

University of Zagreb  
Faculty of Mechanical Engineering and Naval Architecture

Master Thesis

Optimization of waste heat utilization  
in pipeline compressor station

Mentor:  
Doc.dr.sc. Dražen Lončar, dipl.ing.

Author:  
Davor Bišćan, dipl.ing.



Zagreb, 06.03.2009.

## Master Thesis Proposal

Candidate: **Davor Bišćan, dipl.ing.**

Title: **Optimization of waste heat utilization in pipeline compressor station process**

Thesis Contents:

The role of pipeline compressor stations in gas transport system is to maintain the gas pressure at the level required for gas transport to the end-consumers. The gas compressors are driven by gas turbines using so called technical gas taken directly from transport system as a fuel. The gas turbine process utilizes approximately the third of the primary fuel energy while the rest is usually removed via hot flue gases in the environment. In present conditions when the demand for various kind of energy is increasing it is an imperative to evaluate utilization of waste energy from any kind of energy process. Pipeline compressor stations present potential source of additional power and useful heat.

For utilizing the waste heat existing pipeline compressor systems are usually upgraded with a heat recovery steam generator and steam turbine which are used to produce electricity, heat and mechanical energy in conventional combined cycle. In order to define an optimal configuration of the upgraded plant it is necessary to find a compromise between conflicting thermodynamic and economic criteria. Optimal configuration decreases heat losses and enables profitable operation in varying conditions determined by consumption and prices of gas, electricity and heat.

Scope of work:

1. Review of actual investigations, concepts and possibilities of utilizing the waste heat in pipeline compressor stations
2. Defining different configurations of heat recovery steam generator and steam turbines with respect to one, two or three pressure levels.
3. Selection of the preferred configuration by application of multi-attribute decision techniques and with respect to given thermodynamic and economic criteria.
4. Simulation of annual operation of pipeline compressor station and preferred configuration of combined cycle plant.
5. Calculation of profitability indicators and sensitivity analyses with respect to varying investment costs, fuel, electricity, heat and CO<sub>2</sub> emission certificate prices.

Thesis proposed:

Thesis submitted:

Supervisor:

Chairman of Committee for  
postgraduate studies:

Project Director:

Doc.dr.sc. Dražen Lončar

Prof.dr.sc. Tomislav Filetin

Prof.dr.sc. Tonko Čurko

## Acknowledgement

I want to thank my mentor Dr. Dražen Lončar for providing professional support in creation of the thesis in particular at the beginning of the work when the task is focused.

Mr. Bernd Gericke from the company MAN TURBO I want to thank for support in selection of the topic as also for providing relevant input data.

Company ENERSIS I want to thank for providing me a possibility to deal with a realistic problem in this thesis.

Finally I want to thank my family for being my incentive and motivation for completion of this work.

## Abstract

Gas turbine driven pipeline compressor stations employed in existing pipelines and also in those that shall be built due to increasing demand for natural gas present significant potential for production of additional electricity and useful heat by utilizing the waste heat from the gas turbine process. For utilizing the waste heat existing systems are usually upgraded with a heat recovery steam generator, steam turbine and balance of plant equipment, which are used to produce electricity via combined gas and steam cycle process. The thesis analyzes and compares several HRSG configurations both in the terms of thermodynamic and thermoeconomic criteria. In order to find an optimal configuration of a new system an exergo-economic HRSG optimization is conducted for finding the compromise between the exergy losses and investment costs of HRSG surfaces. Initial optimization results show insensibility of the objective function on the cost of surfaces by resulting in zero pinch point. Such result demonstrates low surface costs compared to the cost of exergy losses for given parameters of available flue gases and assumed factors of the objective function. Incorporation of exergy of inlet feed water and outlet gas (which are neglected by author of the selected optimization method) result in max. pinch point and high values of pressure close to critical one. Such result of pressure value is however not feasible for design of the industrial steam turbine. Furthermore tendency of reaching upper limit of pinch point shows inadequacy of this approach. So as to obtain positive but realistic pinch point values a surface exponent is introduced into equation for calculating the HRSG surface cost. Idea is to find the value of the surface exponent obtaining minimal feasible pinch point provided by the heat recovery steam generator manufacturer. Referent minimal feasible pinch point is selected for unfired case. Proper value of the surface exponent is determined on the basic single pressure configuration. Upon defining optimization criteria this value is further used for optimization of other configurations. Predefined single pressure, dual pressure and triple pressure system configurations are further examined in operating regimes of the pipeline compressor station. Thereby are cost of electricity, cash flow and internal rate of return calculated. Comparison of configurations is based on diagram presenting dependency of electricity cost on net system efficiency. Pareto front with respect to both criteria; maximal system efficiency and minimal cost of electricity is constructed. Thereof 2P3 and 3P3 configurations are the closest to the Pareto front whereby the first is characterized by the lowest cost of electricity and the latter by the highest system efficiency. Finally sensitivity analysis has been performed in order to estimate system viability in case of varying fuel prices and CO<sub>2</sub> emission certificates. The results of the performed investigation are based on realistic compressor station data and could be used as a guideline for further research and development in this field.

