency reference value for separate heat production, $\mathbf{RefE_{\eta}}$ is the efficiency reference value for separate electricity production.

2.7. Exergy Analysis

2.7.1. Concept of Exergy and Conventional Exergy Analysis

Exergy, can be defined as 'energy quality'. It is the theoretical maximum of useful work (shaft work or electrical work) obtainable from a thermal system as it is brought into thermodynamic equilibrium with the reference environment while heat transfer occurs with environment only.

Exergy is a measure of the departure of the state of the system from the state of the reference environment. The processes in all real energy conversion systems are irreversible and a part of the exergy supplied to the system is destroyed. Only in a reversible process does the exergy remain constant [25].

Energy process integration encompass techniques based on the thermodynamic and economic analysis of individual components as well as the system as a whole, oriented to design and improve production systems, maximizing the efficiency of consumed resources. Their fundamentals are found in exergy analysis. Exergy analysis combines the first and second law of thermodynamics and informs about the thermodynamic efficiency of a process [25,26]. Through exergy, which is a thermodynamic property that is degrading every time energy is used in any process, it is possible to compare different processes as well as mass and energy flows. Thus, exergy can be considered as a measurement of the thermodynamic value of mass and energy flow-streams and processes. It is not a conservative property and it is destroyed in irreversible processes. As a consequence, exergy destruction provides an evaluation of the consumption and degradation of natural resources that occur in a plant. Exergy analysis locates and quantifies the irreversibilities that appear in a productive process, allowing the identification of the most inefficient processes. The real irreversibilities of a system are exergy destruction, occurring within the system boundaries, and exergy losses, which are exergy transfers out of the system that are not useful. Some of the common causes for exergy destruction include chemical reaction, heat transfer across a finite temperature difference, fluid friction, flow throttling, and mixing of dissimilar fluids[27].

Several authors agree that exergy is the most adequate thermodynamic property to associate with cost, since it contains information from the second law of thermodynamics and accounts for energy quality The production process of a complex energy system can therefore be analyzed in terms of its economic profitability and efficiency with respect to resource consumption. An economic analysis can calculate the cost of fuel, investment, operation and maintenance for the whole plant but does not provide a means to evaluate the single processes taking place in each individual equipment nor how to distribute the costs among them. On the other hand, thermodynamic (exergy) analysis calculates the efficiencies of the subsystems and locates and quantifies the irreversibilities but cannot evaluate their significance in terms of the overall production process. Thus, thermoeconomic assesses the cost of consumed resources, money and system irreversibilities in terms of the overall production process.

2.7.2. Rational Exergy Management Method (REMM)

When sustainability, carbon emissions, and global warming become important issues of concern, the balance among the qualities of the energy in terms of useful work potential at the supply and demand points become critical. The common metric of the exergy balance rationale is provided by the Rational Exergy Management Efficiency, ψ_R , which depends on how the exergy demand (ε_{dem}), exergy supply (ε_{sup}), and destroyed exergy (ε_{dst}) relate to each other on a temperature scale [28].

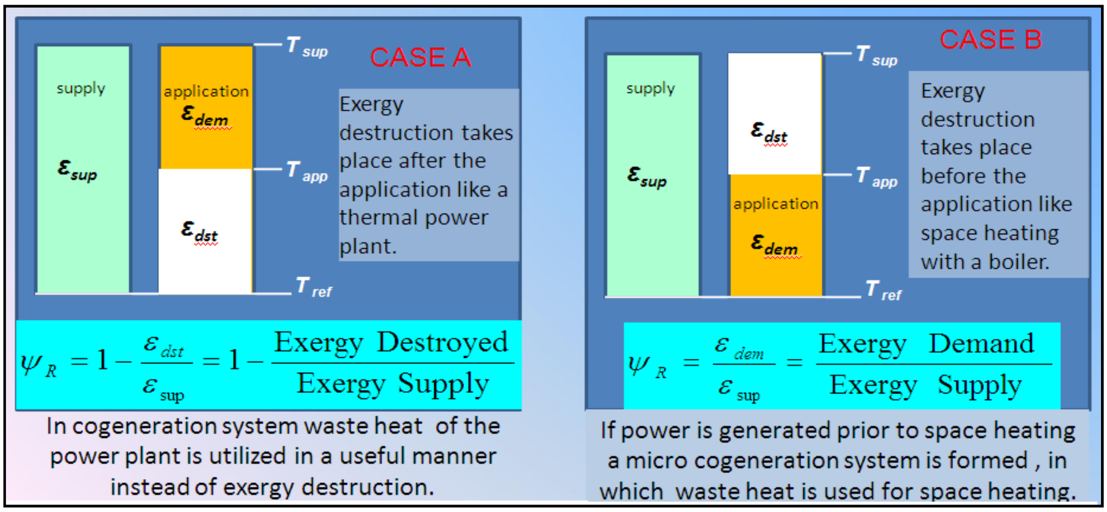


Figure 2.21. REMM Efficiency in Different Conditions of Exergy Destruction

In Figure 2.21, case A corresponds to a thermal power plant where exergy destruction takes place after power generation application in the plant. Case B typically corresponds to a boiler, which only generates heat for indoor space heating, where exergy destruction takes place before the heating application. In this typical case ψ_R is only about 4%.

Equations below are scales about how rational is the match between the supply and demand exergies [28]. If these do not match, fossil fuel based emissions like CO_2 increase. Because, *PES* is related by definition to fossil fuels and the directive aims to lower the harmful emissions, the impact of CHP systems on the environment, in terms of exergy efficiency, must definitely be incorporated in addition to energy efficiency.

$$\psi_R = \frac{\varepsilon_{\min}}{\varepsilon_{\max}} = 1 - \frac{\text{Exergy Destroyed}}{\text{Exergy Supply}} \qquad (2.1)$$

$$\psi_R = \frac{\varepsilon_{\min}}{\varepsilon_{\max}} = \frac{\text{Exergy Demand}}{\text{Exergy Supply}} \tag{2.2}$$

In Equations 2.1 and 2.2, ε_{max} is the maximum exergy of the input fuel for a unit amount of energy for a given application in the consumers area. ε_{max} is a function of the flame temperature T_f of the fuel and the temperature of the environment that the system may become in thermal equilibrium, T_{ref}. T_{ref} may be selected to be the temperature of the ground, sea, or lake, but preferably in the close vicinity of the applications. Because outdoor air temperature is quite variable-even hourly, it should be the last option. ε_{min} is the minimum amount of exergy that could satisfy the same task. For chemical or mechanical systems, an equivalent flame temperature may be defined . Then, According to the sign convention for exergy to be positive both in heating and cooling processes, two cases were distinguished.

$$\psi_R = \frac{\left(1 - \frac{T_{ref}}{T_{app}}\right)}{\left(1 - \frac{T_{ref}}{T_f}\right)} \qquad \{T_{app} > T_{ref}\} \tag{2.3}$$

$$\psi_R = \frac{\left(1 - \frac{T_{app}}{T_{ref}}\right)}{\left(1 - \frac{T_{ref}}{T_f}\right)} \qquad \{T_{app} < T_{ref}\} \tag{2.4}$$

For the special case of $T_{app} = T_{ref}$, T_{ref} may be selected accordingly in order to satisfy one of the above inequality conditions. This seemingly arbitrary selection does not affect the overall rationality of the exergy balance, because all temperatures are referenced to the same T_{ref} (See Figure 2.15). ψ_R also depends upon the kind of application that utilizes the thermal energy supplied from the system at temperature T_{app}.

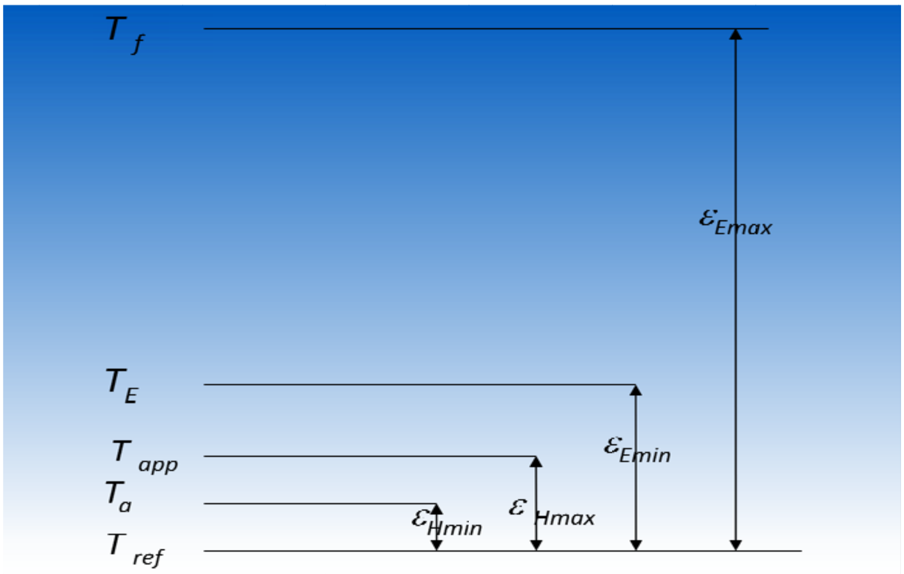


Figure 2.22 Adoption of REMM to CHP Systems for 2004/8/EC Directive [28]

Because Equations 2.3 and 2.4 do not incorporate the exergy component, a new PES_{RCHP} equation was derived. First, Equations 2.3 and 2.4 need to be modified for a CHP system, because there are two different kinds of energy outputs, namely electrical and thermal. According to the principal temperatures defined in Figure 2.22, for a CHP system, Equations 2.1 and 2.2 can be written in the following format, where exergy values for unit energy are given in Equations 2.5 through 2.9.

$$\psi_{RCHP} = \frac{\varepsilon_{\mathrm{Hmin}} + C \times (1 - \varepsilon_{\mathrm{Emin}})}{\varepsilon_{\mathrm{Hmax}} + C \times \varepsilon_{\mathrm{Emax}}} \tag{2.5}$$

$$\varepsilon_{\mathrm{Emax}} = \left(1 - \frac{T_{ref}}{T_f}\right) \tag{2.6}$$

$$\varepsilon_{\mathrm{Emin}} = \left(1 - \frac{T_{ref}}{T_E}\right) \tag{2.7}$$

$$\varepsilon_{\mathrm{Hmax}} = \left(1 - \frac{T_{ref}}{T_{app}}\right) \tag{2.8}$$

$$\varepsilon_{\mathrm{Hmin}} = \left(1 - \frac{T_{ref}}{T_a}\right) \text{ or } \varepsilon_{\mathrm{Hmin}} = \left(1 - \frac{T_a}{T_{ref}}\right) \qquad \{T_a < T_{ref}\} \tag{2.9}$$

Derivation of PES_R and $REF\ \psi_{RCHP}$ are given in Appendix B

PES_R can be given as follows:

$$PES_{RCHP} = \left[1 - \frac{1}{\left(\frac{CHPH\eta}{REFH\eta} + \frac{CHPE\eta}{REFE\eta}\right) * \left(\frac{2 - REF\psi RCHP}{2 - \Psi_{RCHP}}\right)}\right] \tag{2.10}$$

Where;

$$\mathrm{Ref}\psi_{\mathrm{RCHP}} = \frac{\mathrm{H_{CHP}} \times \mathrm{Ref}\psi_{\mathrm{RH}} + \mathrm{E_{CHP}} \times \mathrm{Ref}\psi_{\mathrm{RE}}}{\mathrm{H_{CHP}} + \mathrm{E_{CHP}}} = \frac{\mathrm{Ref}\psi_{\mathrm{RH}} + \mathrm{C} \times \mathrm{Ref}\psi_{\mathrm{RE}}}{1 + \mathrm{C}} \tag{2.11}$$

If the application is indoor space heating, the reference rational exergy efficiency of the heating system based on a ground source heat pump using grid electricity may be chosen 0.10 and for the central power plant a typical exergy value of 0.3 may be chosen. [29]

CHAPTER 3

LITERATURE SURVEY

L. M.Serra et. al. [1] pointed out that with increasing population, consumption of natural resources- mostly water and energy- has been increasing continuously during recent decades and also that this very high consumption endanger the survival of our civilization and the sustainability of current life support systems. To prevent this, they implied the importance of process integration and poly-generation which are promising tools for increasing the efficiency of natural resources and minimizing the environmental impact. They studied the concepts of poly-generation and energy integration and various examples of poly-generation systems. They clarified that, poly-generation systems which include appropriate process integration significantly increase the efficient use of natural resources.

A. Abuşoğlu et. al. [2] studied the energy, exergy, and exergoeconomic analysis of diesel engine powered cogeneration (DEPC). Part 1 presents the formulation developed for such a comprehensive analysis while part 2 is an application of the developed formulation that considers an actual cogeneration power plant installed in Gaziantep, Turkey. Authors considered all components of the DEPC plant and. performed mass, energy, and exergy balances to each system component and subsystem. Various efficiencies based on both energy and exergy methods and the performance assessment parameters are defined for the system components and the entire co-generation plant. They developed formulations for the cost of products, and cost formation and allocation within the system based on both energy and exergy. The cost analyses formulated have significant importance to obtain the optimum marketing price of the product of thermal systems to maximize the benefit and/or minimize the cost. The exergoeconomic analysis is based on specific cost method (SPECO) and it is determined that the specific unit exergy cost of the power produced by the plant is 10.3 $/GJ.

HEGEL Project Team, Fiat- Italy [3] presented the Rankine Cycle Design for FP 6 HEGEL Project (2006) with an emphasis on technological aspects, important parameters and most importantly efficiencies of the steam engine and its components.

Rosen et. al. [5] indicated that, the complex array of energy forms involved in co-generation based district energy systems make them difficult to assess and compare thermodynamically without exergy analysis depending on different nature and quality of the three product energy forms: electricity, heat and cold. Exergy analysis provides important insights into the performance and efficiency for an overall system and its separate components.

Stenhede [6] indicated the effects of emissions on human health. He described some methods for using and converting energy for utility and industry by ICE powered cogeneration and pointed out how to keep emissions low.

J. H. Santoy et. al. [10] presented the design of a tri-generation system as an alternative way of improved energy use in co-generation systems. They observed that fuel consumption is decreased. They studied a regenerative-cycle cogeneration system and a new tri-generation system, showing their benefits as well as the operation criteria for both processes.

M. Badami et. al. [11] studied an innovative natural gas (NG) combined cycle co-generation system. The system is made up combining a reciprocating internal combustion engine co-generator (ICE CHP) as the topping cycle and a Rankine cycle (RC) co-generator which operates as the bottoming cycle on the exhaust gases from the ICE. The ICE is an automotive derived with a high part-load electrical efficiency, due to a variable speed operation strategy and reduced emissions. They defined the main design parameter of the Rankine bottoming cycle (the steam engine) and determined the electrical and thermal efficiency of the new CHP unit at full and part-loads. They introduced some design criteria and energy balance considerations and analyzed some technical solutions.

P. Colonna and S. Gabrielli [12] studied industrial tri-generation 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; tri-generation systems installations (electric generator and absorption refrigeration plant) are increasing nowadays, and ammonia water absorption refrigeration plants is a good solution, if low temperature refrigeration is needed. They presented the thermodynamic system of tri-generation configurations using commercial software integrated with specifically designed modules. In the study, they did a simplified economic assessment for 10 MW_e 7000 h/year).

Yousef S.H. Najjar [14] reviewed ten research investigations in the field of gas turbine co-generation in power and industry that he and his associates carried 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 co-generation with simultaneous production of power and thermal energy; Co-generation 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 co-generation systems.

H. I. Onovwionaa and V. I. Uğursal [15] stated the fact that, increase in energy efficiency due to implementation of co-generation systems can result in lower costs and reduction in greenhouse gas emissions when compared to the conventional methods of generating heat and electricity separately. They provided an up-to-date review of the various cogeneration technologies suitable for residential applications and focus on single-family applications (<10 kW_e). Technologies suitable for residential cogeneration systems include reciprocating internal combustion engine, micro-turbine, fuel cell, and reciprocating external combustion Stirling engine based cogeneration systems. They discussed the state of development and the performance, environmental benefits, and costs of these technologies.

P. Platell [18] stated the fact that, in our daily live we are surrounded with different prime movers as internal combustion engines, jet engines and big power plants. He clarified that, the oldest type of power cycles is the Rankine cycle (steam power) which is mainly used in large power plant generating electricity by burning fossil fuel or splitting atoms in nuclear power plant. He pointed out that, Rankine power cycles has several inherent unique qualities that makes it very attractive as a future propulsion system in mobile applications as well as small scale electric generation and heat at the same time, so-called CHP.

M. R. Muller [19] discussed current applications of steam engines in industrial systems and new technologies which are improving their performance. He gave examples showing cases where it is more cost effective (and efficient) to use steam engines. He pointed out that, for small systems, the steam turbine is not appropriate because of its small turndown ratio, sensitivity to steam quality, and high operating speeds. This results in available steam power being wasted. He stated that, steam engines scale down in size beautifully, work better with wet steam, and operate with very modest operating speeds.

E. Özgirgin [21] developed a computer program, called “Cogeneration Design” 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. She considered Middle East Technical University campus as the primary working area. Before the conceptual design study, she collected detailed information regarding description of the campus, infrastructure, annual electric, water and heat demand covering last 10 years and properties of existing heating plant and examined them in detail. Throughout the study, she developed eight different natural gas fired co-generation power plant designs regarding different gas turbine and steam turbine configurations, for METU Campus by using the "Cogeneration Design" program. Then, by means of a thermoeconomic optimization process, she prepared cost summary reports and discussed the feasibility of the designed power plants.

A. Khaliq et al [22] present thermodynamic methodology for the performance evaluation of combustion gas turbine cogeneration system with reheat. They defined energetic and exergy efficiencies. They investigated effects of process steam pressure and pinch point temperature used in the design of heat recovery steam generator, and reheat on energetic and exergy efficiencies. From the results obtained in graphs they observed that the power to heat ratio increases with an increase in pinch point, but the first-law efficiency and second-law efficiency decreases. Also, the power to heat ratio and second-law efficiency increases significantly with increase in process steam pressure, but the first-law efficiency decreases with the same. They showed that, inclusion of reheat, provide significant improvement in electrical power output, process heat production, fuel-utilization (energetic) efficiency and second-law (exergy) efficiency.

B. Kılkış [24] compared the performance of the bottoming-cycle, high efficiency poly-generation option with co-generation on environmental and technological aspects He presented the results of rational exergy management evaluation of the EU FP6 HEGEL Combined-Cycle, High Efficiency Poly-Generation Project. Results show that while the rational exergy management efficiency increases in poly-generation, the energy efficiency decreases slightly and presents an interesting optimization problem. His research has introduced an environmentally better definition of fuel savings for rating and evaluating CHP systems in terms of both energy and exergy. He has shown that exergy is an important metric to reveal, understand, and appreciate the real advantages of CHP systems, optimize them to minimize their environmental footprint and harmful emissions, maximize their fuel savings.

A. Abuşoğlu [27] performed an exergy and thermo-economic performance analysis and optimization of diesel engine powered cogeneration systems and then applied the analysis to an actual cogeneration system located in Gaziantep, Turkey [Sanko Diesel Engine Powered Cogeneration (DEPC) plant]. She defined the performance parameters based on both the first law and the second law for diesel cogeneration The cost functions of each stream in the plant is obtained using specific exergy costing (SPECO) method. She performed an exergoeconomic optimization of the actual DEPC plant. She presented the results of the system performance and point out that diesel cogeneration systems involve high electrical output compared to process heat and they should be selected for such applications. She also noted that thermoeconomic optimum solution is strongly depended on the cost functions and characteristic coefficient values of the functions defined. Finally, she assessed the exhaust emission characteristics of DEPC plant following both energy and exergy based approaches.

B. Kılkış and Ş.Kılkış [28] updated EU Directive on cogeneration using Rational Exergy Management Model (REMM), which is a new common sustainability metric. The primary energy savings equation of the directive was modified to factor in the rational exergy benefits of cogeneration on the environment and reducing harmful emissions. Using parametric analyses, They drew a new roadmap for better and optimized systems. They presented sample studies which provide the milestones about the potential benefits of cogeneration for sustainable economy, environment, economy, and human welfare.

B. Kılkış and Ş.Kılkış [29] indicated that the Council of the European Union has issued a Cogeneration Directive because the energy savings potential of combined heat and power systems is underestimated. In terms of specific guidelines and rating parameters based on energy efficiency, the objective was to promote "high-energy cogeneration". They make a critical review of the Directive using the Rational Exergy Management Model (REMM),] which is a common sustainability metric that can better bridge the compound benefits of combined heat and power systems with the environment. REMM is an indicator of how supply and demand exergy amounts in the energy sector are balanced. In this respect, they modified the primary energy savings *(PES)* equation of the Directive in order to factor in rational exergy benefits on the environment, which remained hidden so far. The authors carried out sample calculations in the EU FP6 Research Project HEGEL and showed that the rational exergy benefits in conserving primary energy resources are 62% more than what the Directive can indicate. Using parametric analyses, their study draws a new roadmap for better and optimized co-generation systems.

F. Anzioso et. al. [30] studied an innovative NG CHP system, which has been set up at the FIAT Centre of Research (CRF), Turin, Italy. The automotive derived ICE has a high part load electrical efficiency due to a variable speed operation strategy and there is an advanced exhaust gas after-treatment to meet the most stringent pollutant emission regulations in the plant. In the paper, they compared the electrical efficiency and pollutant emissions of the new CHP unit with those of some traditional small scale cogeneration systems.

L. Eriksson and I. Andersson, [31] developed an analytic model for cylinder pressures in SI engines. The method they used is based on a parameterization of the ideal Otto cycle and takes variations in spark advance and air-to-fuel ratio into account. Experimental validation on two engines show that it is possible to describe the in-cylinder pressure of a SI combustion engine operating close to stoichiometric conditions, as a function of crank angle, manifold pressure, manifold temperature and spark timing.

A. Abuşoğlu et al [34] performed the thermodynamic analysis of an existing diesel engine co-generation system located at Gaziantep, Turkey. The exergy analysis was aimed to evaluate the exergy destruction in each component as well as the exergy efficiencies. They analyzed the thermodynamic performance of a 25.32MW electrical power and 8.1 ton/h steam capacity diesel engine cogeneration system at full load conditions. They found the thermal efficiency of the overall plant to be 44.2% and the exergy efficiency to be 40.7%.

C. J. Butcher and B. V. Reddy [37] investigated the performance of a waste heat recovery power generation system based on second law analysis for various operating conditions. They simulated the temperature profiles across the HRSG, network output, second law efficiency and entropy generation number. They also investigated the effect of pinch point on the performance of HRSG and on entropy generation rate and second law efficiency. The second law efficiency of the HRSG and power generation system decreases with increasing pinch point. The first and second law efficiency of the power generation system varies with exhaust gas composition and with oxygen content in the gas. They presented results which contribute further information on the role of gas composition, specific heat and pinch point influence on the performance of a waste heat recovery based power generation system based on first and second law of thermodynamics.

Kanoğlu [38] performed an exergy analysis of an actual binary geothermal power plant. The study describes an easy-to-follow procedure for exergy analysis of binary geothermal power plants and how to apply presented procedure to assess the plant performance by pinpointing sites of primary exergy destruction and thus showing the direction for improvements.

A. Abuşoğlu et. al. [39] studied the energy, exergy, and exergoeconomic analysis of diesel engine powered cogeneration (DEPC). Part 1 presents the formulation developed for such a comprehensive analysis while part 2 is an application of the developed formulation that considers an actual cogeneration power plant installed in Gaziantep, Turkey. [39] refers to Part 2 of the study, refered as [2]

O. Kaya [40] analyzed an internal combustion engine based medium sized co-generation plant according to first and second laws of thermodynamic. He examined energy and exergy efficiency trends for a variable range of operating loads. The system was examined for primary energy savings, power to heat ratio, electrical efficiency, heat efficiency, rational exergy management efficiency, primary energy and exergy savings and rational exergy based carbon emissions performance. He also studied the energy model of the power plant by using RETScreen Clean Energy Analysis Program. He summarized the positive effects of the economical and environmental results of co-generation systems, and found that energy, exergy and primary energy saving performance of the system is increasing at rated performance.

O. Kaya, and E. Bingöl [41] indicated that today, mainly because of the excessive consumptions, primary energy resources decrease continuously and the increasing global competition accelerates this fact. Because of these, new energy conversion systems are being developed in order to secure the quality and continuity of energy inputs with minimum energy costs. The authors imply that poly-generation energy conversion systems gained a great importance due to their economic, efficient and environmentally friendly features. They recorded monthly electric and heat loads (consumptions) for one year, for METU Research and Implementation Centre for Built Environment and Design (RICBED) building and computed the corresponding load characteristic graphs. by using RETScreen computer program. They also carried out productivity, emission and efficiency analysis for natural gas fired boiler, combined heat and power and tri-generation energy conversion techniques.

B. Kılkış [43] developed a procedure for testing and rating poly-generation systems for their technical and economical aspects. Since there are not enough common test standards, rating standards and terminology for poly-generation systems, by this analysis, it will be possible to evaluate the poly-generation systems in terms of basic parameters like inlet temperature, load, power to heat ratio etc. and determine operational characteristics of the system and poly-generation performance.

E. Bingöl et. al. [46] pointed out the importance of poly-generation systems due to their added advantages in terms of efficiency and harmful emissions They stated that a poly-generation system aimed to maximize the utilization of thermodynamic potential of the consumed resources stimulate the economy, ensure reliability of energy supplies and reduce transmission and distribution losses of electrical power grids. The authors imply the importance of rating, energy savings and exery performance, estimating the system efficiencies and controlling emissions of the poly-generation systems. In order to quantify and metricate such parameters, they did a complete first and second law (i.e. exergy) analysis of co-generation systems, a Rational Exergy Management Method (REMM) analysis, which is a linking procedure between exergy supply and demand in a thermodynamic system for optimizing CHP for lower exergy destruction and a carbon footprint measurement.

E. Bingöl et. al. [47] indicated that for poly-generation systems, the primary objective is to maximize the utilization rate of resource exergy. For such technologies, it is important to develop a robust set of metrics in terms of energy savings, exergy performance, efficiency and harmful emissions. In this study, the authors employed a first and second laws analysis, accompanying with Rational Exergy Management Method (REMM) and developed a MATLAB based metrication algorithm for natural gas (NG) fired, internal combustion engine (ICE) powered co-generation system for different operating loads. Results show that even the part loads, efficiencies of the ICE are almost constant, thermal, electrical and second law efficiencies, the PES and PES_R values increase with increasing engine speed. They also examined different energy production systems, pointing out that, poly-generation systems have the least CO_2 emissions and higher *PES* and PES_R values.

J.C. Bruno et. al. [48] 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

A. Khaliq [49] proposed a conceptual tri-generation system based on the conventional gas turbine cycle for the high temperature heat addition while adopting the heat recovery steam generator for process heat and vapor absorption refrigeration for the cold production. He applied first and second law approach and performed a computational analysis to investigate the effects of overall pressure ratio, turbine inlet temperature (TIT), pressure drop in combustor and HRSG and evaporator temperature on the exergy destruction in each component. He also examined first law efficiency, electrical to thermal energy ratio, and second law efficiency of the system. Thermodynamic analysis indicates that exergy destruction in combustion chamber and HRSG is significantly affected by the pressure ratio and TIT, and not at all affected by pressure drop and evaporator temperature. He investigated the exergy destruction rates of all components. He found that first law efficiency, electrical to thermal energy ratio and second law efficiency of the system significantly varies with the change in overall pressure ratio and turbine inlet temperature, but the change in pressure drop, process heat pressure, and evaporator temperature shows small variations in these parameters. Decision makers should find the methodology contained in this paper useful in the comparison and selection of advanced heat recovery systems

M. Badami and M. Mura [50] carried out an exergy analysis of an innovative natural gas (NG) combined cycle cogeneration system. The combined cycle is composed of a reciprocating ICE, which is used as the topping cycle, and a water Rankine cycle (RC), which operates on the exhaust gases from the ICE, as the bottoming cycle. They developed a steady-state model in order to evaluate the exergy irreversibilities of each component of the system and of the whole plant. Furthermore, they did a part-load exergy analysis and a sensitivity analysis for the whole cycle.

P.J. Mago et. al. [51] presented an exergy analysis of a combined engine-organic Rankine cycle configuration (E-ORC) using the exergy topological method. They presented a detailed roadmap of exergy flow using an exergy destruction chart to clearly depict the exergy accounting associated with each thermodynamic process. The analysis indicates that an ORC combined with an engine improves the engine thermal efficiency and increases the exergy efficiency. They evaluated different organic fluids and found that, depending on the organic fluid employed, the thermal and exergy efficiencies could be increased by approximately 10 per cent. They also examined the effect of the pinch-point temperature difference (PPTD) on the E-ORC performance. Results show that if PPTD decreases, thermal and exergy efficiencies increase.

R. Mikalsen et. al. [52] presented an investigation into the feasibility and potential advantages of a small scale Miller cycle natural gas engine for domestic combined heat and power systems. They compared the Miller cycle engine to a standard Otto cycle engine using cycle analyses and multidimensional simulation. They found that the Miller cycle engine has a potential for improved fuel efficiency, but at the cost of a reduced power to weight ratio.

A. Moran et. al. [53] indicated that the number of combinations of components and parameters in a micro-CHP system is too many to be designed through experimental work alone. Therefore, theoretical models for different micro-CHP components and complete micro-CHP systems are needed to facilitate the design of these systems and to study their performance. They presented a model for micro-CHP systems for residential and small commercial applications. Some of the results they obtained included the cost per month of operation of using micro-CHP versus conventional technologies, the amount of fuel per month required to run micro-CHP systems and the overall efficiency, etc. They demonstrated differences in the system performances of micro-CHP systems driven by a natural gas internal combustion engine and a diesel engine. Some of the results show that both systems have similar performance and that system total efficiencies in cooler months of up to 80% could be obtained. Also, modeling results show that there is a limit in fuel price that economically prevents the use of CHP systems.

D.W. Wu et. al. [54] pointed out that, combined cooling, heating and power (CCHP) systems provide an alternative for the world to meet and solve energy-related problems, such as energy shortages, energy supply security, emission control, the economy and conservation of energy. In the first part of this paper, the authors presented the definition and benefits of CCHP systems and then they clarified the characteristics of CCHP Technologies (especially technical performances, as well as the status of utilization and developments. Then they presented diverse CCHP configurations of existing technologies for typical systems of various size ranges. They concluded that, within decades, promising CCHP technologies can flourish with the cooperative efforts of governments, energy-related enterprises and professional associations.

Deng-Chern Sue et. al. [55] presented the engineering design and theoretical exergetic analyses of combustion gas turbine based power generation systems. They performed exergy analysis and realized that the exergy analyses for a steam cycle system predict the plant efficiency more precisely. They also found that; plant efficiency for partial load operation is lower than full load operation; increasing the pinch points will decrease the combined cycle plant efficiency. To evaluate the energy utilization, they analyzed one combined cycle unit and one cogeneration system, consisting of gas turbine generators, heat recovery steam generators, one steam turbine generator with steam extracted for process. They used the analytical results for engineering design and component selection.

G Angelino [56] observed that in a large number of existing steam power plants the energy potential of the cooling medium is not fully utilized owing to turbine limitation in exhaust volume flow-handling capability. He proposed a method by which a fraction of the low-pressure steam is extracted and fed to an auxiliary organic Rankine cycle (ORC) module of small capacity which perfectly suits for exploiting the coldest cooling agent and improves the working conditions of the main turbine by reducing its exhaust volume flow. He analyzed the performance of a typical power station for different cooling situations supplemented by an appropriate computer program he implemented. He considered the characteristics of the ORC module with selection of the new classes of ambient friendly refrigerants. Particular attention is devoted to turbine optimization, leading to high-efficiency low stress two and three-stage turbine configurations. He also performed a preliminary economic analysis.

CHAPTER 4

THERMODYNAMIC MODELING OF POWER SYSTEMS

First law formulation, second-law (exergy) analysis and regarding REMM modeling has been done for two cases, which will be the foundations of the common procedure that will be developed as the aim of the thesis study for testing and rating poly-generation systems under realistic operating conditions with accurate formulae. The cases which are based on HEGEL co-generation system [3], (topping and bottoming cycles) are compared with each other w.r.t system performances, PES and PES_R values and emission characteristics.

The engineering drawings of the HEGEL co-generation system which will provide heat and power to MATPUM building of the Department of Architecture can be found in Appendix C, Figure C.1. Inner and outer views of MATPUM building are also given in Appendix C, Figures C.2 to C.5. Preparation of the system, the installation procedure, establishment of connections and schematics of the HEGEL Room can be found in Appendix D.

4.1. General Continuity, Energy Conservation and Exergy Formulations

Steady flow conditions can be approximated as;

$$\sum \dot{m}_i = \sum \dot{m}_e \tag{4.1}$$

where i=inlet and e=exit.

The conservation of energy and exergy balance for any control volume at steady state with negligible kinetic and potential energy changes can be expressed, respectively,

$$\dot{Q} + \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i \ h_i \tag{4.2}$$

$$\dot{E}_{heat} + \dot{W} = \sum \dot{m}_e e_e - \sum \dot{m}_i \ e_i + \dot{E}_{dest} \tag{4.3}$$

$$\dot{E}_{heat} = \sum (1 - \frac{T_0}{T}) \dot{Q} \tag{4.4}$$

The specific flow exergy and the rate of total exergy are given by

$$e = (h - h_o) - T_0 (s - s_0) \tag{4.5}$$

$$\dot{E} = \dot{m} e \tag{4.6}$$

4.2. Thermodynamic analysis of the cogeneration system –Case-1:Topping Cycle only

Case-1 includes first-law and second-law analysis and PES calculations for the following system, including an ICE and two heat exchangers, for different prime mover (ICE) engine speeds, covering low, medium and high loads (rated conditions). Figure 4.1 shows the schematics of the cogeneration system for this case. As can be seen, engine cooling water (jacket water) of the ICE is also participated in district heating. ICE used in the model is a FPT ENT 60 NG Otto 4T type, 6 cylinder, 5.9 l engine. It has a maximum shaft torque and shaft speed of 650Nm and 1250 rpm respectively. Maximum shaft power is 147 kW corresponding to 125 kW of electrical power [11, 30]. The other technical parameters and properties of the ICE can be found in the Appendix E.

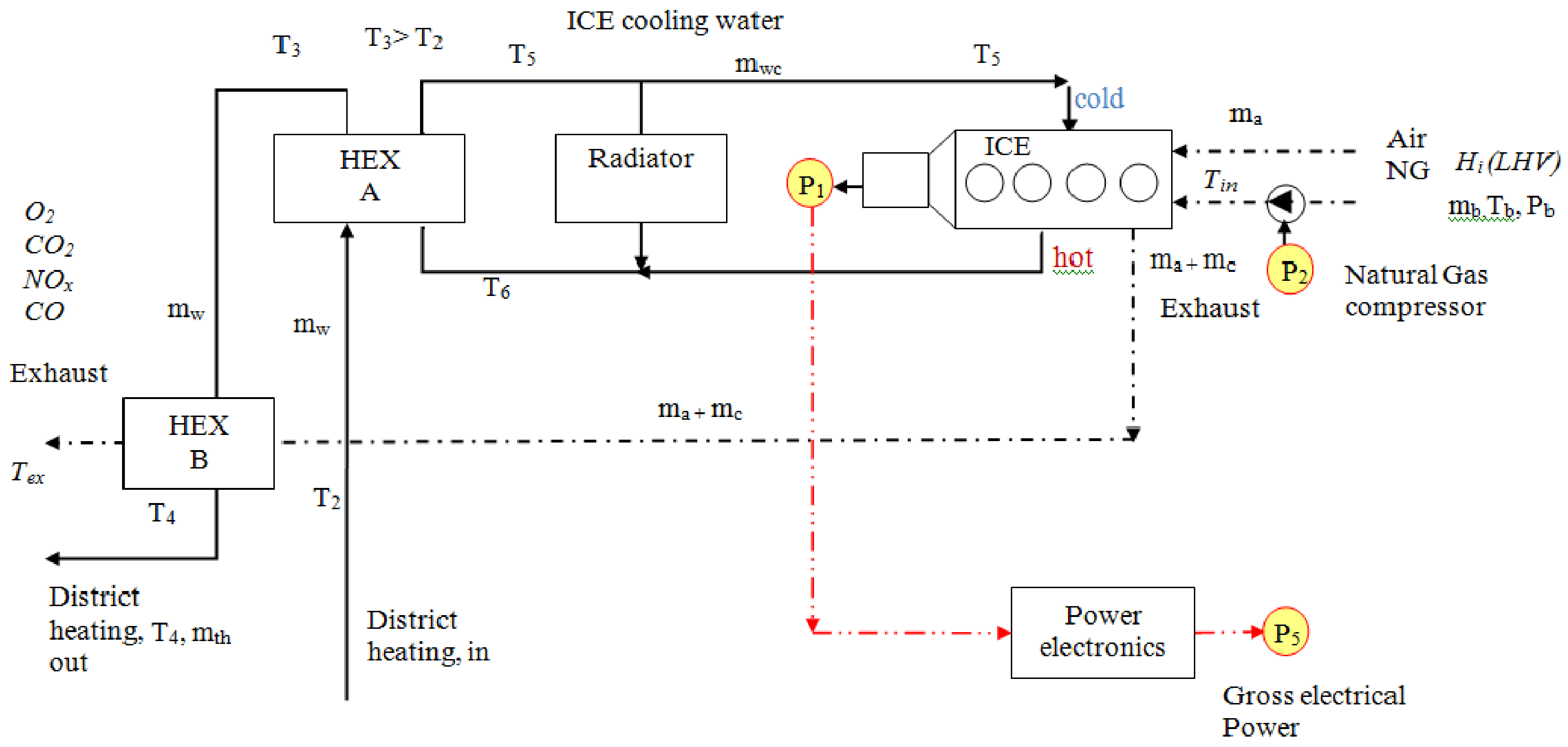


Figure 4.1- Schematical Model for Case-1

4.2.1. ICE Energy Conservation Formulations

Engine Break power can be calculated as seen in equation 4.7, where T is torque, n is rotational speed (rev/s)

$$\mathbb{P}_b = 2 \cdot \pi \cdot T \cdot n \tag{4.7}$$

Engine indicated power:

$$\mathbb{P}_i = \frac{2 \cdot n \cdot i}{j} \cdot P_e \cdot V_s \tag{4.8}$$

where i is cylinder number, j is stroke number, P_e is mean effective pressure and V_s is total swept volume.

Engine mechanical efficiency can be calculated as follows;

$$\eta_m = \frac{\mathbb{P}_b}{\mathbb{P}_i} \tag{4.9}$$

Mass flow rate of exhaust gases;

$$m_{ex} = \frac{N_b}{\eta_m} \cdot LHV \cdot (1 - \alpha) \tag{4.10}$$

where LHV is the lower heating value of fuel and α is the stoichiometric combustion proportion.

$$\mathbb{P}_{el} = \dot{W}_{net} = \mathbb{P}_b \cdot \eta_{inv} - \mathbb{P}_{ng\_comp} \tag{4.11}$$

The thermal efficiency of the ICE is defined as ratio of the net power generated by the engine to the rate of heat input supplied to the engine, and expressed as;

$$\eta_{thermal,ice} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{net}}{\dot{m}_{fuel} \cdot LHV} \tag{4.12}$$

The natural gas compressor energy equation and first law efficiency are given as follows:

$$P_{NGcomp} = \dot{m}_f \cdot (h_{out} - h_{in}) \tag{4.13}$$

$$\eta_{ng\_comp} = \frac{w_s}{w_a} = \frac{h_{e,s} - h_i}{h_e - h_i} \qquad [31, 32] \tag{4.14}$$

Other important physical properties and parameters of design for an ICE are given in Appendix F.

4.2.2. ICE Exergy analysis

The exergy efficiency of the internal combustion engine may be expressed as

$$\eta_{ice,\parallel} = \frac{\dot{W}_{net}}{\dot{E}_{fuel}} = \frac{\dot{W}_{net}}{\dot{m}_{fuel} \cdot e_{fuel}} \tag{4.15}$$

Second law efficiency is;

$$\eta_{ice,\parallel} = \frac{\dot{W}_{net}}{\dot{E}_{fuel}} = \frac{\dot{W}_{net} + \dot{E}_{exhaust} + \dot{E}_6}{\dot{m}_{fuel} \cdot e_{fuel} \cdot + \dot{E}_5} \tag{4.16}$$

Where $\frac{e_{fuel}}{q_{LHV}} = 1.044$ [33] (4.17)

Flow exergies of water and gases are calculated respectively;

$$\dot{E}_6 = \dot{m}_{wc}(h_6 - h_o) - T_0(s_6 - s_0) \tag{4.18}$$

$$\dot{E}_5 = \dot{m}_{wc}(h_5 - h_o) - T_0(s_5 - s_0) \tag{4.19}$$

$$\dot{E}_{exhaust} = \dot{Q}(1 - \frac{T_0}{T_{exaust}}) \tag{4.20}$$

Exergy destruction in the engine is calculated as follows;

$$\dot{E}_{dest=}\dot{E}_5 + \dot{E}_{fuel} - \dot{E}_6 - \dot{E}_{exhaust} - \dot{W}_{net} \tag{4.21}$$

4.2.3. Heat Exchangers Energy Conservation Formulations

Energy equation for HEX A;

$$\dot{m}_w \cdot C_{pw} \cdot (T_3 - T_2) = \dot{m}_{wt} \cdot C_{pw} \cdot (T_5 - T_6) = \dot{Q}_{ICE} \tag{4.22}$$

And energy equation for HEX B:

$$\dot{m}_2 \cdot C_{pw} \cdot (T_4 - T_3) = (\dot{m}_a + \dot{m}_a) \cdot \left(h_{ex} - h_{HRSG,OUT}\right) = \dot{Q}_{EXCH} \tag{4.23}$$

Energy efficiency of the heat exchangers can be calculated with the given formula;

$$\eta_{heat\ exh} = \frac{\dot{m}_{cold}[h_e - h_i]}{\dot{m}_{hot}[h_i - h_e]} \quad [30] \tag{4.24}$$

4.2.4. Heat Exchangers Exergy Analysis

Second law efficiency and exergy destruction of the heat exchangers are given respectively;

$$\eta_{heat\ exh,\text{II}} = \frac{\dot{m}_{cold}[(h_e - h_i) - T_0(s_e - s_i]_{cold}}{\dot{m}_{hot}[(h_i - h_e) - T_0(s_i - s_e)]_{hot}} \tag{4.25}$$

$$\dot{E}_{dest} = \dot{m}_{hot}[(h_i - h_e) - T_0(s_i - s_e)]_{hot} - \dot{m}_{cold}[(h_e - h_i) - T_0(s_e - s_i]_{cold} \tag{4.26}$$

4.2.5. Overall System Energy Conservation Formulations

Heat input of the co-generation system:

$$\dot{Q}_{in} = \dot{m}_{fuel} \cdot H_i \tag{4.27}$$

Gross electrical power, auxiliaries elect power and net electrical power can be calculated as can be seen in the following equations, 4-28 to 4-30:

$$P_{generator} = P_5 \tag{4.28}$$

$$P_{pumps} = P_2 \tag{4.29}$$

$$P_{net} = P_{gross} - P_{aux} = P_5 - P_2 \tag{4.30}$$

Net electrical efficiency is:

$$\eta_{el} = P_{net}/Q_{in} \tag{4.31}$$

Thermal output is the sum of thermal outputs of HEX A and HEX B:

$$Q_{th} = Q_{ICE} + Q_{hex} \tag{4.32}$$

District heating energy input is equal to the thermal output of the system:

$$\dot{m}_w \cdot C_{pw} \cdot (T_4 - T_1) = \dot{Q}_{th} \tag{4.33}$$

While, heat efficiency and energy utilization factor can be given respectively:

$$\eta_{th} = Q_{th}/Q_{in} \tag{4.34}$$

$$EUF = \eta_{el} + \eta_{th} \tag{4.35}$$

Utilization efficiency (first law efficiency) for this cogeneration system may be defined as;

$$\eta_{cogen} = \frac{\dot{W}_{net} + \dot{Q}_{process}}{\dot{Q}_{in}} = \frac{\dot{W}_{net} + \dot{m}_{water}[h_e - h_i - T_0 - (s_e - s_i)]}{\dot{m}_{fuel} q_{LHV}} \tag{4.36}$$

4.2.6. Overall System Exergy Analysis

$$\eta_{cogen,\text{II}} = \frac{\dot{W}_{net} + \dot{E}_{process}}{\dot{E}_{in}} = \frac{\dot{W}_{net} + \dot{m}_{water}[h_e - h_i - T_0 - (s_e - s_i)]_{water}}{\dot{m}_{fuel} e_{fuel}} \quad [34] \tag{4.37}$$

4.2.7. PES and PES_R calculations

4.2.7.1. Calculating PES value

According to [9] primary energy saving can be calculated as follows;

$$PES = \left(1 - \frac{1}{\frac{CHPH_\eta}{REFH_\eta} + \frac{CHPE_\eta}{REFE_\eta}}\right) \cdot 100 \tag{4.38}$$

Where;

$$CHPE_\eta = \left(\frac{P_5 - P_4}{(\dot{m}_b . H_i)}\right) \tag{4.39}$$

$$CHPH_\eta = \left(\frac{\dot{m}_w . c_{pw} . (T_4 - T_2)}{\dot{m}_b . H_i}\right) \tag{4.40}$$

So, PES becomes;

$$PES = \left(1 - \frac{\dot{m}_b . H_i}{\frac{P_{net}}{REFE_\eta} + \frac{Q_{th}}{REFH_\eta}}\right) \tag{4.41}$$

Power to heat ratio is the ratio of heat and electricity efficiencies, thus;

$$CHPE_\eta = C \cdot CHPH_\eta \tag{4.42}$$

For calculating PES, reference values should be carefully chosen from CHP Directive 2004/8/EC Reference Values Matrix [35]. For the current study, $RefE_\eta$ is 52.5% for both cases and when corrected for grid line losses with a factor of 0.925, becomes 48.56%. $RefH_\eta$ is 82% for Case 1, since exhaust gases are directly used for heating and 90% for Case 2, where steam and hot water is produced on site.

4.2.7.2. Calculating PES_R value

PES_R can be calculated as follows:

$$PES_{RCHP} = \left[1 - \frac{1}{\left(\frac{CHPH_\eta}{REFH_\eta} + \frac{CHPE_\eta}{REFE_\eta}\right) * \left(\frac{2 - REF_{\Psi RCHP}}{2 - \Psi_{RCHP}}\right)}\right] \tag{4.43}$$

Where;

Ψ_{RCHP} and $REF_{\Psi_{RCHP}}$ are given in Section 2.7.2 (equations 2.5 and 2.11). $RefH_\eta$ and $RefE_\eta$ are exactly the same as in PES calculation. $Ref\psi_{RCHP}$ is 0.2143 for ICE and heat recovery and 0.2024 for combined cycle with heat recovery.

4.2.8. Calculating Carbon emission values

The relative benefits of cogeneration and poly-generation may also be compared according to the avoidable CO_2 emissions component, ΔCO_2 of the total CO_2 emissions equation provided by REMM. Emission values are estimated using fuel consumption data and CHP efficiencies.

$$\sum CO_2 = \frac{ci}{\eta}(1 - \psi_{R,CHP}) \quad [28] \tag{4.44}$$

Where ci is carbon content of the fuel (kg CO_2/kWh) and η is the energy efficiency of the system.

4.3. Thermodynamic analysis for the cogeneration system –Topping and Bottoming Cycles

Case-2 includes first-law analysis and PES calculations for the following system for different prime mover (ICE) engine speeds, covering low, medium and high loads (rated conditions). Figure 4.2 shows the schematics of the cogeneration system for this case. The same ICE is used in the model, with a novel bottoming steam cycle composed of an HRSG, SE, condenser and a FWP added. SE is a 3 cylinder radial type with 300 cm3 displacement, a volumetric ratio of 30 and 2300 rpm maximum speed. Other technical parameters and properties of the SE and the condenser can be found in the Appendix G, Tables G.1 and G.2 respectively. [36]

4.3.1. ICE Energy Conservation Formulations

Formulations and calculation are the same as in section 4.2.1.

4.3.2. ICE Exergy analysis

Formulations and calculation are the same as in section 4.2.2.

4.3.3 Heat Exchangers Energy Conservation Formulations

Formulations and calculation are the same as in section 4.2.3.

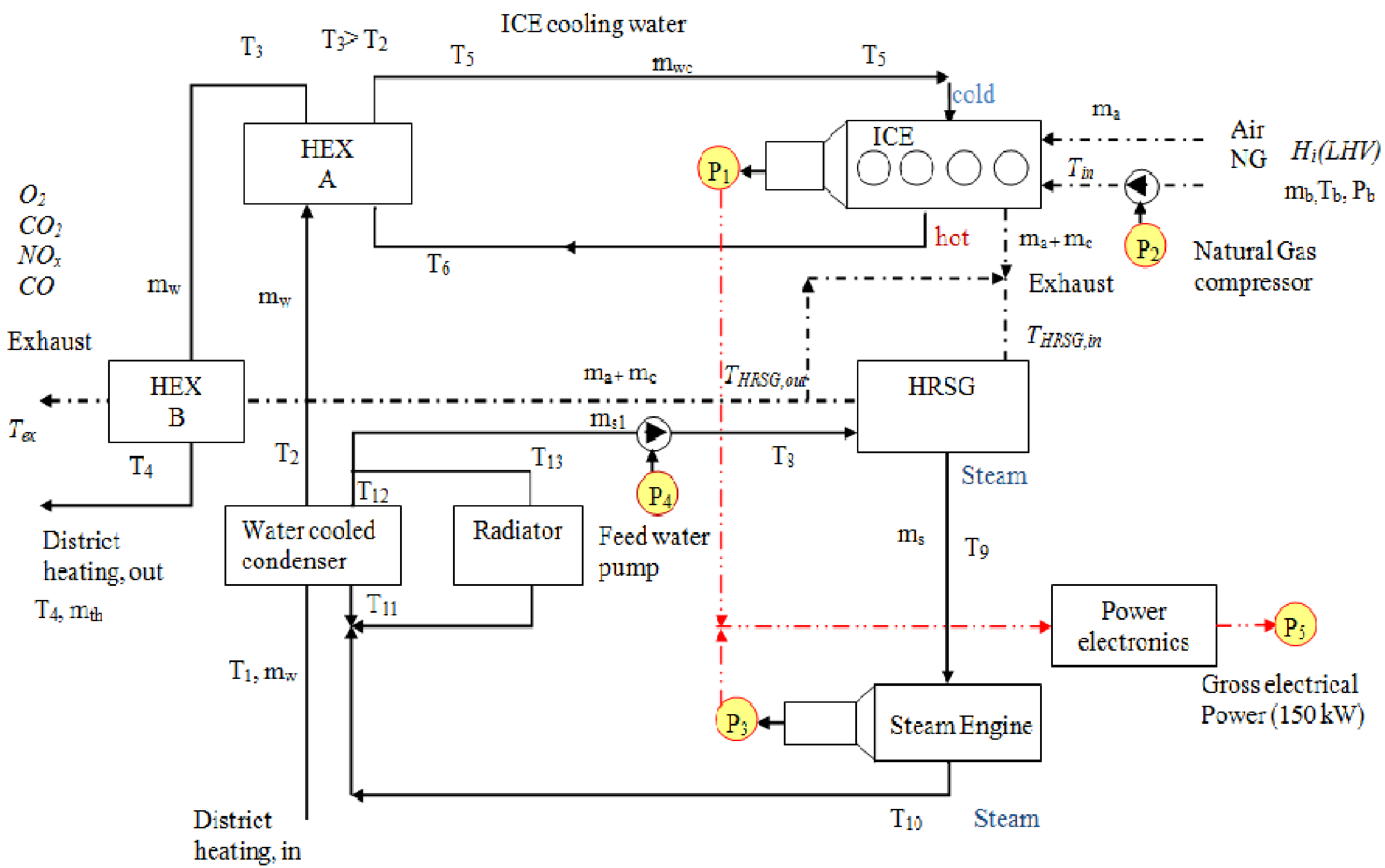


Figure 4.2 Schematical Model for Case-2

4.3.4. Heat Exchangers Exergy Analysis

Formulations and calculation are the same as in section 4.2.4.

4.3.5. HRSG Energy Conservation Formulations

Below, in Figure 4.3, heat transfer diagram of the HRSG can be seen. Viewing this generalized sketch that shows the relationship between the heat absorbed and the heat given up, it is important to consider the "pinch" at the evaporator outlet. Pinch point (PP), as already mentioned in Chapter 2, is the difference between the gas temperature leaving the evaporator and the saturation temperature of steam at HRSG pressure. The evaporator pinch, or approach temperature is what limits the amount of heat that can be recovered in most HRSG designs. [37]

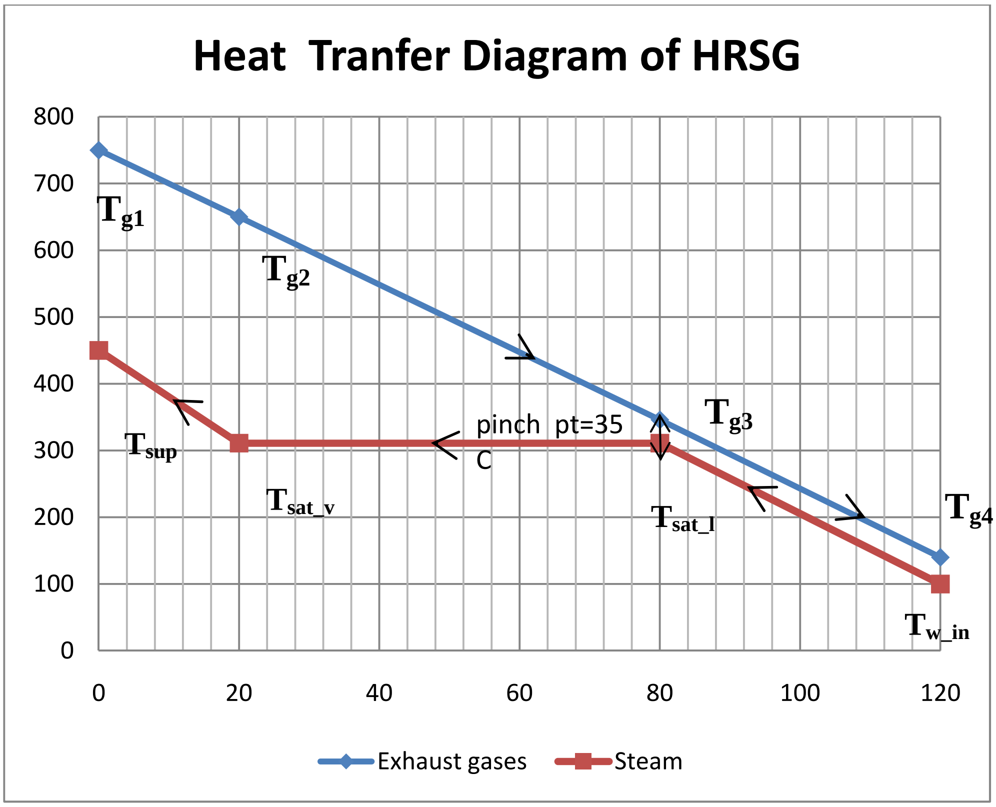


Figure 4.3 Heat Transfer Diagram of HRSG

So, first of all, pinch point temperature difference should be applied. In the current study, pinch of 35 C will be considered. In the following equations, calculation of the critical temperatures and enthalpy values are given;

$$T_{g3} = T_{sat} + PP \tag{4.45}$$

$$T_{g4} = T_{g3} - \frac{\dot{m_s}.(h_{sat\_l} - h_{w\_in})}{\dot{m_g}.cp_3 \cdot \eta_{HEX}} \tag{4.46}$$

$$h_{sup} = -\frac{\dot{m_g}.cp_{1-3} \cdot (T_{g1} - T_{g3}) \cdot \eta_{HEX}}{\dot{m_s}} + h_{sat\_l} \tag{4.47}$$

$$T_{sup} = f(P, h_{\mathrm{sup}}) \ \ [37] \tag{4.48}$$

4.3.6. HRSG Exergy Analysis

The specific flow exergies and the rate of total exergies for exhaust gases are given by;

$$\dot{E}_{exhaust\_out} = \dot{m_g}.cp \cdot \left(\mathrm{T}_{\mathrm{exhaust\_out}} - T_o\right) - T_0 \cdot cp \cdot log\left(\mathrm{T}_{\mathrm{exhaust_{out}}}/T_0\right) \qquad 4.49)$$

$$\dot{E}_{exhaust\_in} = \dot{m_g}.cp \cdot \left(\mathrm{T}_{\mathrm{exhaust\_in}} - T_o\right) - T_0 \cdot cp \cdot log\left(\mathrm{T}_{\mathrm{exhaust_{in}}}/T_0\right) \tag{4.50}$$

$$e_{in} = (h_{in} - h_o) - T_0(s_{in} - s_0) \tag{4.51}$$

$$e_{out} = (h_{out} - h_o) - T_0(s_{out} - s_0) \tag{4.52}$$

The exergy destruction and the second law efficiency of the HRSG are given respectively;

$$\dot{E}_{dest} = (\dot{E}_{exhaust_{in}} - \dot{E}_{exhaust_{out}}) - \dot{m_g}.(e_{out} - e_{in}) \tag{4.53}$$

$$\eta_{second_{HRSG}} = \frac{\dot{m_g}.(e_{out} - e_{in})}{(\dot{E}_{exhaust_{in}} - \dot{E}_{exhaust_{out}})} \qquad [37,38] \tag{4.54}$$

4.3.7. Steam Engine Energy Conservation Formulations

First law efficiency of the SE is;

$$\eta_{SE} = \frac{\dot{W}_{SE}}{\dot{m}_{steam} * (h_{in,SE} - h_{out,SE})} \tag{4.55}$$

Other important formulation regarding technical parameters, and energy considerations of the steam engine can be found in Appendix H.

4.3.8. Steam Engine Exergy Analysis

Input and output flow exergies of the SE are calculated as follows;

$$e_{in} = (h_{in} - h_o) - T_0(s_{in} - s_0) \tag{4.56}$$

$$e_{out} = (h_{out} - h_o) - T_0(s_{out} - s_0) \tag{4.57}$$

Second law efficiency is calculated as;

$$\eta_{SE,II} = \frac{\dot{w}_e}{\dot{m}_s * (e_{in} - e_{out})} \tag{4.58}$$

$$\dot{E}_{dest} = \dot{m}_s \cdot (e_{in} - e_{out}) \tag{4.59}$$

4.3.9. Condenser Energy Conservation Formulations

Condenser heat balance is as follows;

$$\dot{m}_w \cdot C_{pw} \cdot (T_2 - T_1) = \dot{m}_s \cdot (h_{11} - h_{12}) = \dot{Q}_{COND} \tag{4.60}$$

4.3.10. Condenser Exergy Analysis

User side input and output flow exergies, and total flow exergy of the condenser are calculated respectively;

$$e_{in,user} = \left(h_{in,user} - h_o\right) - T_0\left(s_{user,in} - s_0\right) \tag{4.61}$$

$$e_{out,user} = \left(h_{out,user} - h_o\right) - T_0\left(s_{out,user} - s_0\right) \tag{4.62}$$

$$\dot{E}_{CONDENSER,user} = \dot{m}_{s,user} * (\, e_{in,user} - e_{out,user} \,) \tag{4.63}$$

Plant side input and output flow exergies and total flow exergy of the condenser are calculated respectively;

$$e_{in} = (h_{in} - h_o) - T_0(s_{in} - s_0) \tag{4.64}$$

$$e_{out} = (h_{out} - h_o) - T_0(s_{out} - s_0) \tag{4.65}$$

$$\dot{E}_{CONDENSER} = \dot{m}_{steam} * (\, e_{in} - e_{out} \,) \tag{4.66}$$

Exergy destruction of the condenser and the second law efficiency can be calculated as;

$$\dot{E}_{CONDENSER,destroyed} = \dot{E}_{CONDENSER} - \dot{E}_{CONDENSER,user} \tag{4.67}$$

$$\eta_{CONDENSER,II} = \frac{\dot{E}_{CONDENSER,user}}{\dot{E}_{CONDENSER}} \quad [2,39] \tag{4.68}$$

4.3.11. Pump Energy Conservation Formulations

$$P_4 = \dot{m}_{s1} \cdot (h_8 - h_{13}) = \dot{m}_{s1} \cdot v_{f13}(P_8 - P_{13}) \tag{4.69}$$

4.3.12. Pump Exergy Conservation Formulations

Input and output flow exergies of the pump are calculated respectively;

$$\mathrm{e_{in}} = (\mathrm{h_{in}} - \mathrm{h_o}) - \mathrm{T_0}(\mathrm{s_{in}} - \mathrm{s_0}) \tag{4.70}$$

$$\mathrm{e_{out}} = (\mathrm{h_{out}} - \mathrm{h_o}) - \mathrm{T_0}(\mathrm{s_{out}} - \mathrm{s_0}) \tag{4.71}$$

Reversible work input, exergy destruction rate and the second law efficiency of the pump can be found using equations 4.73- 4.75.

$$\dot{W}_{PUMP,reversible} = \dot{m}_{flowrate} * (e_{in} - e_{out}) \quad (4.72)$$

$$\dot{E}_{PUMP,destroyed} = \dot{W}_{PUMP,reversible} - \dot{W}_{PUMP,actual} \quad (4.73)$$

$$\eta_{PUMP,II} = \frac{\dot{E}_{PUMP,reversible}}{\dot{E}_{PUMP,actual}} \quad (4.74)$$

4.3.13. Overall System Energy Conservation Formulations

Heat input of the co-generation system:

$$\dot{Q}_{in} = \dot{m}_b \cdot H_i \quad (4.75)$$

Gross electrical power, auxiliaries elect power and net electrical power can be calculated as can be seen in the following equations, 4-76 to 4-78:

$$P_{generator} = P_5 \quad (4.76)$$

$$P_{pumps} = P_2 + P_4 \quad (4.77)$$

$$P_{net} = P_{gross} - P_{aux} = P_5 - (P_2 + P_4) \quad (4.78)$$

Net electrical efficiency is:

$$\eta_{el} = P_{net}/Q_{in} \quad (4.79)$$

Thermal output is the sum of thermal outputs of HEX A, HEX B and the condenser:

$$Q_{th} = Q_{cond} + Q_{ICEC} + Q_{hex} \quad (4.80)$$

Heat efficiency and the energy utilization ratio are given respectively;

$$\eta_{th} = Q_{th}/Q_{in} \quad (4.81)$$

$$EUF = \eta_{el} + \eta_{th} \quad (4.82)$$

District heating energy input is equal to the thermal output of the system:

$$\dot{m}_w \cdot C_{pw} \cdot (T_4 - T_1) = \dot{Q}_{th} \tag{4.83}$$

Then, utilization efficiency for this cogeneration system may be defined as;

$$\eta_{cogen} = \frac{\dot{W}_{net} + \dot{Q}_{process}}{\dot{Q}_{in}} = \frac{\dot{W}_{net} + \dot{m}_{water}[h_e - h_i - T_0 - (s_e - s_i)]}{\dot{m}_{fuel} q_{LHV}} \tag{4.84}$$

4.3.14. Overall System Exergy Analysis

$$\eta_{cogen,\text{II}} = \frac{\dot{W}_{net} + \dot{E}_{process}}{\dot{E}_{in}} = \frac{\dot{W}_{net} + \dot{m}_{water}[h_e - h_i - T_0 - (s_e - s_i)]_{water}}{\dot{m}_{fuel} e_{fuel}} \quad [34] \tag{4.85}$$

4.3.15. PES and PES_R calculations

Primary energy saving;

$$PES = \left(1 - \frac{1}{\frac{CHPH_\eta}{REFH_\eta} + \frac{CHPE_\eta}{REFE_\eta}} \right) \tag{4.86}$$

Where in this case;

$$CHPE_\eta = \left(\frac{P_5 - (P_2 - P_4)}{(\dot{m}_b . H_i)} \right) \text{ and} \tag{4.87}$$

$$CHPH_\eta = \left(\frac{\dot{m}_w . c_{pw} . (T_4 - T_1)}{\dot{m}_b . H_i} \right) \tag{4.88}$$

So *PES* becomes;

$$PES = \left(1 - \frac{\dot{m}_b . H_i}{\frac{P_{net}}{REFE_\eta} + \frac{Q_{th}}{REFH_\eta}} \right) \tag{4.89}$$

Power to heat ratio is;

$$CHPE_\eta = C \cdot CHPH_\eta \tag{4.90}$$

Primary energy exergy savings value can be calculated as before;

$$PES_{RCHP} = \left[1 - \frac{1}{\left(\frac{CHPH\eta}{REFH\eta} + \frac{CHPE\eta}{REFE\eta}\right) * \left(\frac{2 - REF\Psi RCHP}{2 - \Psi_{RCHP}}\right)}\right] \tag{4.90}$$

4.3.16. Calculating Carbon Emission Values

Formulations and calculation are the same as in section 4.2.

CHAPTER 5

MATLAB SIMULINK MODELING OF THERMODYNAMIC SYSTEMS

After the thermodynamic analysis for all kinds of cogeneration system equipment is done using the accurate formulation, Simulink Module of MATLAB R2007b is used for constructing the mathematical models for power systems. It is chosen because Simulink models are flexible, more user friendly then most of the other compilers, it is easy to add new modules and functions, and has an easy to follow user interface.

Using Simulink, first of all a general Library model is constructed, enabling the user to build his own thermodynamic system to work with. Figure 5.1 shows the Library model which contains all the necessary mathematical models for the thermodynamic equipment for constructing thermodynamic systems, including the cases previously referred.

MATLAB Simulink models, as already mentioned, have user interfaces for the user to fill in the necessary parameters. Examples of the user interfaces for ICE, pipe, HEX A, HEX B, condenser, HRSG and feed water pump models can be found in Appendix I , Figures I.1- I.7.

Simulink models also hide the related calculations and functions underneath, allowing the user to reach them when necessary. The hidden schematical details of some of the constructed models are given in Appendix I, Figures I.8-I.11 as examples, which include ICE, HEX A and HRSG model, exergy and REMM analysis.

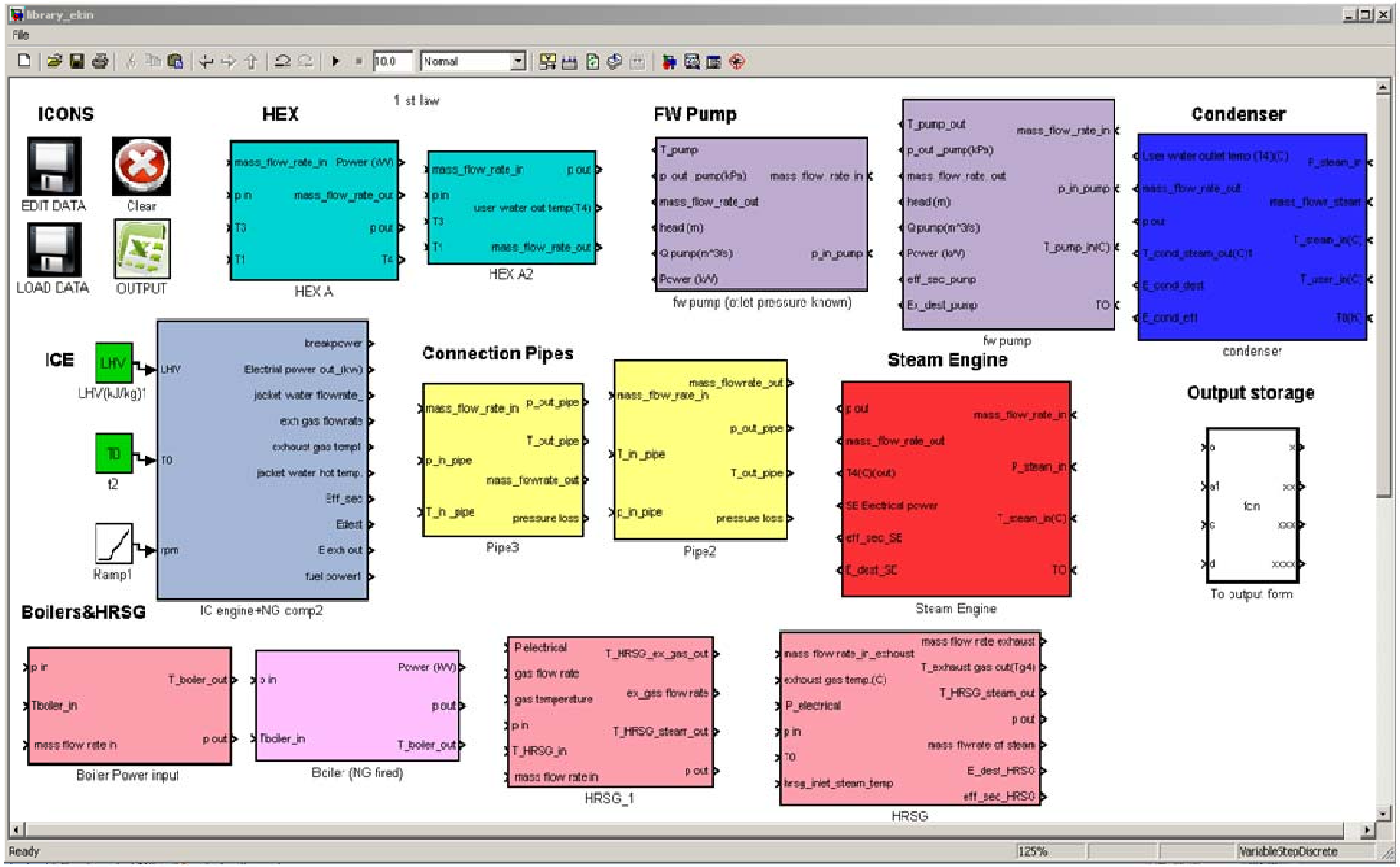


Figure 5.1 Library Model

5.1. Case 1- ICE Co-generator System

Figure 5.2 shows the constructed MATLAB Simulink model of the system in Case-1. Model is built by choosing the necessary equipments in the Library model and connecting them by using Simulink. System simulation is done considering an environment of 15 ºC ambient temperature and 1 atm ambient pressure for different ICE speeds ranging from 1000 rpm to 2150 rpm, in intervals of 115 rpm.

After the model is run, all input values and outputs of the results are written on excel files, which later can be run from main program. Graphical representations are also activated from the main model. All input values for this simulation and corresponding result sheet for the rated conditions can be found in the output file shown in Figure 5.4. Graphical representations and tabulated results from simulation of the case for all engine speeds are given in section 5.3.1 and are checked with the theoretical values given in EU HEGEL Project preliminary studies report [41].

5.2. Case 2- ICE Co-generator System with SE bottoming cycle

Figure 5.3 shows the constructed MATLAB Simulink model of Case-2. System simulation is done considering an environment of 15 ºC ambient temperature and 1 atm ambient pressure for different ICE speeds ranging from 1000 rpm to 2150 rpm, in intervals of 115 rpm.

All input values for this simulation and corresponding result sheet for the rated conditions can be found in the output file shown in Figure 5.5. Graphical representations and tabulated results from simulation of the case for all engine speeds are given in section 5.3.2 and are checked with the theoretical values given in EU HEGEL Project preliminary studies report [41].

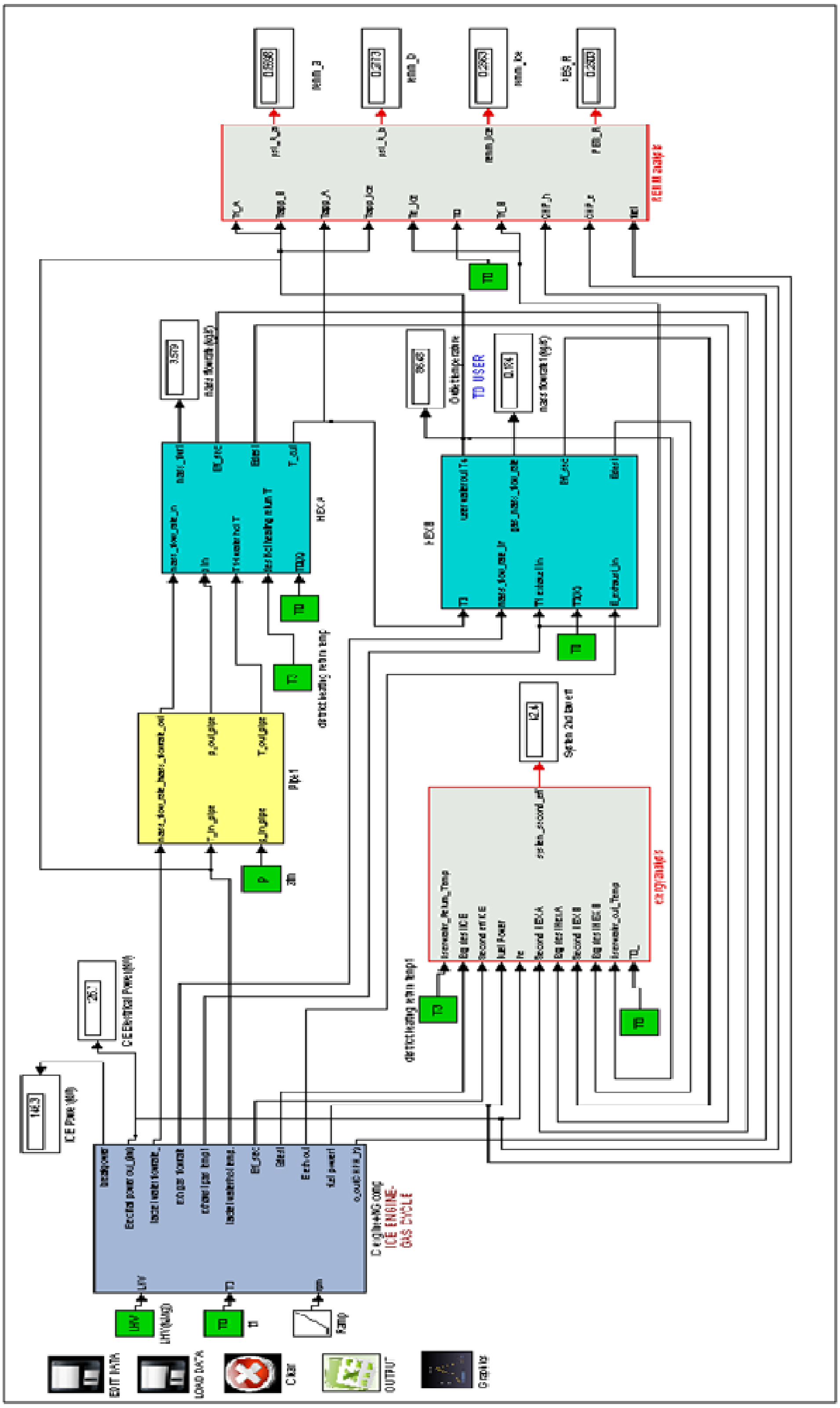


Figure 5.2 Case-1 Model

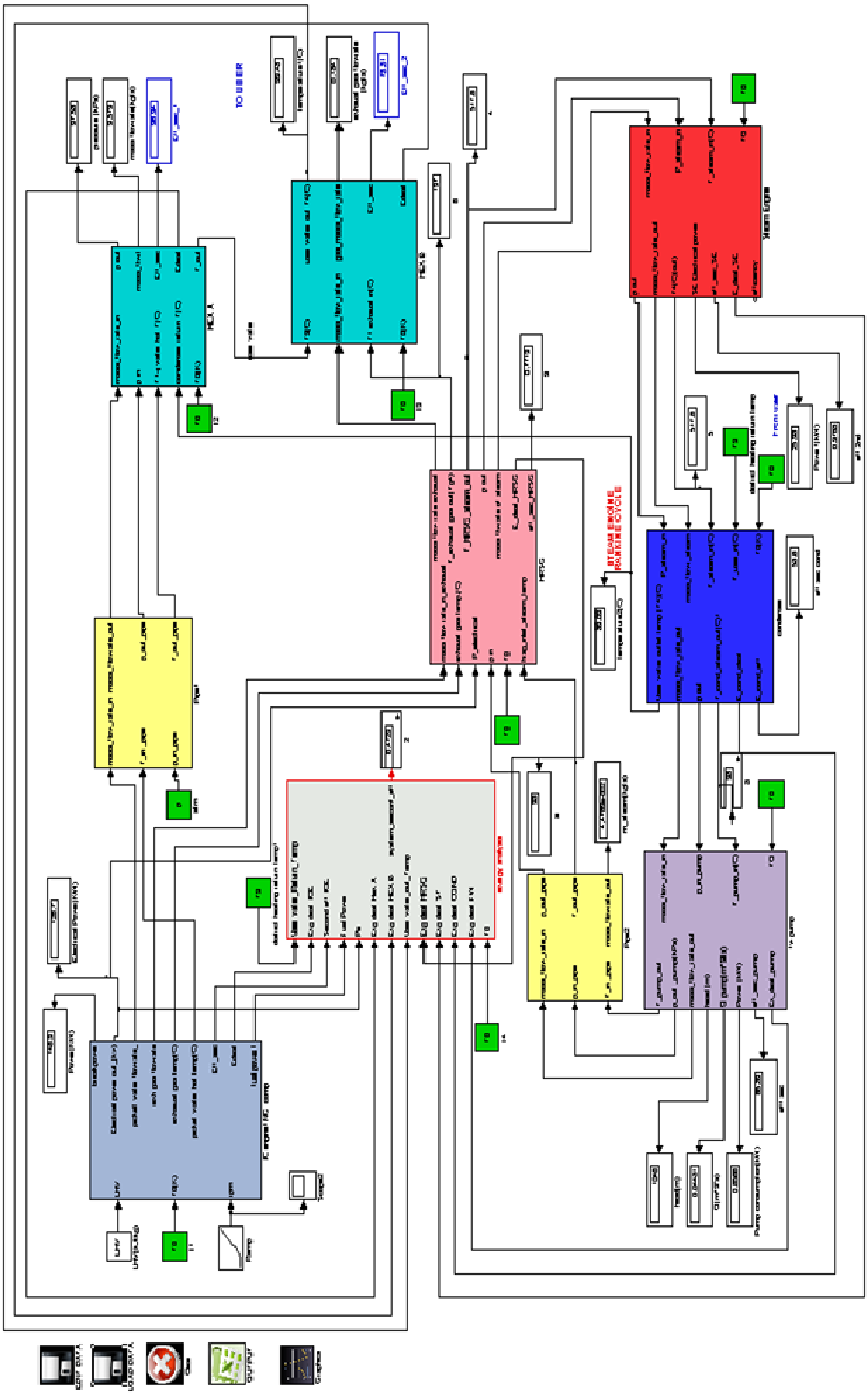


Figure 5.3 Case-2 Model

ICE cogeneration system output file
Cogenertion System: Case-1
Prepared by: Ekin Bingöl

Internal Combustion Engine:

Inputs:

Engine Speed (Nm):	2150
Engine Torque (rpm):	650
Sthociometric α:	16,3
Jacket water outlet temperature(C)	92

Outputs:

Break Power(kW):	146,3
Efficiency (%):	33,6
Electrical Power output(kW):	126,7
Jacket water flowrate (m^3/s)	3,6
Exhaust gas flowrate(kg/s):	0,178
Exhaust Gas temperature(C):	745

User Water

Inputs:

User Water mass flowrate (m^3/s):	2,5
User Water inlet Temperature (m^3/s):	70,0

Outputs:

Temperature after HEX A (C):	79,7
Temperature after HEX B (C):	86,6

Exergy Analysis

ICE 2nd law efficiency	69,23
HEX A 2nd law efficiency	81,83
HEX B 2nd law efficiency	22,31
System Overall 2nd law efficiency	41,00

ICE exergy loss (%)	34,44
HEXA exergy loss (%)	0,99
HEXB exergy loss (%)	12,31
other exergy loss (%)	11,26
Electrical output (%)	32,20
Output heat exergy (%)	8,80

Natural Gas properties:

Calorific Value (kJ/kg):	42500
Density (kg/Nm^3):	0,717
Mass flow rate (kg/s) :	0,3089
Natural gas pump consumption (kW):	5,0

PES, FES_R and EfficiencyCalculation:

CHPHη (%):	46,1
CHPEη(%):	33,6
ηCHP (%):	79,7
Power to heat ratio-C:	0,73
PES(%)	19,91
Heat Power (kW):	173,89
PES_RCHP(%)	22,29

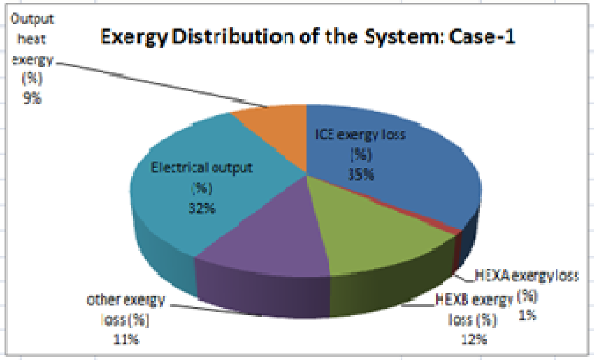


Figure 5.4 Case-1 Output File at Rated Engine Conditions

ICE cogeneration system output file
Cogenertion System: Case-2
Prepared by: Ekin Bingöl

Internal Combustion Engine:

Inputs:

Engine Torque (Nm):	2150
Engine Speed (rpm):	650
Sthociometric α:	16,3
Jacket water outlet temperature(C):	92

Outputs:

Break Power(kW):	146,3
Efficiency (%):	33,6
Electrical Power output(kW):	126,7
Jacket water flowrate (m^3/s):	3,6
Exhaust gas flowrate(kg/s):	0,178
Exhaust Gas temperature(C):	745

HRSG

Outputs:

Steam outlet Temperature(C):	487
Steam Outlet Pressure (kPa):	10000
Steam Mass flowrate (g/s):	44,8

Steam Engine

Outputs:

Steam outlet Temperature(C):	326
Steam Outlet Pressure (kPa):	100
Electrical Power (KW):	25,4

Feed Water Pump

Inputs:

Pump inlet pressure (kPa):	100
Pump working temperature (C):	98

Outputs:

Pump outlet pressure (kPa):	10000
Pump head (m):	1046
Pump Flowrate (m^3/s):	98
Pump Power Consumption(kW):	0,66

Natural Gas properties:

Calorific Value (kJ/kg):	42500
Density (kg/Nm^3):	0,717
Mass flow rate (kg/s) :	0,0089
Natural gas pump consumption (kW	5,0

User Water

Inputs:

User Water mass flowrate (m^3/s)	2,5
User Water inlet Temperature (m^3	70,0

Outputs:

Temperatur after condenser (C):	79,4
Temperature after HEX A (C):	88,7
Temperature after HEX B (C)	89,2

PES and EfficiencyCalculation:

CHPHη (%)	53,4
CHPEη(%)	40,3
η_{CHP} (%)	93,7
Power to heat ratio-C :	0,76
PES (%)	29,76
Heat Power (kW):	201,17

Exergy Analysis

ICE 2nd law efficiency (%)	69,23
HEX A 2nd law efficiency (%)	88,48
HEX B 2nd law efficiency (%)	75,90
HRSG 2nd law efficiency (%)	76,84
Condenser 2nd law efficiency (%)	55,26
Pump 2nd law efficiency (%)	65,89
System Overall 2nd law efficiency (	42,67
SE 2nd law efficiency (%)	38,95

ICE exergy loss (%)	34,44
HEXA exergy loss (%)	0,63
HEXB exergy loss (%)	0,08
HRSG exergy loss(%)	4,81
Condenser+Pump exergy loss (%)	3,50
other exergy loss (%)	5,64
Electrical output (%)	32,20
Output heat exergy (%)	10,47
SE exergy loss (%)	8,17

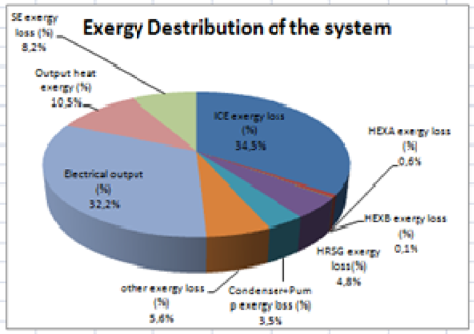


Figure 5.5 Case-2 Output File at Rated Engine Conditions

5.3. Experiments and Collecting the Experimental Data

For measuring all necessary parameters like temperature and pressure values of the processes, ambient characteristics, voltage and power outputs of the HEGEL co-generator, different instruments were used of which their list and corresponding measurement ranges are given in Appendix J, Table J.1. All analog outputs were collected and digitized by using a data logger and transferred by an interface of a computer program, LabVIEW. All output data of the program which were compiled as MS Excel spreadsheets were used for simulation purposes in the MATLAB Program.

Experiments for operation of the HEGEL co-generator were carried out in Italy, Politecnicio di Torino (PdT- University of Torino) and because of some technical difficulties, could not be repeated on the co-generation system installed in MATPUM Building, METU. Both poly-generation systems were identical. A sample spreadsheet of the gathered data in Torino with the same HEGEL co-generator module is also given in Appendix J as Table J.2.

In the Table 5.1 bellow, a summary of the results of the experiments for different engine speeds can be found. The ICE power output and other properties are not significantly affected by altitude like they are affected in a gas turbine fired plant, but nevertheless, the altitude, thus ambient pressure of Torino (215 m corresponding to 99 kPa) is considered during simulations. For investigating the effects of change of altitude on performance, a table of comparison of results of the plant performance in Torino and results of the plant performance if it was in Ankara are presented in Chapter 8. The properties of natural gas supplied to METU was also taken into account.

Table 5.1 Summary of Experimental Data gathered from PdT

Electrical power set point [kW]	0	40	50	60	70	80	90	100	110	120
Primary power [kW]	0	150,0	182,2	214,2	247,6	277,7	309,8	344,1	384,1	415,0
Engine speed [rpm]	1000	1000	1100	1200	1295	1465	1635	1805	1975	2155
Engine torque [Nm]	0	488,1	553,2	604,5	650	650	650	650	650	650
Mechanical engine efficiency	0	0,34	0,35	0,35	0,36	0,36	0,36	0,36	0,35	0,35
Jacket water flow rate [kg/s]	1,73	1,73	1,91	2,09	2,26	2,55	2,87	3,18	3,47	3,58
Fuel Mass flow rate (kg/s)*1000	3,97	4,85	5,55	6,02	6,50	6,97	7,45	7,92	8,40	8,87

5.4. Tabulation of Results From MATLAB and Graphical Representations for Cases

5.4.1. Case 1

Table 5.2 below shows the basic parameters and characteristics of the system for entire range of engine speeds for Case-1. When the table is examined, it can be seen that, maximum electrical output is 127 kW and corresponding efficiencies are calculated as; $CHPH_{\eta}$ 46.1 %, $CHPE_{\eta}$ 33.6%,and η_{CHP} 79.7 %. The highest value of the district heating temperature is 87 °C and fuel mass flow rate is 0.0088 kg/s. Maximum value of the power to heat ratio (C) is 0.9, corresponding to half of the maximum speed but at rated conditions, C is 0.73. PES value is 20 % and PES_R is 22.3 % for the rated engine conditions, referring to 2150 rpm of engine speed and 650 Nm of engine torque. System second law efficiency is 41 % at rated conditions.

Figure 5.6 shows CHP efficiency, CHP second law efficiency and *REMM* efficiency (ψ_R) variation for the ICE cogeneration system for different engine speeds and Figure 5.7 represents second law efficiencies of all the components and the overall second law efficiency w.r.t increasing engine speed.

Table 5.2 Basic Parameters and Characteristics of the System for Case-1 with respect to Increasing Engine Speed

RPM	1000	1115	1230	1345	1460	1575	1690	1805	1920	2035	2150
Electrical Power Output (kW)	44,3	56,7	69,2	79,3	86,1	92,8	99,6	106,4	113,2	119,9	126,7
CHPHη (%)	42,3	40,0	37,4	37,3	41,9	42,7	43,6	44,6	45,4	45,9	46,1
CHPEη(%)	33,6	33,6	33,6	33,6	33,6	33,6	33,6	33,6	33,6	33,6	33,6
ηCHP (%)	75,9	73,6	71,0	70,9	75,5	76,3	77,2	78,2	79,0	79,5	79,7
Power to heat ratio-C	0,8	0,8	0,9	0,90	0,80	0,79	0,77	0,75	0,74	0,73	0,73
PES(%)	16,8	14,8	12,4	12,4	16,5	17,1	17,9	18,7	19,3	19,7	19,9
Heat Power output (kW)	55,7	67,5	77,0	88,0	107,3	117,9	129,3	141,0	152,7	163,8	173,9
PES_RCHP (%)	23,7	21,0	18,7	18,2	20,2	20,6	21,1	21,6	22,1	22,3	22,3
ICE 2nd law efficiency (%)	66,3	66,9	67,3	67,6	68,4	68,5	68,6	68,7	68,8	69,0	69,2
HEX A 2nd law efficiency (%)	78,9	79,3	79,6	79,9	80,2	80,5	80,9	81,2	81,5	81,8	81,8
HEX B 2nd law efficiency (%)	12,8	12,3	10,8	11,4	16,3	17,2	18,2	19,4	20,5	21,5	22,3
System 2nd law efficiency (%)	39,2	38,9	38,6	38,6	39,6	39,8	40,1	40,4	40,7	40,9	41,0
Fuel Mass flow rate (kg/s)*1000	3,1	3,97	4,8	5,55	6,02	6,50	6,97	7,45	7,92	8,40	8,87
Jacket water mass flowrate (kg/s)	1,7	1,9	2,1	2,4	2,5	2,7	3,0	3,2	3,4	3,5	3,6
District heating out temperature(C)	75,3	76,4	77,4	78,4	80,3	81,3	82,4	83,5	84,6	85,7	86,6

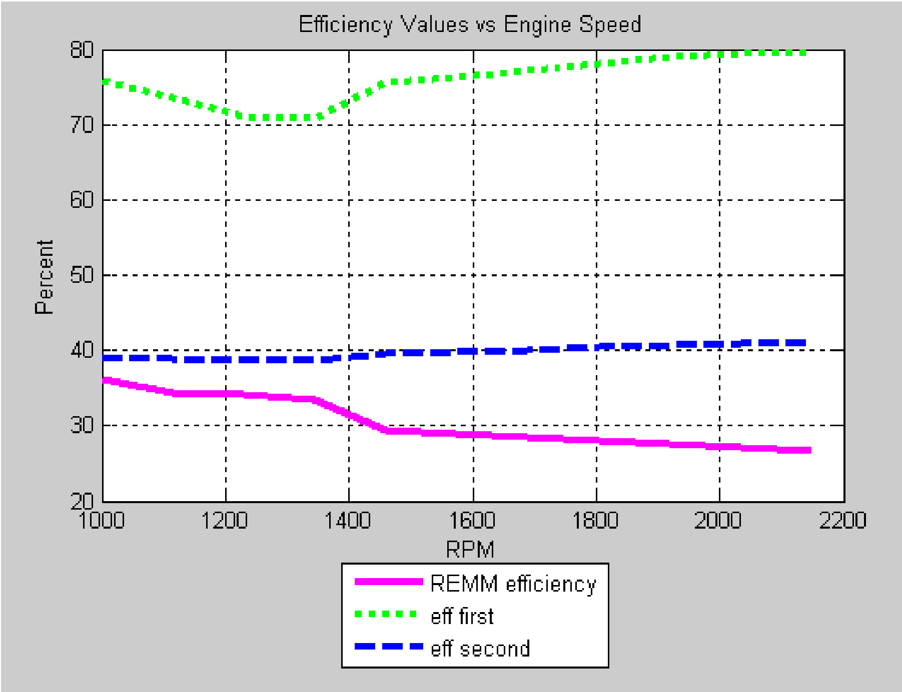


Figure-5.6 *REMM*, First Law and Second Law Efficiencies versus Engine Speed for Case-1

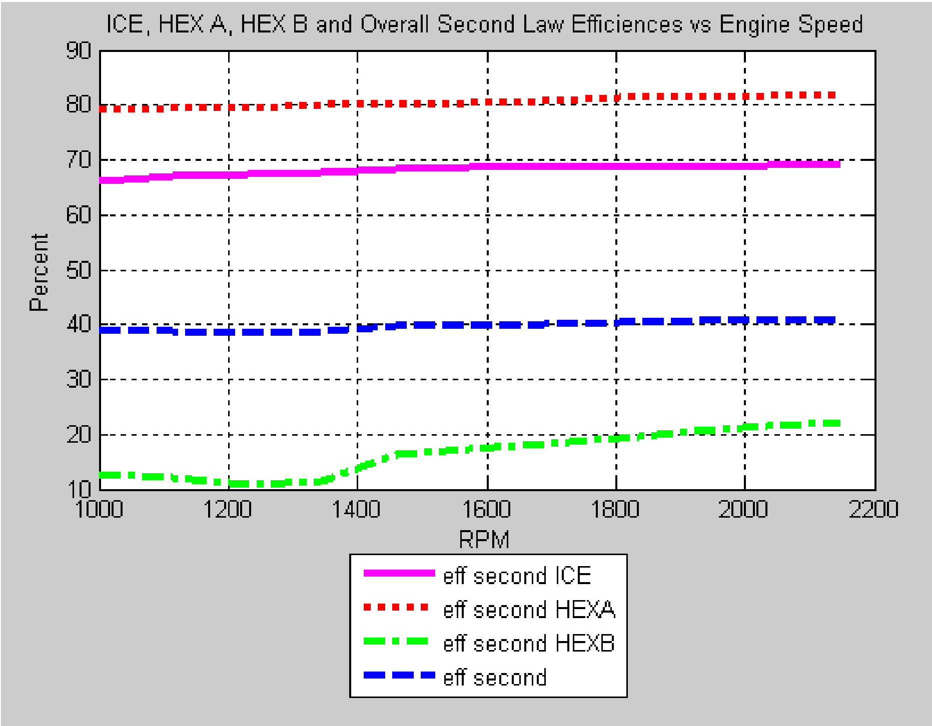


Figure 5.7 Second law Efficiencies of ICE, HEX_A, HEX_B and the System versus Engine Speed for Case-1

Figure 5.8 shows *PES*, PES_R and *REMM* efficiency values for the entire range of engine speeds, and Figure 5.9 shows the exergy distribution of the system for the rated conditions, 2150 rpm.

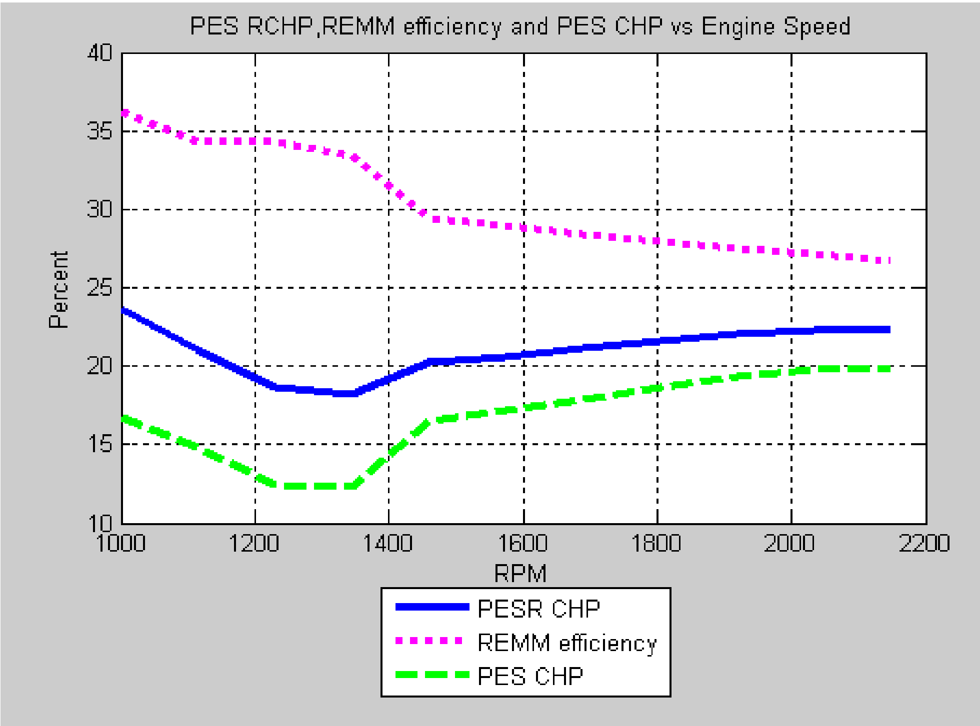


Figure 5.8 PES_R, *PES* and *REMM* Efficiency versus Engine Speed for Case-1

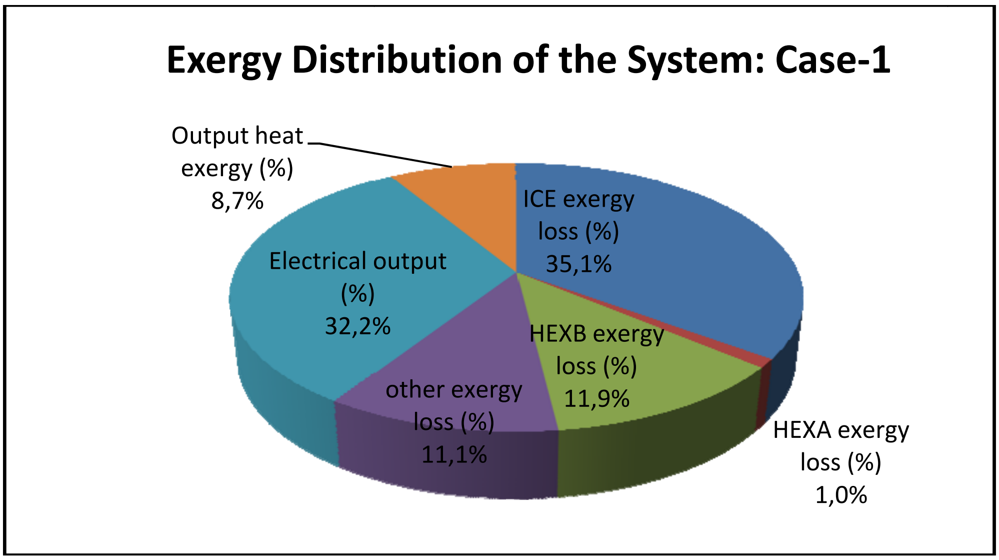


Figure 5.9 Exergy Distribution of the System for Rated Conditions for Case-1

5.4.2. Case-2

Table 5.3 shows the basic parameters and characteristics of the system for entire range of engine speeds for Case-2. When the table is examined, it can be seen that , maximum total electrical output (ICE+SE) is 148.5 kW and corresponding efficiencies are calculated as; $CHPH_\eta$ 48.6 %, $CHPE_\eta$ 39.4%,and η_{CHP} 87.9 %. The highest value of the district heating temperature is 87.5 °C and fuel mass flow rate is 0.0088 kg/s. Maximum value of the power to heat ratio (C) is 0.84, corresponding to half of the maximum speed but at rated conditions, C is 0.81. PES value is 25.3% and PES_R is 48.1 % for the rated engine conditions, referring to 2150 rpm of engine speed and 650 Nm of engine torque. System second law efficiency is 42.6 % at rated conditions.

Figure 5.10 shows CHP efficiency, CHP second law efficiency and *REMM* efficiency (ψ_R) for the ICE cogeneration system and Figure 5.11 represents second law efficiencies of all the components and the overall second law efficiency w.r.t increasing engine speed.

Table 5.3 Basic Parameters and Characteristics of the System for Case-2 with respect to Increasing Engine Speed

RPM	1000	1115	1230	1345	1460	1575	1690	1805	1920	2035	2150
Electrical Power Output ICE(kW)	44,3	56,7	69,2	79,3	86,1	92,8	99,6	106,4	113,2	119,9	126,7
Electrical Power Output SE (kW)	0,0	1,4	2,0	3,2	12,2	13,0	13,7	14,4	15,6	17,8	21,8
CHPHη (%)	53,1	49,0	46,0	45,3	45,5	46,2	47,0	47,9	48,6	48,9	48,6
CHPEη(%)	33,6	34,4	34,6	35,0	38,4	38,3	38,2	38,2	38,2	38,6	39,4
ηCHP (%)	86,7	83,5	80,6	80,3	83,9	84,5	85,3	86,1	86,8	87,5	87,9
Power to heat ratio-C :	0,63	0,70	0,75	0,77	0,84	0,83	0,81	0,80	0,79	0,79	0,81
PES (%)	0,0	19,7	17,7	17,6	22,2	22,6	23,0	23,5	24,0	24,6	25,3
Heat Power (kW):	70,0	82,6	94,8	106,8	116,5	127,5	139,3	151,6	163,6	174,4	183,1
PES_RCHP (%)	0,00	43,41	42,37	42,50	46,13	46,31	46,52	46,76	47,05	47,47	48,10
ICE 2nd law efficiency (%)	66,3	66,9	67,3	67,6	68,4	68,5	68,6	68,7	68,8	69,0	69,2
HEX A 2nd law efficiency (%)	77,2	78,1	79,0	79,9	80,7	81,7	82,8	83,9	85,0	86,0	86,8
HEX B 2nd law efficiency (%)	47,9	51,0	52,6	54,8	57,1	60,1	63,8	68,0	72,1	75,1	75,7
HRSG 2nd law efficiency (%)	0,0	77,4	77,4	77,3	78,1	77,8	77,5	77,4	77,4	77,5	77,5
Condenser 2nd law efficiency (%)	58,7	59,0	59,2	59,5	59,7	59,9	60,2	60,6	60,9	61,3	61,6
Pump 2nd law efficiency (%)	65,9	65,9	65,9	65,9	65,9	65,9	65,9	65,9	65,9	65,9	65,9
SE 2nd law efficiency (%)	0,0	10,6	11,8	15,1	37,9	35,8	33,4	31,2	30,0	30,4	33,0
System 2nd law efficiency (%)	41,2	40,6	40,2	40,2	40,3	40,6	40,8	41,1	41,4	41,5	41,5
Steam mass flow rate (g/s)	7,6	11,3	14,3	17,5	20,3	23,7	27,7	32,0	36,6	40,9	44,8
Fuel Mass flow rate (kg/s)	3,1	4,0	4,8	5,5	6,0	6,5	7,0	7,4	7,9	8,4	8,9
Jacket water mass flowrate (kg/s)	1,73	1,94	2,14	2,35	2,54	2,75	2,97	3,20	3,39	3,53	3,58
District heating out temperature(C)	76,7	77,9	79,1	80,2	81,1	82,2	83,3	84,5	85,6	86,7	87,5

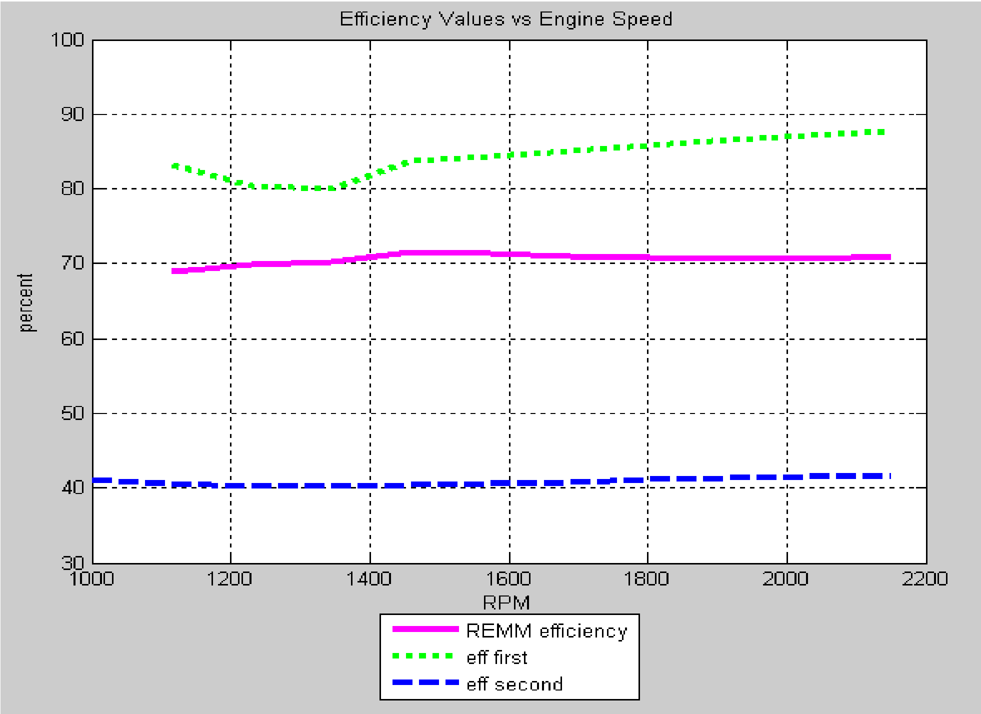


Figure 5.10 *REMM*, First law and Second law Efficiencies versus Engine Speed for Case-2

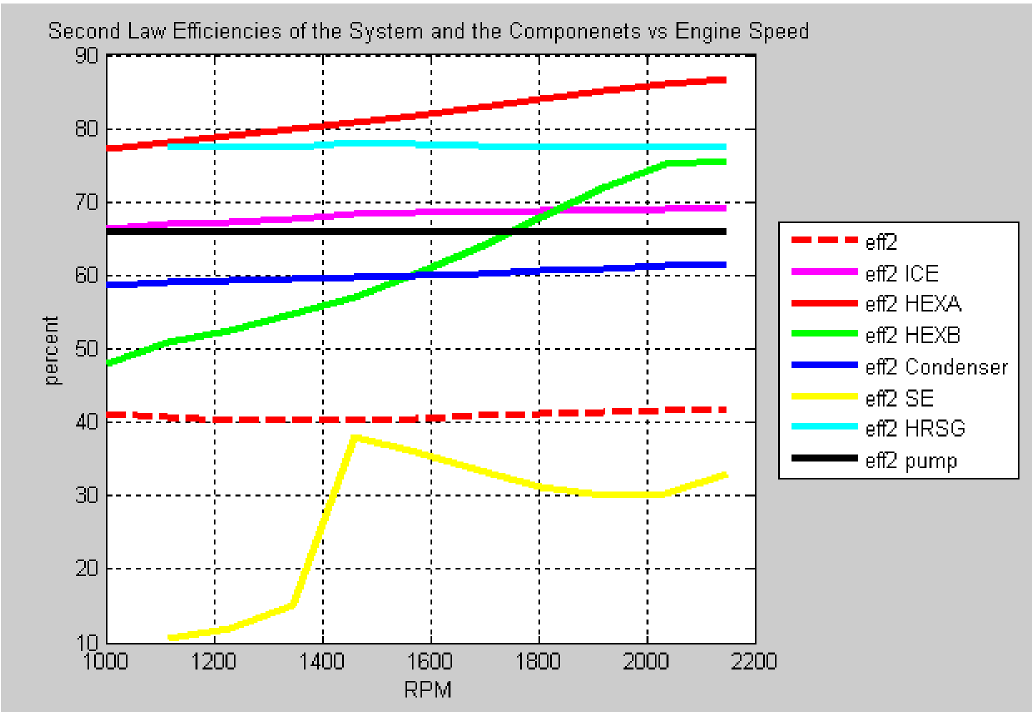


Figure 5.11 Second law Efficiencies of ICE, HEX_A, HEX_B and the System versus Engine Speed for Case-2

Figure 5.12 shows *PES, PES_R* and *REMM* efficiency values for the entire range of engine speeds, and Figure 5.13. shows the exergy distribution of the system of Case-2 for the rated conditions, 2150 rpm.

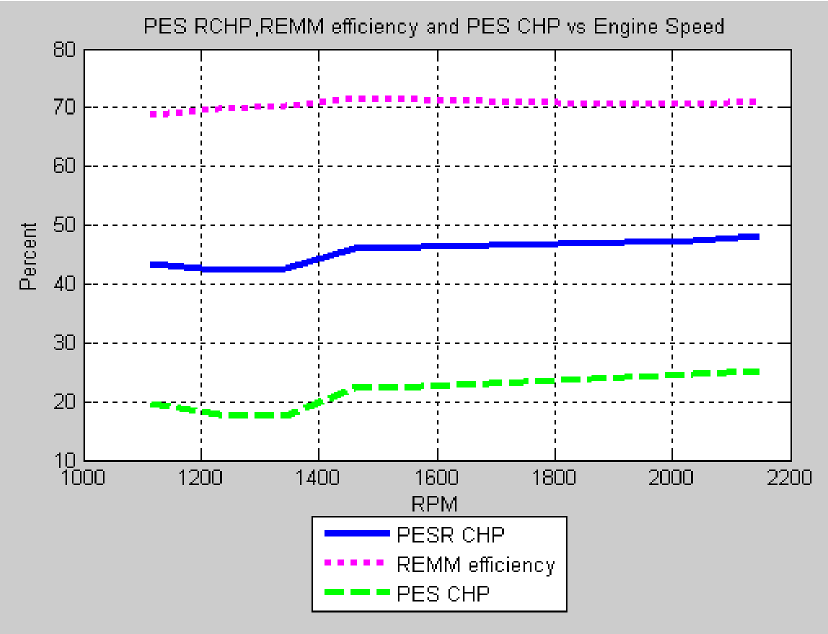


Figure 5.12 *PES_R*, *PES* and *REMM* Efficiency versus Engine Speed for Case-2

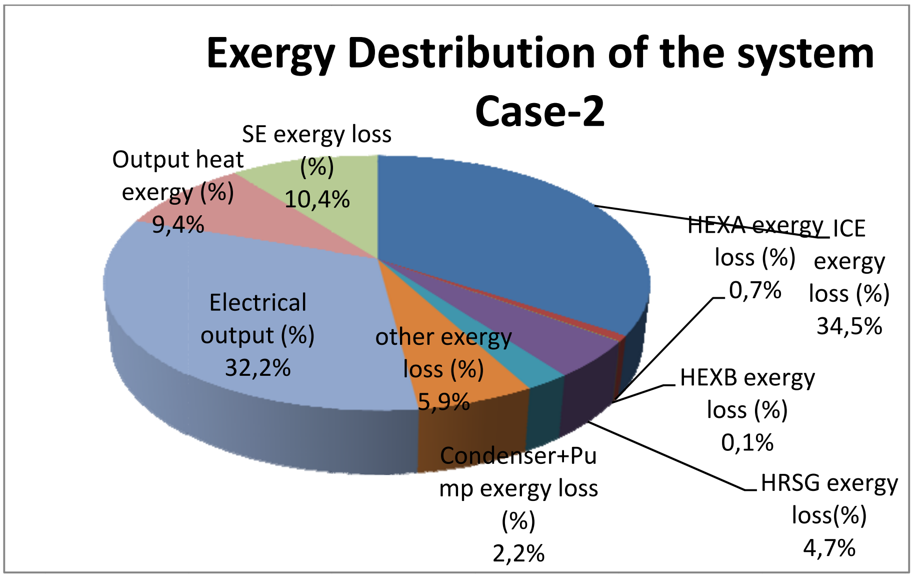


Figure 5.13 Exergy Distribution of the System for Case-2

5.5. Discussion of Results and Comparison of the Cases

When all the results are examined, it is clear that both systems may provide sustainable, high performance energy generation.

Generally, in ICEs, at low engine speeds, mechanical efficiency decreases, affecting the overall efficiency, but in this case, the particular engine has almost constant part load efficiency, which is very advantageous for a CHP system. (Appendix K, Figure K.2)

Performance characteristics obtained in this study reveal that, after a certain point which is 1300 rpm, first and second law efficiencies increase and power to heat ratio decreases, with increasing engine speed. When the ICE characteristics are examined, after 1300 rpm, engine torque, which increases up to that point remains constant and around that engine speed, the thermal efficiency drops. That is why before that point, performance trends are different and at that point, overall efficiencies drop.

Regarding Figures 5.6 for Case-1 and Figure 5.10 for Case-2, it can be seen that second law and first law efficiency trends are similar, but *REMM* efficiency does not show same trend for Case-1. This is mainly because; *REMM* analysis is sensitive to the process temperatures. *REMM* is a very effective tool for balancing demand and supply temperatures, thus has an important role in effective utilization of natural resources and lessening harmful emissions. In Case 1, a higher content of energy of the exhaust gases is thrown away, and w.r.t increasing engine speed, temperature of the gases increase thus *REMM* efficiency, which is sensitive to the temperature matches during the process decreases. But in Case-2, Temperature of the exhaust gases are lower, since they enter HRSG and steam is produced, *REMM* efficiency increases w.r.t increasing engine speed. Trend of the exhaust gas temperature of the ICE w.r.t. engine speed can be found in Appendix K, Figure K.4.

In Figures 5.7 and 5.11, system components and overall system second law efficiency trends can be seen, respectively for Case-1 and Case-2. In Figure 5.7, ICE, HEX A and HEX B second law efficiencies show similar trends. All the second law efficiencies increase w.r.t. increasing engine speed. The reason why HEX_A has an average of 80% efficiency while HEX B has about 25% is that, HEX_B has a very high exhaust gas temperature, causing high exergy destruction, since the hot gases are wasted. In Figure 5.11, again all second law efficiency trends are found to be increasing except for Steam engine. SE component shows first an increasing, then slightly decreasing behavior. In this case, condenser has the lowest exergy efficiency among other components except for SE, since the energy content of the steam is wasted to surroundings. HEX B second law efficiency increases rapidly w.r.t increasing engine speed, and at higher engine speeds, since the exhaust gas temperature is lower in this case, exergy destruction is lower in HEX B, compared to Case-1. HEX B, pump, HRSG and ICE second law efficiencies show similar trends.

When the *PES* values are considered, it can be said that, primary energy savings are very high considering the lowest permissible value of 10 % in the EU directive for the qualification of co-generation, which indicates the efficiency. Results show that in the particular case study, even at part loads, *PES* values are at least 14%, when Figures 5.8 and 5.12 are considered. PES_R usually decreases with *REMM* efficiency, but for Case-1, increasing electrical and thermal efficiencies of the system w.r.t. increasing engine speed compensates the fall. Figure 5.8 also shows that, heat recovery is a major fuel saving method. If the heat which is to be wasted at high temperatures can be recovered instead, REMM efficiency increases, influencing PES_R value to increase, indicating that, cogeneration system is more effective, qualified and environmentally friendly. The difference between *PES* and PES_R values are because of the contribution of the exergy components. For Case-2, trends are similar except for *REMM* efficiency, the reason of which was explained before. In this case, *PES* and PES_R values are obviously higher.

Exergy distribution in Figure 5.9 indicates that, for Case-1, 32% of the fuel exergy is converted to electricity, 8.7 % is converted to useful heat power and the rest is destroyed in the system. For Case-2, Figure 5.13 indicates that, 32.2 % of the fuel exergy is converted to electricity, 9.4 % is converted to useful heat power and the rest is destroyed in the system.

CHAPTER 6

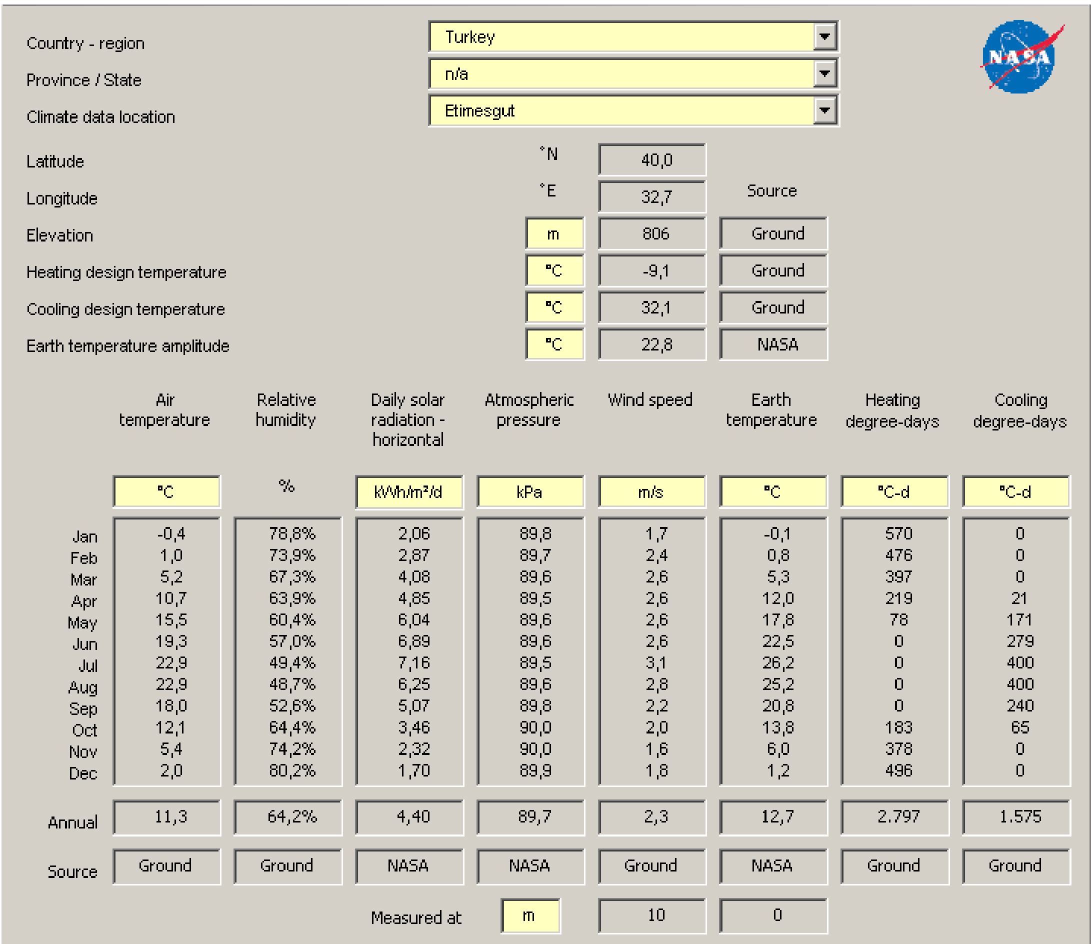
ENERGY MODELING AND ECONOMICAL ANALYSIS OF POWER SYSTEMS USING RETSCREEN PROGRAM

Energy characteristics of the building should be evaluated, where poly-generation will take place. This may be done by determination of previous consumption data for heat and electricity (and cold requirements). Correct definition of electrical and heating loads of the building is significantly important for evaluating the pay-back time, operation cost and initial investment costs. To investigate the behavior of poly-generation systems, during partial load operation, a detailed energy analysis had been done and energy model of MATPUM building is studied for different cases, by RETScreen Clean Energy Project Analysis Software. For each case, electrical and thermal loads have been evaluated and graphically represented, pay back analysis and emissions analysis had been done.

For evaluating the models, it is assumed that;

- Electrical and thermal energies are 100% self consumed. (The plant capacities are lower than the user loads of the Middle East Technical University, Turkey);
- Peak load operation at winter season is at rated power;
- The thermal energy recovered from the prime movers during the winter season is fully used for heating purposes. [40]

For all cases, the co-generator is connected to local electrical grid to sell or buy electricity when the energy produced is higher or lower than the user load. The heat from the co-generator is used to satisfy the thermal load. In the configuration considered, a boiler is also present to supply heat in the building when the co-generator is switched-off, or unsatisfactory. RETScreen generates normalized profiles starting from data about weather conditions, which are taken from NASA (Figure 6.1) and the constructional features of the MATPUM building (Figure 6.2) some of which are to be given by the user.



Country - region		Turkey	
Province / State		n/a	
Climate data location		Etimesgut	
Latitude	°N	40,0	
Longitude	°E	32,7	Source
Elevation	m	806	Ground
Heating design temperature	°C	-9,1	Ground
Cooling design temperature	°C	32,1	Ground
Earth temperature amplitude	°C	22,8	NASA

	Air temperature	Relative humidity	Daily solar radiation - horizontal	Atmospheric pressure	Wind speed	Earth temperature	Heating degree-days	Cooling degree-days
	°C	%	kWh/m²/d	kPa	m/s	°C	°C-d	°C-d
Jan	-0,4	78,8%	2,06	89,8	1,7	-0,1	570	0
Feb	1,0	73,9%	2,87	89,7	2,4	0,8	476	0
Mar	5,2	67,3%	4,08	89,6	2,6	5,3	397	0
Apr	10,7	63,9%	4,85	89,5	2,6	12,0	219	21
May	15,5	60,4%	6,04	89,6	2,6	17,8	78	171
Jun	19,3	57,0%	6,89	89,6	2,6	22,5	0	279
Jul	22,9	49,4%	7,16	89,5	3,1	26,2	0	400
Aug	22,9	48,7%	6,25	89,6	2,8	25,2	0	400
Sep	18,0	52,6%	5,07	89,8	2,2	20,8	0	240
Oct	12,1	64,4%	3,46	90,0	2,0	13,8	183	65
Nov	5,4	74,2%	2,32	90,0	1,6	6,0	378	0
Dec	2,0	80,2%	1,70	89,9	1,8	1,2	496	0
Annual	11,3	64,2%	4,40	89,7	2,3	12,7	2.797	1.575
Source	Ground	Ground	NASA	NASA	Ground	NASA	Ground	Ground
			Measured at	m	10	0		

Figure 6.1 Weather Condition of Ankara, TURKEY

Heating project	Unit		
Base case heating system	Single building - space heating		
Heated floor area for building	m²	3.000	
Fuel type		Natural gas - m³	
Seasonal efficiency	%	67%	
Heating load calculation			
Heating load for building	W/m²	65,0	
Domestic hot water heating base demand	%	5%	
Total heating	MWh	400	
Total peak heating load	kW	195,0	
Fuel consumption - annual	m³	63.296	
Fuel rate	$/m³	0,400	
Fuel cost		$	25.318
Proposed case energy efficiency measures			
End-use energy efficiency measures	%		
Net peak heating load	kW	195,0	
Net heating	MWh	400	

Figure 6.2 Constructional Features MATPUM Building

The energy modeling of the building can be seen below, in Figure 6.3, which basically consists of the electrical and thermal power demanded by the building covering a whole year of operation and electricity costs. In Figure 6.4, graphical representation of heating and electrical loading of MATPUM building can be seen.

Base case load characteristics				
Month		Power gross average load kW	Power net average load kW	Heating average load kW
January		65	65	122
February		64	64	98
March		67	67	67
April		67	67	35
May		68	68	13
June		63	63	2
July		62	62	2
August		61	61	2
September		64	64	2
October		64	64	29
November		68	68	62
December		65	65	90
System peak electricity load over max monthly average		0,0%		
Peak load - annual		68	68	195
Electricity	MWh	568	568	
Electricity rate - base case	$/kWh	0,220	0,220	
Total electricity cost		$ 125.045	$ 125.045	

Figure 6.3-Energy Model of MATPUM Building

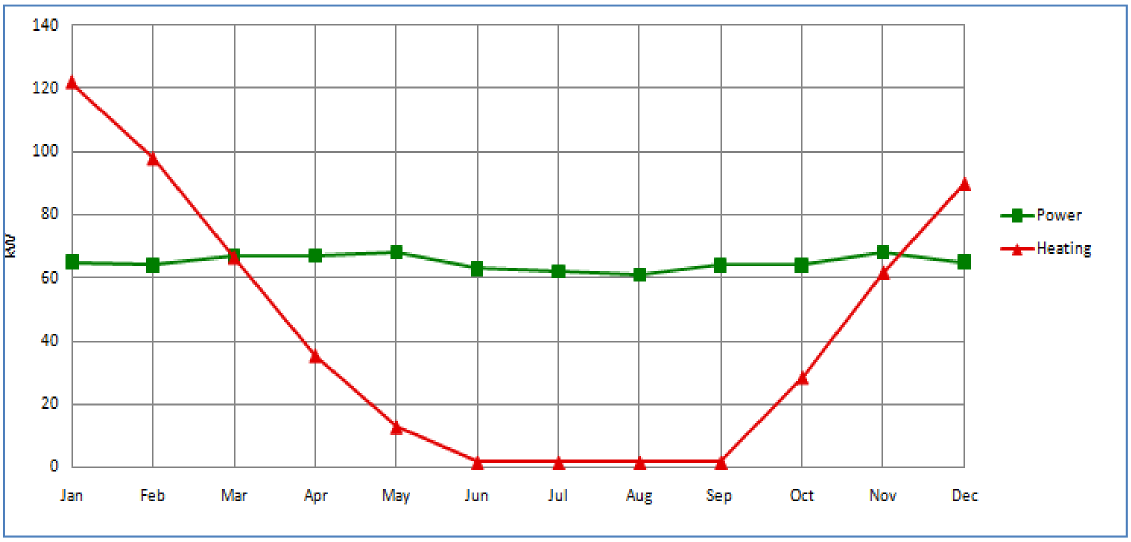


Figure 6.4-MATPUM Heating and Electrical Loading

6.1. Energy Model for Case-1

Energy modeling of MATPUM building (only topping cycle, which is schematically represented in Figure 4.1) is studied as Case-1 for partial load operation. [3]

Details about the base load power system, and operating strategy for the co-generation system are summarized in Figure 6.5. As can be seen, ICE availability is chosen to be 95%, minimum capacity of the engine is 25%, heat rate is 12800kJ/kWh and heating capacity is 149 kW.

In poly-generation systems, load following can be done in three ways: Heat load following, electrical load following and full throttle operation. In applications where load is variable, full throttle operation is not effective and pay-back period is longer. If the heat demand of the building is below full throttle load, there is unnecessary fuel consumption. In heat load following and electrical load following operations, more effective solutions can be found. Usually, the load which is higher is chosen to follow. In the current study, heat load following has been chosen.

Proposed case power system					Incremental initial costs	
System selection	Base load system					
Base load power system						
Technology	Reciprocating engine					
Availability	%		95,0%	8.322 h		
Fuel selection method	Single fuel					
Fuel type	Natural gas - m³					
Fuel rate	$/m³	0,400				
Reciprocating engine						
Power capacity	kW	120	176,5%		$ 364.000	See product database
Minimum capacity	%	25,0%				
Electricity delivered to load	MWh	211	37,1%			
Electricity exported to grid	MWh	45				
Manufacturer						
Model						
Heat rate	kJ/kWh	12.800				
Heat recovery efficiency	%	48,5%				
Fuel required	GJ/h	1,5				
Heating capacity	kW	148,7	76,3%			

Operating strategy - base load power system		
Fuel rate - base case heating system	$/MWh	63,25
Electricity rate - base case	$/MWh	220,00
Fuel rate - proposed case power system	$/MWh	42,38
Electricity export rate	$/MWh	0,12
Electricity rate - proposed case	$/MWh	0,38

Operating strategy	Electricity delivered to load MWh	Electricity exported to grid MWh	Remaining electricity required MWh	Heat recovered MWh	Remaining heat required MWh	Power system fuel MWh	Operating profit (loss) $	Efficiency %
Full power capacity output	540	459	28	374	26	3.551	-1.737	38,7%
Power load following	540	0	28	319	82	1.320	63.826	44,7%
Heating load following	**211**	**45**	**358**	**316**	**84**	**908**	**106.465**	**63,0%**

Select operating strategy	Heating load following

Figure 6.5 Base Load Power System and Operating Strategy for Case-1

For the current system, seen from Figure 6.6, 211 MWh_e is the base load for electricity supplied to the building, 358 MWh_e additional energy is necessary for suplying the peak demand. When electricity demand is below the base load, 45 MWh_e electrical energy is sold to the grid. This is graphically represented in Figure 6.7 When heat load demand is considered, 316 MWh_h is the base demand of the building and 84 MWh_h extra energy is needed for peak loads, as can be seen in Figure 6.8.

Fuel type	Fuel consumption - unit	Fuel consumption	Capacity (kW)	Energy delivered (MWh)
Natural gas	m^3	96.161	120	211
Electricity	MWh	358	152	358
				45
		Total	**272**	**613**
Recovered heat			149	316
Natural gas	m^3	13.264	300	84
		Total	**449**	**400**

Figure 6.6 Electrical and Heating Capacities Regarding Operation

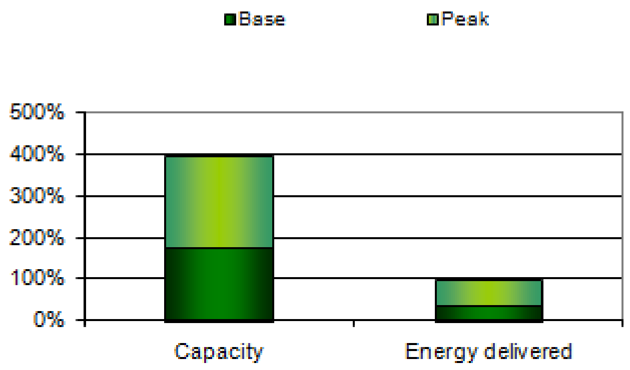


Figure 6.7 Building Electrical Energy Demand and Delivered Electrical Energy Percentages

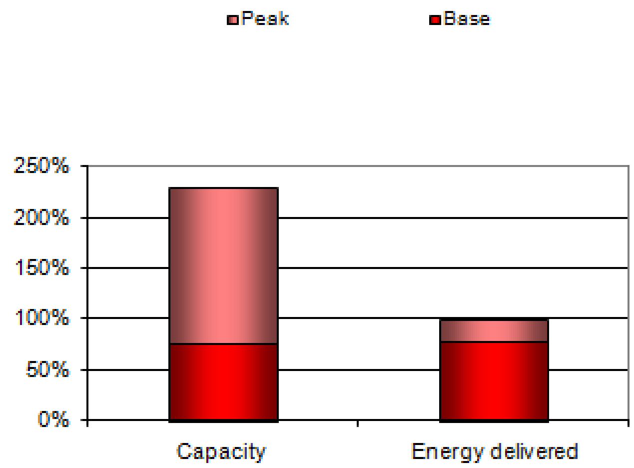


Figure 6.8 Building Heating Energy Demand and Delivered Heating Energy Percentages

Next step is the emission analysis for the system. Poly-generation systems have effective influence on decreasing emission rates. RETScreen program can evaluate green house emissions. In Figure 6.9, Greenhouse gas emission values of the current system is summarized. For determination of gas emissions, 0,445 ton CO2/MWh is the accepted value.

Base case electricity system (Baseline) **Country - region**	**Fuel type**	**GHG emission factor (excl. T&D)** **tCO2/MWh**	**T&D losses** **%**	**GHG emission factor** **tCO2/MWh**
Turkey	All types	0,445	15,0%	0,523
Electricity exported to grid	MWh	45	T&D losses	15,0%
GHG emission				
Base case	tCO2	438,5		
Proposed case	tCO2	394,5		
Gross annual GHG emission reduction	tCO2	44,0		
GHG credits transaction fee	%	0,0%		
Net annual GHG emission reduction	tCO2	44,0	is equivalent to	44,0
GHG reduction income				
GHG reduction credit rate	$/tCO2	0,00		

Figure 6.9-CHP Greenhouse Gas Emission Values

Finally, a Pay Back Period (PBP) analysis has been developed assuming investment costs and other regarding parameters.

Payback period in capital budgeting refers to the period of time required for the return on an investment to "repay" the sum of the original investment, usually measured in years. Payback period as a tool of analysis is often used because it is conceptually simple, easy to apply and easy to understand. When used carefully to compare similar investments, it can be quite useful.

For poly-generation systems, payback period is affected by cost per unit power, investment cost of the poly-generation system and operational costs, which mainly includes fuel consumption, labor and maintenance costs.

The income of the poly-generation system is basically electrical energy sold to grid when production of electrical energy is higher than internal consumption.

Fuel consumption cost is evaluated by including natural gas costs. In the current study, for energy modeling, unit natural gas costs are considered to be 0.6 TL/m^3 for city of Ankara.

Electrical energy costs differ by end user types and cities. In this study, for Ankara, electrical consumption cost is evaluated as 0.22 TL/kWh and electricity selling price to the grid is taken as 0.12 TL/kWh. [40,41]

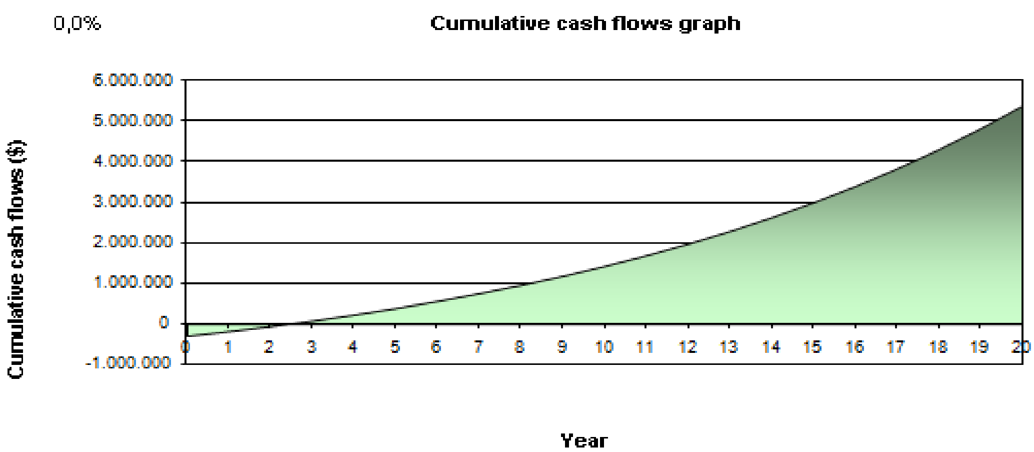


Figure 6.10-Cumulative Cash Flows and Pay-Back Graph

If electrical cost / fuel cost ratio becomes higher for a poly-generation system, The PBP decreases. If this ratio is smaller than an optimum value, system would never pay itself back. The corresponding PBP for partial load operation for Case-1 is about 2.5 years due to the fact that the investment cost for HEGEL unit is higher than investment costs for conventional boiler which is the most currently used heating system for buildings. Above is the cumulative cash flows and pay-back graph (Figure 6.10).

6.2. Energy Model for Case-2

Energy modeling of MATPUM building (topping and bottoming cycles, which is schematically represented in Figure 4.2) is studied as Case-2 for partial load operation. [3]

Details about the base load power system, and operating strategy for the co-generation system are summarized in Figure 6.11. As can be seen, ICE availability is chosen to be 95%, minimum capacity of the engine is 25%, heat rate is 12800kJ/kWh and heating capacity is 197 kW. Operating strategy is heat load following.

Proposed case power system					Incremental initial costs	
System selection		Base load system				
Base load power system						
Technology		Reciprocating engine				
Availability	%	95,0%		8.322 h		
Fuel selection method		Single fuel				
Fuel type		Natural gas - m³				
Fuel rate	$/m³	0,600				
Reciprocating engine						
Power capacity	kW	157	230,9%		$ 464.000	See product database
Minimum capacity	%	25,0%				
Electricity delivered to load	MWh	210	36,9%			
Electricity exported to grid	MWh	47				
Manufacturer						
Model						
Heat rate	kJ/kWh	12.800				
Heat recovery efficiency	%	49,2%				
Fuel required	GJ/h	2,0				
Heating capacity	kW	197,4	101,2%			

Operating strategy - base load power system		
Fuel rate - base case heating system	$/MWh	63,25
Electricity rate - base case	$/MWh	220,00
Fuel rate - proposed case power system	$/MWh	63,57
Electricity export rate	$/MWh	0,12
Electricity rate - proposed case	$/MWh	0,22

Operating strategy	Electricity delivered to load MWh	Electricity exported to grid MWh	Remaining electricity required MWh	Heat recovered MWh	Remaining heat required MWh	Power system fuel MWh	Operating profit (loss) $	Efficiency %
Full power capacity output	540	767	28	380	20	4.646	-145.137	35,3%
Power load following	540	0	28	321	79	1.920	23.306	44,9%
Heating load following	**210**	**47**	**359**	**323**	**78**	**912**	**87.377**	**63,5%**

Select operating strategy: Heating load following

Figure 6.11 Base Load Power System and Operating Strategy for Case-2

For the current system, as can be seen from Figure 6.12, 210 MWh_e is the base load for electricity supplied to the building. 359 MWh_e extra energy is necessary for the peak electrical energy. When electricity demand is below the base load, 47 MWh_e electrical energy is sold to the grid. Graphical representation of electrical energy demand and supply can be seen in Figure 6.13. When heat load demand is considered, 323 MWh_h is the base demand of the building and 78 MWh_h extra energy is needed for peak loads, as can be seen in Figure 6.14.

Fuel type	Fuel consumption - unit	Fuel consumption	Capacity (kW)	Energy delivered (MWh)
Natural gas	m^3	96.676	157	210
Electricity	MWh	359	152	359
				47
		Total	**309**	**615**
Recovered heat			197	323
Natural gas	m^3	12.270	300	78
		Total	**497**	**400**

Figure 6.12 Electrical and Heating Capacities Regarding Operation

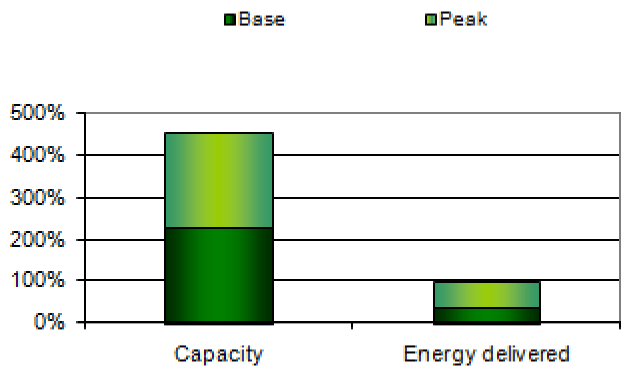


Figure 6.13 Building Electrical Energy Demand and Delivered Electrical Energy Percentages

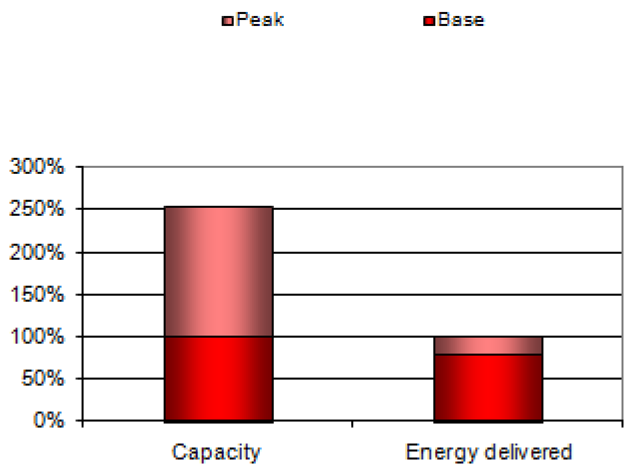


Figure 6.14 Building Heating Energy Demand and Delivered Heating Energy Percentages

Next step is the emission analysis for the system. In Figure 6.9, Greenhouse gas emission values of the current system is summarized.

Base case electricity system (Baseline) Country - region	Fuel type	GHG emission factor (excl. T&D) tCO2/MWh	T&D losses %	GHG emission factor tCO2/MWh
Turkey	All types	0,445	15,0%	0,523
Electricity exported to grid	MWh	47	T&D losses	15,0%
GHG emission				
Base case	tCO2	439,8		
Proposed case	tCO2	394,4		
Gross annual GHG emission reduction	tCO2	45,4		
GHG credits transaction fee	%	0,0%		
Net annual GHG emission reduction	tCO2	45,4	is equivalent to	45,4
GHG reduction income				
GHG reduction credit rate	$/tCO2	0,00		

Figure 6.15 CHP Greenhouse Gas Emission Values

The corresponding PBP for partial load operation for Case 2 is about 3.8 years due to the fact that the investment cost for HEGEL unit which consists of topping and bottoming cycles is higher than investment costs for conventional boiler and only the topping cycle. Below is the cumulative cash flows and pay-back graph (Figure 6.16)

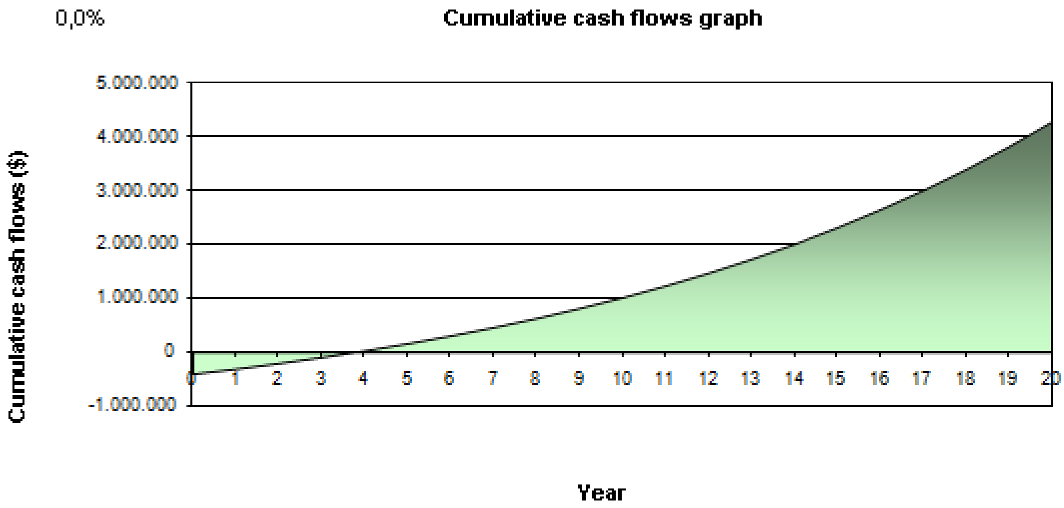


Figure 6.16 Cumulative Cash Flows and Pay-Back Graph

6.3. Discussion and Comparison of the Cases

When Figures 6.5 and 6.11 are examined, it can be seen that, for both cases heat load following is the most efficient and full capacity usage is the least efficient scheme. But since heat load is followed, during summer months electricity is supplied from main grid, which decreases the feasibility of the poly-generation systems for both cases. If both load following scheme could be chosen, and during summer months electrical load would be followed and excess heat could be used for cooling (for instance an absorption cooling module, which scheme is discussed later), both cases would have been more feasible and economical. Regarding these information, since investment cost for Case-2 is higher, payback period becomes longer.

For both systems, heat recovery efficiencies are almost same (49%) but for Case-2, unit fuel consumption is higher, as it could be predicted.

For Case-2, more electricity is produced, so when sold, it would mean electricity export income would be higher for this case.

Greenhouse gas emission values are same, since same amount of fuel is wasted in both cases. For both cases, emission values could have been reduced, by again using load following schemes. Below, in Table 6.1, an illustrative comparison is given for different load following schemes for Case-1 [41].

Table 6.1 Comparison for Different Load Following Schemes for Case-1

Energy modeling	**Case-1 (Ton CO_2/ year)**	**Change in emission value (ΔTon CO_2/year)**
Full throttle	759,8	+104,7
Electrical load following	418,1	+2,9
Heat load following	394,5	-44,0

6.4 Simple PBP Analysis and Comparison with RETScreen Results

Money saving (MS) of a poly-generation system is evaluated as the difference between the fuel cost in case of using a conventional boiler for heating and buying electricity from interconnected grid (FC) and the cost of operating a poly-generation unit (FC_{CHP}) [42].

$$FC = h \cdot \left[\frac{a}{0.85} \cdot Q_H + b \cdot C \cdot Q_H\right] \tag{6.1}$$

$$FC_{CHP} = h \cdot \left[\frac{a}{CHPH\eta} \cdot Q_H\right] \tag{6.2}$$

When FC_{CHP} is subtracted from FC and multiplied by availability factor (AF) and Base Load Factor (BLF), MS is evaluated as represented in equation 6.3.

$$MS = h\left\{\left[\frac{a}{0.85}\cdot Q_H + b\cdot C\cdot Q_H\right] - \left[\frac{a}{CHPH\eta}\cdot Q_H\right]\right\}\cdot AF\cdot BLF \qquad (6.3)$$

Where h is hours of operation, Q_H is heat power of the system (kW), C is power to heat ratio, $CHPH_\eta$ is heat efficiency of the system, a is unit fuel cost (TL/ kWh) and b is unit electricity cost (TL/ kWh).

With the above equation, MS can be evaluated in terms of h. After that, a simple PBP can be found by balancing MS with an initial cost of the poly-generation system (IC), as can be seen in equation 6.4.

$$h = \frac{IC}{MS\cdot(1-x)} \qquad (6.4)$$

where x is maintenance cost in terms of hours.

As already mentioned, a is 0.06 TL/ kWh and b is 0.22 TL/ kWh. For both systems, BLF can be taken as 0.9 and x as 0.05 [42].

For Case-1, Q_H is 174 kW, IC is 546,000 TL, $CHPH_\eta$ is 46%, C can be taken as 0.75 and AF is 0.95. If the system is considered to be operating 8320 hours in one year, PBP is calculated as 4.3 years.

For Case-2, Q_H is 183 kW, IC is 696,000 TL, $CHPH_\eta$ is 49.6%, C can be taken as 0.81 and AF is 0.85. If the system is considered to be operating 8320 hours in one year, PBP is calculated as 5 years.

In this simple PBP analysis, results show correspondence with the ones evaluated by the RETScreen program. Only for Case-1, PBP is calculated higher than the previous value. The differences are because of the fact that, electricity selling and buying prices are different in practice, but the simple analysis does not take them into account. Also, for RETScreen program, electrical consumption values are evaluated monthly throughout the year, but for the simple analysis, an average value of fuel consumption regarding the electrical consumption is assumed, which results less accurate findings.

CHAPTER 7

TESTING STANDARDS AND RATING PARAMETERS

7.1. Introduction

The lack of reliable information and transparency may be considered as a barrier to the further development of the cogeneration technology. As mentioned before, to remove the ambiguity resulting from a lack of standardized procedures, a set of widely accepted determination rules is needed which will create greater certainty that the basic concept of poly-generation is understood and determined in the same way.

In order to achieve the aim of this study, a standard procedure is developed with a set of widely accepted, exergy based rating and testing parameters for poly-generation design and evaluation. Important parameters for demonstrating the electrical performance, CHP thermal performance including thermal output, heat efficiency, coefficient of performance and energy utilization factor, atmospheric emissions performance, acoustic emissions performance, and other variables like heat input, gross electrical power, auxiliaries electrical power, net electrical power and efficiency, heat flows, primary energy savings (PES) and primary energy exergy savings (PES_R) are used to determine energy and exergy rating, or so called; technical performance of the cogeneration system. [43] Below, rating analysis for Case-1 and Case-2 can be found in Tables 7.1 and 7.2 and technical points scale and the results are in Figures 7.1 and 7.2 respectively.

Table 7.1 Rating Parameters and Technical Grading of Co/ Poly Generation Systems – CASE-1

	Parameter	Weight	Referans Value	Formulae	Explanations	INPUTS		OUTPUT	
1	Power to heat ratio at rated power	5	Cref=1	$C = \frac{P_e}{P_h}$		Pe Ph	126,7 173,89	C=	0,73
2	Primary Energy Savings	5	PESref=0,3	$PES = \left[1 - \frac{1}{\frac{CHPH\eta}{RefH\eta} + \frac{CHPE\eta}{RefE\eta}}\right] \times 100$		CHPHη CHPEη Ref H_η RefE_η	46,1 33,6 82 49	PES=	19,91
3	REMM efficiency	5		$\Psi_R = \frac{\varepsilon_{Hmin} + C \times (1 - \varepsilon_{Emin})}{\varepsilon_{Hmax} + C \times \varepsilon_{Emax}}$		ε_{Hmax} ε_{Emax} ε_{Hmin} ε_{Emin}	0,211 0,856 0,0171 0,7171	ψ_R=	0,27
4	1st law efficiency	5	ηref=0,9	$\eta = \frac{P_e + P_h}{G}$		G	372,5	η=	80,9
5	Primary Energy Exergy Savings	5	PESRref=0,27	$PES_R = \left[1 - \frac{1}{\left(\frac{CHPH\eta}{RefH\eta} + \frac{CHPE\eta}{RefE\eta}\right) \times \frac{(2 - Ref\psi_R)}{(2 - \psi_R)}}\right] \times 100$	Efficiencies are average of ¼, ½, ¾, 1 loads)	Ref$_{\psi R}$	0,2143	PES_R=	23,31
6	Multiple-fuel capability and Conversion factor	2	ηref=2	F = 0.5 + (number of total available fuel types/2)		Fuel type number	1	F=	1
7	Maximum Supply Temperature factor	5	Tref=363,15	$TF = \frac{\left(1 - \frac{T_{ref}}{T_{sup}}\right)}{0.22}$		T_{SUP}	2000	TF=	3,72
8	CO2 emissions degradation factor	3	Fuel:NG ηbiler=0,85 η elect_prod=0,33 ηIIDG_combi=0,04	$\Delta CO_2 = \frac{\left(\left(\frac{(P_e + P_h)}{\eta} \times 0.2\right)\left(\frac{1}{\psi_R}\right) - \left(\frac{P_e \times 0.6}{0.33} + \frac{P_h \times 0.2}{0.85}\right)(2 - 0.04)\right)}{(P_e + P_h) \times (-2.5)}$	Carbon emm: NG- 0.2 kg CO_2/kW-h, coal fired- 0.6 kg CO_2/kW-h			ΔCO2=	0,74

Table 7.1 Rating Parameters and Technical Grading of Co/ Poly Generation Systems – CASE-1 (cont.)

9	Quality of heat and power (Q), Reliability (R)	3	Q,R	$QR = \frac{(Q+R)}{(0.99+0.95)}$		Q	0,95	QR=	0,95
						R	0,9		
10	On board control, software and hardware factor	5	X: 0 none, X: 5 all	S = (1+ X/5)	full automatic load following, water temperature modulation, outer air compansation, frequancy control, cos φ	X	2	S=	1,40
11	Easy Control and operations factor	5	Ref 3 shifts	OC = 1/[0.5 + (I/2)]	I=avarage man power per shift	I	0,5	OC=	1,33
12	Accessories included in the Initial cost factor	5		A = 0.5 + (Aksesuar sayısı/7)	UPS, accumulator, monitoring, presurizing, balancer,expansion tank, exhaust catalitic converter, remote control	Accessories number	4	A=	1,07
13	Heat following factor	5	Z : 2	HF = 0.5 + (Z/2)	Only heat or electrical load following; Z = 1	Z	1	HF=	1,00
14	Technological convenience factor	4		T	engine; T = 1, Gas turbine T = 0.75		1	T=	1,00
15	Alternatifve energy ratio	1		$AEC = \frac{\Psi_R}{1-\left(\frac{Q_A}{Q}\right)}$	Q=total heat demand , QA=alternatif energy	QA	0	AEC=	0,27
16	Primary Energy Ratio	4	PER	PER=Electrical production transmission efficiency*COP		COP=	1	PER=	0,85
						Trans. Eff	0,85		
17	Unit CO2 emissions factor	3	100 kg CO_2/h	$CO_2 = \left(\frac{\left(\frac{(P_e+P_h)}{\eta}\times 0.2\right)\left(\frac{1}{\psi_R}\right)}{100}\right)$				CO2=	2,76
18	System weight and size factor	3	ρREF= 160 kg/m3	$S = \frac{\rho / (P_e + P_h)}{0.6} \quad \rho = \frac{T}{L \times H \times W}$	T = total weight, kg, L =Length, m H = Height, m, W = width m	T	2700	S=	0,74
						L	4		
						H	2,8		
						W	1,8		
19	Noice level factor	5	40dB 7 m away from the motor	NL= noise level 7 m away from the motor/40	noise level 7 m away from the motor		65	NL=	1,625
20	Engine life expectancy	5	50,000 h	EL= Catalog Engine Life Value/ 50,000	Engine life		65000	EL=	1,3

Table 7.2 Rating parameters and technical grading of Co/ Poly Generation Systems – CASE-2

	Parameter	Weight	Referans Value	Formulae	Explanations	INPUTS		OUTPUT	
1	Power to heat ratio at rated power	5	Cref=1	$C=\frac{P_e}{P_h}$		Pe	148,5	C=	0,81
						Ph	183,05		
2	Primary Energy Savings	5	PESref=30	$PES=\left[1-\frac{1}{\frac{CHPH\eta}{RefH\eta}+\frac{CHPE\eta}{RefE\eta}}\right]x100$		CHPHη	48,6	PES=	25,31
						CHPEη	38,4		
						Ref H_η	90		
						RefE_η	49		
3	REMM efficiency	5		$\Psi_R=\frac{\varepsilon_{Hmin}+C\times(1-\varepsilon_{Emin})}{\varepsilon_{Hmax}+C\times\varepsilon_{Emax}}$	$\varepsilon_{H,max}=\left(1-\frac{T_0}{T_{app}}\right)$ $\varepsilon_{E,max}=\left(1-\frac{T_0}{T_{fuel}}\right)$ $\varepsilon_{H,min}=\left(1-\frac{T_0}{T_a}\right)$ $\varepsilon_{E,min}=\left(1-\frac{T_0}{T_{El}}\right)$	ε_{Hmax}	0,211	ψ_R=	0,71
						ε_{Emax}	0,856		
						ε_{Hmin}	0,0171		
						ε_{Emin}	0,2279		
4	1st law efficiency	5	ref=?	$\eta=\frac{P_e+P_h}{G}$		G	372,5	η=	93
5	Primary Energy Exergy Savings	5	PESRref=27 ΨRref: 0,6	$PES_R=\left[1-\frac{1}{\left(\frac{CHPH\eta}{RefH\eta}+\frac{CHPE\eta}{RefE\eta}\right)\times\frac{(2-Ref\psi_R)}{(2-\psi_R)}}\right]\times 100$	Efficiencies are average of ¼, ½, ¾, 1 loads)	$Ref_{\psi R}$=	0,2024	PES_R=	48,1
6	Multiple-fuel capability and Conversion factor	2	ref=2	F̄ = 0.5 + (number of total available fuel types/2)		Fuel type number	1	F̄=	1
7	Maximum Supply Temperature factor	5	Tref=363,15	$TF=\frac{\left(1-\frac{T_{ref}}{T_{sup}}\right)}{0.22}$		T_{SUP}	2000	TF=	3,72
8	CO2 emissions degradation factor	3	Fuel:NG ηbiler=0,85 η elect_prod=0,33 ηIIDG_combi=0,04	$\Delta CO_2=\frac{\left(\left(\frac{(P_e+P_h)}{\eta}\times0.2\right)\left(\frac{1}{\psi_R}\right)-\left(\frac{P_e\times0.6}{0.33}+\frac{P_h\times0.2}{0.85}\right)(2-0.04)\right)}{(P_e+P_h)\times(-2.5)}$	Carbon emm: NG- 0.2 kg CO_2/kW-h, coal fired- 0.6 kg CO_2/kW-h			ΔCO2=	0,69

Table 7.2 Rating parameters and technical grading of Co/ Poly Generation Systems – CASE-2 (cont.)

9	Quality of heat and power (Q), Reliability (R)	3	Q,R	$QR = \frac{(Q+R)}{(0.99+0.95)}$		Q R	0,95 0,9	QR=	0,95
10	On board control, software and hardware factor	5	X: 0 none, X: 5 all	S = (1+ X/5)	full automatic load following, water temperature modulation, outer air compansation, frequancy control, cos φ	X	2	S=	1,4
11	Easy Control and operations factor	5	Ref 3 shifts	OC = 1/[0.5 + (I/2)]	I=avarage man power per shift	**I**	**0,5**	**OC=**	1,33
12	Accessories included in the Initial cost factor	5		A = 0.5 + (Aksesuar sayısı/7)	UPS, accumulator, monitoring, presurizing, balancer,expansion tank, exhaust catalitic converter, remote control	**Accessories number**	**4**	**A=**	1,07
13	Heat following factor	5	Z : 2	HF = 0.5 + (Z/2)	Only heat or electrical load following; Z = 1	**Z**	**1**	**HF=**	1
14	Technological convenience factor	4		T	engine ; T = 1, Gas turbine T = 0.75		1	**T=**	1
15	Alternatifve energy ratio	1		$AEC = \frac{\Psi_R}{1-\left(\frac{Q_A}{Q}\right)}$	Q=total heat demand , QA =alternatif energy	**QA**	**2**	**AEC=**	2,13
16	Primary Energy Ratio	4	PER	PER=Electrical production transmission efficiency*COP		**COP=** **Trans. Eff**	**1** **0,85**	**PER=**	0,85
17	Unit CO2 emissions factor	3	100 kg CO_2/h	$CO_2 = \frac{\left(\frac{(P_e+P_h)}{\eta}\times 0.2\right)\left(\frac{1}{\psi_R}\right)}{100}$				**CO2=**	1,05
18	System weight and size factor	3	ρREF= 160 kg/m3	$S = \frac{\rho / (P_e + P_h)}{0.6}$ $\rho = \frac{T}{L \times H \times W}$	T = total weight, kg, L =Length, m H = Height, m, W = width m	**T** **L** **H** **W**	**2700** **4** **2,8** **1,8**	**S=**	0,67
19	Noice level factor	5	40dB 7 m away from the motor	NL= noise level 7 m away from the motor/40	noise level 7 m away from the motor		**65**	**NL=**	1,625
20	Engine life expectancy	5	50,000 h	EL= Catalog Engine Life Value/ 50,000	Engine life		**65000**	**EL=**	1,3

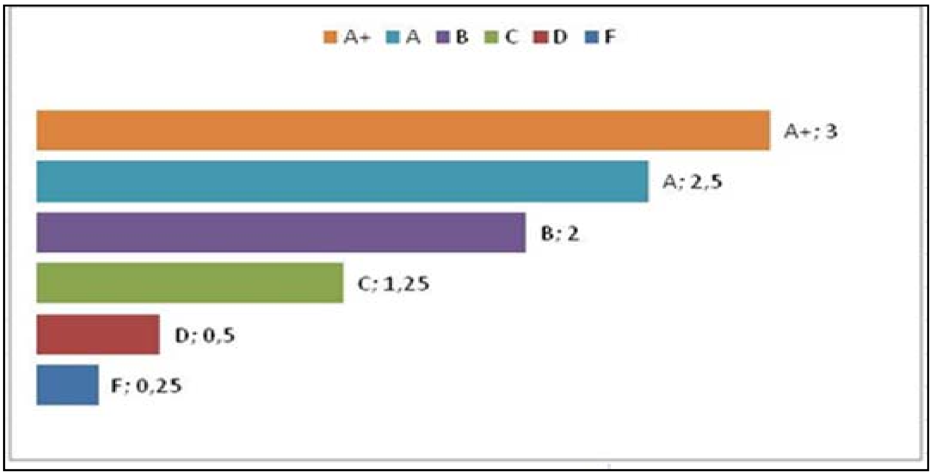


Figure 7.1 Technical Points Scale [43]

TECHNICAL POINTS- CASE-1	**1,55**
TECHNICAL POINTS- CASE-2	**3,06**

Figure 7.2 Technical Points for Case-1 and 2

7.2. Discussion and Comparison of the Cases

Technical points are highly affected by 1st and 2nd law efficiencies, Power to heat ratio, *PES* and PES_R values and *REMM* efficiency. Also, maximum supply temperature factor, initial cost factor, load following and engine life expectancy do effect the technical points of the evaluated systems. The other factors are of secondary importance.

When the rating parameter analysis is applied, Case-1 gets C and Case-2 gets A+ technical points from the Technical Points Scale in Figure 7.1. This is mainly because the exhaust gases wasted in Case-1 are used in the HRSG in Case-2 for further steam production to be used in a SE Rankine cycle. Because of that, *PES*, PES_R and *REMM* efficiency values are higher in Case-2, as already described in the previous chapters, which considerably increases the average technical points.

CHAPTER 8

RESULTS AND CONCLUSIONS

8.1. Internal Combustion Engine Characteristics and Operation

First of all, it should be noted that, the choice of ICE over conventional gas turbine system is due to some technical and thermodynamical properties of the ICE co-generator, such as; high electrical power generation efficiency, high first-law and second-law (exergy) efficiencies, uniform constant part load efficiency; low pollutant emissions, high flexibility and easy integration with the energy grid. Also, ICE systems have ease of maintenance and operation when compared to gas turbine system which has a comparable high efficiency loss during summer months due to high inlet air temperatures.

All ICE working parameters and output parameters have been studied and detailed graphical representations are given in Appendix K. As can be seen in Figure K.1 ICE engine has a linear power to speed correspondence. ICE has almost a constant part load efficiency, which is uniform after engine break power of 50 kW, corresponding to engine speed between 1000 and 2150 rpm, which are the lowest and the highest speeds for the co-generator as seen in Figure K.2. It is clearly seen that efficiency does not fall in part load, or even at very low speed working conditions, which enhances cogeneration efficiency and decreases transmission, transformation, heat and electrical power losses, which is very advantageous for a CHP systems.

The behavior of exhaust gas flow rate from the ICE shows an increasing manner with increasing engine speed, which is almost linear and can be seen in Figure K.3

in Appendix K. Also the exhaust gas temperature shows an increasing behavior, only which has a characteristic curve, not linear and demonstrated in Figure K.4.

Jacket water flow rate characteristics can be found in Figure K.5 showing an increasing behavior with engine break power.

The last graphical representation in Figure K.6 shows the linear correspondence of engine break power and engine electrical power output. From the graph, it is seen that electrical power production of the ICE is almost 80 % of the engine break power.

As already mentioned, performance characteristics obtained in this study reveal that, after a certain point which is 1300 rpm, engine torque, which increases up to that point remains constant and around that engine speed, the thermal efficiency drops. Also, till that point, first and second law efficiencies drop. After about 1350 rpm, efficiencies increase and power to heat ratio decreases, with increasing engine speed. This obviously indicates a change on the trends of the graphics.

When the fuel mass flow rate versus engine speed graph is examined (Figure 8.1), after 1300 rpm the slope of the linear line of the graph is discovered to be decreasing. This shows that, after that point, the engine operates in a way that enhances air admission into the cylinders of the engine. This may be accomplished by a compressor or a turbo charger or an intercooler. The decrease in the slope of the fuel mass flow rate is an indication that after this threshold, the engine works more efficiently, more economically and releases less emissions (w.r.t. power output). It also proves that, engine working below 25% of its rated speed is not efficient and economical, and it should be avoided.

As explained before in Chapter 5, for investigating the effects of change of altitude on performance, a table for comparing the results of the plant performance in Torino (215 m corresponding to 99 kPa) and results of the plant performance if it was in Ankara (840 m corresponding to 91 kPa) is developed. Table L.1 in Appendix L is formed by considering average air and soil temperatures to be same in both cities. The last column of the table shows the maximum differences of percent values taking the performance values of Torino as the base. Regarding this, a maximum of 4.5 % difference is observed on the results.

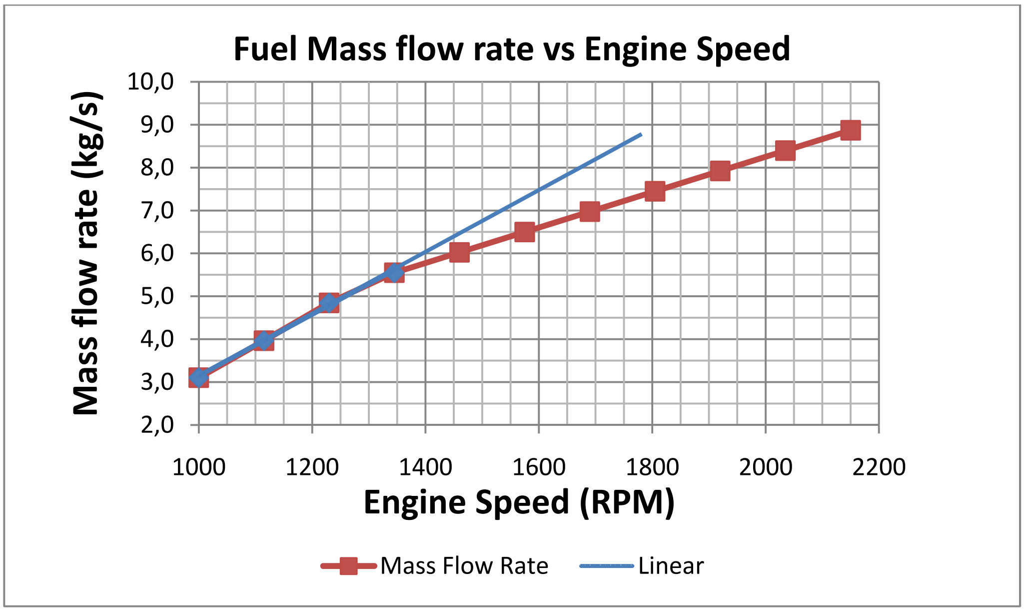


Figure 8.1 Fuel Mass Flow Rate vs Engine Speed

8.2. Steam Engine Characteristics and Operation

The choice of steam engine over conventional organic cycle (steam turbine) is due to its high electrical power generation efficiency, high flexibility and reliability and low flame temperatures in the system, causing very little nitrogen oxide, thus protecting the environment.

Steam engine working parameters and characteristics are studied and results are given in Appendix H as already mentioned in previous chapters.

The increasing manner of steam flow rate vs ICE break power is represented in Figure H.1. In Figure H.2, temperature of outlet exhaust gases and HRSG efficiency as function of ICE power can be found. Exhaust gases temperature decreases and HRSG efficiency increases with ICE power, which again proves that, at full load, engine is more efficient.

In Figure H.3 Steam Engine isentropic efficiency as a function of ICE Power is represented. Throughout the operation range, efficiency seems to be constant, and as high as 90%. Figure H.4 shows SE power as a function of ICE power, which is almost linearly increasing.

8.3. Poly-generation System Characteristics and Results

For poly generation systems, it is important to rate the performance and energy savings potential, estimate the system efficiency, power to heat ratio, emissions performance and temperature compatibility. This study not only evaluates such parameters and compares them for different systems, but also shows different procedures for rating such systems, and compares the procedures.

Two ICE poly-generation models with different system characteristics are studied as cases, mentioned before. As can be seen from the output files (Figures 5.4 and 5.5 in Chapter 5), a great variety of properties besides those of ICE's are calculated and presented for the user, as outputs of the program in either MATLAB graphics file or in MS Excel spreadsheets. These include natural gas and user water properties for both cases, SE, Condenser and HRSG properties added for Case-2 and most importantly, *PES*, PES_R, first law and second law efficiency calculations for again both cases.

When the results are examined, it is clear that both systems may provide high performance energy generation. When the *PES* values are considered, it can be said that, even without the bottoming cycle of Case 2, primary energy savings are very high considering the limiting lowest value of 10 % in the EU Directive, which indicates the efficiency, reliability and feasibility of the cogeneration systems. Even at part loads in Case-1, PES values are minimum 12.4% and in Case-2, 17.7 %.

In Figure 8.2 Plant Efficiencies vs Engine Speed can be found. When Figure 8.3 is examined, *PES* and PES_R Values vs Engine Speed for two cases can be seen.

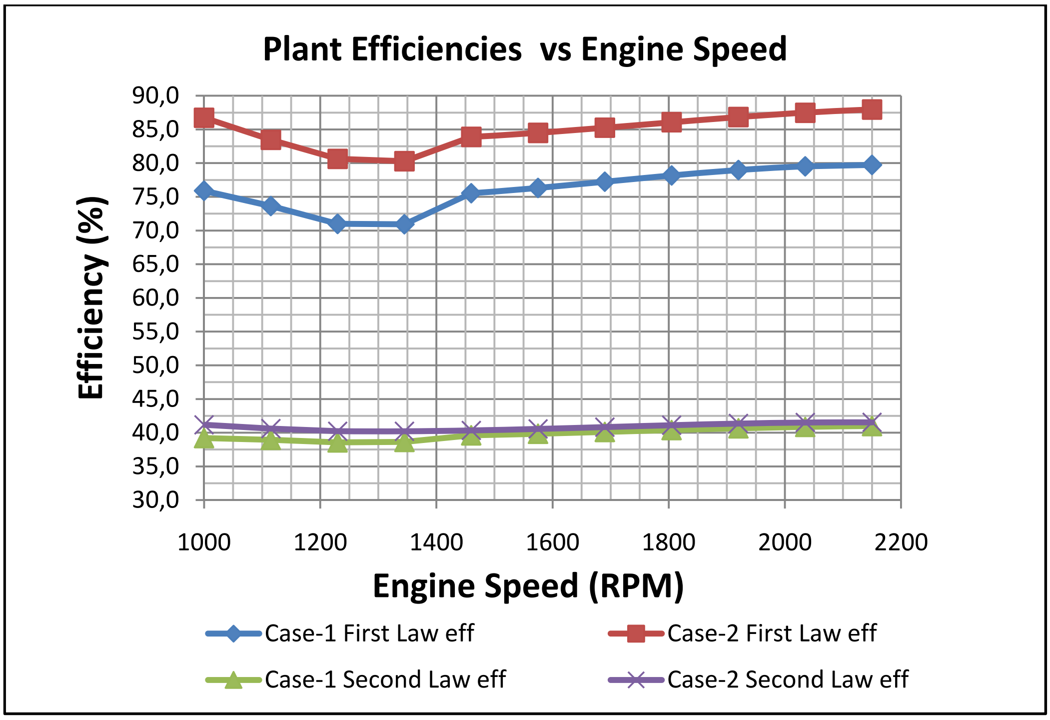


Figure 8.2 Plant Efficiencies vs Engine Speed

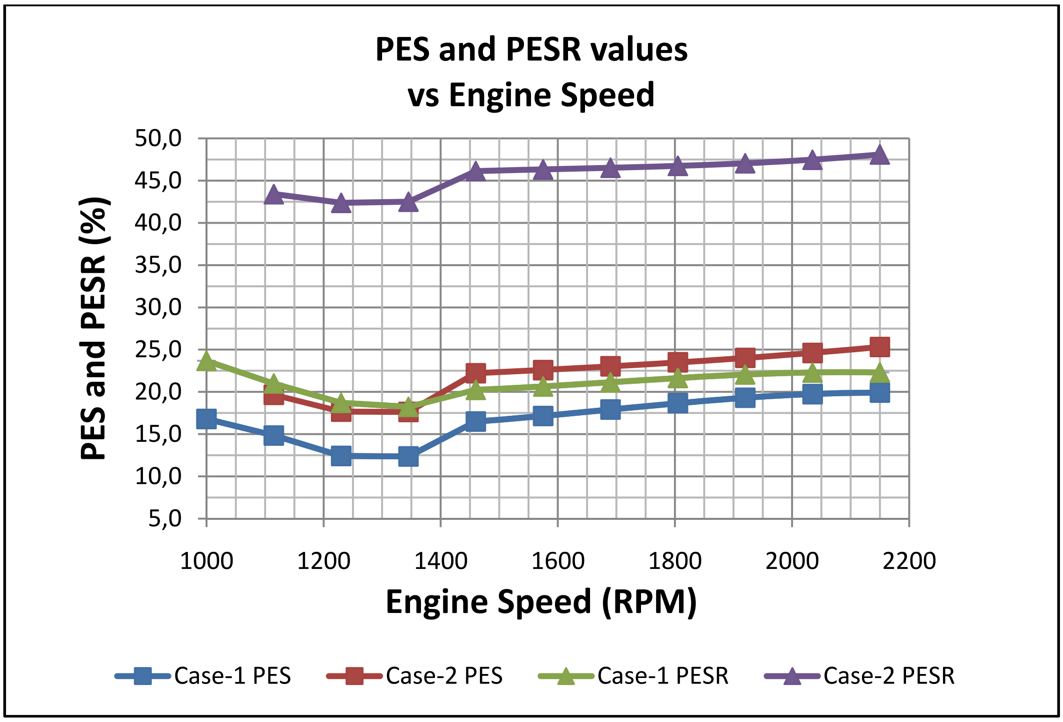


Figure 8.3 *PES* and *PES*$_R$ Values vs Engine Speed

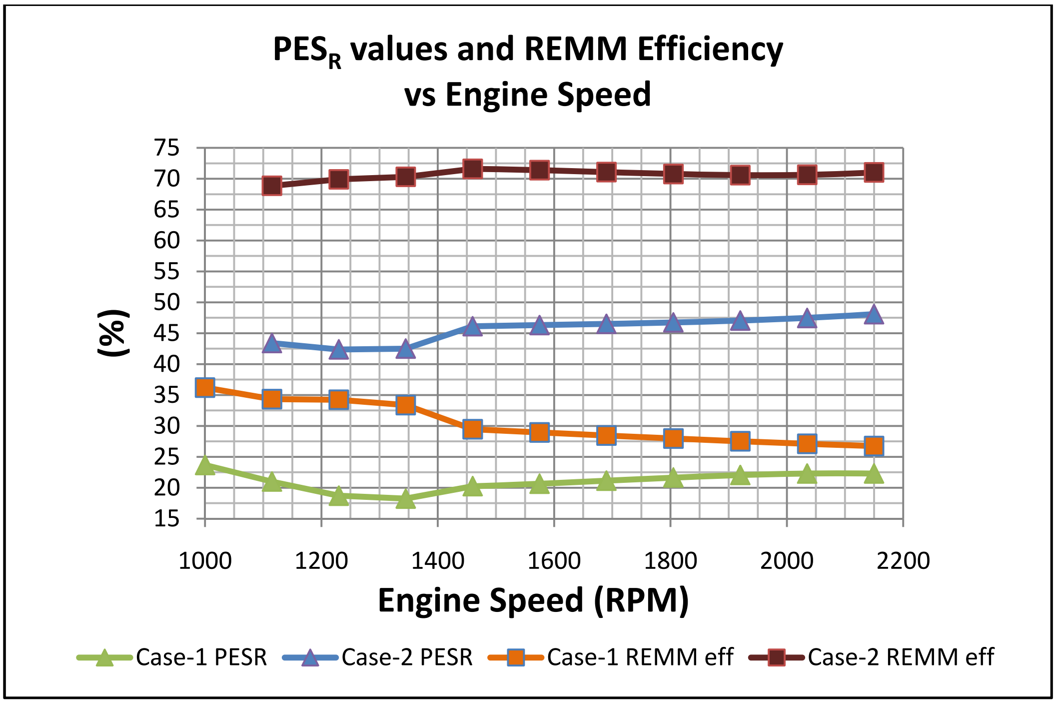


Figure 8.4 PES_R and $REMM$ efficiency values vs Engine Speed

In Figure 8.4, *REMM* efficiency (ψ_R) values and PES_R values for both cases are compared with each other. As already mentioned, for Case-1 and Case-2, second law efficiency trends are similar, but *REMM* efficiency does not show same trend for Case-1. This is because; *REMM* analysis is sensitive to the process temperatures and in Case-1, a higher content of energy of the exhaust gases is thrown away, and w.r.t increasing engine speed, temperature of the gases increase thus *REMM* efficiency decreases. But in Case-2, temperature of the exhaust gases are lower, since they enter HRSG and HRSG efficiency increases w.r.t engine speed, so *REMM* efficiency increases as well. When a higher energy content is wasted, exergy destruction is higher, so this shows that, temperature values of the processes do effect the efficiency, environmental footprint and even payback period of the investments.

Graphical representation in Figure 8.5 shows the relation between power to heat ratio (C) of the systems for both cases. For Case-1, after the admission of the intercooler, C seems to be decreasing with increasing engine speed, but for Case-2, C is increasing throughout, and is higher for the rated engine speeds.

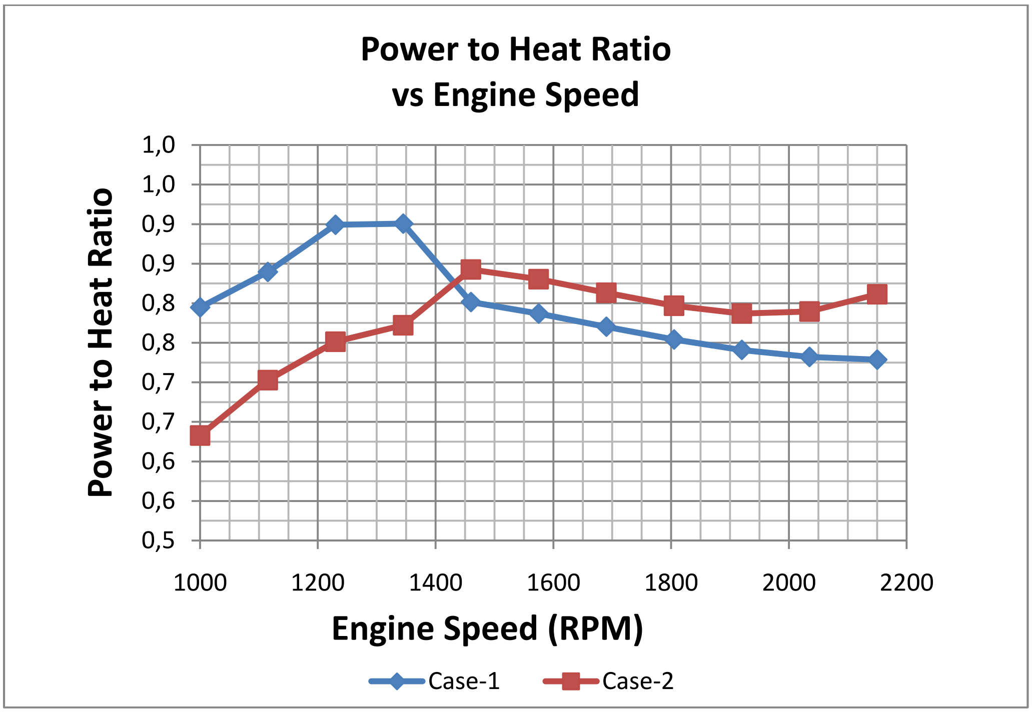


Figure 8.5 Power to Heat Ratio vs Engine Speed

In Figure 8.6, district heating temperature outputs of the cases are represented graphically. The output temperature increases linearly with increasing engine speed, and for Case-2, values are always higher.

In Figure 8.7, electrical and heat power vs engine speed for both cases can be found. Electrical power output of Case-2 is at any instant, higher than that of Case-1 as much as the value of SE electrical power output. Heat power output of Case-2 shows a linear trend, and after 1300 rpm, the difference between the cases heat power outputs decrease. This is probably because, ICE performance, thus heat rate is very much effected by the ICE torque while SE electrical production rate does not.

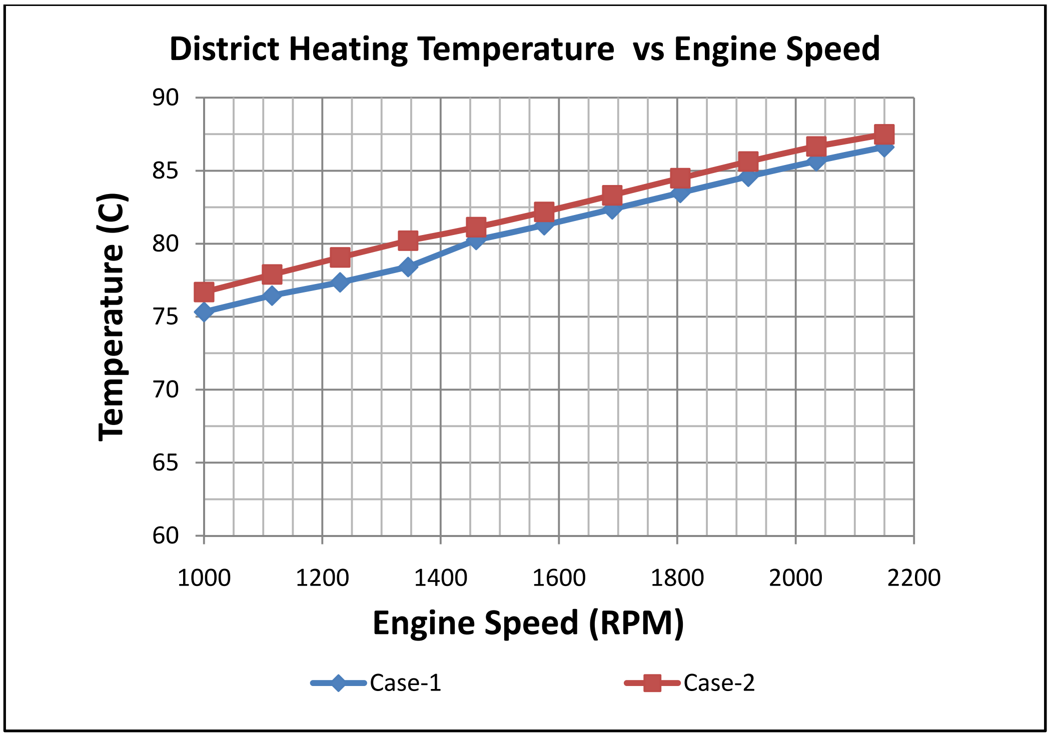


Figure 8.6 District Heating Temperature vs Engine Speed

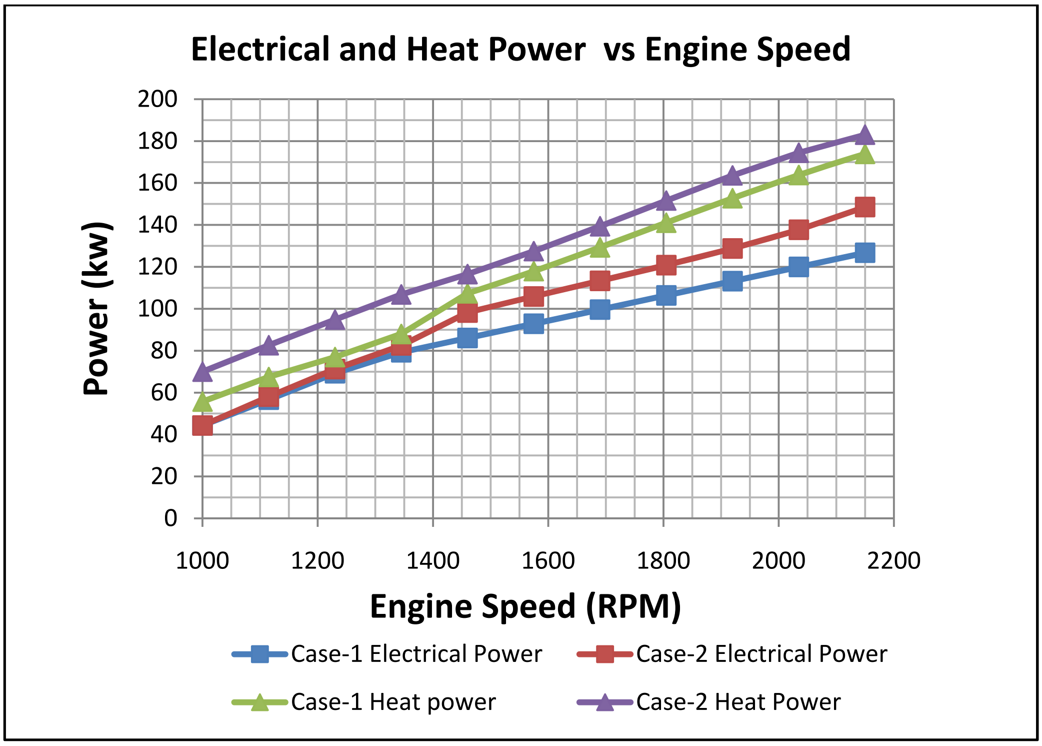


Figure 8.7 Electrical and Heat Power Outputs vs Engine Speed

Finally, below in Figure 8.8, harmful carbon emissions graphics are presented. As can be seen, Case-1 has comparably high emissions value, because of the excess temperature of the exhaust gases thrown to atmosphere, thus wasted exergy content of the fuel.

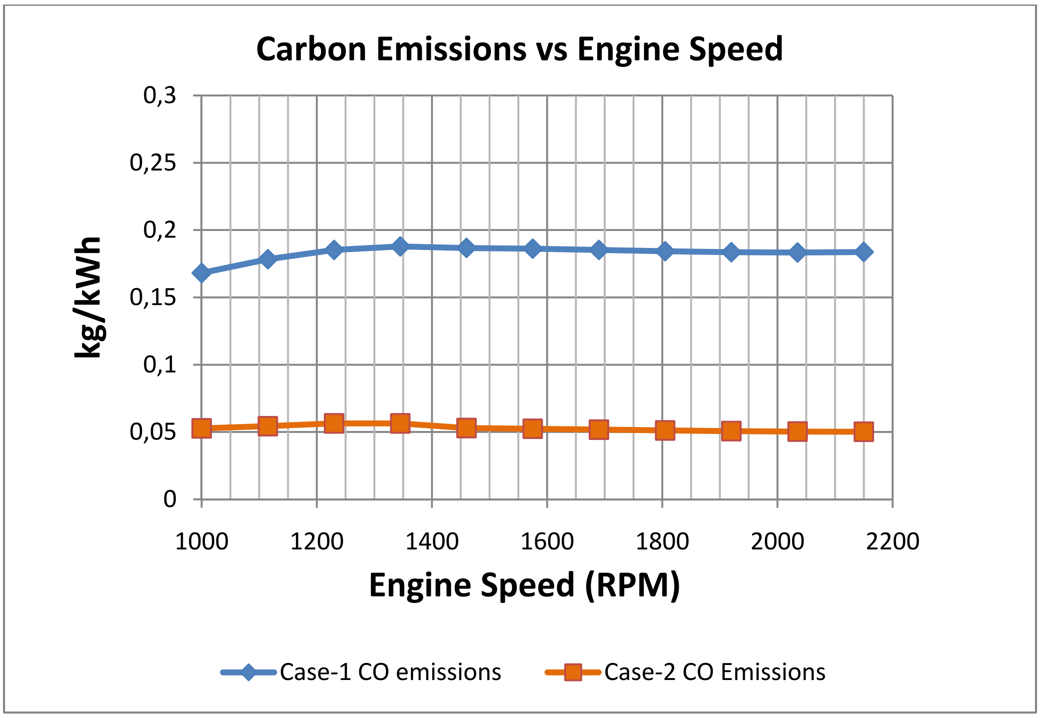


Figure 8.8 Carbon Emissions vs Engine Speed

8.4. Discussion and Recommendations

Even though the poly-generation systems examined in the scope of this thesis work may provide sustainable, high performance energy generation, they might be more effective, environmentally friendly and profitable if some additional equipment can be used with further installments. For instance, for a better implementation of the fuel resources and for increasing the exergy efficiencies, the plant can be integrated with a refrigeration unit which will work during summer months, when heating will not be necessary and/or with a ground source heat pump (GSHP) when electrical energy exceeds the demanded value, yet heating load is below.

In this last part of the thesis studies, theoretical analysis is done for both cases; a GSHP integrated poly-generation system and an absorption chiller added to the tri-generation unit. First of all, a GSHP integrated poly-generation system will be considered.

Ground source heat pumps utilize pipes buried underground to extract heat from the ground. This is usually used to heat radiators or under floor heating systems and hot water. Beneath the surface, the ground stays at a fairly constant temperature, so a ground source heat pump can be used throughout the year - even in the middle of winter. [44] The only energy used by Ground Source Heat Pump systems is electricity to power the pumps.

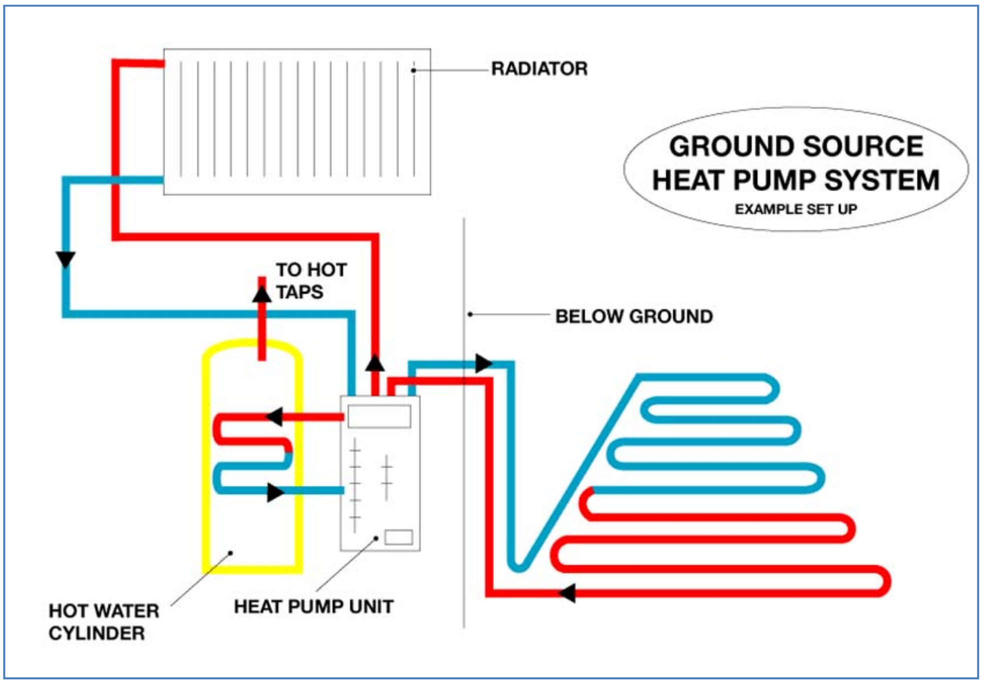


Figure 8.9 Schematical view of a Ground Source Heat Pump

Typically, a GSHP will deliver 3 or 4 times as much thermal energy (heat) as electrical energy used to drive the system. They make use of renewable energy stored in the ground, providing one of the most energy efficient ways of heating buildings. They are suitable for a wide variety of building types and are particularly appropriate for low environmental impact projects. There are reverse-cycle heat pumps that can deliver both heating and cooling [45]. Figure 8.9 shows a typical GSHP system.

Poly-generation system may be easily and practically combined with a GSHP such that additional heat can be provided to the consumer simply by allocating part of the electrical (or mechanical) power produced to drive the heat pump. Calculations show that, allocating part of the electric power output to a GSHP has special benefits as long as there is more heat demand than electricity in the consumers' area. In this case, PES_R may be maximized in terms of C by varying the amount of electric power output allocated to the GSHP. The poly-generator must provide enough electric power to drive the GSHP. While C increases, the size of GSHP increases, thus its power demand increases. [28]

Below in Figure 8.10, there is a schematical representation of a tri-generation system integrated with GSHP [28]. Here GSHP is driven with part of the CHP power output. Remaining electricity is provided to the consumers. The tri-generation system may be used for all seasons, because GSHP may also be used for summer indoor comfort cooling and air conditioning.

On the consumer site, part of the heat delivered from the CHP unit may be used for absorption cooling and/or liquid desiccant cooling. Absorption cooling technology and added benefits will be explained in the following paragraphs.

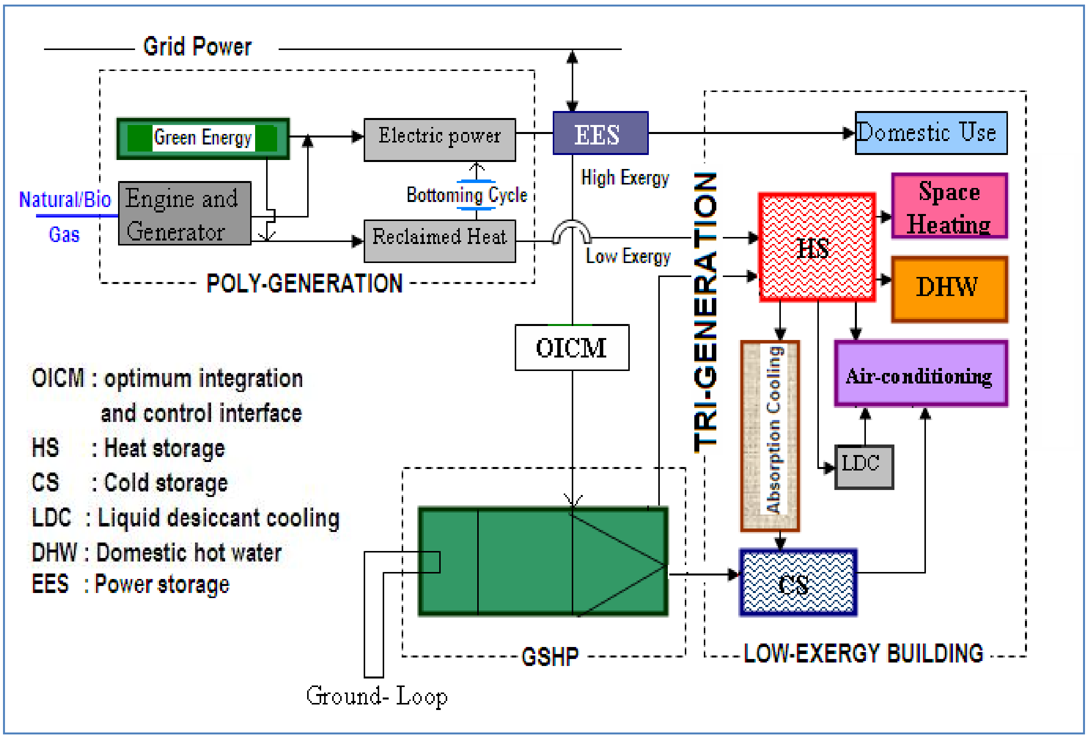


Figure 8.10 Poly-Generation and High Performance Building Coupling [28]

Below are some results of simulations done by building modules using MATLAB R2007b, Simulink, with GSHP added to Case-2, where there is an ICE topping and a SE Rankine bottoming cycle.

Comparison is done for different energy production systems, respectively; a boiler, a topping cycle, topping + bottoming cycles and topping+ bottoming cycles driving a ground source heat pump. With the analysis, the energy savings and exergy performance is rated, *C* (power to heat ratio), *PES*, PES_R and efficiencies values are calculated and the emissions are estimated and compared among the cases.

In Table 8.1, System parameters are represented for Case-1 and Case-2 which are already studied, for a conventional boiler and a theoretical case where GSHP is integrated to Case-2. When examined, thermal efficiencies of the systems are found to be quite similar, but exergy efficiency of the systems are varying. For instance, for a conventional boiler, it is considerably lower than the other systems. *PES* and

PES_R values increase as the system is upgraded and carbon emissions decrease. Figure 8.11 shows the graphical representation of the efficiency values for this comparison while Figure 8.12 shows the power to heat ratio and carbon emission values for all cases.

Table 8.1 Comparison of different cases [46]

	Boiler	Topping Cycle	Topping + Bottoming Cycle	Topping + Bottoming Cycle+GSHP
Efficiency (%)	85	79,7	87,9	87,9
Power to heat ratio(C)	-	0,73	0,81	0,81
Exergy efficiency (%)	4	41	42	46
PES (%)	0	20	25,3	30
PES_R (%)	6	22,3	48,1	53
Carbon emmisions (kg/kWh)	0,4	0,18	0,05	0,04

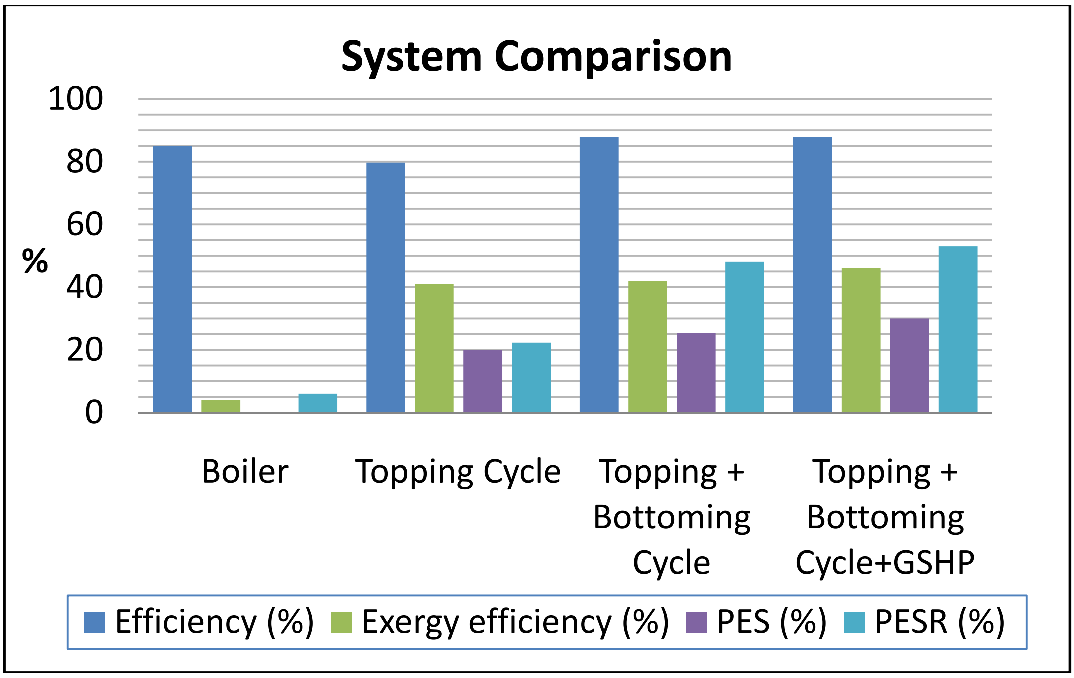


Figure 8.11 Graphical Representation for Efficiencies [46]

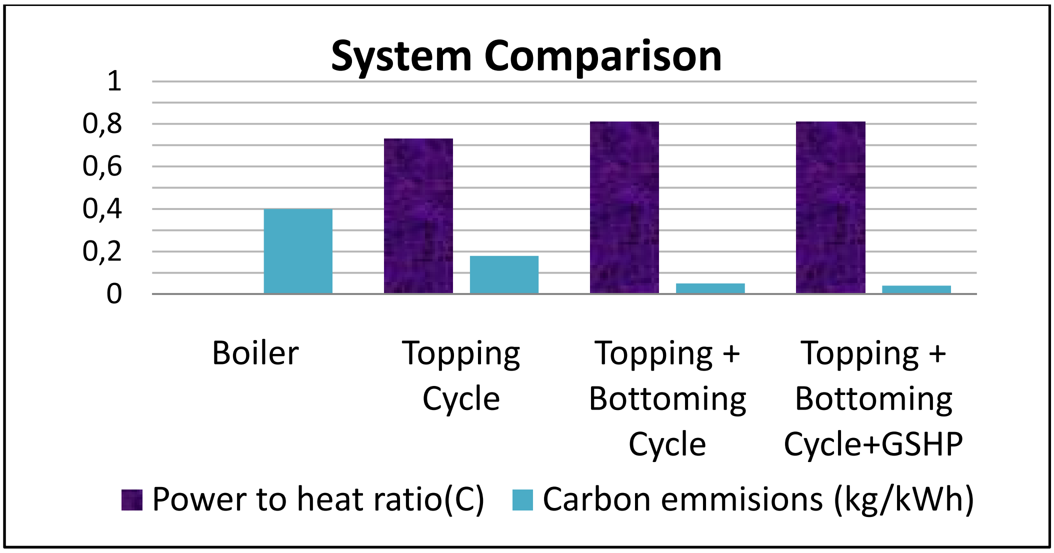


Figure 8.12 Graphical Representation for C and CO_2 Emissions [46]

If exergy efficiency is higher, thus PES_R values, it means that the system has less exergy destruction, thus natural resources are used in a more efficient manner and wasted energy is low. Also considering carbon footprint, it is obvious that, emissions decrease with increasing exergy efficiency. As easily realized, a GSHP integration is considerably efficient, economical, less harmful with a high energy and exergy performance.

Emission analysis for these systems, for whole working range of the ICE are represented in Figure 8.13. The graphic shows CO_2 emission values for different energy production systems. A conventional boiler has the maximum emission value, 0.4 kg/kWh, with $\psi_R = 0.06$ and thermal efficiency of 85 %. In Case-1, carbon emissions shows a significant decrease (0.18 kg/kWh) where ψ_R is 0.27 and thermal efficiency is 79.5 %. In Case-2, carbon emissions further decrease to about 0.05 kg/kWh where ψ_R is 0.7 and thermal efficiency is 87.9 %. If a GSHP is added with same thermal efficiency but increased *REMM* efficiency, emissions would fall below 0.05 kg/kWh.

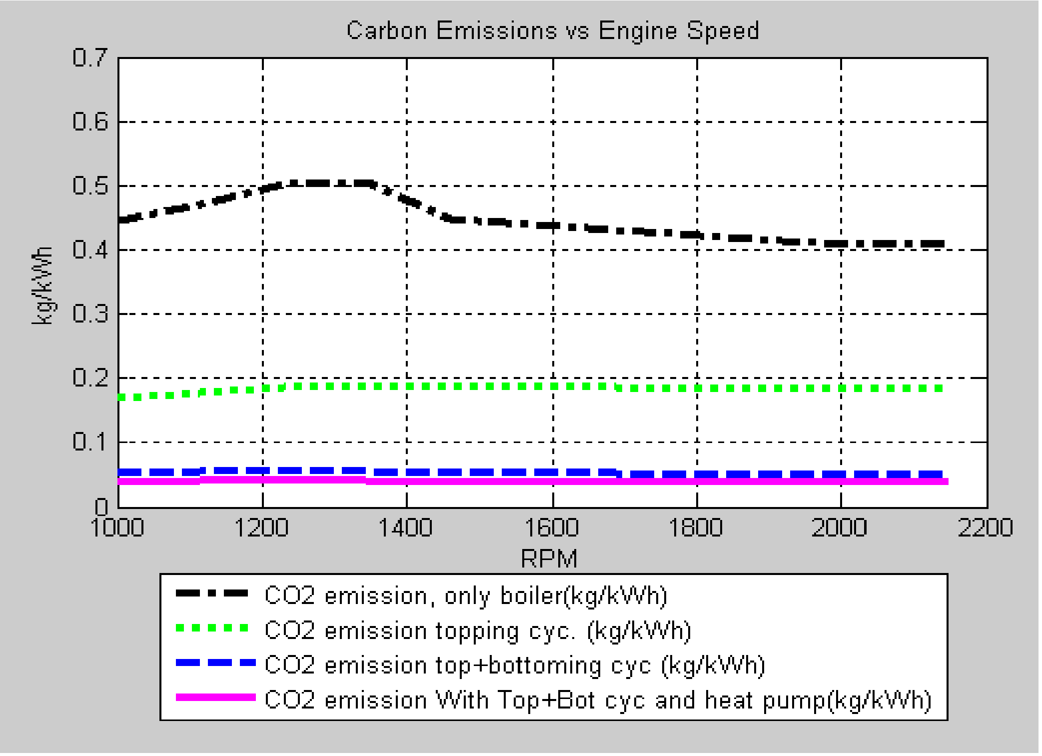


Figure 8.13 CO_2 Emission Values for Different Energy Production Systems [47]

For further increasing efficiency, exergy and emissions performance of the system, as a second theoretical case, an absorption chiller might be added as described above.

Absorption chiller technology is more complicated than a GSHP. They are driven by low temperature waste heat, and usually have low operational cost. But usually investment cost for these chillers are quite high. Below, absorption chillier technology is described in quite detail.

The available heat modelled by a heat rejection process from the prime mover drives the absorption chiller. The heated water or exhaust enters the absorption chiller generator, which acts as a heat exchanger and evaporates the working fluid out of the generator solution. With a known outlet temperature of the exhaust or water from the absorption chiller generator, the heat rate dissipated in the generator is the difference in the heat rate available from the exhaust or water heat rate still available in ambient conditions after leaving the generator. The heat rate dissipated in the generator is calculated using either of the following equations where $T_{exhaust;gen}$ is the temperature of the exhaust exiting the absorption chiller generator and $T_{exhaust;PM}$ is the temperature of the exhaust exiting the prime mover. $T_{w;in;gen}$ and $T_{w;out;gen}$ are the temperatures of the water entering and exiting the absorption chiller generator, respectively.

$$\dot{Q}_{\mathrm{gen}} = \dot{m}_{\mathrm{mix}} \cdot c_{p,\mathrm{cp}}(T_{\mathrm{exhaust,gen}} - T_{\mathrm{exhaust,PM}})$$
$$\dot{Q}_{\mathrm{gen}} = \dot{m}_{\mathrm{water}} \cdot c_{p\mathrm{water}} \cdot (T_{\mathrm{w,in,gen}} - T_{\mathrm{w,out,gen}}) \tag{8.1}$$

The COP of an absorption chiller is the ratio of thermal power (cooling) produced by the absorption chiller evaporator to the thermal power required to drive the generator. The COP is defined in the following equation when pump work is neglected [17] where ’Q_{evap} is the cooling power output of the absorption chiller evaporator.

$$\mathrm{COP} = \frac{\dot{Q}_{\mathrm{evap}}}{\dot{Q}_{\mathrm{gen}}} \tag{8.2}$$

In vapor -absorption systems, 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. [9]

Below, a vapor absorption refrigeration system, which uses water as the absorbent and ammonia as the refrigerant is examined (Figure 8.13) as to clearly illustrate the refrigeration cycle. [48]

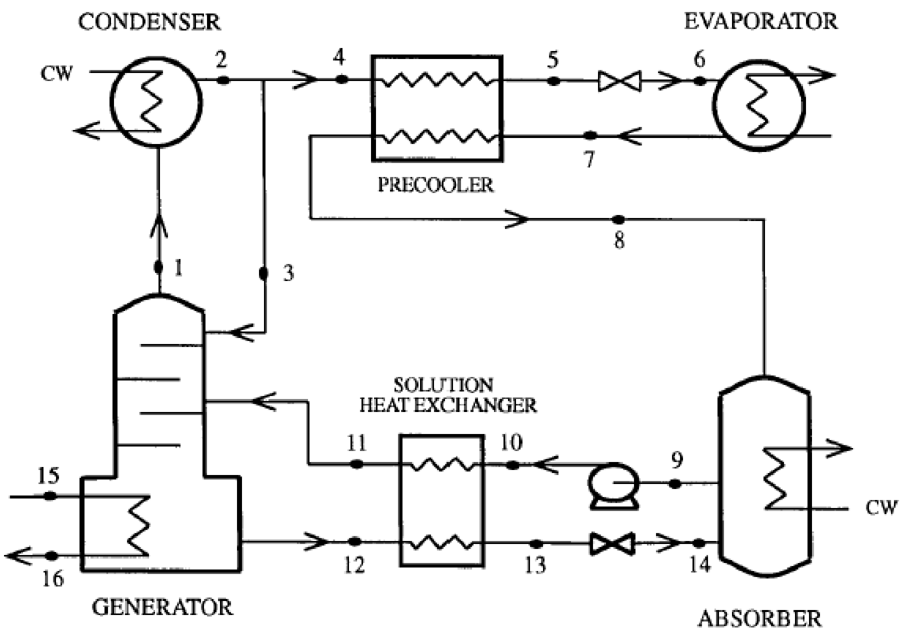


Figure 8.14 A single Stage Ammonia Absorption Chiller

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 (Q_a) 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 (Q_c) 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 (Q_{ev}). The steam generated in this process (7) flows to the subcooler and finally to the absorber to dilute the weak solution. [48]

When a refrigeration unit is added; from thermodynamic point of view, the combination of ICE with absorption chiller proves to be highly efficient, because the flue gas from heat recovery steam generator and/or hot water/steam produced may be used as a heat source for absorption refrigeration as described in this study [49].

When heat load following scheme is considered, which is the most efficient scheme for operating the poly-generation system as discussed before, during summer months where heating is not needed, hot water output of the poly-generation system can be used in this absorption chiller unit. Outlet temperature which is about 90 ºC is enough for single effect cooling for air conditioning or indoor comfort cooling. This way, almost all useful energy is extracted from the system and the system would be very efficient, will have cost effective operation and very low harmful emissions with exergy efficiency above 60%.

8.5. Conclusion

This study introduces a better definition for rating and evaluating poly-generation systems in terms of both energy and exergy. Besides energy, exergy is an important metric to reveal and understand the real advantages of poly-generation systems, optimize them to minimize their environmental footprint and harmful emissions, maximize their fuel savings, and thus to accomplish an optimum sustainability metric among the factors of environment, energy, human needs, and economics.

A sustainable poly-generation may mean an integration of several green components, which can be optimized using energy, exergy and Rational Exergy Management Model (REMM) analysis. For proposing a poly-generation system, alternative types of energy conversion and used components need to be compared and optimized, not only from economic point of view but also from the environmental point of view at large. Thus, the internal combustion engine in the HEGEL poly-generator used in all of the cases discussed in this study is speculated both from exergy and energy points of view, which cumulatively effect the emissions and profitability of the project.

ICE poly-generator used in HEGEL project has high electrical power generation efficiency, high first-law and second-law (exergy) efficiencies, uniform constant part load efficiency; which enhances thermal efficiency and decreases power losses; low pollutant emissions, high flexibility and is easily integrated with the energy grid. Also, ICE systems have ease of maintenance and operation when compared to other energy conversion systems. All these properties are very advantageous for poly-generation. It should also be noted that, ICE powered poly-generation applications have a low power to heat ratio, and thus it is more reasonable to use them when heat demand is high and power demand is lower.

The choice of steam engine over conventional steam turbine is due to its high electrical power generation efficiency, high flexibility and reliability and low flame temperatures in the system, causing very little nitrogen oxide, thus protecting the environment.

The current study reveals that when ICE poly-generation systems are coupled to exergy-conscious HVAC systems, exergy efficiency may increase up to 60 % and beyond. This is very important to achieve high performance buildings for a better and green future.

When all the results of the MATLAB Simulink analysis done in the scope of the thesis study are examined and compared to a conventional boiler utilization for heating a building and supplying electricity of the building from the energy grid supplied by government, it is clear that all poly-generation systems described in Case-1, Case-2 and other theoretical cases provide sustainable, high performance and green energy generation.

It can be seen that second law and first law efficiencies both increase a particular amount from Case-1 to the last case described in the previous part (Case-2 + GSHP+ absorption chiller) and trends are similar, but *REMM* efficiency does not show same trend. This is mainly because *REMM* analysis is sensitive to the process temperatures and *REMM* efficiency increases considerably w.r.t temperature changes in the system.

When exhaust gases are more efficiently used, for instance if the heat which is to be wasted at high temperatures can be recovered instead keeping exergy destruction lower, (as on site steam production, absorption), *REMM* efficiency increases a significant amount, indicating the effective utilization of natural resources and lessening harmful emissions. So, it also shows that, heat recovery is a major fuel saving method. These all prove that, *REMM* is a powerful tool since temperature values of the processes do affect the efficiency, environmental footprint and even payback period of the investments.

When the primary energy savings are considered, it can be said that for all cases *PES* are very high considering the lowest permissible value of 10 % in the EU directive for the qualification of co-generation. When *REMM* efficiency increases, PES_R values increase. For instance, for a conventional boiler, second law efficiency and *REMM* efficiency are considerably lower than the other systems although thermal efficiency is quite similar, and does not indicate the effectivity of the system. PES and PES_R values increase as the system is upgraded and carbon emissions decrease.

The difference between PES and PES_R values are because of the contribution of the process temperature based exergy components. Throughout the thesis study, it is proved that, PES_R is a better indicator of the performance for poly-generation systems. If PES_R values are higher, it means that the system has less exergy destruction, thus natural resources are used in a more efficient manner and wasted energy is low. Also considering carbon footprint, it is obvious that, emissions decrease with increasing exergy efficiency. As easily realized, a GSHP integration is considerably efficient, economical, less harmful with a high energy and exergy performance. Furthermore, when an absorption chillier is added to the system, almost all useful energy of the fuel input to the system can be utilized with correct installation and arrangement of the system components.

RETScreen energy modeling analysis is an important tool for calculating energy flow for buildings where poly-generation system is to be installed. Payback periods and the feasibility of the investment can be easily calculated by using RETScreen modules.

The results of the rating analysis which is a helpful and a powerful tool, provides cogeneration plant investors, designers and engineers some key information for specifying and comparing general performance of different poly-generation schemes from economic and environmental point of view. This is very important for proposing a poly-generation system which will be the best alternative investment for a particular situation.

As to conclude, it may be stated that the energy and exergy rating methods (including REMM) and performance analysis of ICE powered poly-generation systems can be effectively used as a guide study to analyze and rate the performance of different ICE poly-generation systems and power plants both thermodynamically and environmentally providing rational comparison to other cogeneration applications. The modules prepared using MATLAB Simulink are flexible, and with a little rearrangement, they can be adapted to other systems for thermodynamic evaluation.

The results of the present thesis study are also expected to give a modern and unique direction to scientists, academicians, engineers, and energy policy makers in application of energy planning studies in the light of energy and exergy based methodologies.

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APPENDIX A

HRSG PROPERTIES

A.1.Types and Configurations of HRSG According to evaporator layout

A.1.1.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 not advantageous for units having a large gas flow. A schematic of the D-frame evaporator layout can be seen in Figure A.1.

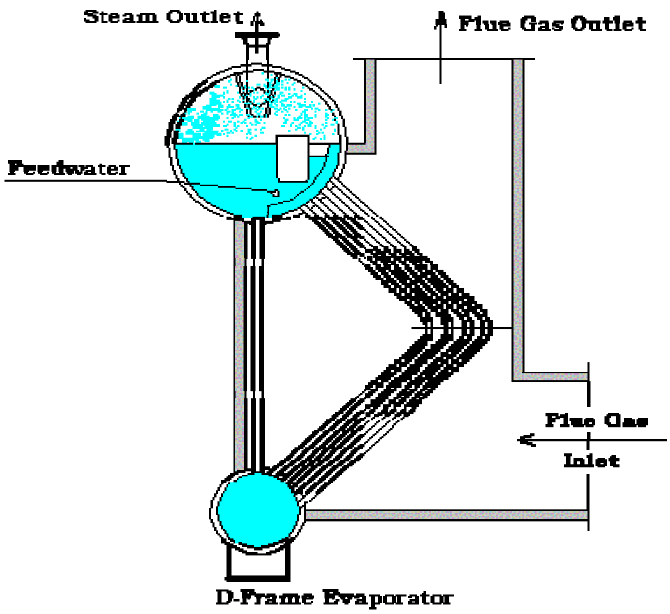

Figure A.1 D-Frame Evaporator Layout

A.1.2. 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 Figure A.2.

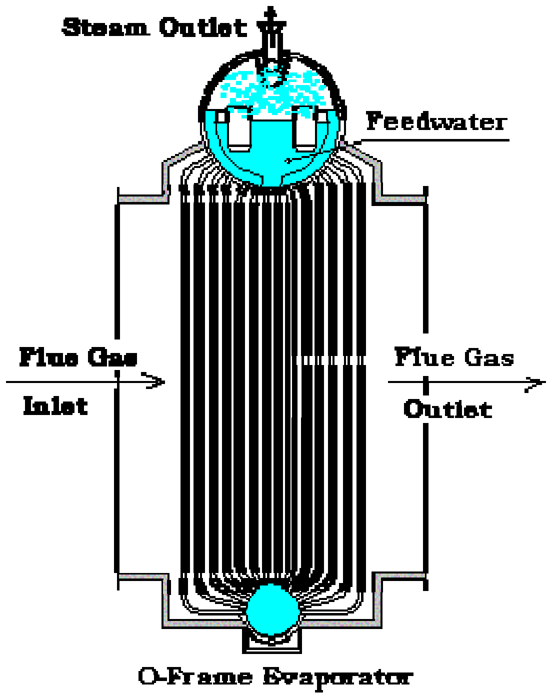

Figure A.2 O-Frame Evaporator Layosut

A.1.3. 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 Figure A.3.

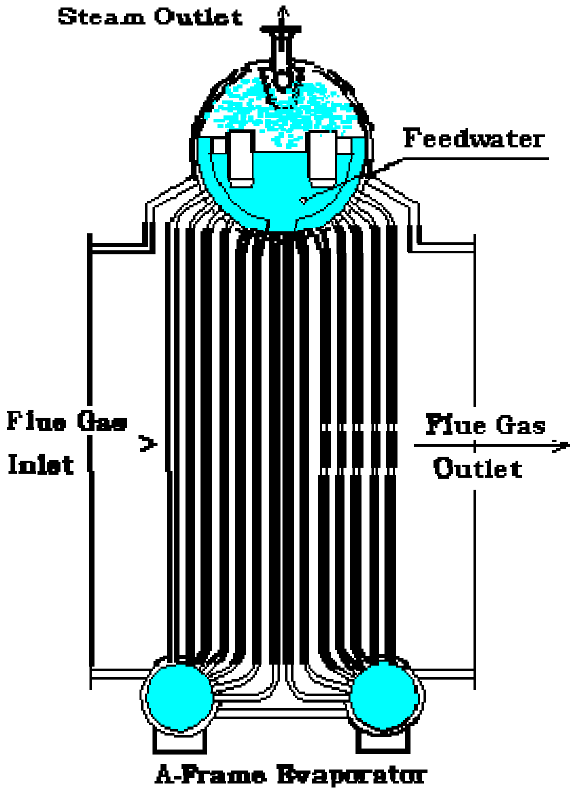

Figure A.3 A-Frame Evaporator Layout

A.1.4. 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 Figure A.4.

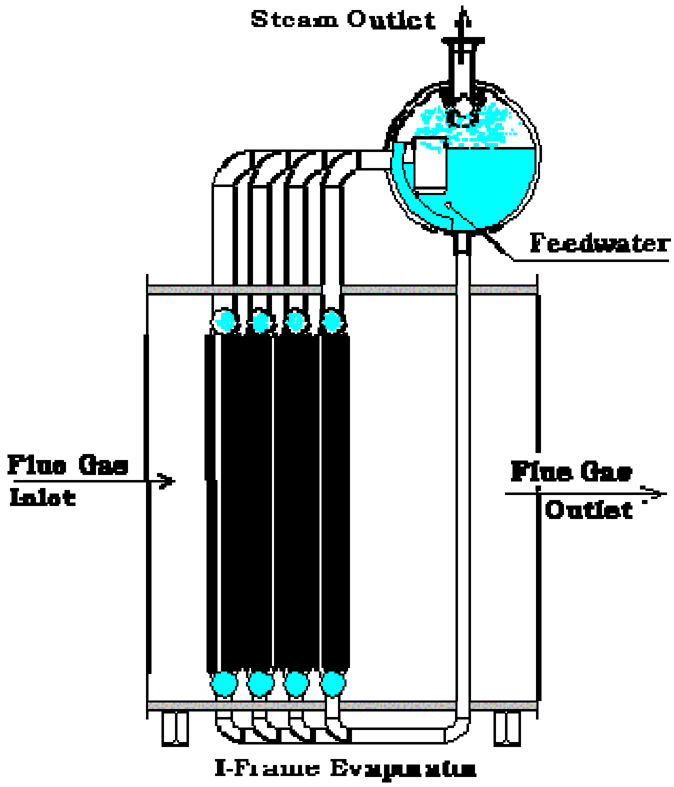

Figure A.4 I-Frame Evaporator Layout

A.1.5. 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 Figure A.5.

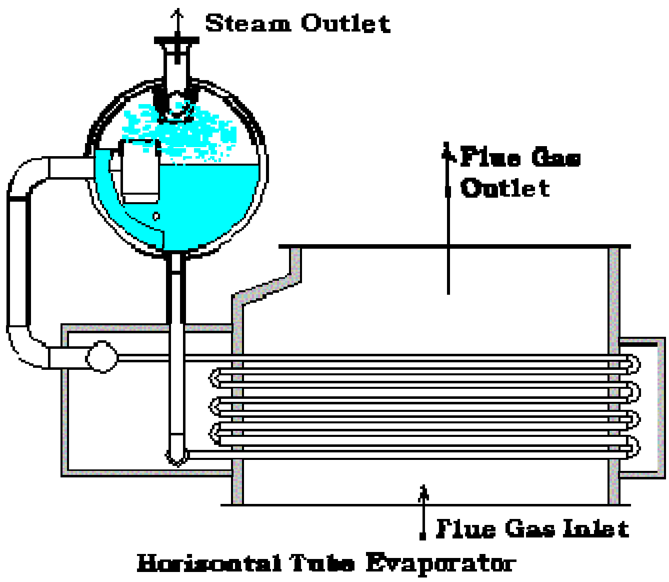

Figure A.5 Horizontal Tube Evaporator Layout

A.2. Types and Configurations of HRSG According to Superheater Layouts

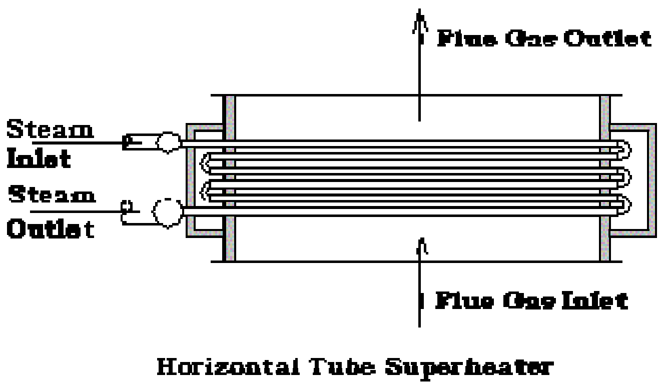

Figure A.6. Horizontal Tube Type Superheater Layout

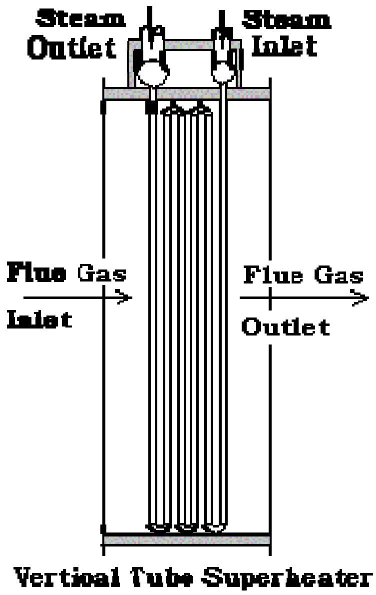

Figure A.7. Vertical Tube Type Superheater Layout

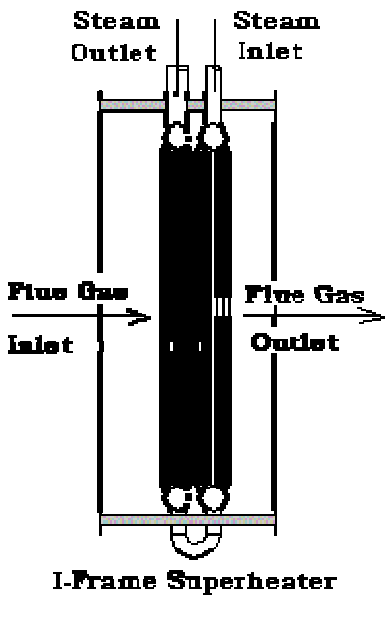

Figure A.8 I-Frame Type Superheater Layout[20]

APPENDIX B

REMM ANALYSIS, BASIC CONCEPTS

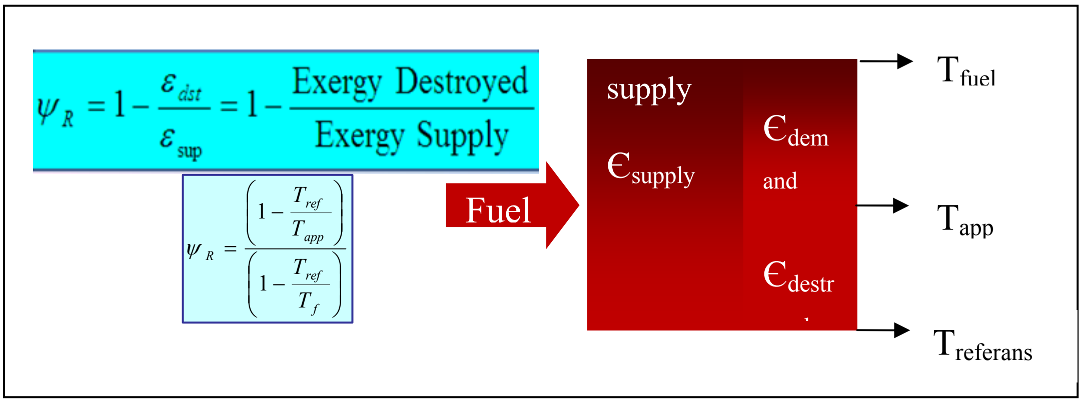

Figure B.1 Schematical Representation of REMM [28]

5-a and 5-b is a scale about how rational is the match between the supply and demand exergies (Kılkış-a,b, 2007). If these do not match, fossil fuel based emissions like CO_2 increase. Because, *PES* is related by definition to fossil fuels and the directive aims to lower the harmful emissions, the impact of CHP systems on the environment, in terms of exergy efficiency, must definitely be incorporated in addition to energy efficiency.

$$\psi_R = \frac{\varepsilon_{\min}}{\varepsilon_{\max}} = 1 - \frac{\text{Exergy Destroyed}}{\text{Exergy Supply}} \qquad (5\text{-a})$$

$$\psi_R = \frac{\varepsilon_{\min}}{\varepsilon_{\max}} = \frac{\text{Exergy Demand}}{\text{Exergy Supply}} \qquad (5\text{-b})$$

In Equation 5-a, ε_{max} is the maximum exergy of the input fuel for a unit amount of energy for a given application in the consumers area. For the thermal energy conversion system like the one shown in Figure 2, ε_{max} is a function of the flame temperature T_f of the fuel and the temperature of the environment that the system may become in thermal equilibrium, T_{ref}. T_{ref} may be selected to be the temperature of the ground, sea, or lake, but preferably in the close vicinity of the applications. Because outdoor air temperature is quite variable-even hourly, it should be the last option. ε_{min} is the minimum amount of exergy that could satisfy the same task. For chemical or mechanical systems, an equivalent flame temperature may be defined (Kılkış-a, b 2007). Then, According to the sign convention for exergy to be positive both in heating and cooling processes, two cases were distinguished.

$$\psi_R = \frac{\left(1 - \frac{T_{ref}}{T_{app}}\right)}{\left(1 - \frac{T_{ref}}{T_f}\right)} \qquad \{T_{app} > T_{ref}\} \tag{6-a}$$

$$\psi_R = \frac{\left(1 - \frac{T_{app}}{T_{ref}}\right)}{\left(1 - \frac{T_{ref}}{T_f}\right)} \qquad \{T_{app} < T_{ref}\} \tag{6-b}$$

For the special case of $T_{app} = T_{ref}$, T_{ref} may be selected accordingly in order to satisfy one of the above inequality conditions. This seemingly arbitrary selection does not affect the overall rationality of the exergy balance, because all temperatures are referenced to the same T_{ref} (See Figure 1). ψ_R also depends upon the kind of application that utilizes the thermal energy supplied from the system at temperature T_{app}. Because Equations 3 and 4 do not incorporate the exergy component, a new PES_{RCHP} equation was derived. First, Equation 6 needs to be modified for a CHP system, because there are two different kinds of energy outputs, namely electrical and thermal. According to the principal temperatures defined in Figure 1 for a CHP system, Equation 5 can be written in the following format, where exergy values for unit energy are given in Equations 8 through 11.

In Equation 11, the minimum exergy demand is for ispace heating at a comfort indoor air temperature of T_a. A low ψ_{RCHP} means that most of the exergy of the fuel inputted (FC_1) to the system (*1*) per unit energy is destroyed and opportunities of deriving more useful work from the same amount of fuel are missed. Therefore, additional fuel, FC_2 is required to make-up this destruction by another plant (*2*).

$$\psi_{RCHP} = \frac{\varepsilon_{\text{Hmin}} + C \times (1 - \varepsilon_{\text{Emin}})}{\varepsilon_{\text{Hmax}} + C \times \varepsilon_{\text{Emax}}} \tag{7}$$

$$\varepsilon_{\text{Emax}} = \left(1 - \frac{T_{ref}}{T_f}\right) \tag{8}$$

$$\varepsilon_{\text{Emin}} = \left(1 - \frac{T_{ref}}{T_E}\right) \tag{9}$$

$$\varepsilon_{\text{Hmax}} = \left(1 - \frac{T_{ref}}{T_{app}}\right) \tag{10}$$

$$\varepsilon_{\text{Hmin}} = \left(1 - \frac{T_{ref}}{T_a}\right) \text{ or } \varepsilon_{\text{Hmin}} = \left(1 - \frac{T_a}{T_{ref}}\right) \qquad \{T_a < T_{ref}\} \tag{11}$$

The first term in Equation 12 is the direct fuel spending in plant (*1*) with efficiency η_1. The second term, is a result of compounded fuel spending in plant (*2*) with efficiency η_2. If ψ_{RCHP} was high, the second term could be greatly avoided. If the consumers area and plant (*2*) are close or integrated like in a CHP system, η_T may be assumed unity. Then, Equation 12 may be simplified, if η_1 and η_2 are close to each other

$$\sum FC = FC_1 + FC_2 = \left(\frac{1}{\eta_1}\right) + \left(\frac{1}{\eta_2 \times \eta_T}\right)(1 - \psi_{RCHP}) \tag{12}$$

$$\sum FC = FC_1 + FC_2 = \left(\frac{1}{\eta}\right)(2-\psi_{RCHP}) \qquad \{\eta_T = 1\} \qquad (13)$$

Because the energy efficiency terms are already included in Equation 3, the exergy efficiency component may be easily incorporated in the form of a multiplier. The new *Rational Energy-Exergy Savings Factor PES$_{RCHP}$* becomes:

$$PES_{RCHP} = \left[1 - \frac{1}{\left(\frac{\mathrm{CHPH\eta}}{\mathrm{RefH\eta}} + \frac{\mathrm{CHPE\eta}}{\mathrm{RefE\eta}}\right) \times \frac{(2-\mathrm{Ref}\psi_{RCHP})}{(2-\psi_{RCHP})}}\right] \times 100 \qquad (14)$$

$$PES_{RCHP} = \left[1 - \frac{1}{\mathrm{CHPH\eta}\left(\frac{1}{\mathrm{RefH\eta}} + \frac{C}{\mathrm{RefE\eta}}\right) \times \frac{(2-\mathrm{Ref}\psi_{RCHP})}{(2-\psi_{RCHP})}}\right] \mathrm{x}100 \qquad (15)$$

In Equation 14, if ψ_{RCHP} is equal to Refψ_R (no rational exergy improvement), then PES_R will be equal to *PES*. Refψ_{RCHP} and ψ_{RCHP} are calculated according to Equations 16. Because all equations are based on unit exergies and unit energy and unit power values, Equations 14 and 15 may directly be employed to calculate compound reduction in carbon emissions, if the fuel type is known.

$$\mathrm{Ref}\psi_{RCHP} = \frac{H_{CHP} \times \mathrm{Ref}\psi_{RH} + E_{CHP} \times \mathrm{Ref}\psi_{RE}}{H_{CHP} + E_{CHP}} = \frac{\mathrm{Ref}\psi_{RH} + C \times \mathrm{Ref}\psi_{RE}}{1+C} \qquad (16)$$

When sustainability, carbon emissions, and global warming become important issues of concern, the balance among the qualities of the energy in terms of useful work potential at the supply and demand points become critical. The common metric of the exergy balance rationale is provided by the **Rational Exergy Management Efficiency** [17], ψ_R which depends on how the exergy demand (ε_{dem}), exergy supply (ε_{sup}), and destroyed exergy (ε_{dst}) relate to each other on a temperature scale. Schematical representation of REMM can be seen in figure 2. ψ_R is a scale about how rational is the match between the supply and demand exergies

So, PES_R can be calculated as follows:

$$PES_{RCHP} = \left[1 - \frac{1}{\left(\frac{CHPH\eta}{REFH\eta}+\frac{CHPE\eta}{REFE\eta}\right)*\left(\frac{2-REF\psi RCHP}{2-\Psi_{RCHP}}\right)}\right] \tag{17}$$

Where;

$$\psi_{RCHP} = \frac{\varepsilon_{\text{Hmin}} + C \times (1-\varepsilon_{\text{Emin}})}{\varepsilon_{\text{Hmax}} + C \times \varepsilon_{\text{Emax}}} \tag{18}$$

and

$$\varepsilon_{\text{Emax}} = \left(1-\frac{T_{ref}}{T_f}\right),\ \varepsilon_{\text{Emin}} = \left(1-\frac{T_{ref}}{T_E}\right),\ \varepsilon_{\text{Hmax}} = \left(1-\frac{T_{ref}}{T_{app}}\right),\ \varepsilon_{\text{Hmin}} = \left(1-\frac{T_{ref}}{T_a}\right) \tag{19}$$

$$\text{Ref}\psi_{RCHP} = \frac{H_{CHP} \times \text{Ref}\psi_{RH} + E_{CHP} \times \text{Ref}\psi_{RE}}{H_{CHP} + E_{CHP}} = \frac{\text{Ref}\psi_{RH} + C \times \text{Ref}\psi_{RE}}{1+C} \tag{20}$$

If the application is indoor space heating, the reference rational exergy efficiency of the heating system based on a ground source heat pump using grid electricity may be chosen 0.10 and for the central power plant a typical exergy value of 0.3 may be chosen [39] Temperature and ε values representation can be seen in Figure B.2.

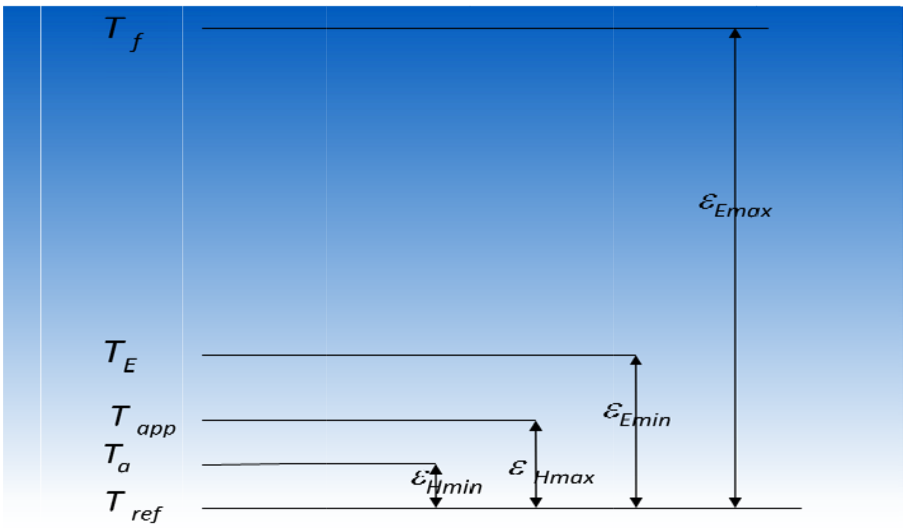

Figure B.2 Adoption of REMM to CHP Systems for 2004/8/EC Directive [28]

APPENDIX C

ENGINEERING DRAWINGS OF THE HEGEL CO-GENERATOR AND MATPUM BUILDING

Below, Technical drawings of the ICE and the pictures of the MATPUM Building, which is the host for FP 6 EU HEGEL Project in METU, Ankara, Turkey.

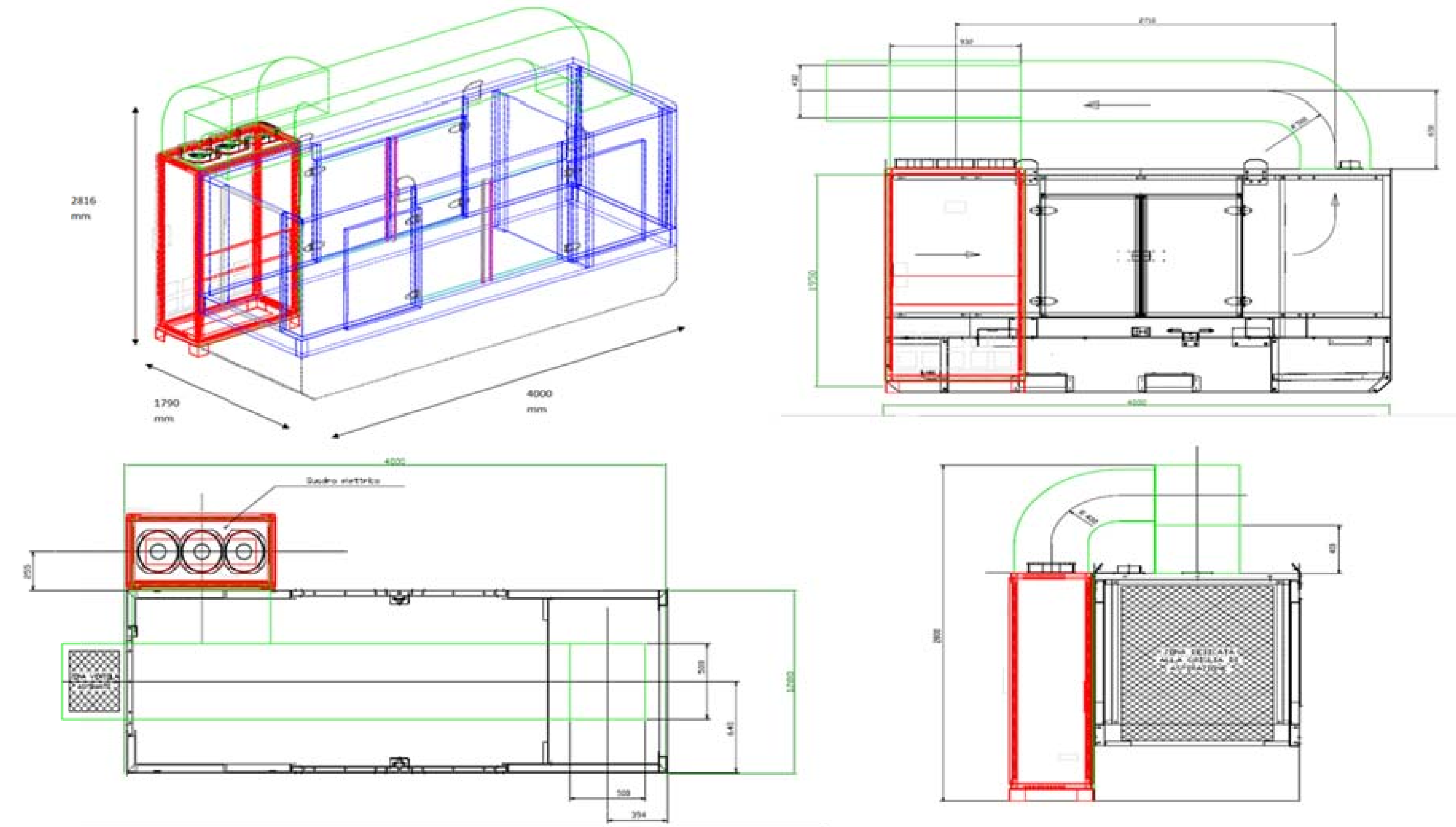

Figure C.1 Engineering Drawings of the HEGEL Co-generator

Figure C.2 MATPUM Building, Front View

Figure C.3 MATPUM Building, Side View

Figure C.4 MATPUM Building Inner View

Figure C.5 MATPUM Building Inner View, Work Stations

APPENDIX D

ESTABLISHMENT OF HEGEL ICE CO-GENERATOR AND ACCOMPANYING CONSTRUCTIONAL WORK

A team of CRF (Centre Recherche Fiat, Torino, Italy) workers arrived in Ankara, METU on the 14th of December, 2009 for preparing the system for start up. CRF and METU teams work in cooperation for preparation of the ICE, but testing and operation of the CHP system will not be finished before two months.

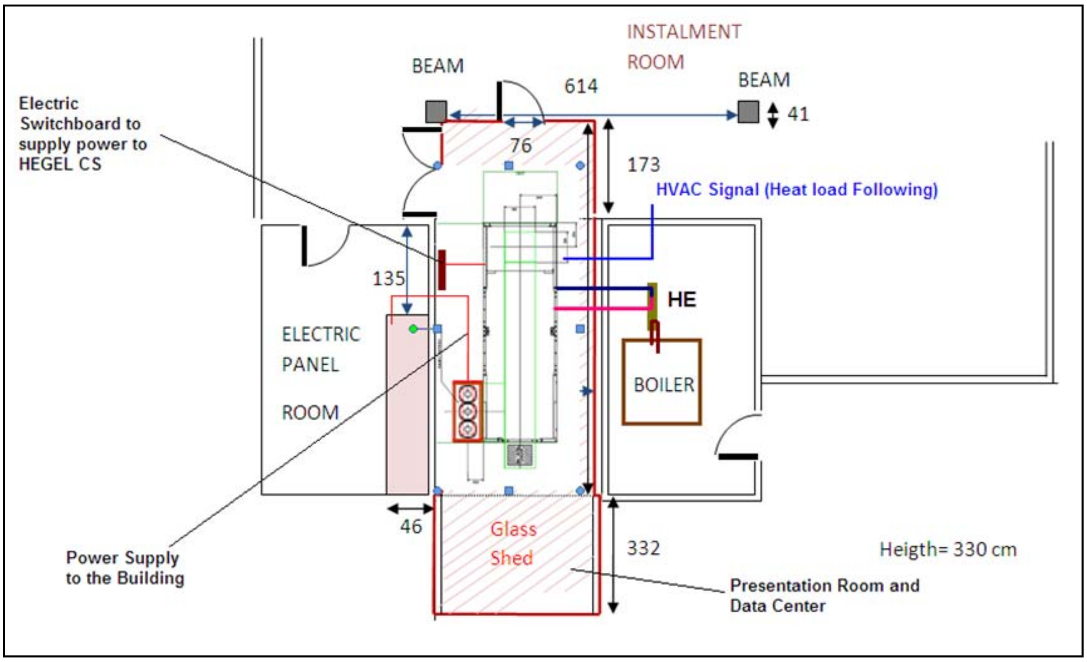

Figure D.1 . Schematics for the HEGEL Room

Work done for installation of HEGEL System in METU start with construction of the HEGEL room and positioning the ICE with the help of cranes. After that, inlet and outlet water pipe connections and natural gas inlet pipe connection were done, generator cooling water system installation had been carried out and lastly, electrical connections of the system with grid had been finished. Schematics for the HEGEL room can be found in Figure D.1 below. In the following figures from Figure D.2 to D.6, the installation progress can be seen.

Figure D.2 Beginning of the Establishment Procedure for ICE Co-generator

Figure D.3 Unwrapping the ICE Co-generator

Figure-D.4 HEGEL ICE Co-generator

Figure D.5 ICE Outside View

Figure D.6 ICE Exhaust Piping Assembly

Figure D.7 ICE After the Construction- Outside View

Figure D.8 ICE After the Construction- Inside the HEGEL Room

APPENDIX E [11,30,50]

TECHNICAL PROPERTIES OF ICE

Table E.1 Technical Properties of ICE

Model	FPT ENT 60 NG
Engine Type	Otto 4T
Fuel	Natural Gas
Air to Fuel Ratio	Stoicihiometric
Feeding	Multi point injection
Cylinder arrangement	Six in line
Displacement	5883cm3
Compression ratio	11:01
Electrical power output	126 kW
Thermal output	220 kW
Supplied fuel power	427 kW
Electrical efficiency	29.5%
Heat efficiency	51.5%
EUF	81.0%
Maximum shaft torque	650Nm
Maximum shaft speed	2150 rpm
Maximum shaft power	141 kW
Mean piston speed at rated speed	10.8 m/s
bmep at maximum torque	14.0 bar
Dry weight	520 kg
Emission reduction	TWC
CO	<589mg/kWhe
NOx	<236mg/kWhe

APPENDIX F

ICE OPERATING AND PERFORMANCE CHARACTERISTICS [51,52,53]

*C*haracteristics of the engine and various engine performance parameters are defined below.

The crank offset a is given as

a = S 2 (F.1)

Average piston speed is

Up = 2SN (F.2)

Average piston speed for all engines will normally be in the range of 5 to 20 m/sec, with large diesel engines on the low end and high-performance automobile engines *on the high end* Displacement, or displacement volume Vd is the volume displaced by the piston as it travels from bottom dead center (BDC) to top dead center (TDC).

Vd =VBDC −VTDC (F.3)

Displacement can be given for an engine with Nc number of cylinders one cylinder or the entire engine. For one cylinder

$$V_d = \frac{\pi}{4} B^2 S N_c \tag{F.4}$$

Minimum cylinder volume occurs when the piston is at TDC and called the clearance volume VC

$$V_C = V_{TDC} = V_d - V_{BDC} \tag{F.5}$$

The compression ratio is defined as

$$R_C = \frac{V_{BDC}}{V_{TDC}} = \frac{(V_C + V_d)}{V_C} \tag{F.6}$$

Force due to gas pressure on the moving piston generates the work in an internal combustion engine cycle

$$W = \int PdV \tag{F.8}$$

It is convenient to analyze engine cycles per unit mass of gas m within the cylinder. To do so, volume V is replaced with specific volume υ and work is replaced with specific work:

$$w = \frac{W}{m}, \quad v = \frac{V}{m}, \quad w = \int Pdv \tag{F.9}$$

Specific work is equal to the area under the process lines on the P-υ coordinates of indicator diagram. The areas shown in the indicator diagram gives the work inside the combustion chamber and called as indicated work W_i, but the work delivered by crankshaft is less than indicated one due to mechanical friction and parasitic loads of the engine W_f . Actual work available at the crankshaft is called brake work.

$$W_b = W_i - W_f \tag{F.10}$$

The ratio of brake work to indicated work defines the mechanical efficiency of an engine. Mechanical efficiencies will be on the order of 75 % to 95 %,

$$\eta_m = \frac{W_b}{W_i} \tag{F.11}$$

An average or mean effective pressure (*mep*) is defined by

$$mep = \frac{w}{\Delta v} = \frac{W}{V_d} \tag{F.12}$$

Mean effective pressure is a good parameter to compare engines for design or output because it is independent of engine size and /or speed. Various *mep* can be defined by using different work terms. If brake work is used brake mean effective pressure is obtained:

$$bmep = \frac{W_b}{V_d} \quad \text{(F.13)}$$

Indicated work gives indicated mean effective pressure:

$$imep = \frac{W_i}{V_d} \quad \text{(F.14)}$$

Typical maximum values of *bmep* for naturally aspirated SI engines are in the range of 850 to 1050 kPa. For CI engines, they are 700 to 900 kPa for naturally aspirated engines and 1000 to 1200 kPa for turbocharger engines. Torque is a good indicator of an engine's ability to do work. It is defined as force acting at a moment distance and has units of N-m. Torque is related to work by:

$$2\pi\tau = W_b = (bmep)\frac{V_d}{n} \quad \text{(F.15)}$$

$$\tau = (bmep)\frac{V_d}{4\pi} \quad \text{(F.16)}$$

for four stroke cycle.

Power is defined as the rate of work of the engine. If n is the number of revolutions per cycle and N is the engine speed, then

$$\dot{W} = W\frac{N}{n} \quad \text{(F.17)}$$

Other characteristic parameters for an engine are:

Specific power;

$$SP = \frac{\dot{W}_b}{A_p} \tag{F.18}$$

Output per displacement;

$$OPD = \frac{\dot{W}_b}{V_d} \tag{F.19}$$

Specific volume;

$$SV = \frac{V_d}{\dot{W}_b} \tag{F.17}$$

Specific weight;

$$SW = \frac{(Weight_{engine})}{\dot{W}_b} \tag{F.18}$$

Energy input to an engine Qin comes from the combustion of a hydrocarbon fuel. Air is used to supply the oxygen needed for this chemical reaction. For combustion reaction to occur, the proper relative amounts of air (oxygen) and fuel must be present. Air-fuel ratio (AF) and fuel–air ratio (FA) are parameters used to describe the mixture ratio:

$$AF = \frac{m_a}{m_f} = \frac{\dot{m}_a}{\dot{m}_f} \tag{F.19}$$

$$FA = \frac{m_f}{m_a} = \frac{\dot{m}_f}{\dot{m}_a} \tag{F.20}$$

where ma = mass of air, m& a = mass flow rate air, mf = mass of fuel, m& f = mass fuel rate of fuel. The ideal stoichiometric AF for many gasoline type hydrocarbon fuels is very close to 15, with combustion possible for value in the range 6 to 19

Equivalence ratio is defined the actual ratio of fuel–air to ideal or stoichiometric fuel–air:

$$\Phi = \frac{(FA)_{\text{act}}}{(FA)_{\text{stoich}}} = \frac{(AF)_{\text{stoich}}}{(AF)_{\text{act}}} \tag{F.21}$$

Fuel consumption of an engine is calculated from the consumption per power generated; this is also called specific fuel consumption (sfc) and can be derived for all types of works. The specific fuel consumption equations for brake power and indicated power are given as

$$bsfc = \frac{\dot{m}_{\text{f}}}{\dot{W}_{\text{b}}} \tag{F.22}$$

$$isfc = \frac{\dot{m}_{\text{f}}}{\dot{W}_{\text{i}}} \tag{F.23}$$

In order to measure or comment on an engine performance, different types of efficiencies related with engine parameters must be known. These efficiencies are:

Combustion efficiency ηc is defined as the fraction of fuel which burns. It has values in the range 0.95 to 0.98 and it is given as

$$\eta_{\text{c}} = \frac{\dot{Q}_{\text{in}}}{Q_{\text{HV}}\dot{m}_{\text{f}}} \tag{F.24}$$

where Qin is the rate of net heat input to the engine and QHV is the lower heating value of the fuel. Thermal efficiency is defined as where Qin is the rate of net heat input to the engine and QHV is the lower heating value of the fuel. Thermal efficiency is defined as

$$\eta_{th} = \frac{W}{Q_{in}} = \frac{\dot{W}}{\dot{Q}_{in}} = \frac{\dot{W}}{\dot{m}_f \dot{Q}_{HV} \eta_c} \tag{F.25}$$

Thermal efficiency can be given as indicated or brake depending on whether indicated power or brake power is used. Engines can have indicated thermal efficiencies in the range of 50% to 60% with brake thermal efficiency usually about 30%. Some large slow CI engines can have brake thermal efficiencies greater than 50% Fuel conversion efficiency is defined as

$$\eta_f = \frac{\dot{W}}{\dot{m}_f Q_{HV}} \tag{F.26}$$

Volumetric efficiency is defined as

$$\eta_v = \frac{n\dot{m}_a}{\rho_a V_d N} \tag{F.27}$$

where ρa is the density of atmospheric air. Volumetric efficiency is a measure of how much air is ingested into the engine and it could be greater than one for turbocharged engines.

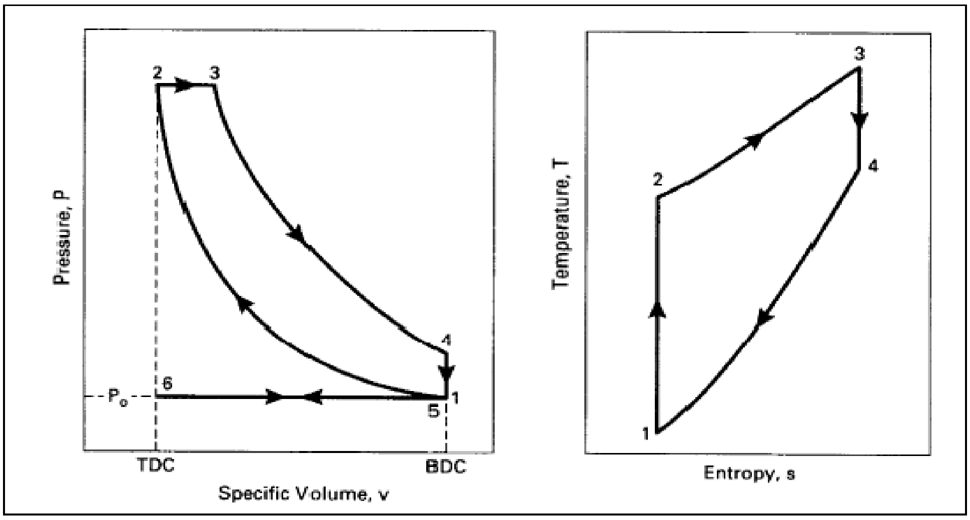

Figure F.1 P-s and T-s Diagrams of the ICE Cycle

Combustion Equation and calculation of Specific Heats

$$C_1H_{3.8}N_{0.1} + 16\left(1 + \frac{3.8}{4} - \frac{0.1}{2}\right).(O_2 + 3.773\ N_2)$$
$$\xrightarrow{Heat} 1CO_2 + 3.8H_2O + \left(1 + \frac{3.8}{4} - \frac{0.1}{2}\right).(15.3\ O_2) + \left(1 + \frac{3.8}{4} - \frac{0.1}{2}\right).(3.773O_2).(16.3) \quad \text{(F.28)}$$

$$C_V = \frac{\sum C_{v,i}.m_i}{\sum m_i} \quad \text{(F.29)}$$

APPENDIX G

TECHNICAL PROPERTIES OF EQUIPMENT [11,50]

Table G.1 Technical Properties of Steam Engine

Configuration	3 cylinder radial
Bore/stroke	50/50mm
Displacement	300 cm3
Number of valves per cylinder	2
Volumetric ratio	30
Cut-off	0.11
Mean piston speed at nominal speed	3.8 m/s
Nominal inlet pressure	100 bar
Nominal inlet temperature	450 C
Nominal outlet pressure	1 bar
Steam flow rate	44 g/s
bmep	23 bar
Maximum shaft power	26 kW
Thermal power	64 kW
Maximum shaft speed	2300 rpm
Maximum shaft torque	130 Nm
Thermodynamical efficiency	32%
SE isoentropic efficiency	65%
SE organic efficiency	80%

Table G.2 Technical Properties of Condenser

Maximum pressure (safety)	100 bar
Maximum temperature (safety)	450 C
Operating inlet pressure	1 bar
Operating temperature condensation	100 C
Steam flow rate	5–44 g/s
Inlet temperature water	70 C
Outlet temperature water	80 C
Mean coeff. of exchange	2200W/m2 K
Tube per passage	31
Passage	4
Tube diameter	3/8 in, 9.525mm
Tube length	605mm
Shell diameter	170mm

APPENDIX H

TECHNICAL PARAMETERS AND DETAILED ENERGY FORMULATION OF THE STEAM ENGINE CYCLE [56]

Steam flow rate of the HRSG can be calculated

$$\dot{m}_{steam} = \dot{m}_{exh} \cdot c_{exh} \cdot \frac{(T_4 - T_6)}{h_e - h_c} \tag{H.1}$$

with

$$T_6 = T_c + \Delta T_{pp} \tag{H.2}$$

where *he*, *hc* are the enthalpies of points *e* and *c*; T_c is the temperature of point *c* (see Figures 2.9), T_4 is the inlet exhaust gases temperature and T_{pp} is the temperature difference at pinch point;

HRSG specific steam flow rate can be calculated as ratio of steam flow rate and ICE's exhaust gases flow rate:

$$\frac{\dot{m}_{steam}}{\dot{m}_{exh}} = c_{exh} \cdot \frac{T_4 - T_6}{h_e - h_c} \tag{H.3}$$

The calculation of HRSG efficiency can be performed as follows:

$$\varepsilon_{HRSG} = \frac{\dot{Q}_{steam}}{\dot{Q}_{exh}} = \frac{\dot{m}_{steam}}{\dot{m}_{exh}} \cdot \frac{\Delta h_{HRSG}}{c_{exh} \cdot (T_4 - T_{ref})} \tag{H.4}$$

where *T ref* is the reference temperature (20°C).

The HRSG outlet temperature is:

$$T_7 = T_4 - \frac{\dot{m}_{steam}}{\dot{m}_{exh}} \cdot \frac{\Delta h_{HRSG}}{c_{exh}} \tag{H.5}$$

where T_7 is the temperature of outlet exhaust gases.

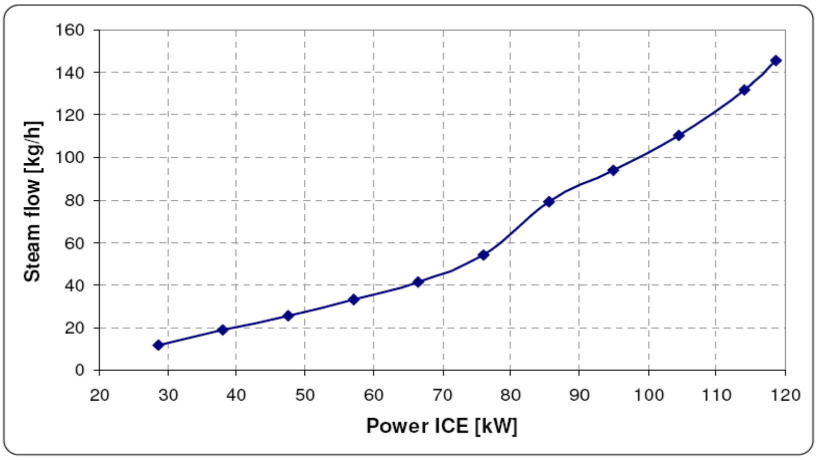

Figure H.1 Steam flow rate as function of normalized ICE power

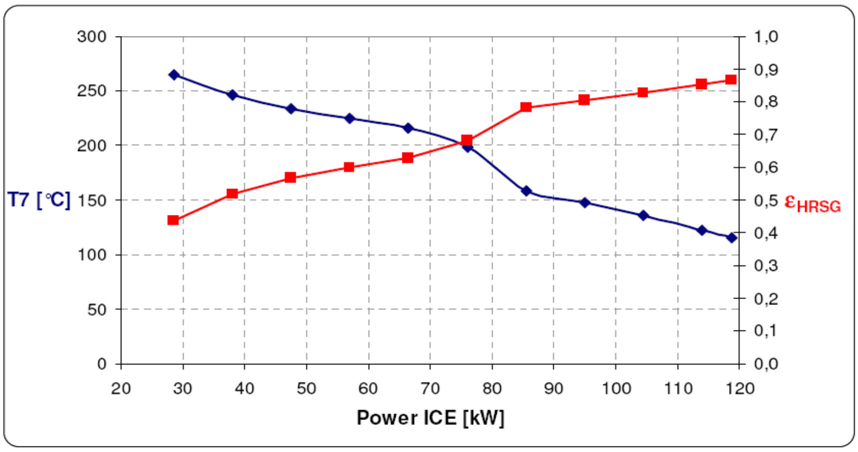

Figure H.2 T_7 and HRSG Efficiency as Function of ICE Power

The ideal efficiency of the Rankine cycle is:

$$\eta_{id} = \frac{P_{id}}{\dot{Q}_{steam}} = \frac{\dot{m}_{steam}}{\dot{m}_{steam}} \cdot \frac{L_{id}}{\Delta h_{HRSG}} \tag{H.6}$$

where:

$$\Delta h_{HRSG} = h_e - h_b ; \tag{H.7}$$

$$L_{id} = h_e - h_f .$$

This evaluation neglects the power of the pump, that will be considered through organic efficiency. The isentropic efficiency of the steam engine can be defined as follows:

$$\eta_{is} = \frac{P_i}{P_{id}} = \frac{\dot{m}_{steam}}{\dot{m}_{steam}} \cdot \frac{L_i}{L_{id}} = \frac{L_i}{h_e - h_f} \tag{H.8}$$

the internal work per mass unit

$$L_i = \frac{L_{cycle}}{m_{cycle}} \tag{H.9}$$

The work per cycle produced by steam engine is given by

$$L_{cycle} = \oint P \cdot dV = \int_1^2 P \cdot dV + \int_2^3 P \cdot dV + \int_4^5 P \cdot dV + \int_5^6 P \cdot dV \tag{H.10}$$

According to engine geometry the following parameters can be defined

compression ratio (>1)

$$\varepsilon = \frac{V_{\max}}{V_{\min}} \tag{H.11}$$

filling ratio (>1)

$$\Phi = \frac{V_5}{V_{min}} \tag{H.12}$$

recompression ratio (>1)

$$\chi = \frac{V_2}{V_{min}} \tag{H.13}$$

cut-off ratio;

$$\varphi = \frac{\Phi - 1}{\varepsilon - 1} \tag{H.14}$$

For: $(0 \le \varphi \le 1)$ it is 0 for no admission, 1 for maximum admission.

The work per cycle can be calculated by means of previously describer parameters

$$L_{cycle} = -P_{min} \cdot V_{min} \cdot (\varepsilon - \chi) - \frac{1}{k-1} \cdot P_{min} \cdot V_{min} \cdot \chi \cdot (\chi^{k-1} - 1) + P_{max} \cdot V_{max} \cdot \varphi \cdot (\varepsilon - 1) +$$
$$+ P_{max} \cdot V_{min} \cdot \frac{1 + \varphi \cdot (\varepsilon - 1)}{k - 1} \cdot \left[1 - \left(\frac{1 + \varphi \cdot (\varepsilon - 1)}{\varepsilon}\right)^{k-1}\right]. \tag{H.15}$$

Where

$$V_{min} = \frac{V}{\varepsilon - 1} \tag{H.16}$$

Steam mass per cycle is given by:

$$m_{cycle} = m_{max} - m_{min} \tag{H.17}$$

Where;

$$m_{max} - \frac{V_5}{v_5} - \frac{V_{min}}{v_5} \cdot [1 + \varphi \cdot (\varepsilon - 1)] \tag{H.18}$$

and

$$m_{\min} = \frac{V_2}{v_2} = \frac{V_{\min}}{v_1} \cdot \chi \tag{H.19}$$

So;

$$m_{cycle} = \frac{V_{\min}}{v_5} \cdot [1 + \varphi \cdot (\varepsilon - 1)] - \frac{V_{\min}}{v_1} \cdot \varepsilon \tag{H:20}$$

and

$$\eta_{is}(\varphi, \varepsilon, \chi) = \frac{1}{h_e - h_f} \cdot \frac{L_{cycle}}{m_{cycle}} \tag{H.21}$$

is a function only of geometric parameters is not dependent on the steam engine displacement.

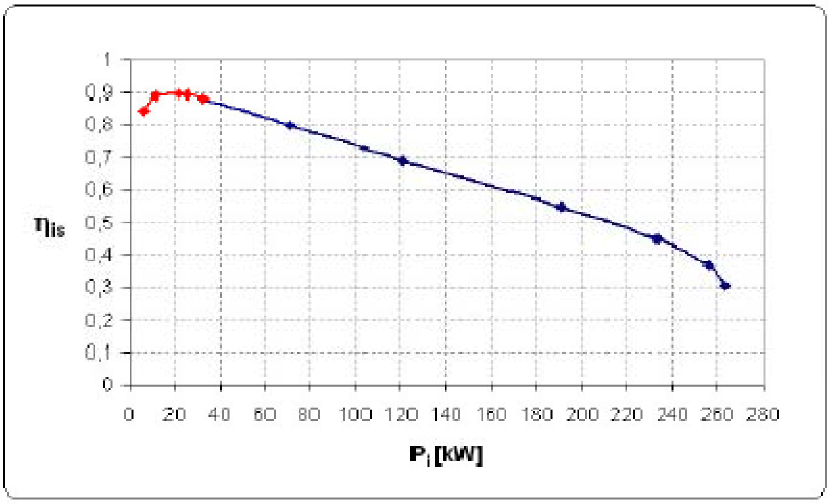

Figure H.3 Steam Engine Isentropic Efficiency as a Function of Power

The red line is referred to operative conditions. The electrical conversion efficiency is defined by:

$$\eta_{ec} = \frac{P_{el}}{P_u} \tag{H.22}$$

This value is relative to the generator and it has been assumed equal to 0,94. Figure F.4 shows electrical power produced by the SE.

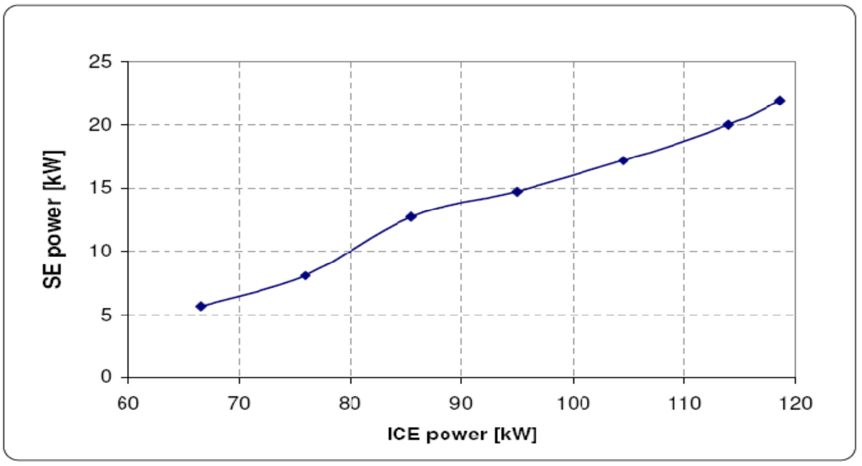

Figure H.4 SE Power as a Function of ICE power

Table H.1 Characteristics of SE

	SE 2			
P_{i_max} [kW]	62			
Speed [rpm]	1500			
Displacement [cc]	270			
Mean speed [m/s]	2,391			
Steam flow rate [kg/s]	P_i [kW]	P_u [kW]	$\eta_{is}*\eta_{org}$	VOT [ms]
0,011	8.53	6.87	0.703	1.98
0,015	11.69	10.02	0.724	2.64
0,027	18.95	17.28	0.709	4.17
0,031	21.58	19.91	0.695	4.71
0,041	26.40	24.73	0.666	5.67

APPENDIX I

MATLAB SIMULINK USER INTERFACES AND MODELS

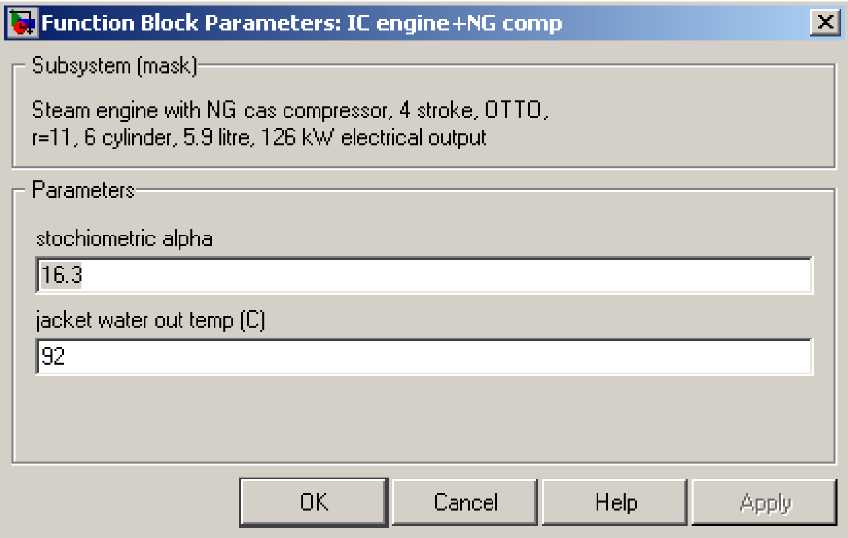

Figure I.1 User Interface Function Block-ICE

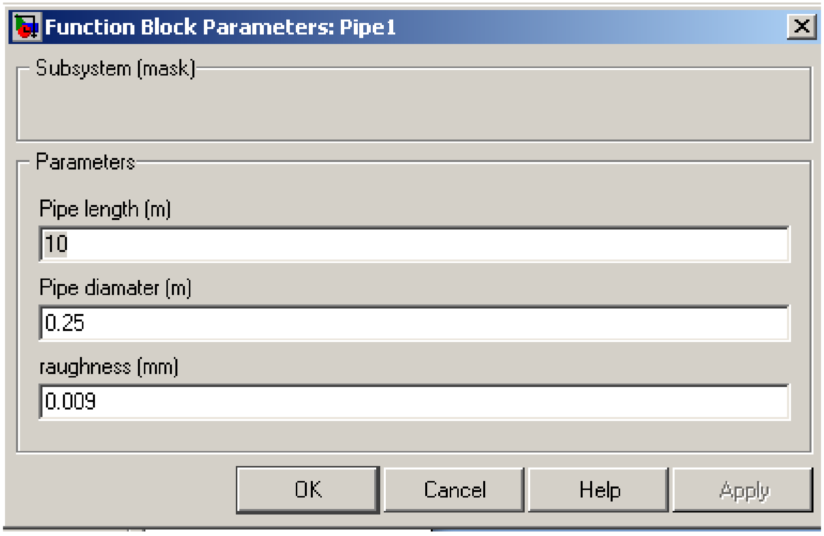

Figure I.2 User Interface Function Block –Pipe

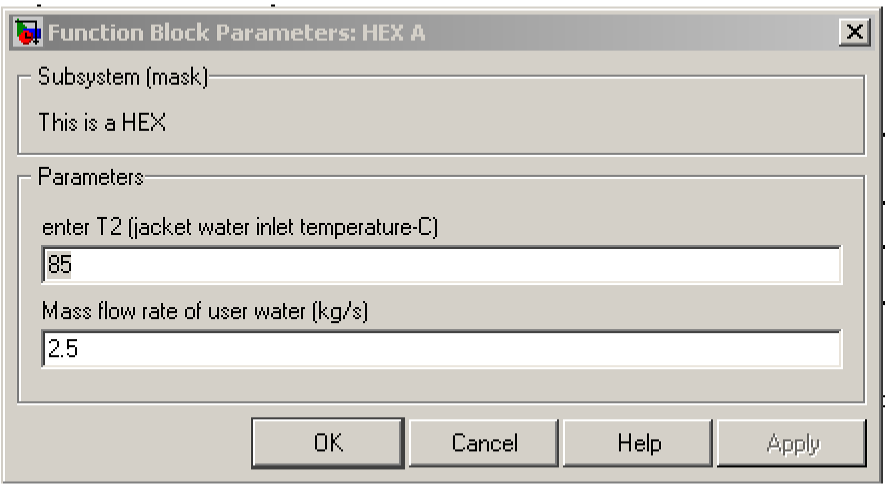

Figure I.3 User Interface Function Block-HEX A

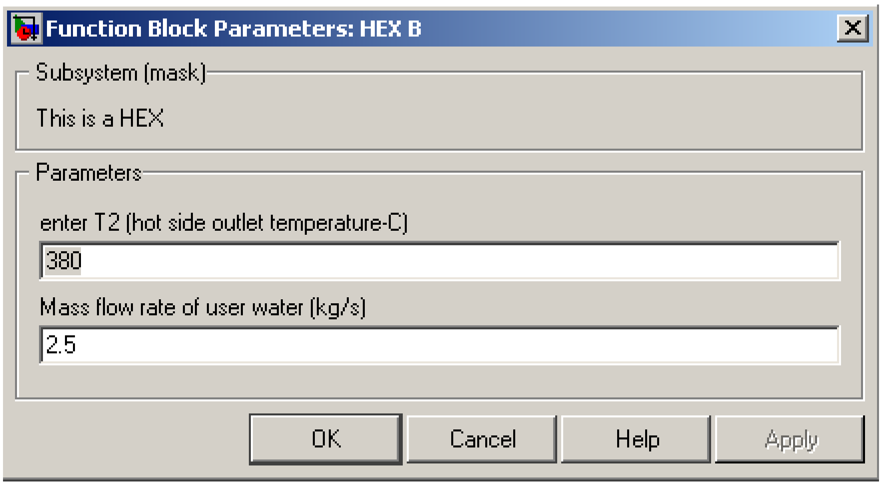

Figure I.4 User Interface Function Block-HEX B

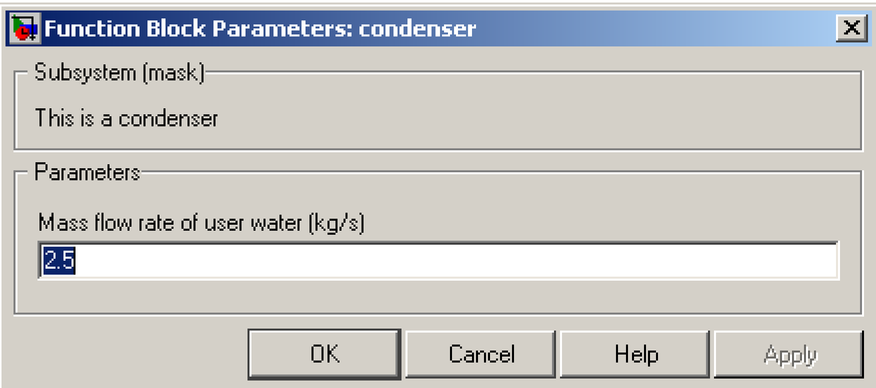

Figure I.5 User Interface Function Block –Condenser

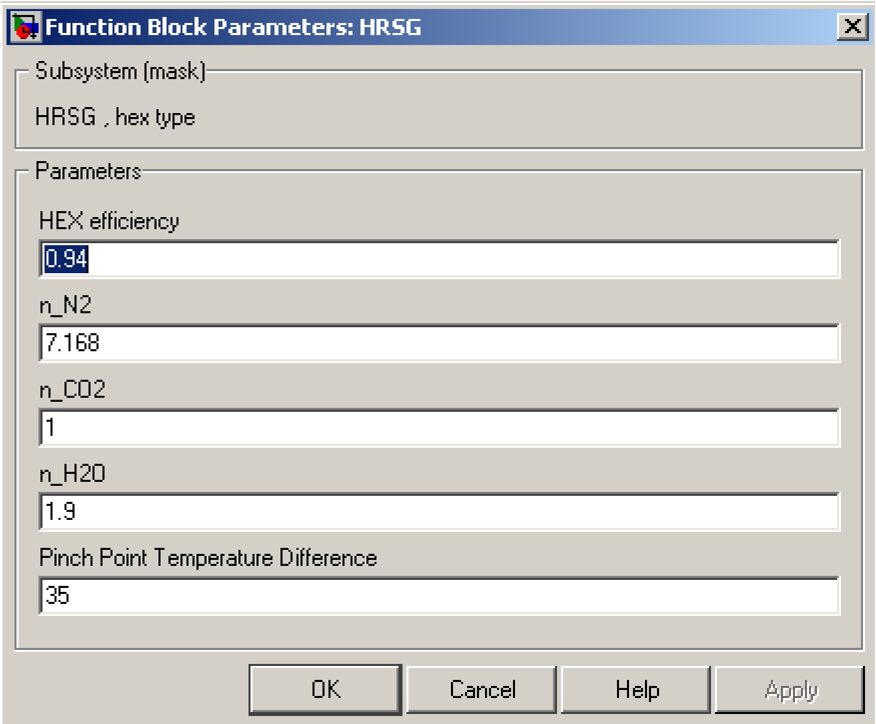

Figure I.6 User Interface Function Block-HRSG

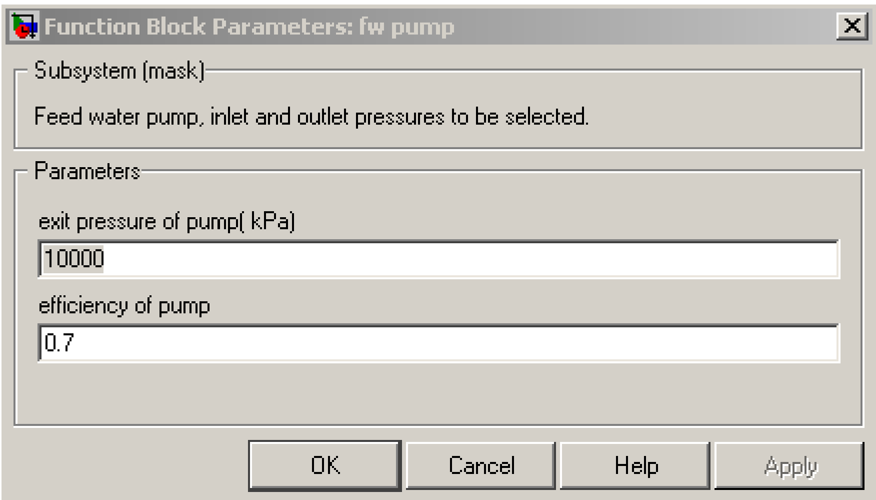

Figure I.7 User Interface Function Block –Feed Water Pump

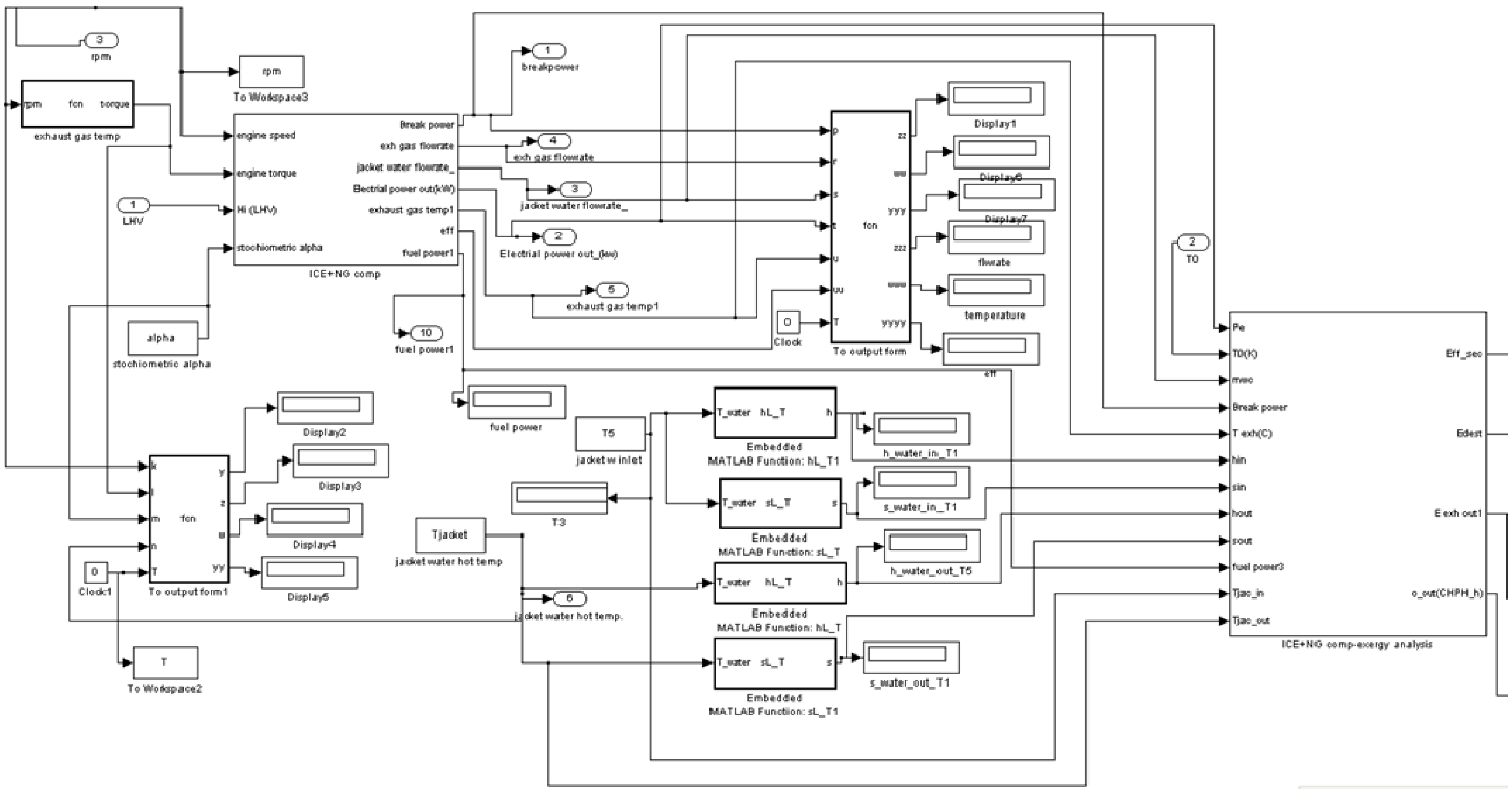

Figure I.8 Details of MATLAB Simulink Modeling of ICE (CASE 1-2)

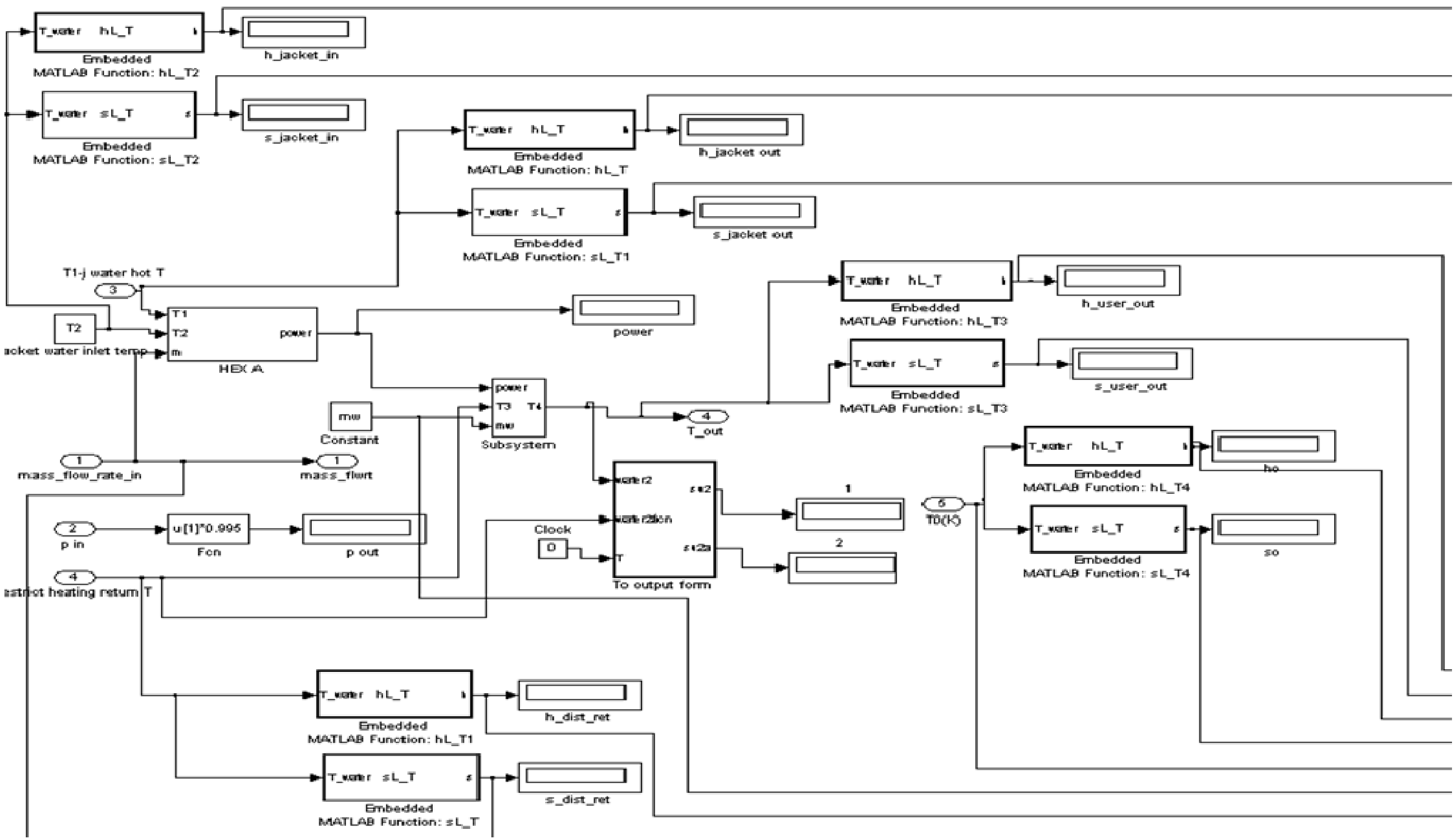

Figure I.9 Details of MATLAB Simulink Modeling of HEX A (CASE 1-2)

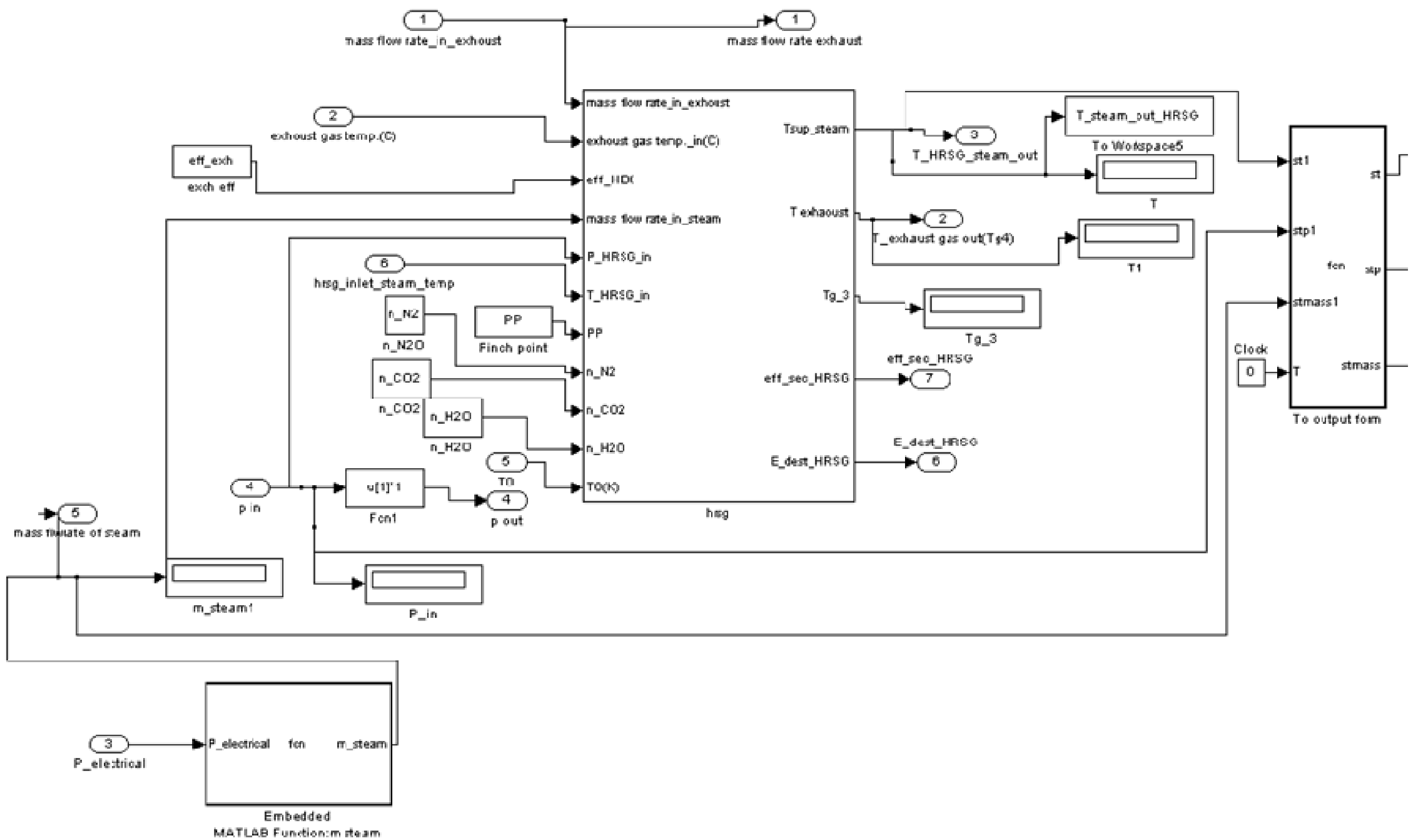

Figure I.10 Details of MATLAB Simulink Modeling of HRSG (CASE 2)

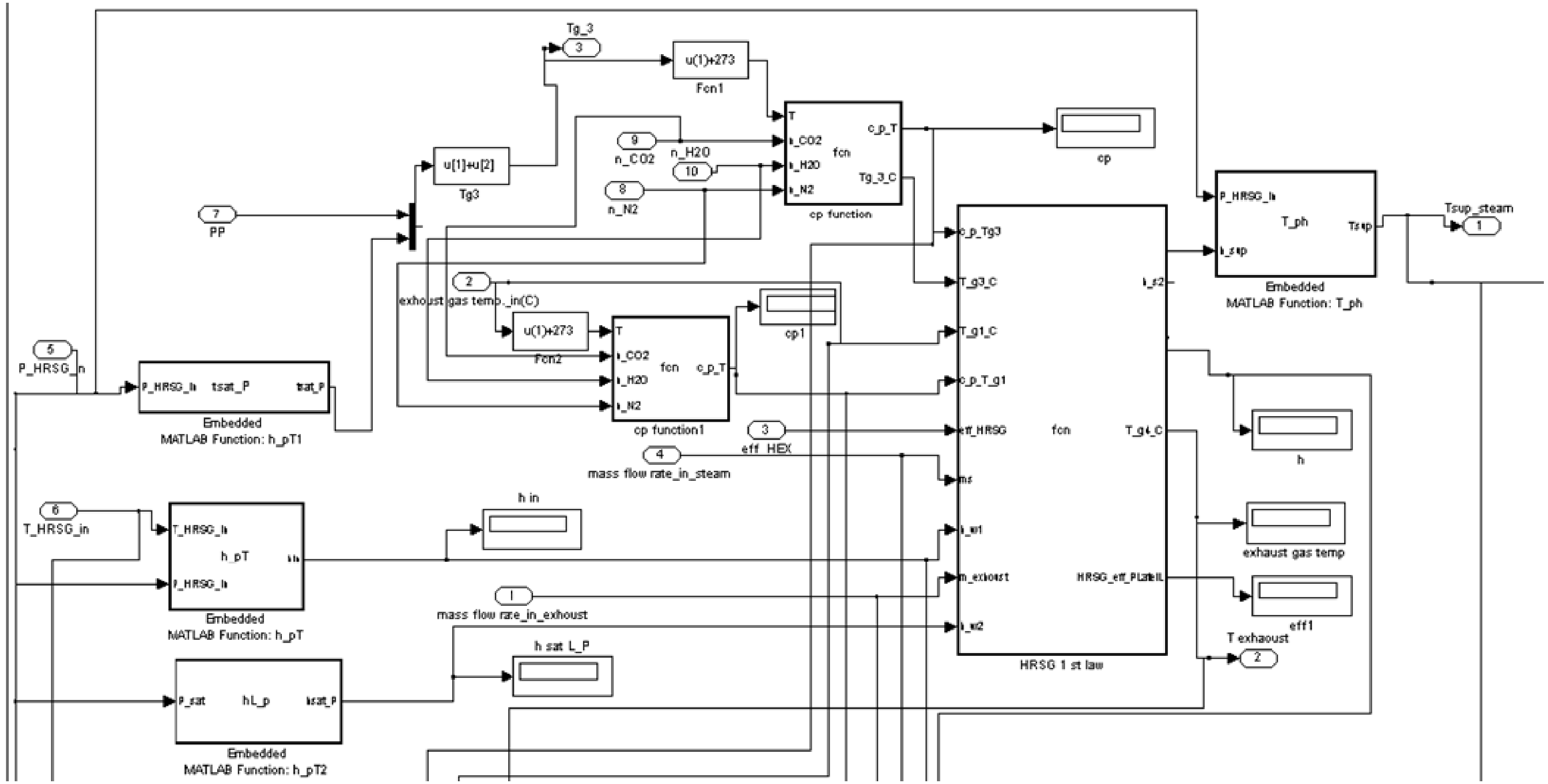

Figure I.11 Details of MATLAB Simulink Modeling of HRSG-1st Law analysis (CASE 2)

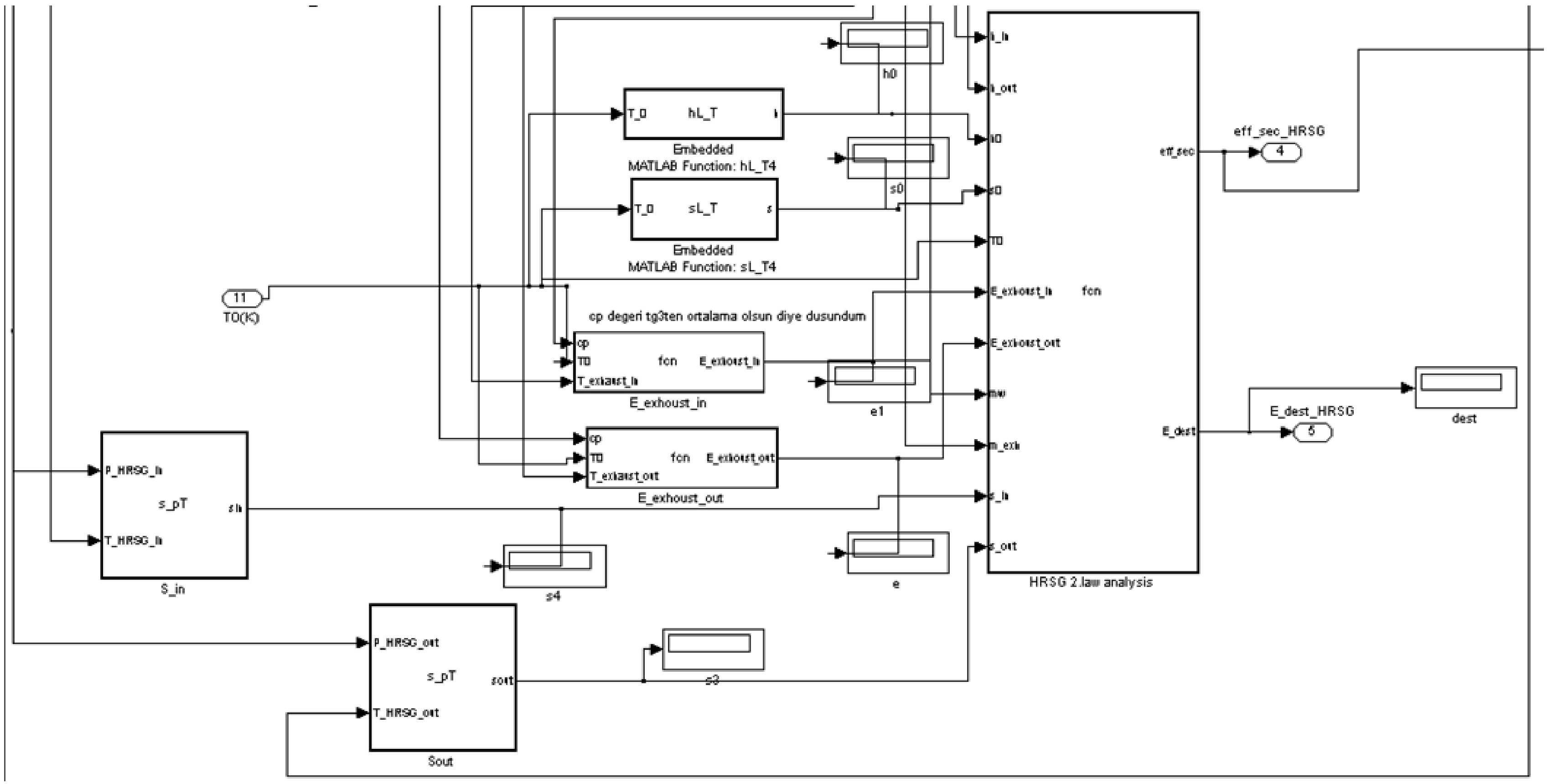

Figure I.12 Details of MATLAB Simulink Modeling of HRSG-2nd Law analysis (CASE 2)

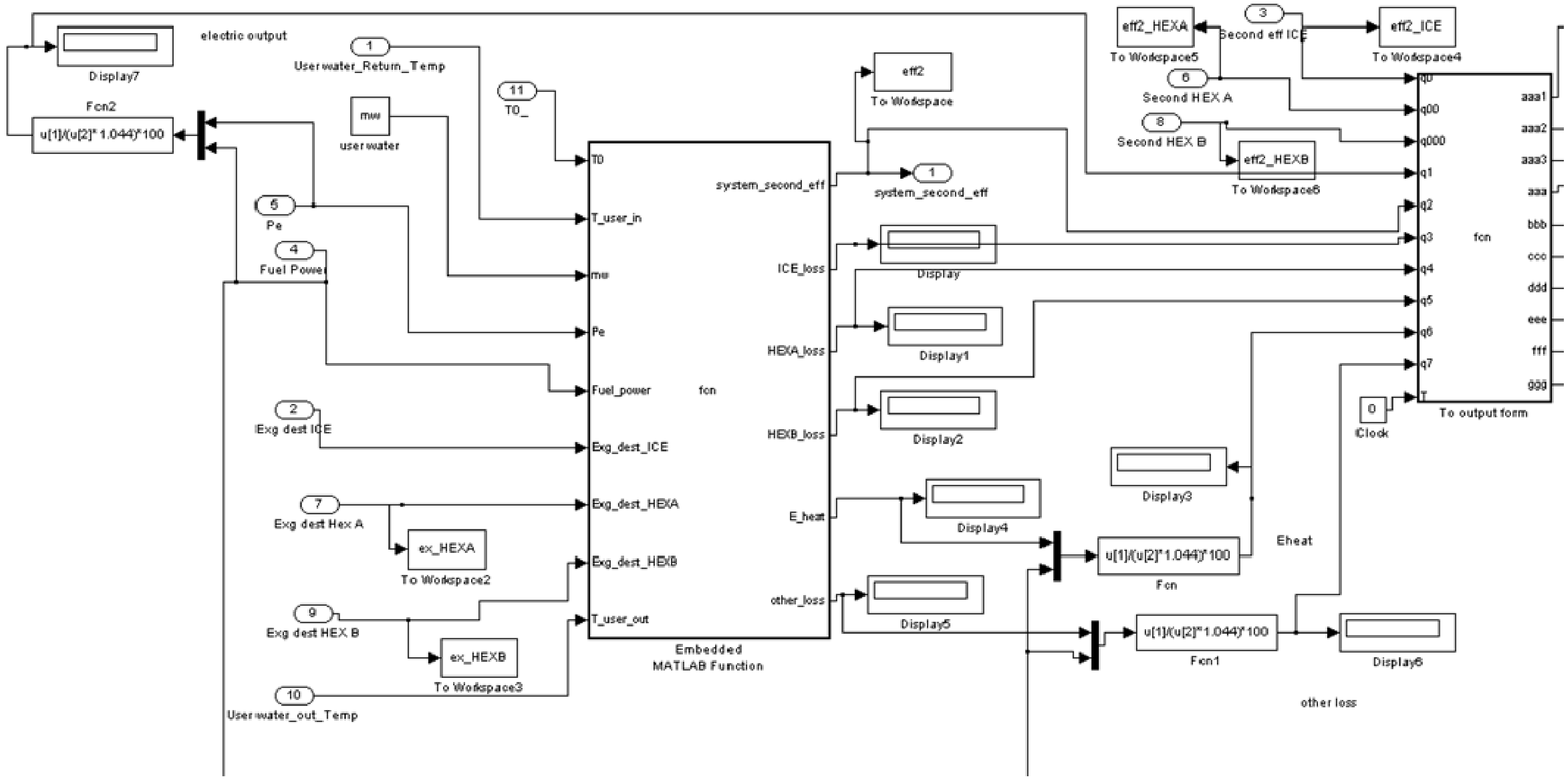

Figure I.13 Details of MATLAB Simulink Modeling –Exergy Analysis (CASE 1)

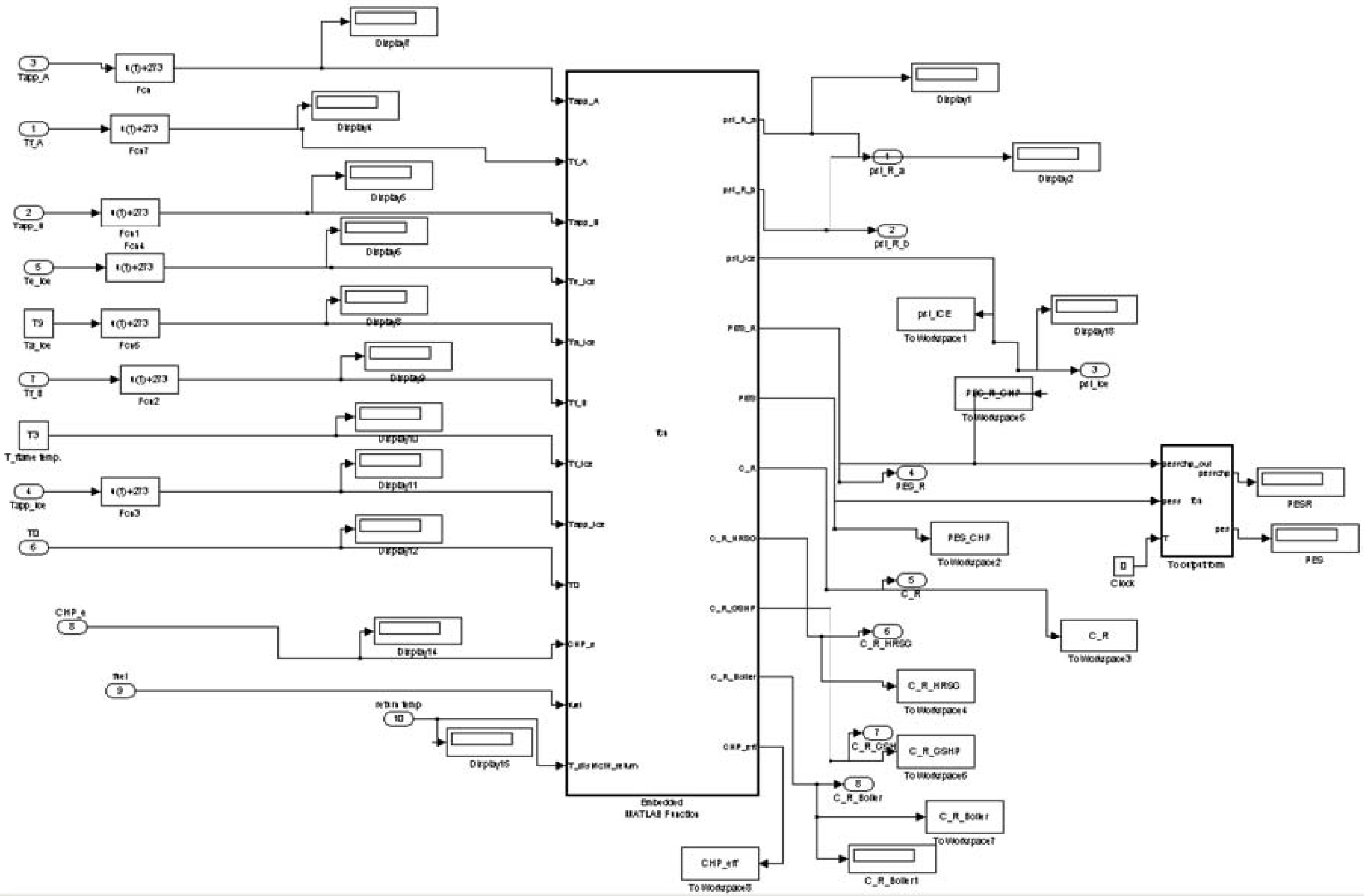

Figure I.14 Details of MATLAB Simulink Modeling –REMM Analysis (CASE 1)

APPENDIX J

MEASUREMENTS, AUXILIARY EQUIPMENT LIST AND COLLECTED DATA

Table J.1. Measurements and Auxiliary Equipment List

Symbol	Control Parameter	Instrument	Nominal range	Value Min÷Max
T_1	User water inlet temperature	RTD	70 °C	35 ÷ 80 °C
T_2	Condenser Outlet temperature	RTD	80 °C	35 ÷ 90 °C
T_3	Jacket water outlet temperature	RTD	95 °C	35 ÷ 100 °C
T_4	User water outlet temperature	RTD	95 °C	55 ÷ 110 °C
$\dot{m}_{th}$	User water mass flow rate	Flow meter	2.5 kg/s	-
$\dot{m}_b$	NG mass flow rate	Gas Flow meter	43 Nm³/h	11÷ 43 Nm³/h
H_i	NG LHV	Gas chromatograph	48900 kJ/kg	-
P_1	Engine electrical production	Wattmeter	126 kW	20 ÷ 130 kW
P_2	NG compressor consumption	Wattmeter	5 kW	1 ÷ 5 kW
$P_3 = 0$	SE compressor consumption	Wattmeter	24 kW	3 ÷ 25 kW
$P_4 = 0$	FWP consumption	Wattmeter	1 kW	0 ÷ 1 kW
P_5	Gross electrical power	grid analyser	150 kW	23 ÷ 155 kW
T_a, rh_a	Ambient temperature and humidity	temperature and humidity sensor	20 °C, 60%	-10 ÷ 50 °C 0 ÷ 100%
p_a	Ambient pressure	Pressure transmitter	1.01bar	0.8 ÷ 1.2 bar
T_b	Fuel temperature	RTD	20 °C	-10 ÷ 50 °C
p_b	Fuel Pressure	Pressure transmitter	1.01bar	0.8 ÷ 1.2 bar

Table J.2 Collected Data

Voltage battery [V] vbatt	Voltage bus [V] vbus	Electrical Generator cooling water temperature [°C] temp	ICE speed [rpm] wm	ICE Oil pressure [Bar] Polio	ICE cooling water temperature inlet [°C] Ta in	ICE exhaust gases temperature outlet [°C] Tf out	Exhaust gases temperature after exchanger [°C] Tf aria	User water temperature outlet [°C] T ut out	User water temperature inlet [°C] T ut in
559	729	31,7	663	1,32	26,35	108,33	35,00	20,53	20,42
557,5	731	31,7	663	2,72	26,34	116,42	37,47	20,51	20,42
557,5	730	31,6	680	5,16	26,41	142,87	38,42	20,61	20,42
557,5	729	31,7	681	5,2	26,38	147,10	39,02	20,61	20,38
557,5	730	31,7	681	5,24	26,39	152,57	39,52	20,58	20,36
559	730	31,6	674	5,24	26,40	167,64	40,55	20,64	20,37
559	730	31,5	672	5,2	26,39	170,02	40,86	20,65	20,44
557,5	728	31,5	672	5,16	26,43	173,25	40,61	20,65	20,39
557,5	729	31,5	672	5,12	26,47	180,90	41,02	20,63	20,42
557,5	730	31,6	676	5,16	26,51	183,20	41,19	20,60	20,45
557,5	729	31,4	676	5,16	26,50	189,33	41,52	20,60	20,33
559	729	31,3	689	5,12	26,55	190,37	41,15	20,65	20,45
556	730	31,2	736	5,36	26,52	191,00	40,26	20,60	20,43
557,5	730	19,1	760	5,44	26,53	191,08	41,13	20,58	20,35
557,5	730	18,9	770	5,48	26,54	189,08	41,53	20,64	20,40
557,5	730	20	934	5,68	26,58	196,81	43,63	20,68	20,40
556	729	20,7	976	5,72	26,62	199,27	43,56	20,74	20,45
557,5	731	21,8	1020	5,8	26,59	209,60	46,02	20,68	20,33
557,5	731	21,8	1018	5,8	26,59	211,64	46,72	20,66	20,38
557,5	730	20,9	1015	5,8	26,61	213,04	46,93	20,67	20,34
557,5	730	22	1013	5,8	26,64	214,21	47,45	20,67	20,38
556	731	22,3	1011	5,8	26,66	219,17	49,32	20,76	20,44
556	731	22,2	1011	5,8	26,67	220,41	49,73	20,71	20,35
559	730	21,7	1013	5,8	26,58	221,32	49,79	20,67	20,31
557,5	730	22,1	1011	5,8	26,65	222,24	50,39	20,71	20,38
557,5	731	21,7	1012	5,8	26,61	222,88	50,56	20,66	20,28
557,5	731	21,9	1009	5,8	26,58	223,91	51,19	20,64	20,31
557,5	730	21,3	1004	5,76	26,66	224,28	51,32	20,69	20,37
557,5	729	22,3	1006	5,8	26,73	224,95	51,55	20,69	20,35
557,5	732	22,4	1001	5,76	26,60	225,84	51,70	20,66	20,32
557,5	729	21,5	1001	5,8	26,71	226,27	52,02	20,71	20,38
559	729	20,9	984	5,76	26,72	230,54	52,32	20,73	20,35
559	729	22,4	984	5,76	26,73	231,03	52,38	20,63	20,31
560,5	729	22,2	981	5,76	26,78	231,34	52,50	20,67	20,36
560,5	728	21,7	983	5,76	26,81	231,66	52,72	20,71	20,37
569	731	20	992	5,76	27,11	250,91	53,11	20,81	20,35
569	730	21,3	989	5,72	27,21	251,05	53,05	20,89	20,34
570	726	21,3	986	5,72	27,19	251,37	53,09	20,84	20,41
569	727	20	987	5,72	27,15	251,80	53,24	20,84	20,31
569	728	20,6	989	5,72	27,13	252,48	53,21	20,84	20,28

Table J.2 **Collected Data, cont.**

570	730	20,1	990	5,72	27,18	252,60	53,32	20,88	20,30
571,5	730	19,3	992	5,72	27,16	253,04	52,92	20,85	20,26
571,5	730	19,9	991	5,72	27,18	253,55	52,96	20,83	20,33
571,5	731	20,1	988	5,72	27,21	253,79	52,99	20,83	20,31
571,5	729	19	989	5,72	27,24	253,97	53,28	20,84	20,28
571,5	727	21,2	991	5,72	27,18	254,15	53,37	20,89	20,33
571,5	731	21,4	995	5,76	27,15	254,49	53,29	20,93	20,37
571,5	730	20	994	5,68	27,13	254,82	53,59	20,83	20,26
571,5	729	19,6	993	5,76	27,24	254,99	53,50	20,95	20,35
574,5	730	20,6	990	5,72	27,29	255,28	53,52	20,89	20,28
574,5	730	21,8	983	5,72	27,18	255,58	53,49	20,86	20,32
574,5	729	19,1	981	5,72	27,24	255,55	53,29	20,93	20,34
574,5	729	19,6	986	5,68	27,30	254,97	53,75	20,89	20,34
574,5	730	21,3	990	5,72	27,36	254,70	53,99	20,99	20,37
574,5	730	19,5	988	5,72	27,34	254,81	54,07	20,98	20,37
576	729	19,5	987	5,72	27,28	254,83	54,20	20,86	20,36
577	728	21,4	992	5,68	27,28	253,91	54,57	20,84	20,23
577	730	20,7	991	5,68	27,35	253,78	54,66	20,90	20,31
578,5	729	19,1	993	5,72	27,26	253,99	54,58	20,77	20,12
578,5	730	18,8	993	5,72	27,40	253,75	55,04	20,95	20,34
583	730	20	988	5,68	27,44	253,22	55,36	20,99	20,38
581,5	730	19,4	988	5,68	27,40	252,62	55,36	20,94	20,27
583	730	18,6	987	5,68	27,38	252,48	55,42	20,99	20,29
583	730	18,5	995	5,68	27,40	251,74	55,15	20,95	20,27
584,5	728	18,5	994	5,72	27,38	251,82	55,36	20,93	20,27
584,5	729	20,8	989	5,68	27,51	251,37	55,50	21,01	20,37
584,5	729	21,3	992	5,68	27,46	251,19	55,62	20,98	20,27
594	729	20,7	1006	5,56	28,57	251,17	60,19	21,03	20,32
595,5	729	20	1006	5,56	28,81	255,85	60,57	21,03	20,38
595,5	728	19,9	1003	5,56	28,78	256,33	60,77	20,97	20,30
594	729	20,3	1006	5,52	28,84	256,46	60,75	21,01	20,37
594	728	22,2	1007	5,56	28,84	256,86	60,93	21,02	20,32
597	730	19,8	1003	5,56	28,83	257,22	61,01	21,00	20,37
598,5	734	19,8	997	5,56	28,79	257,56	60,95	21,03	20,31
598,5	730	21,9	995	5,6	28,90	257,62	61,26	21,07	20,41
597	730	19,4	1002	5,52	28,84	257,92	61,48	20,96	20,35
598,5	731	21,2	1002	5,52	28,86	258,26	61,41	21,03	20,37
597	730	19,4	997	5,6	28,95	258,47	61,82	21,03	20,35
595,5	729	19,9	1004	5,6	28,90	258,88	62,14	21,02	20,34
594	729	22,3	1008	5,56	29,12	262,21	62,58	21,09	20,44
594	730	22,5	1010	5,52	29,12	262,57	62,31	21,02	20,43
594	727	20,8	1005	5,52	29,10	262,92	62,17	21,06	20,43
597	729	20,2	1002	5,52	29,16	263,09	62,44	21,04	20,42
597	729	20,1	1000	5,56	29,13	263,61	62,85	20,96	20,33
597	729	21,6	1002	5,56	29,13	263,97	63,00	21,00	20,35
595,5	729	21,5	1009	5,52	29,12	264,31	63,14	20,99	20,38
576	722	19,5	1004	5,52	29,19	264,80	63,03	21,01	20,39
587	731	19,5	1007	5,52	29,27	264,90	63,11	21,02	20,42
588,5	729	20,5	1003	5,52	29,18	265,52	63,13	20,99	20,32
576	726	21,8	996	5,48	29,26	265,70	63,62	21,02	20,34
573	729	21,5	999	5,48	29,29	266,14	63,59	21,11	20,39
573	728	21	997	5,52	29,33	266,41	63,75	21,13	20,50
571,5	730	19,9	966	5,44	29,43	265,17	65,96	21,02	20,39
571,5	730	19,9	968	5,44	29,47	264,41	66,73	21,15	20,45

Table J.2 Collected Data, cont.

571,5	728	23,1	969	5,44	29,47	263,35	67,70	21,14	20,47
571,5	729	20,2	964	5,44	29,41	262,89	68,38	21,11	20,37
571,5	729	21,9	969	5,44	29,50	262,12	69,28	21,14	20,39
573	733	19	964	5,44	29,50	261,93	69,74	21,15	20,39
573	730	22,3	962	5,44	29,47	261,60	70,60	21,12	20,37
573	730	21,3	969	5,4	29,44	261,77	70,85	21,13	20,37
573	729	19,7	982	5,44	29,46	261,97	71,28	21,12	20,45
574,5	730	20,1	976	5,44	29,63	262,60	71,97	21,21	20,41
574,5	738	20,8	971	5,44	29,63	263,02	71,98	21,25	20,44
574,5	734	20,8	969	5,44	29,59	263,49	72,74	21,21	20,38
574,5	734	20,8	975	5,44	29,67	263,62	73,16	21,22	20,41
574,5	734	21,9	982	5,44	29,66	264,13	73,49	21,24	20,43
576	730	19,6	980	5,44	29,61	264,57	73,68	21,18	20,39
581,5	731	19	981	5,4	29,77	267,84	74,85	21,26	20,48
580	732	21,9	984	5,4	29,77	268,84	75,01	21,28	20,38
580	730	20,9	981	5,44	29,81	269,32	74,94	21,29	20,41
581,5	731	22,5	978	5,44	29,87	269,91	75,41	21,31	20,40
581,5	732	22,5	982	5,44	29,92	270,54	75,65	21,45	20,49
581,5	733	19,4	988	5,4	29,86	271,92	75,14	21,31	20,41
583	733	22,2	989	5,44	29,88	272,72	75,07	21,32	20,39
584,5	731	21,7	987	5,44	29,92	273,47	74,75	21,38	20,40
584,5	730	21,1	990	5,4	29,97	274,55	74,77	21,33	20,39
584,5	728	21,5	996	5,44	29,97	275,49	74,71	21,33	20,41
593	727	18,8	996	5,4	30,17	283,19	75,24	21,39	20,40
595,5	729	21,6	995	5,4	30,32	283,78	75,59	21,48	20,53
593	729	22,3	995	5,4	30,22	284,35	75,67	21,49	20,50
593	729	21,3	1000	5,4	30,21	285,03	75,65	21,36	20,37
593	727	23,5	1001	5,44	30,21	285,46	75,65	21,43	20,41
594	730	20	996	5,44	30,25	286,16	75,75	21,48	20,46
594	730	19,3	994	5,44	30,28	286,60	75,69	21,39	20,39
595,5	731	23,6	995	5,36	30,50	287,01	76,11	21,52	20,50
597	729	23,6	999	5,4	30,37	287,55	76,78	21,40	20,45
600	730	22	993	5,36	30,41	287,96	77,22	21,44	20,41
598,5	725	20,4	996	5,44	30,45	288,58	77,46	21,50	20,43
601,5	724	23,7	1005	5,36	30,69	290,39	78,37	21,57	20,47
600	727	23,7	1004	5,36	30,62	291,24	78,39	21,53	20,55
601,5	730	23,4	999	5,4	30,59	291,74	78,50	21,51	20,47
601,5	730	22,1	993	5,4	30,53	292,54	78,71	21,47	20,55
602,5	729	22,1	994	5,32	30,63	292,73	78,90	21,48	20,50
604	731	19,9	1004	5,4	30,67	293,03	78,94	21,57	20,45
602,5	726	22,4	1002	5,32	30,63	293,83	79,64	21,44	20,45
602,5	726	23,3	998	5,36	30,66	294,16	79,61	21,45	20,48
607	730	22,4	988	5,32	30,71	294,92	79,44	21,41	20,35
605,5	731	22,6	1008	5,32	30,77	294,97	79,62	21,47	20,47
598,5	726	19,6	1013	5,36	30,79	295,52	79,68	21,52	20,45
600	728	22,6	1009	5,4	30,80	296,04	79,47	21,45	20,48
602,5	730	23,6	1002	5,36	30,79	296,71	79,26	21,53	20,46
602,5	729	21,8	1003	5,4	30,89	297,03	79,87	21,52	20,52
604	730	20,8	1000	5,28	30,91	297,45	79,84	21,52	20,49
602,5	730	20,8	1000	5,24	30,92	298,06	79,99	21,57	20,53
601,5	729	21,1	1007	5,32	30,92	298,52	80,18	21,58	20,44
601,5	730	23,7	1008	5,32	30,89	299,16	79,97	21,49	20,44
601,5	728	22,5	1007	5,32	31,05	299,62	79,90	21,55	20,50
601,5	730	18,8	1007	5,32	30,97	300,42	79,97	21,48	20,42

Table J.2 Collected Data, cont.

602,5	730	19,2	1003	5,36	31,09	300,78	80,19	21,55	20,51
601,5	733	21,9	1008	5,28	31,07	301,36	80,27	21,49	20,42
600	729	20,7	1009	5,32	31,08	301,75	80,74	21,57	20,46
600	729	20,3	1011	5,32	31,12	302,15	80,86	21,58	20,50
598,5	730	23	1012	5,32	31,19	302,71	80,83	21,54	20,50
602,5	730	24,3	1003	5,32	31,09	303,21	80,87	21,46	20,46
604	727	24,3	1000	5,32	31,19	303,89	81,35	21,53	20,49
602,5	730	21,1	1005	5,32	31,25	304,11	81,50	21,55	20,50
600	729	21,2	1012	5,28	31,22	304,66	81,49	21,52	20,48
601,5	732	23,2	1007	5,32	31,15	305,10	81,50	21,50	20,45
601,5	729	20,4	1005	5,24	31,24	305,80	81,70	21,50	20,47
601,5	728	19,8	1007	5,28	31,27	306,09	81,94	21,54	20,44
604	730	20,1	1000	5,24	31,29	306,63	82,00	21,58	20,45
602,5	731	23,1	1000	5,24	31,30	307,35	82,69	21,57	20,48
607	729	23,3	994	5,2	31,48	309,47	83,93	21,62	20,49
608,5	729	19,7	993	5,16	31,51	309,76	84,42	21,59	20,56
601,5	729	22,7	1011	5,24	31,49	310,17	84,49	21,54	20,50
600	729	23,3	1013	5,24	31,55	310,61	84,61	21,61	20,54
598,5	729	20,3	1014	5,24	31,51	311,32	84,39	21,56	20,47
593	730	22,3	1016	5,24	31,55	312,16	84,30	21,57	20,43
594	729	20,9	1087	5,28	31,55	312,64	84,02	21,48	20,46
593	729	20,2	1098	5,36	31,64	314,96	84,25	21,59	20,51
594	728	21,5	1153	5,4	31,68	316,84	83,91	21,57	20,54
591,5	730	22,7	1166	5,48	31,64	319,59	83,66	21,60	20,42
591,5	730	22,6	1161	5,52	31,71	320,84	83,65	21,64	20,47
593	730	22,6	1164	5,52	31,79	322,45	83,55	21,69	20,53
591,5	730	21,7	1155	5,56	31,75	323,47	83,81	21,70	20,58
594	728	23,6	1158	5,52	31,79	324,40	83,64	21,65	20,49
591,5	729	20,1	1159	5,52	31,77	325,79	83,83	21,58	20,42
593	730	19,1	1155	5,52	31,82	326,41	84,12	21,61	20,45
593	730	23,3	1152	5,52	31,86	327,66	84,11	21,71	20,47
591,5	731	19,2	1158	5,52	31,89	328,44	83,87	21,63	20,44
591,5	728	20,1	1159	5,52	31,95	329,55	84,37	21,61	20,44
590	729	23,4	1158	5,52	31,96	330,14	84,51	21,68	20,49
591,5	730	22,6	1151	5,52	32,01	331,05	84,58	21,67	20,55
593	730	24	1151	5,48	31,93	331,73	84,65	21,62	20,43
594	730	24	1147	5,48	32,06	332,44	84,74	21,77	20,48
597	730	23,1	1143	5,48	32,05	333,49	84,78	21,65	20,51
595,5	729	22,6	1145	5,48	32,07	334,17	84,44	21,66	20,47
595,5	728	19,4	1148	5,48	32,02	335,17	83,97	21,73	20,46
595,5	729	20,8	1150	5,48	32,20	337,21	83,78	21,78	20,51
594	729	23,4	1160	5,48	32,20	338,11	83,85	21,71	20,44
595,5	727	19,8	1148	5,48	32,26	339,64	84,43	21,73	20,46
602,5	730	23,2	1135	5,48	32,29	340,77	84,59	21,78	20,47
591,5	729	24,9	1162	5,48	32,38	342,15	84,12	21,75	20,50
598,5	727	23,6	1150	5,48	32,42	343,13	84,63	21,73	20,49
595,5	728	22,7	1151	5,48	32,48	345,44	84,78	21,72	20,49
594	729	20,8	1153	5,48	32,50	346,42	84,78	21,75	20,43
600	731	23,5	1147	5,44	32,55	346,98	85,19	21,80	20,49
598,5	729	23,5	1152	5,44	32,62	349,07	84,47	21,80	20,48
595,5	729	21,1	1162	5,48	32,67	349,73	84,74	21,72	20,42
598,5	730	23,5	1149	5,48	32,66	350,47	85,44	21,72	20,42
598,5	730	21,2	1151	5,48	32,67	352,10	86,25	21,73	20,46
598,5	730	21,3	1150	5,48	32,80	353,64	86,07	21,79	20,50

Table J.2 Collected Data, cont.

600	729	23,5	1148	5,44	32,82	355,10	85,86	21,72	20,39
597	730	24	1155	5,44	32,87	355,87	86,04	21,76	20,51
595,5	730	21,1	1153	5,44	32,93	358,02	87,52	21,72	20,41
600	730	25,3	1148	5,48	32,94	358,51	87,94	21,69	20,42
602,5	730	21,4	1140	5,44	33,09	359,21	88,52	21,73	20,52
593	731	23,3	1157	5,44	33,60	372,57	92,60	21,69	20,46
595,5	728	25	1148	5,44	33,67	374,29	92,78	21,76	20,39
598,5	726	21,8	1149	5,4	33,83	374,77	92,86	21,69	20,47
591,5	729	22,6	1158	5,4	33,75	375,18	92,99	21,73	20,40
591,5	731	25,1	1156	5,4	33,91	376,46	93,03	21,74	20,45
594	725	21	1152	5,4	33,91	377,29	93,19	21,74	20,48
601,5	730	21,8	1139	5,36	34,05	378,57	93,00	21,76	20,44
598,5	727	23,6	1144	5,36	33,98	379,12	93,89	21,68	20,36
594	730	24,7	1159	5,36	34,08	379,55	94,09	21,77	20,50
591,5	727	20,9	1162	5,4	34,10	381,05	94,52	21,80	20,37
593	730	23,1	1151	5,36	34,24	382,41	94,71	21,72	20,44
595,5	729	20,5	1147	5,36	34,26	383,67	94,80	21,73	20,40
594	728	23,9	1155	5,36	34,35	383,94	95,15	21,73	20,47
593	728	22,4	1156	5,36	34,28	384,67	95,59	21,72	20,42
597	730	25,3	1150	5,36	34,43	385,60	95,47	21,81	20,42
595,5	729	23,2	1148	5,36	34,35	386,10	95,93	21,75	20,42
593	729	20,9	1152	5,32	34,42	386,22	96,35	21,75	20,40
593	730	22	1157	5,36	34,47	387,52	96,69	21,64	20,34
593	730	21,9	1154	5,32	34,59	388,41	96,92	21,79	20,40
593	730	22,3	1150	5,32	34,57	388,81	96,57	21,78	20,38
595,5	729	22,9	1150	5,32	34,66	389,57	96,41	21,77	20,47
594	729	21,9	1151	5,32	34,62	389,94	96,50	21,69	20,39
594	730	21,9	1150	5,32	34,72	390,62	96,95	21,80	20,41
597	730	22,6	1148	5,32	34,77	391,73	97,82	21,77	20,35
595,5	730	22	1151	5,32	34,75	391,96	98,12	21,70	20,37
594	730	23,5	1156	5,32	34,85	392,60	98,81	21,81	20,44
595,5	729	23,6	1148	5,32	34,89	393,13	98,74	21,76	20,46
595,5	730	22,3	1151	5,28	34,87	393,98	97,79	21,79	20,38
594	729	20,9	1154	5,32	34,91	394,35	98,40	21,72	20,36
594	731	25,7	1153	5,32	34,95	394,72	98,54	21,73	20,47
595,5	731	24,8	1148	5,28	35,01	395,41	99,01	21,74	20,45
594	730	24	1150	5,28	35,10	395,63	99,07	21,76	20,45
594	729	25,9	1150	5,28	35,07	396,37	99,34	21,70	20,34
595,5	730	24,9	1153	5,28	35,11	396,70	99,49	21,70	20,37
593	729	22,9	1159	5,24	35,58	402,86	101,70	21,81	20,42
602,5	729	21,8	1135	5,16	35,69	403,46	101,89	21,80	20,42
595,5	729	21,7	1153	5,12	35,59	403,78	102,48	21,71	20,34
595,5	730	24,7	1159	5,2	35,67	404,42	103,38	21,74	20,40
597	731	22,6	1146	5,2	35,74	404,92	103,18	21,82	20,38
604	731	24,5	1135	5,12	35,75	405,22	103,25	21,72	20,36
594	729	24,3	1158	5,12	35,84	405,69	104,58	21,76	20,37
595,5	730	25,7	1154	5,16	35,87	406,28	104,45	21,75	20,41
598,5	730	24,9	1157	5,16	36,05	409,18	105,45	21,76	20,35
598,5	730	22,2	1155	5,12	36,09	409,55	105,77	21,80	20,37
598,5	728	24,7	1150	5,12	36,22	409,91	105,61	21,78	20,37
601,5	727	22,2	1150	5,12	36,15	410,18	105,29	21,76	20,35
600	727	23	1153	5,12	36,22	410,34	105,86	21,80	20,37
597	730	21,3	1154	5,08	36,24	410,63	106,15	21,74	20,36
597	730	25,9	1160	5,12	36,35	410,94	106,54	21,76	20,37

Table J.2 Collected Data, cont.

597	730	22,1	1157	5,12	36,37	411,73	106,84	21,82	20,42
595,5	730	23,2	1155	5,12	36,35	412,15	106,59	21,74	20,36
595,5	726	24	1160	5,12	36,39	412,71	106,00	21,71	20,32
595,5	726	22,8	1153	5,12	36,46	413,16	105,41	21,73	20,33
567,5	728	23,7	1151	5,08	36,71	415,87	106,06	21,78	20,31
566	728	21,9	1156	5,08	36,79	415,95	106,14	21,70	20,30
566	730	24	1144	5,04	36,78	416,42	106,27	21,75	20,33
566	730	20,7	1127	4,96	36,87	416,06	106,66	21,67	20,42
566	730	23,6	1133	4,92	36,94	415,87	107,42	21,78	20,28
563	732	23,2	1132	4,92	37,00	416,15	109,35	21,81	20,35
563	731	23,5	1140	4,92	37,02	416,29	110,06	21,78	20,35
564,5	730	22,6	1124	4,88	37,17	416,64	112,10	21,88	20,39
564,5	729	23,9	1127	4,88	37,26	416,02	111,39	21,91	20,39
564,5	730	21,1	1131	4,88	37,13	416,07	111,65	21,88	20,37
562	732	26,3	1134	4,88	37,24	416,12	112,68	21,84	20,44
563	730	23,1	1138	4,92	37,24	416,50	113,36	21,85	20,34
564,5	731	22,7	1138	4,88	37,31	416,77	113,84	21,99	20,41
563	730	25,9	1141	4,88	37,39	416,91	114,66	21,96	20,38
562	727	24,9	1140	4,88	37,29	417,28	115,50	21,86	20,30
564,5	728	24,5	1129	4,84	37,41	418,26	117,11	21,90	20,29
564,5	729	24,9	1125	4,84	37,46	418,07	115,82	21,90	20,34
563	731	25,4	1135	4,84	37,52	417,95	114,86	22,00	20,36
563	730	22,7	1137	4,84	37,60	418,33	115,05	21,96	20,42
564,5	724	27,1	1134	4,84	37,63	419,05	115,30	22,01	20,31
566	729	21,6	1129	4,84	37,63	419,49	115,48	21,97	20,35
566	731	25,9	1133	4,8	37,73	420,40	115,37	22,02	20,35
566	729	23,1	1139	4,84	37,81	420,75	115,61	22,04	20,43
566	726	26,2	1141	4,84	37,88	421,55	117,20	22,05	20,32
566	726	21,8	1141	4,8	37,83	422,17	117,90	22,09	20,33
566	730	21,6	1140	4,84	37,95	422,89	118,63	22,09	20,40
566	729	25,5	1137	4,8	37,94	423,36	118,79	22,07	20,30
564,5	730	22,9	1147	4,84	37,95	423,76	118,99	22,03	20,34
566	729	24,8	1151	4,84	38,11	425,32	121,22	22,04	20,33
567,5	730	23,1	1139	4,84	38,09	425,83	121,73	22,05	20,32
570	731	22,3	1130	4,8	38,24	425,80	121,80	22,17	20,42
569	728	25,6	1129	4,76	38,29	425,71	120,30	22,23	20,47
567,5	732	25,9	1152	4,8	38,20	426,86	120,73	22,05	20,34
569	729	25,3	1139	4,8	38,32	427,47	121,38	22,16	20,35
571,5	728	25,8	1123	4,76	38,33	427,60	121,01	22,18	20,38
570	733	25,7	1141	4,72	38,42	427,26	119,31	22,13	20,33
566	729	25,6	1162	4,76	38,52	427,81	119,71	22,16	20,35
567,5	731	22,7	1167	4,8	38,57	429,80	121,97	22,29	20,35
574,5	730	25,8	1129	4,76	38,61	430,50	122,90	22,21	20,38
576	731	26,5	1129	4,72	38,63	430,05	121,69	22,18	20,37
574,5	730	27,5	1140	4,68	38,65	429,92	121,03	22,15	20,35
570	730	23,4	1148	4,72	38,75	430,36	121,41	22,25	20,36
574,5	730	27,7	1141	4,72	38,77	430,96	122,04	22,23	20,36
574,5	731	23,4	1135	4,72	38,80	431,33	121,71	22,26	20,30
573	724	24,5	1147	4,68	38,84	431,58	121,47	22,18	20,36
573	729	22,4	1151	4,72	38,87	432,51	122,29	22,32	20,35
576	727	27,2	1141	4,72	38,88	433,40	123,02	22,21	20,32
577	728	28,2	1137	4,68	38,95	433,93	123,57	22,25	20,34
577	730	26,6	1138	4,68	39,01	433,91	123,39	22,25	20,42

Table J.2 **Collected Data, cont.**

578,5	730	26,6	1134	4,68	39,11	433,84	123,59	22,32	20,41
574,5	732	27,7	1155	4,76	39,14	435,78	125,70	22,26	20,31
576	730	26,4	1146	4,68	39,12	436,37	126,65	22,22	20,26
580	730	23,8	1135	4,68	39,26	436,66	126,48	22,32	20,38
583	731	22,9	1133	4,64	39,35	436,32	125,76	22,30	20,34
574,5	729	23	1160	4,64	39,29	436,73	125,21	22,26	20,28
576	729	24,2	1161	4,68	39,41	437,37	126,64	22,26	20,35
578,5	730	23,9	1156	4,72	39,39	438,73	128,22	22,28	20,30
583	729	25,5	1134	4,68	39,49	438,96	128,59	22,38	20,34
584,5	728	26	1131	4,64	39,53	438,68	127,34	22,26	20,40
580	729	22,5	1146	4,6	39,60	438,77	126,61	22,31	20,33
580	730	26,7	1157	4,64	39,61	439,63	127,54	22,35	20,34
585,5	729	26,5	1135	4,64	39,59	440,30	127,79	22,34	20,28
584,5	728	27,7	1138	4,6	39,67	440,30	127,19	22,35	20,41
581,5	727	26,7	1156	4,6	39,75	440,43	126,87	22,37	20,37
580	730	24,5	1159	4,64	39,74	441,16	127,73	22,37	20,35
581,5	730	27,6	1158	4,64	39,81	442,15	129,59	22,41	20,39
580	730	26,3	1158	4,64	39,80	442,93	130,22	22,31	20,32
580	729	24,1	1156	4,68	39,88	444,08	131,07	22,36	20,37
583	731	28,1	1148	4,64	39,94	444,40	131,44	22,39	20,39
585,5	727	28,3	1132	4,6	39,93	444,50	130,57	22,32	20,33
581,5	729	22,9	1162	4,6	40,31	446,07	129,86	22,37	20,35
583	731	23,2	1157	4,6	40,29	447,52	131,38	22,42	20,39
587	730	24,3	1147	4,56	40,31	448,10	132,03	22,44	20,44
585,5	729	26,9	1143	4,56	40,36	448,55	131,36	22,40	20,37
585,5	729	26,2	1147	4,56	40,34	448,66	131,02	22,37	20,36
585,5	731	23,1	1151	4,56	40,47	450,35	131,58	22,40	20,39
585,5	730	24,4	1151	4,56	40,58	450,95	131,79	22,35	20,42
583	730	24,2	1159	4,56	40,67	451,48	132,11	22,41	20,38
580	728	26,9	1164	4,56	40,72	452,53	133,11	22,43	20,42
581,5	728	24,7	1160	4,56	40,68	453,34	134,06	22,34	20,36
580	731	24,9	1162	4,56	40,72	454,11	134,93	22,39	20,29
583	731	23,4	1149	4,56	40,80	454,56	134,97	22,38	20,35
588,5	730	25,2	1136	4,48	40,94	454,35	133,61	22,42	20,39
584,5	727	26,9	1147	4,48	40,90	454,33	133,21	22,36	20,32
580	727	22,9	1164	4,48	40,97	454,94	133,42	22,44	20,47
581,5	730	26,7	1156	4,52	41,00	456,92	134,79	22,40	20,39
583	728	23,6	1156	4,52	41,03	457,19	134,42	22,32	20,27
587	729	26,8	1143	4,44	41,11	457,71	134,31	22,43	20,36
585,5	728	26,5	1147	4,48	41,19	457,72	133,98	22,30	20,33
581,5	730	27,5	1155	4,48	41,28	458,85	133,71	22,40	20,38
583	730	23,8	1159	4,52	41,30	459,41	134,10	22,51	20,44
585,5	729	27,1	1144	4,48	41,34	459,86	133,67	22,40	20,30
590	730	24,5	1135	4,44	41,34	459,89	133,97	22,54	20,35
590	729	28,5	1138	4,44	41,56	461,63	134,44	22,44	20,38
588,5	731	24,2	1141	4,4	41,73	461,64	134,53	22,51	20,40
594	730	24,5	1136	4,4	41,64	461,85	134,18	22,49	20,44
594	729	24,4	1133	4,36	41,64	462,00	134,17	22,38	20,40
593	730	23,4	1137	4,36	41,67	462,44	134,43	22,44	20,37
591,5	730	27,1	1150	4,36	41,76	462,74	134,21	22,48	20,47
590	730	27,1	1155	4,36	41,82	463,51	134,28	22,46	20,36
598,5	731	25,2	1132	4,36	42,00	466,49	136,80	22,47	20,37
601,5	730	27,4	1130	4,32	42,07	466,47	136,66	22,52	20,41

Table J.2 Collected Data, cont.

595,5	728	26,3	1147	4,36	42,12	466,60	136,62	22,43	20,40
587	729	27,2	1170	4,4	42,08	467,32	136,67	22,45	20,38
591,5	728	24,1	1157	4,4	42,23	468,21	137,51	22,51	20,43
605,5	731	27,3	1130	4,36	42,23	469,24	138,00	22,48	20,43
611	734	27,8	1117	4,32	42,23	469,13	137,91	22,50	20,39
601,5	729	27	1133	4,28	42,32	469,64	136,65	22,53	20,43
585,5	728	24,7	1178	4,32	42,43	470,11	136,44	22,57	20,53
585,5	731	26,6	1179	4,4	42,43	472,16	139,44	22,51	20,41
593	727	24,9	1155	4,44	42,43	472,72	140,22	22,47	20,42
604	729	26,7	1133	4,36	42,51	472,67	140,13	22,46	20,45
598,5	731	28,3	1142	4,32	42,58	472,40	139,08	22,52	20,40
595,5	733	24,5	1149	4,32	42,56	472,57	139,16	22,49	20,42
601,5	725	26,3	1149	4,36	42,67	473,69	138,73	22,45	20,38
597	730	27,2	1144	4,36	42,74	474,02	138,74	22,47	20,41
601,5	729	26	1147	4,28	42,83	474,13	138,73	22,52	20,45
591,5	732	24,9	1149	4,28	42,77	474,81	139,28	22,48	20,37
597	728	27,5	1142	4,28	42,92	475,32	139,12	22,46	20,45
604	732	27,6	1131	4,28	43,08	475,59	139,36	22,59	20,50
611	727	27,9	1128	4,24	43,13	476,28	138,33	22,61	20,55
604	729	28	1142	4,24	43,24	476,74	137,81	22,53	20,57
598,5	729	24,9	1152	4,28	43,26	477,18	137,93	22,56	20,54
617	734	27,9	1131	4,24	43,43	477,89	139,17	22,54	20,66
612,5	729	27,3	1132	4,2	43,60	478,40	138,12	22,64	20,70
605,5	728	28	1151	4,2	43,58	479,07	137,51	22,53	20,67
595,5	730	24,6	1177	4,36	43,70	480,72	139,61	22,50	20,66
595,5	727	27,9	1158	4,32	43,90	481,90	140,66	22,49	20,71
594	729	26,7	1160	4,36	44,01	481,99	141,24	22,52	20,76
594	730	29,4	1162	4,36	44,11	482,34	141,35	22,54	20,76
594	729	27,2	1157	4,36	44,09	482,83	141,94	22,53	20,77
594	729	25,5	1155	4,36	44,18	482,96	141,99	22,57	20,74
591,5	731	25,7	1164	4,36	44,24	483,35	142,31	22,52	20,77
593	731	25,7	1156	4,4	44,31	483,90	142,61	22,56	20,82
600	730	25,3	1141	4,4	44,36	484,16	142,82	22,50	20,74
598,5	727	25,7	1136	4,36	44,55	484,16	142,77	22,60	20,80
598,5	730	24,3	1141	4,32	44,56	484,52	143,26	22,63	20,74
601,5	728	28,2	1141	4,36	44,55	484,75	143,56	22,54	20,69
601,5	731	25,8	1148	4,36	44,82	485,39	143,19	22,72	20,79
593	730	28	1160	4,36	44,77	485,77	142,84	22,60	20,66
595,5	727	29,4	1163	4,4	44,80	486,61	142,76	22,61	20,70
591,5	725	29,4	1180	4,44	44,95	487,08	143,29	22,65	20,74
594	731	25,6	1169	4,48	44,98	488,55	144,99	22,67	20,72
597	730	25,5	1159	4,48	45,05	488,95	144,96	22,65	20,72
588,5	728	28,3	1177	4,52	45,23	489,47	144,81	22,65	20,70
593	729	23,9	1161	4,48	45,25	490,18	144,97	22,67	20,68
594	737	27,4	1150	4,48	45,32	490,47	145,18	22,65	20,65
605,5	730	26	1132	4,4	45,37	490,27	145,25	22,66	20,76
600	729	24,4	1140	4,44	45,43	490,06	146,12	22,65	20,63
595,5	728	24,4	1155	4,4	45,52	490,28	146,15	22,66	20,64
594	731	26,1	1162	4,44	45,55	490,71	145,95	22,62	20,62
595,5	729	28,3	1162	4,48	45,68	491,39	145,74	22,72	20,71
598,5	731	24,5	1148	4,48	45,74	491,75	145,96	22,75	20,70
597	728	26,5	1153	4,44	45,82	491,77	145,95	22,76	20,66
594	729	25,7	1164	4,44	45,79	492,39	146,54	22,70	20,62

Table J.2 Collected Data, cont.

594	730	27	1154	4,44	45,85	493,11	146,60	22,71	20,60
597	730	25,8	1157	4,44	45,90	493,45	146,32	22,74	20,58
601,5	730	28,5	1149	4,4	45,89	493,69	146,86	22,58	20,53
598,5	729	26,3	1149	4,4	46,03	493,83	147,03	22,71	20,65
595,5	729	25,9	1157	4,4	46,15	493,92	146,84	22,79	20,62
597	730	28,5	1157	4,44	46,10	494,56	146,36	22,76	20,61
594	728	28,3	1162	4,44	46,13	494,90	147,17	22,66	20,54
595,5	728	27,3	1154	4,44	46,24	495,54	147,58	22,77	20,64
594	730	25,8	1152	4,44	46,30	495,70	147,97	22,74	20,55
594	730	26,2	1157	4,44	46,25	496,24	148,35	22,76	20,63
594	730	25,7	1154	4,44	46,32	496,50	148,69	22,72	20,62
601,5	730	28,5	1141	4,4	46,48	496,81	149,38	22,67	20,62
608,5	730	26,7	1125	4,36	46,47	497,19	149,50	22,73	20,61
600	729	28,3	1140	4,32	46,45	497,69	149,19	22,61	20,57
602,5	730	24,8	1143	4,36	46,52	497,93	149,01	22,71	20,54
601,5	730	24,8	1148	4,36	46,58	498,18	149,29	22,73	20,56
600	731	28,3	1156	4,36	46,56	498,67	149,16	22,68	20,46
614	731	26,4	1130	4,4	46,71	498,99	149,01	22,74	20,60
618	728	27,1	1125	4,32	46,69	499,35	148,94	22,70	20,54
611	729	26,3	1144	4,28	46,71	500,21	146,19	22,71	20,60
617	729	28	1139	4,36	46,75	501,07	148,47	22,62	20,37
611	730	26,1	1144	4,36	46,86	501,26	148,84	22,79	20,55
600	730	26,1	1166	4,36	46,97	501,61	148,65	22,80	20,49
602,5	727	26,4	1164	4,4	46,97	502,09	149,15	22,80	20,51
604	729	28,1	1159	4,4	46,97	502,45	149,38	22,72	20,50
604	729	25	1157	4,4	47,00	502,74	149,91	22,76	20,53
608,5	731	26,6	1152	4,4	47,03	503,14	149,94	22,81	20,44
611	732	29,6	1148	4,36	46,98	503,31	150,47	22,84	20,47
610	728	30,4	1147	4,36	47,11	503,48	150,64	22,71	20,48
601,5	728	30,5	1163	4,36	47,12	503,65	150,09	22,76	20,44
602,5	728	28,4	1164	4,4	47,21	504,06	151,45	22,73	20,56
602,5	727	25,8	1161	4,4	47,23	504,53	151,64	22,79	20,48
601,5	730	25,1	1162	4,4	47,25	504,93	151,05	22,66	20,47
595,5	728	25	1170	4,4	47,31	505,23	151,69	22,74	20,51
595,5	730	28,8	1223	4,44	47,30	505,96	151,63	22,66	20,49
600	729	27,5	1234	4,56	47,34	509,45	155,22	22,71	20,47
612,5	731	29,8	1275	4,64	47,44	511,30	156,16	22,91	20,57
611	729	29,7	1298	4,72	47,47	515,13	157,44	22,83	20,43
601,5	728	28,4	1322	4,8	47,42	517,20	157,61	22,94	20,47
663,5	733	29,1	1301	4,96	47,54	519,84	156,01	22,94	20,37
624	695,5	25,7	1370	4,92	47,61	520,53	155,55	23,05	20,49
569	725	26	1522	5,2	47,63	520,65	155,31	22,96	20,47
557,5	728	28,8	1555	5,32	47,66	517,43	153,07	22,56	20,44
556	728	27,2	1536	5,36	47,75	516,08	153,21	22,44	20,45
554,5	727	27,6	1524	5,36	47,76	514,42	153,19	22,25	20,46
552	730	28	1497	5,32	47,83	513,05	153,79	22,11	20,45
547,5	728	27,3	1248	4,64	49,15	517,76	155,09	22,16	20,32
546	730	28,4	1252	4,6	49,20	518,08	156,61	22,23	20,42
547,5	728	25,9	1259	4,6	49,27	518,72	158,05	22,26	20,42
549	731	25,7	1257	4,6	49,31	519,18	158,79	22,23	20,46
547,5	730	24,6	1264	4,6	49,28	519,84	160,68	22,31	20,32
547,5	729	28,6	1273	4,64	49,30	520,26	161,06	22,31	20,37
549	732	28	1271	4,64	49,40	520,90	160,99	22,36	20,40
549	731	29,8	1266	4,64	49,39	521,33	160,88	22,39	20,34

Table J.2 **Collected Data, cont.**

550,5	730	28,4	1269	4,64	49,45	521,88	160,89	22,39	20,39
549	729	25,6	1276	4,64	49,46	522,27	160,36	22,45	20,44
549	727	27,1	1288	4,68	49,51	522,78	160,17	22,49	20,35
550,5	729	26,1	1280	4,64	49,53	523,14	159,95	22,51	20,34
550,5	727	26,9	1280	4,64	49,51	523,45	160,24	22,53	20,38
549	725	26,9	1290	4,64	49,58	523,67	160,46	22,48	20,37
549	728	28,8	1291	4,68	49,52	524,04	160,33	22,46	20,30
552	731	30,6	1271	4,68	49,69	524,22	160,41	22,48	20,45
554,5	730	26,4	1264	4,6	49,70	524,39	160,03	22,57	20,36
553,5	732	29,1	1263	4,56	49,67	524,50	160,48	22,45	20,33
553,5	730	28,7	1275	4,56	49,73	524,68	160,43	22,57	20,34
553,5	729	30,5	1279	4,6	49,77	525,12	160,05	22,62	20,30
553,5	731	29,2	1281	4,6	49,81	525,50	159,50	22,62	20,37
553,5	730	29,3	1282	4,6	49,90	526,15	159,39	22,76	20,36
556	729	26,9	1278	4,6	49,92	526,43	159,30	22,66	20,31
557,5	727	30,3	1275	4,56	49,89	526,71	159,14	22,67	20,40
557,5	728	30,3	1270	4,56	49,91	527,13	159,63	22,79	20,44
556	728	27,2	1283	4,56	49,93	527,50	159,86	22,74	20,34
554,5	729	26,8	1297	4,64	50,05	528,06	159,56	22,86	20,41
557,5	733	29,9	1286	4,6	50,09	528,36	159,05	22,79	20,31
559	730	28,1	1274	4,6	50,02	528,83	158,93	22,77	20,39
559	730	29,2	1275	4,56	50,09	528,90	159,78	22,81	20,38
557,5	729	27,5	1281	4,56	50,15	528,95	161,08	22,88	20,39
556	728	27,1	1305	4,6	50,23	529,18	161,80	22,95	20,38
557,5	731	25,3	1290	4,6	50,20	529,63	161,53	22,88	20,33
563	731	31,5	1268	4,56	50,21	529,96	161,08	22,79	20,40
560,5	731	27	1272	4,52	50,33	529,89	161,82	22,91	20,37
557,5	727	27	1263	4,52	50,30	529,74	163,62	22,93	20,40
557,5	732	25,2	1294	4,52	50,31	529,91	164,90	22,84	20,32
557,5	730	27,3	1304	4,6	50,47	530,56	164,28	22,99	20,54
562	731	25,5	1279	4,64	50,52	530,88	163,96	22,90	20,46
563	730	27	1268	4,56	50,59	530,73	165,13	22,90	20,53
562	731	30,8	1272	4,48	50,58	530,69	165,53	22,82	20,44
560,5	732	28,7	1284	4,48	50,63	530,74	165,96	22,85	20,49
562	730	27,4	1302	4,56	50,65	531,12	166,21	22,78	20,38
562	730	27,1	1295	4,64	50,83	531,30	166,29	22,95	20,59
560,5	732	26,9	1293	4,52	50,89	531,36	166,26	22,85	20,56
564,5	733	27,3	1280	4,56	50,92	531,39	166,08	22,83	20,59
564,5	731	27,3	1278	4,52	50,92	531,22	166,46	22,92	20,62
563	730	29,3	1297	4,52	51,02	531,17	167,13	23,02	20,67
559	727	26,7	1311	4,6	50,98	531,39	167,55	22,84	20,59
559	730	28,4	1315	4,68	50,98	531,37	166,83	23,03	20,63
560,5	730	28,9	1298	4,68	51,12	531,36	166,21	22,94	20,60
560,5	727	29,5	1296	4,56	51,14	531,09	165,49	22,88	20,59
578,5	730	28	1312	4,68	52,96	531,28	170,03	23,29	20,32
578,5	732	31,9	1307	4,68	52,86	531,74	171,06	23,22	20,26
581,5	730	28,1	1302	4,76	52,92	532,07	171,98	23,24	20,42
581,5	731	31,9	1298	4,72	53,12	532,50	171,78	23,33	20,38
578,5	729	30,5	1309	4,64	53,06	533,01	171,85	23,34	20,34
576	728	31,1	1318	4,68	53,06	533,70	172,30	23,26	20,27
578,5	730	31,1	1306	4,72	53,15	533,96	173,19	23,22	20,34
588,5	731	30,1	1280	4,68	53,12	534,25	173,12	23,30	20,30
590	729	28,4	1269	4,6	53,10	534,58	172,39	23,19	20,30
588,5	729	27,3	1273	4,56	53,23	534,77	170,60	23,20	20,33

Table J.2 Collected Data, cont.

585,5	730	29,9	1290	4,56	53,25	535,54	169,06	23,32	20,36
588,5	730	27,5	1295	4,64	53,22	536,41	169,07	23,31	20,32
585,5	729	27,4	1305	4,72	53,24	537,49	169,37	23,33	20,34
574,5	727	27,4	1333	4,72	53,30	538,24	169,24	23,27	20,31
580	731	30,1	1318	4,76	53,26	538,83	170,66	23,26	20,18
594	732	30,2	1273	4,64	53,36	538,94	171,69	23,39	20,29
597	727	28,6	1261	4,56	53,41	539,22	170,97	23,26	20,29
587	726	28,2	1290	4,56	53,44	539,39	169,29	23,34	20,37
577	727	28,2	1316	4,52	53,35	540,66	169,31	23,22	20,21
580	729	30,1	1328	4,68	53,43	541,30	170,51	23,29	20,25
585,5	730	26,3	1305	4,68	53,45	541,62	171,19	23,27	20,23
588,5	730	26,9	1291	4,72	53,51	542,33	171,66	23,32	20,23
583	727	30,8	1302	4,64	53,53	542,75	172,12	23,22	20,28
580	729	27,4	1323	4,68	53,55	543,61	173,10	23,30	20,27
587	732	28,3	1302	4,68	53,59	543,86	173,84	23,25	20,24
593	730	30,7	1282	4,64	53,61	544,39	174,46	23,38	20,26
580	726	27,7	1305	4,64	53,66	544,68	173,46	23,32	20,30
580	728	26,9	1318	4,68	53,68	545,41	174,52	23,25	20,26
584,5	730	28,8	1299	4,72	53,63	545,79	175,24	23,25	20,20
585,5	730	28,8	1298	4,72	53,66	546,12	176,03	23,29	20,22
583	730	30,2	1299	4,68	53,74	546,47	175,77	23,34	20,21
588,5	730	32,5	1296	4,64	53,84	546,94	175,78	23,30	20,24
583	729	26,8	1302	4,52	53,78	547,57	175,57	23,27	20,20
593	731	30,7	1285	4,52	53,80	547,94	176,26	23,21	20,23
602,5	731	30,6	1262	4,48	53,89	548,12	175,20	23,25	20,26
581,5	727	31	1304	4,56	53,89	548,10	174,25	23,31	20,28
581,5	729	29,8	1331	4,64	53,94	549,38	174,01	23,28	20,21
588,5	730	29	1308	4,64	53,93	549,78	175,38	23,31	20,24
598,5	732	28,5	1282	4,52	53,93	550,39	176,08	23,35	20,26
597	732	28,3	1274	4,48	53,96	550,29	175,59	23,33	20,20
587	730	32,3	1299	4,52	54,12	550,49	174,28	23,37	20,27
585,5	730	28,2	1319	4,56	54,09	551,29	174,59	23,27	20,21
588,5	730	28,2	1309	4,6	54,04	552,14	176,30	23,35	20,25
588,5	728	30,8	1303	4,6	54,14	552,45	176,27	23,37	20,22
590	730	30,9	1303	4,6	54,11	552,77	176,34	23,36	20,26
597	730	30,4	1286	4,56	54,25	552,94	176,85	23,41	20,26
594	729	31,8	1286	4,52	54,20	553,21	175,92	23,37	20,20
600	731	32	1280	4,52	54,22	553,66	175,03	23,33	20,25
604	730	32,1	1276	4,44	54,35	553,69	174,27	23,42	20,22
593	727	28,4	1302	4,48	54,32	554,44	173,24	23,42	20,31
583	730	28,1	1335	4,56	54,27	555,17	173,31	23,32	20,20
583	731	29,8	1335	4,64	54,31	555,68	174,62	23,37	20,32
593	729	29,9	1292	4,56	54,39	555,76	176,26	23,41	20,25
593	726	28,9	1288	4,52	54,40	555,85	177,49	23,50	20,29
595,5	728	28,9	1285	4,52	54,40	556,22	177,41	23,25	20,18
600	725	29	1279	4,44	54,42	556,20	176,99	23,30	20,23
598,5	730	31,5	1284	4,44	54,38	556,84	175,48	23,17	20,06
595,5	729	27,5	1290	4,48	54,55	557,07	174,57	23,36	20,26
588,5	727	30,8	1319	4,52	54,62	557,78	174,43	23,39	20,28
590	727	32	1316	4,52	54,59	558,27	176,07	23,33	20,28
583	728	30,7	1324	4,56	54,57	558,90	177,34	23,41	20,25
580	729	27,1	1336	4,6	54,68	558,95	179,01	23,45	20,33
587	731	30,1	1312	4,6	54,67	559,06	179,95	23,44	20,26
597	727	28,5	1283	4,6	54,62	559,24	181,92	23,35	20,27

Table J.2 **Collected Data, cont.**

588,5	723	29,1	1294	4,48	54,69	559,26	181,23	23,29	20,25
591,5	725	29,1	1296	4,52	54,64	559,78	180,05	23,24	20,20
595,5	727	30,4	1290	4,36	54,76	559,90	179,87	23,34	20,26
600	724	32,9	1284	4,4	54,71	560,37	178,58	23,29	20,18
591,5	731	33	1291	4,44	54,82	560,61	177,46	23,28	20,26
584,5	728	28,5	1323	4,4	54,85	561,47	177,54	23,33	20,24
578,5	727	29,5	1314	4,52	54,83	561,79	179,67	23,33	20,26
577	725	29,4	1321	4,56	54,90	561,74	180,77	23,46	20,31
584,5	732	29,5	1300	4,56	55,00	561,89	181,99	23,47	20,38
585,5	730	32,5	1284	4,48	54,97	562,04	182,60	23,30	20,40
583	730	28,5	1288	4,48	55,02	562,08	181,07	23,33	20,38
597	728	32,6	1263	4,4	55,14	562,19	180,75	23,26	20,43
595,5	727	28,6	1265	4,32	55,14	561,79	178,77	23,25	20,42
594	728	28,6	1271	4,32	55,16	561,69	177,97	23,30	20,48
584,5	729	30,2	1299	4,36	55,22	562,60	177,63	23,23	20,48
584,5	729	28,6	1313	4,48	55,26	563,72	177,52	23,19	20,46
595,5	731	30,3	1296	4,44	55,27	564,34	178,93	23,24	20,48
604	730	27,5	1265	4,36	55,35	564,60	179,06	23,27	20,55
594	731	32,9	1276	4,36	55,42	564,03	178,46	23,26	20,57
578,5	726	29,9	1326	4,44	55,45	564,90	178,76	23,36	20,55
581,5	730	31	1328	4,56	55,42	565,70	180,34	23,32	20,53
581,5	732	31	1316	4,6	55,56	566,12	182,61	23,44	20,64
587	733	31,7	1300	4,6	55,54	566,31	183,27	23,33	20,62
584,5	732	31,3	1292	4,56	55,55	566,58	183,21	23,32	20,60
587	733	31,3	1291	4,6	55,61	566,69	183,02	23,28	20,60
588,5	729	30,8	1285	4,52	55,54	566,81	182,01	23,26	20,53
587	729	30,8	1287	4,48	55,63	567,18	180,55	23,27	20,56
587	730	29,9	1296	4,44	55,64	567,56	180,46	23,29	20,57
590	732	30,9	1292	4,48	55,73	567,99	180,20	23,33	20,59
585,5	730	28,8	1303	4,44	55,83	568,28	179,51	23,32	20,62
585,5	729	31	1306	4,48	55,79	569,05	179,79	23,34	20,56
578,5	729	28,8	1328	4,56	55,87	569,30	180,30	23,42	20,53
583	730	29,2	1315	4,64	55,94	569,57	182,09	23,43	20,60
588,5	729	33,1	1298	4,64	55,96	569,61	183,01	23,46	20,66
585,5	730	28,8	1289	4,56	56,02	569,66	182,81	23,35	20,57
590	730	28,8	1292	4,56	55,99	569,78	182,15	23,31	20,53
593	730	32,4	1297	4,56	56,09	569,78	181,60	23,41	20,53
584,5	729	30,4	1321	4,64	56,12	570,65	181,56	23,46	20,53
584,5	731	31,5	1321	4,64	56,11	570,97	183,00	23,43	20,50
585,5	727	31,7	1312	4,68	56,20	571,07	184,56	23,27	20,53
587	728	29,2	1309	4,68	56,33	571,11	184,70	23,46	20,62
600	731	29,9	1280	4,56	56,33	571,26	185,64	23,52	20,62
587	728	28,6	1292	4,52	56,26	571,19	184,34	23,41	20,43
580	730	29,6	1322	4,6	56,37	571,85	183,48	23,44	20,49
580	730	32,1	1336	4,68	56,38	572,18	184,03	23,51	20,50
584,5	725	29,7	1313	4,72	56,40	572,38	185,18	23,44	20,50
584,5	727	29,7	1300	4,68	56,42	572,46	187,05	23,41	20,43
587	729	32,4	1290	4,68	56,41	572,32	187,42	23,39	20,53
594	731	32,1	1274	4,56	56,60	572,18	186,01	23,45	20,52
583	730	33	1300	4,52	56,59	572,33	184,85	23,54	20,49
587	728	31,8	1308	4,6	56,60	573,05	185,91	23,50	20,47
596,5	726	31	1283	4,68	56,68	573,40	185,46	23,52	20,49
584,5	728	32,4	1298	4,56	56,67	573,60	185,08	23,46	20,46
583	729	29,3	1317	4,6	56,64	574,04	184,39	23,42	20,42

Table J.2 Collected Data, cont.

587	726	31,2	1306	4,6	56,74	574,26	185,61	23,56	20,44
601,5	729	29,5	1274	4,64	56,81	574,51	186,84	23,50	20,51
598,5	729	31,6	1266	4,52	56,85	574,18	186,07	23,49	20,53
590	726	31,6	1290	4,52	56,80	574,02	183,86	23,42	20,45
588,5	730	31,3	1307	4,52	56,81	574,77	183,84	23,46	20,46
593	730	33	1300	4,6	56,81	575,60	184,15	23,43	20,36
591,5	728	33	1293	4,68	56,88	575,66	184,42	23,50	20,44
597	729	31,8	1296	4,64	56,89	576,21	183,93	23,49	20,36
602,5	731	29,7	1288	4,56	56,92	576,35	183,77	23,56	20,42
588,5	727	32,8	1302	4,52	56,93	576,40	183,27	23,54	20,38
583	729	31,8	1335	4,6	56,88	576,87	184,56	23,48	20,36
588,5	732	29,9	1321	4,68	56,96	576,99	186,68	23,50	20,38
578,5	727	29,8	1329	4,68	57,02	576,92	189,02	23,43	20,34
580	731	31,8	1326	4,72	56,97	576,92	190,46	23,41	20,34
585,5	731	31,8	1301	4,68	56,99	576,73	192,01	23,53	20,36
601,5	731	29,5	1261	4,6	57,04	576,43	191,59	23,42	20,36
602,5	730	31,5	1254	4,48	57,02	575,19	187,98	23,40	20,34
600	729	32,5	1264	4,44	57,09	574,78	185,79	23,48	20,42
591,5	729	32,1	1293	4,52	57,09	575,12	184,79	23,43	20,35
591,5	728	29,9	1309	4,6	57,07	576,67	183,60	23,44	20,33
588,5	730	29,7	1311	4,56	57,14	577,49	185,12	23,47	20,34
588,5	725	33,5	1312	4,6	57,18	578,10	186,87	23,47	20,34
587	728	30,5	1313	4,6	57,21	578,37	188,08	23,57	20,32
590	730	30,1	1307	4,56	57,18	578,51	188,86	23,44	20,31
597	730	28,8	1291	4,6	57,19	578,52	189,65	23,45	20,32
594	729	28,8	1287	4,56	57,26	578,49	188,39	23,46	20,42
590	729	31,9	1299	4,48	57,20	578,64	187,81	23,51	20,40
590	729	31	1305	4,52	57,26	578,78	187,67	23,51	20,40
587	729	30,1	1315	4,56	57,24	579,19	188,12	23,52	20,32
583	727	29,8	1318	4,6	57,25	579,24	189,54	23,46	20,33
593	731	28,5	1297	4,64	57,26	579,23	190,50	23,41	20,28
588,5	729	31,2	1296	4,56	57,32	579,14	190,78	23,48	20,40
587	729	31,5	1301	4,6	57,36	579,03	190,37	23,40	20,36
588,5	730	29,7	1302	4,64	57,36	579,14	189,94	23,44	20,32
591,5	731	32,6	1300	4,68	57,38	579,24	189,90	23,49	20,33
585,5	730	29,9	1307	4,56	57,38	579,47	189,93	23,56	20,36
584,5	730	29,9	1313	4,56	57,37	579,58	190,09	23,57	20,37
593	732	33,1	1293	4,56	57,38	579,49	192,23	23,38	20,27
590	729	30,3	1289	4,44	57,52	579,16	191,91	23,44	20,35
590	730	29,4	1296	4,56	57,63	579,31	191,43	23,55	20,37
593	729	29,5	1294	4,48	57,43	579,62	191,13	23,50	20,35
587	729	32,2	1309	4,44	57,50	579,98	190,23	23,47	20,35
591,5	734	29,9	1298	4,56	57,55	580,23	190,85	23,51	20,32
602,5	732	32,3	1270	4,52	57,61	580,00	191,29	23,48	20,39
605,5	728	31,2	1259	4,4	57,59	579,11	190,11	23,53	20,35
590	728	29,9	1292	4,36	57,59	578,24	189,50	23,41	20,33
590	729	30,8	1321	4,44	57,62	579,25	188,07	23,49	20,37
587	729	30,8	1322	4,48	57,72	580,10	189,28	23,49	20,40
590	730	30,4	1318	4,56	57,71	580,57	192,12	23,49	20,33
594	729	33,3	1299	4,44	57,68	580,89	192,13	23,49	20,28
583	729	30,4	1328	4,48	57,78	581,03	192,11	23,50	20,38
587	729	30,4	1314	4,56	57,72	581,27	192,96	23,55	20,28
601,5	729	29,1	1276	4,44	57,85	580,76	193,87	23,50	20,42
594	729	30,5	1281	4,4	57,86	580,42	192,55	23,50	20,39

Table J.2 **Collected Data, cont.**

602,5	730	30,2	1272	4,36	57,82	579,92	191,37	23,54	20,37
605,5	725	34,1	1274	4,36	57,86	579,67	189,75	23,54	20,31
593	726	33,5	1305	4,4	57,85	580,05	189,32	23,46	20,31
595,5	729	30,3	1311	4,48	57,98	580,86	189,34	23,60	20,37
593	730	29,5	1312	4,44	57,93	581,90	190,16	23,53	20,29
584,5	729	30,4	1327	4,52	57,99	582,19	190,81	23,54	20,37
584,5	730	30,4	1329	4,52	57,94	582,44	192,11	23,50	20,34
594	730	33,7	1298	4,56	58,00	582,33	193,53	23,52	20,34
601,5	726	31,1	1275	4,44	58,04	581,90	193,43	23,51	20,39
601,5	729	28,8	1274	4,4	58,13	581,24	191,31	23,55	20,39
594	730	30,1	1289	4,36	58,00	581,25	190,47	23,47	20,35
595,5	730	28,8	1299	4,44	58,09	581,82	190,45	23,56	20,36
602,5	730	30,4	1297	4,44	58,14	582,32	190,08	23,58	20,37
594	728	31,1	1304	4,28	58,13	582,78	190,11	23,58	20,34
578,5	726	29,1	1357	4,4	58,07	583,22	190,83	23,52	20,32
593	733	32,5	1319	4,56	58,19	583,31	192,43	23,65	20,34
604	731	33,2	1272	4,44	58,17	583,20	193,70	23,52	20,34
608,5	731	33,2	1260	4,4	58,18	582,52	192,57	23,48	20,33
605,5	730	31	1264	4,32	58,23	581,34	190,39	23,53	20,37
602,5	729	30,4	1287	4,36	58,30	581,24	189,00	23,49	20,36
601,5	730	28,9	1299	4,4	58,29	582,39	189,09	23,58	20,30
593	729	30,5	1328	4,44	58,27	583,28	188,68	23,60	20,35
584,5	729	30,2	1316	4,48	58,37	583,88	190,62	23,67	20,45
590	731	32	1305	4,44	58,40	584,31	190,99	23,62	20,48
597	730	30,3	1282	4,44	58,41	584,40	191,71	23,54	20,44
594	731	34	1282	4,4	58,49	583,98	190,41	23,57	20,44
591,5	728	31,4	1298	4,32	58,51	583,83	189,35	23,61	20,49
591,5	731	30,7	1307	4,28	58,54	584,41	189,63	23,42	20,45
601,5	731	31,5	1288	4,4	58,56	584,90	190,29	23,46	20,52
608,5	731	31,5	1265	4,36	58,64	584,39	189,13	23,56	20,56
601,5	729	29,2	1273	4,28	58,70	583,39	188,70	23,52	20,61
595,5	730	29	1299	4,36	58,61	583,67	189,42	23,54	20,49
588,5	730	30,5	1324	4,44	58,75	584,77	190,19	23,60	20,58
588,5	729	30,8	1323	4,48	58,70	585,65	190,70	23,51	20,62
590	730	29,1	1314	4,48	58,78	585,95	192,14	23,61	20,70
585,5	730	30,5	1319	4,48	58,84	586,09	193,05	23,56	20,65
595,5	729	32,7	1288	4,48	58,81	586,12	194,10	23,49	20,59
591,5	729	29,6	1292	4,44	58,88	585,75	193,36	23,50	20,60
605,5	732	32,5	1269	4,36	58,87	585,68	192,25	23,51	20,61
610	731	30,8	1257	4,32	58,91	585,01	191,57	23,55	20,59
605,5	729	30,8	1262	4,28	58,98	583,84	190,71	23,51	20,62
587	728	29,6	1321	4,36	59,06	584,43	190,00	23,62	20,68
590	729	32,4	1324	4,48	59,01	585,75	190,49	23,51	20,61
600	730	34,6	1294	4,48	59,01	586,63	192,08	23,54	20,62
608,5	730	31,1	1270	4,44	59,13	586,43	191,82	23,68	20,66
601,5	729	30,3	1281	4,36	59,14	585,82	190,47	23,51	20,65
590	725	32,1	1318	4,32	59,12	586,04	189,84	23,57	20,61
590	726	30,8	1317	4,48	59,09	587,02	190,21	23,62	20,58
598,5	728	32,4	1304	4,52	59,15	587,40	190,22	23,65	20,66
595,5	730	30,7	1299	4,6	59,30	587,55	189,49	23,59	20,66
587	729	30,4	1314	4,64	59,28	587,65	188,80	23,64	20,62
588,5	726	30,4	1326	4,6	59,30	588,02	188,92	23,59	20,62
593	731	34,1	1304	4,6	59,47	588,39	195,24	23,68	20,58
590	730	34,1	1309	4,64	59,49	588,27	195,42	23,64	20,58

Table J.2 Collected Data, cont.

601,5	726	31,1	1285	4,6	59,70	588,61	195,95	23,63	20,58
593	732	32,9	1311	4,6	60,16	591,39	196,93	23,77	20,44
593	730	33,8	1316	4,56	60,04	591,64	198,25	23,65	20,35
591,5	729	33	1311	4,6	60,04	591,54	199,02	23,62	20,34
591,5	730	31	1301	4,56	60,11	591,43	201,80	23,51	20,30
611	730	32,3	1262	4,48	60,20	591,29	201,43	23,53	20,36
598,5	728	34,3	1293	4,4	60,54	593,14	202,23	23,56	20,25
601,5	729	32,9	1287	4,4	60,60	592,98	201,58	23,56	20,30
593	729	34	1306	4,4	60,62	592,69	200,01	23,46	20,26
591,5	730	32,8	1316	4,48	60,55	593,28	199,96	23,58	20,22
601,5	732	33,5	1298	4,48	60,68	593,41	201,93	23,64	20,33
591,5	725	32,5	1307	4,4	60,60	593,48	201,32	23,54	20,21
591,5	729	33,3	1313	4,48	60,61	593,61	200,47	23,53	20,22
588,5	726	32,2	1317	4,48	60,70	593,61	202,37	23,61	20,28
587	728	34,1	1321	4,48	60,68	593,49	203,02	23,35	20,18
587	730	30,3	1317	4,52	60,68	593,56	205,28	23,50	20,25
593	731	30,8	1288	4,48	60,80	593,28	206,61	23,55	20,33
601,5	728	30,3	1272	4,36	60,80	592,61	205,84	23,45	20,32
601,5	728	33,6	1268	4,32	60,69	592,24	203,01	23,33	20,16
595,5	728	35	1277	4,32	60,70	591,78	201,21	23,37	20,22
588,5	729	31,5	1314	4,4	60,81	592,10	200,34	23,51	20,22
590	728	33,9	1319	4,44	60,86	592,77	201,30	23,55	20,34
591,5	729	33,5	1309	4,48	60,85	593,53	202,50	23,41	20,21
595,5	728	30,2	1300	4,4	60,80	593,61	203,80	23,38	20,16
600	730	35,3	1286	4,32	60,94	593,43	203,98	23,35	20,26
595,5	731	31,6	1287	4,4	60,89	593,26	202,55	23,53	20,25
591,5	729	31,4	1310	4,28	60,90	593,59	203,23	23,52	20,20
587	728	32,8	1317	4,36	60,93	593,84	203,25	23,45	20,25
587	729	33,2	1319	4,48	60,98	594,08	204,60	23,48	20,29
587	731	30,9	1307	4,48	60,96	594,17	204,97	23,49	20,26
585,5	729	33,7	1303	4,44	61,03	593,97	206,10	23,51	20,23
587	727	31,9	1305	4,4	61,01	593,87	207,28	23,39	20,21
598,5	730	33,6	1277	4,36	61,07	593,65	207,67	23,41	20,23
591,5	731	35,3	1288	4,32	61,03	593,07	206,28	23,32	20,24
594	731	34,1	1291	4,28	61,05	593,32	206,17	23,39	20,23
588,5	729	35,8	1321	4,52	62,61	598,70	208,18	23,33	20,04
597	731	33,7	1298	4,56	62,69	598,73	209,11	23,35	20,17
611	728	36,3	1262	4,44	62,71	598,35	208,59	23,33	20,10
588,5	730	33,4	1313	4,52	62,86	599,00	210,27	23,36	20,09
591,5	731	34,7	1297	4,56	62,97	598,75	212,05	23,35	20,16
594	730	34,6	1289	4,56	62,99	598,34	213,19	23,33	20,15
600	733	36,5	1287	4,6	66,25	604,33	215,59	23,60	20,40
594	730	37,4	1307	4,56	66,45	604,50	215,92	23,42	20,20
597	730	38,4	1291	4,36	66,49	604,07	217,97	23,46	20,32
594	731	35,9	1312	4,52	66,53	604,87	217,24	23,43	20,16
590	729	36,4	1309	4,52	66,54	604,62	217,89	23,45	20,18
594	729	35,9	1299	4,52	66,60	604,31	219,48	23,45	20,17
594	732	35,7	1304	4,36	66,56	604,43	220,21	23,48	20,13
590	730	32,9	1309	4,4	66,49	604,42	220,88	23,32	20,18
594	730	33,7	1298	4,36	66,58	604,29	220,61	23,43	20,20
604	731	34,5	1283	4,36	66,69	603,96	220,31	23,38	20,24
598,5	730	35,5	1289	4,36	66,60	603,78	216,67	23,40	20,20
590	730	36,5	1304	4,4	66,58	603,71	214,78	23,39	20,10
588,5	729	32,9	1322	4,44	66,59	604,36	216,22	23,39	20,13

APPENDIX K

ICE CHARACTERISTICS

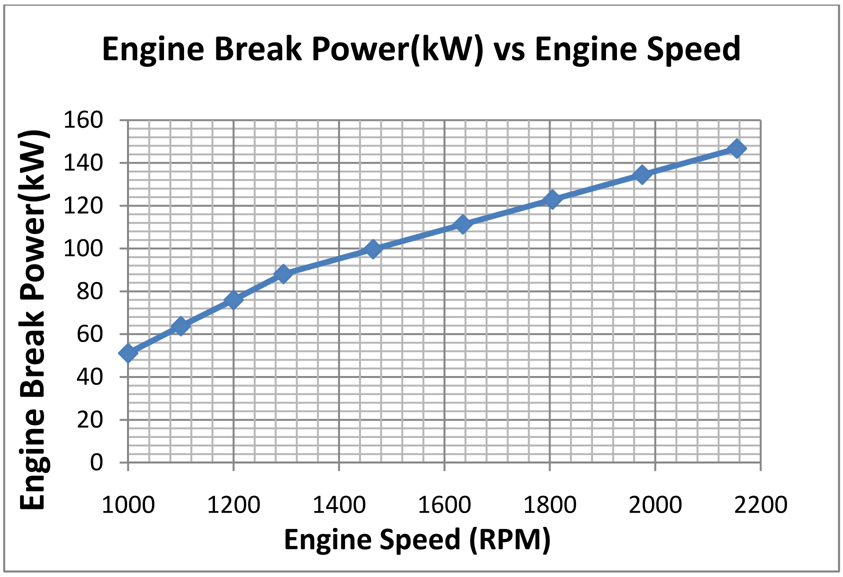

Figure K.1 Engine Break Power vs Engine Speed

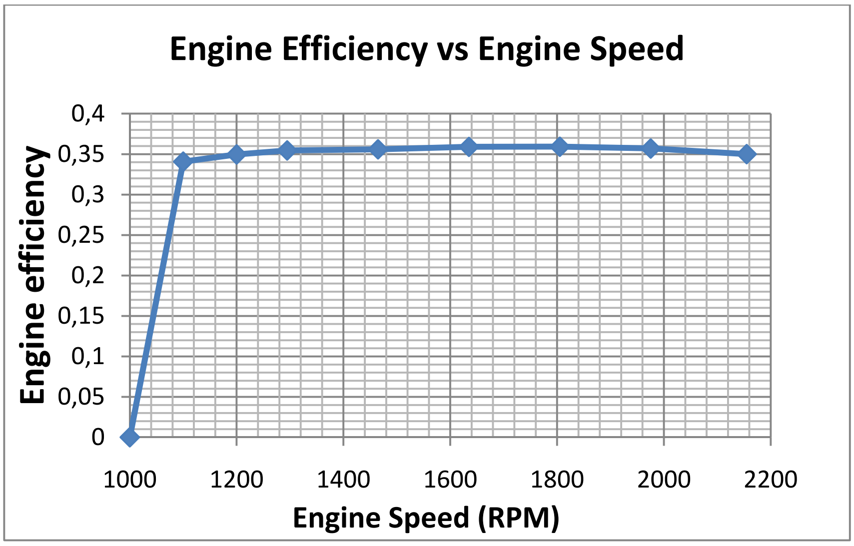

Figure K.2 Engine Efficiency vs Engine Speed

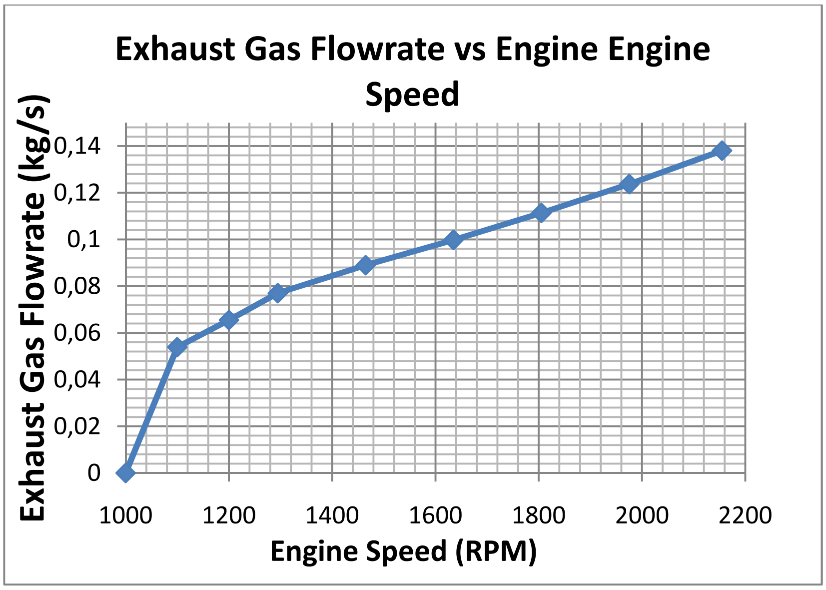

Figure K.3 Exhaust Gas Flowrate vs Engine Speed

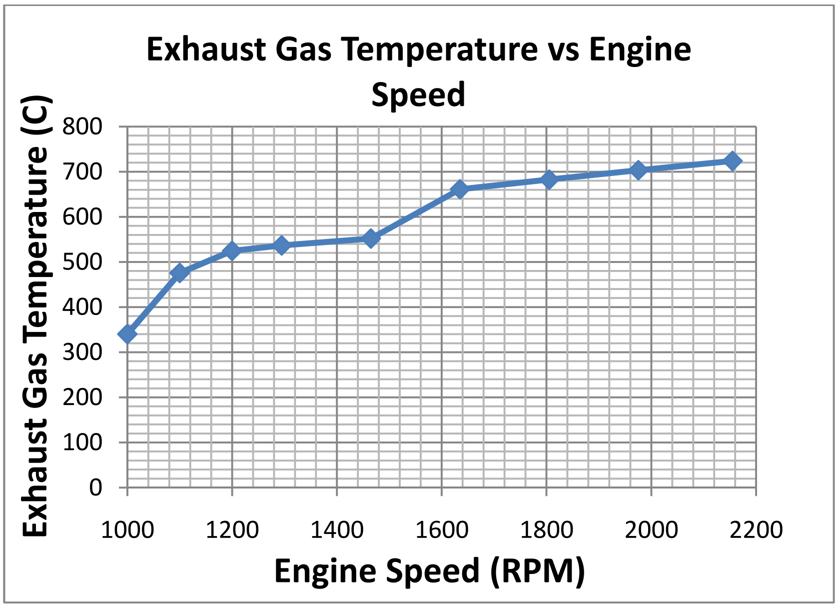

Figure K.4 Exhaust Gas Temperature vs Engine Speed

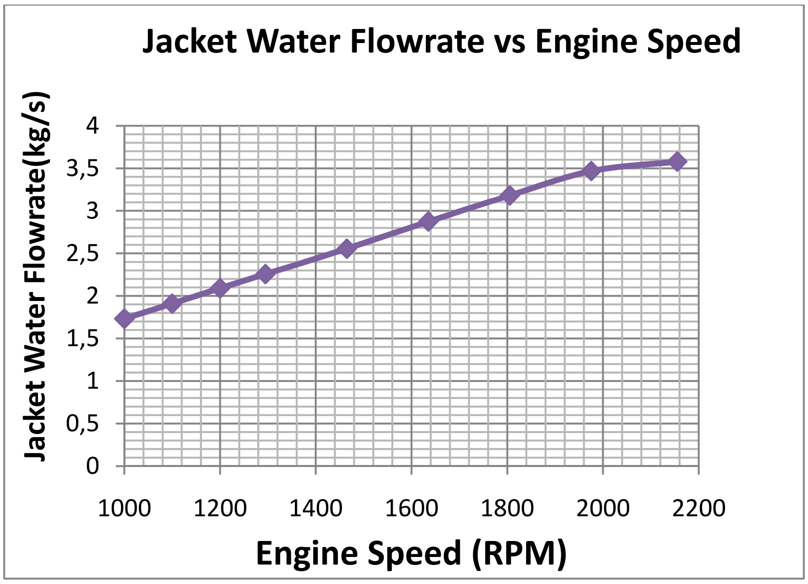

Figure K.5 Jacket Water Flowrate vs Engine Speed

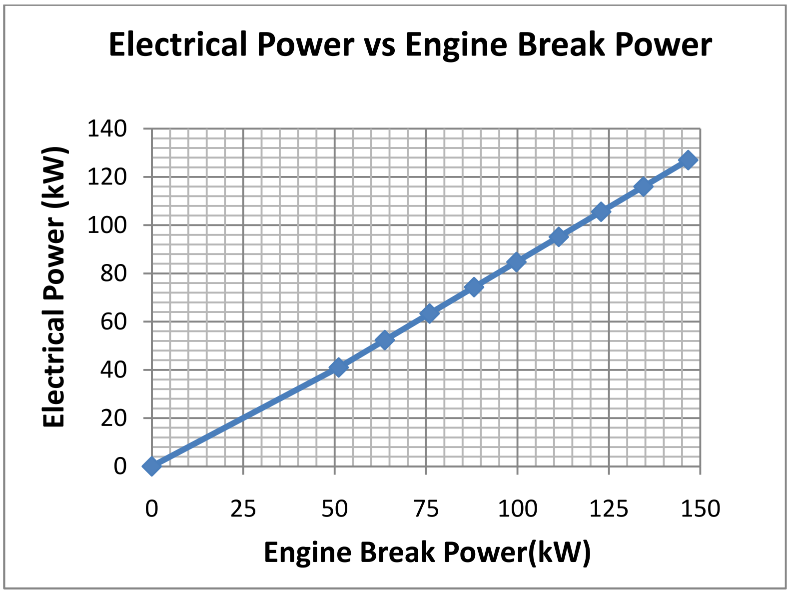

Figure K.6 Electrical Power output of ICE vs Engine Break Power

APPENDIX L

COMPARISON OF RESULTS OF PLANT PERFORMANCE IN TORINO AND IN ANKARA

Table L.1. Comparison of Plant Performances in Torino and in Ankara

	Ankara Case-1	Torino Case-1	Ankara Case-2	Torino Case-2	Maximum difference (%)
maximum heating load (kW)	122	128	122	128	4,4
Total energy capacity(electricity) (kW)	272	272	309	309	0,0
Total energy Capacity (heating) (kW)	449	449	497	497	0,0
Total energy Delivered (electricity) (MWh)	613	625	615	627	1,9
Total energy Delivered (heating) (MWh)	400	419	400	419	4,5
Emissions (tCO2/MWh)	394	401	394	404	2,5
Simple Pay back (year)	2,9	2,9	4,6	4,7	2,1
Equity Pay back (year)	2,5	2,5	3,8	3,8	0,0

CURRICULUM VITAE

PERSONAL INFORMATION

Surname, Name: Ekin Bingöl

Nationality: Turkish (TC)

Date and Place of Birth: 05.03.1979, Ankara

Marital Status: Married with one daughter

Phone: +90 312 2358514

email: ekinbingol@gmail.com

EDUCATION

Degree	Institution	Year of Graduation
MS	METU Mechanical Engineering	2004
BS	METU Mechanical Engineering	2001
High School	Collage Ayşeabla High School	1997

WORK EXPERIENCE

Year	Place	Enrollment
2009-2010	HEGEL Project, 6th Frame work	R&D Engineer
2008-2009	Taru Engineering INC	R&D Engineer
2001-2008	METU Mechanical Engineering	Teaching Assistant
2000 July	ENRON Europe Ltd-London and Teesside Power Plant-Middlesbrough / UK	Intern Engineering Student

1999 August	MAN Bus and Truck Factory, Ankara	Intern Engineering Student

FOREIGN LANGUAGES

Advanced English

PUBLICATIONS

"Comparison of Poly-Generation Systems For Energy Savings, Exergetic Performance And Carbon Footprint", Ekin Bingöl, Ozan Kaya, Birol Kılkış, Cahit Eralp,16th International Energy and Environment Fair ICCI 2010, 12-14 May, İstanbul

"Computer Aided Modeling of Co-generation Systems in High Performance Buildings for Efficiency, Economical and Environmental Aspects", Ozan Kaya, Ekin Bingöl, Birol Kılkış, Cahit Eralp, 16th International Energy and Environment Fair ICCI 2010, 12-14 May, İstanbul

"Different Exergy Analysis Techniques for High Efficiency Poly/Co-Generation Systems for Various Operating Loads", Ekin Bingöl, Ozan Kaya, Birol Kılkış, Cahit Eralp, REHVA World Congress CLIMA 2010 Sustainable Energy Use, R1-TS17-OP05, (pg 17) 9-12 May, Antalya

"Co-generation and Combined Cycle Applications and Utilization of Natural Gas", Ekin (Özgirgin) Bingöl, O. Cahit Eralp, Haluk Direskeneli, II. Natural Gas and Energy Management Congress, (2003), s.231-243

PROJECTS INVOLVED

April 2007-	For BOTAŞ A.Ş.; Effect of Environmental Conditions on Gas Turbine and Compressor Power at Natural Gas Compressor Stations; Ardahan–Hanak Compressor Stations Power Evaluation, İskra – Metot Buisness Partnership
December 2006 -	Internal and External Pressure Resistance Tests for Polyethylene Pipes and Fittings, SENKRON PLASTİK İnş. Tek. San. Tic. Ltd
June 2006-	Nh_{10} Gas Storage Tank Pressure Resistance Test, Türk Telekom Ankara İl Telekom Müdürlüğü

March 2006-	High Pressure Hose Resistance Experiment for ASKİ, GÜNERİ MAKİNA San. Tic. A.Ş.
December 2004- January 2005	GRE Pipe Test Project, BOTAŞ, BTC.
July 2004- September 2004	Feasibility Project Preparation, JTI Factory, ENSİDA

COURSES/ SEMINARS:

EU FP 6 HEGEL Project, Developing, demonstrating and comparing three high efficiency micro poly-generation plants- Final Dissemination Workshop, 17 December 2009, Brussels, BELGIUM

6 Sigma in Design, ARÇELİK, April 2005, ANKARA

Taguchi Method, Improvement of Quality in Production and Process Design, April 2005, ANKARA

E.C.A. Heat Loss and Product Selection for Heating Systems Seminar, May 2000, ANKARA

World Quality Industrial Lubricants Seminar, July 1999, ANKARA

English Course, Regent Brighton Language School, June-July 1998. BRIGHTON, UK

HOBBIES

Swimming, tennis, poetry, piano playing, painting.