Keywords: pipeline compressor station, booster station, waste heat utilization, combined cycle, HRSG, exergo-economic optimization

Symbols	Unit	Description
Data		
A	m <sup>2</sup>	surface area
B	kg/s	fuel consumption
c	kJ/kgK	specific heat capacity
C <sub>eco</sub>	\$/m <sup>2</sup>	specific surface cost economizer
C <sub>eva</sub>	\$/m <sup>2</sup>	specific surface cost evaporator
C <sub>sh</sub>	\$/m <sup>2</sup>	specific surface cost superheater
C <sub>eco</sub>	\$	surface cost economizer
C <sub>eva</sub>	\$	surface cost evaporator
C <sub>sh</sub>	\$	surface cost superheater
Δt <sub>m</sub>	°C	mean logarithmic temp. difference
dt <sub>1</sub>	°C	greater temperature difference
dt <sub>2</sub>	°C	smaller temperature difference
exp		surface exponent
E <sub>gas,in</sub>	kW	exergy of gas at HRSG inlet
E <sub>gas,out</sub>	kW	exergy of gas at HRSG outlet
E <sub>Q</sub>	kW	exergy losses due to heat exchange with environment
E <sub>w,in</sub>	kW	feed water exergy at HRSG inlet
E <sub>w,out</sub>	kW	superheated steam exergy at HRSG outlet
fc		system complexity factor (2P, 3P HRSG)
g <sub>o</sub>		heat loss due to heat transfer from HRSG surfaces to environment
g <sub>k</sub>		heat loss due to incomplete combustion caused by chemical reasons
g <sub>m</sub>		heat loss due to incomplete combustion caused by mechanical reasons
g <sub>out</sub>		heat loss of flue gases sensitive heat
g <sub>slag</sub>		heat loss due to exiting physical heat of slag
h	kJ/kg	specific enthalpy
h <sub>g</sub>	kJ/kg	specific enthalpy of flue gas
h <sub>s</sub>	kJ/kg	specific enthalpy of steam
h <sub>L</sub>	kJ/m <sup>3</sup>	air specific enthalpy
h <sub>L0</sub>	kJ/m <sup>3</sup>	air specific enthalpy at ambient conditions
H	kJ	enthalpy
H <sub>d</sub> , H <sub>low</sub> (LHV)	kJ/kg	lower heating value
i	%	discount rate
IRR	%	internal rate of return
k	kW/m <sup>2</sup> K	overall heat transfer coefficient
k <sub>i</sub>	\$/kWh	specific cost of exergy losses
K <sub>i</sub>	\$	cost of exergy losses
K <sub>HRSG</sub>	\$	HRSG surface cost
K <sub>total</sub>	\$	total annualized HRSG cost
$\dot{m}$	kg/s	mass flow
$\dot{m}_{GTtotal}$	kg/s	total flue gas mass flow at GT exit
$\dot{m}_{HP}$	kg/s	steam mass flow
$\dot{m}_{sf}$	kg/s	supplementary firing fuel mass flow
n		number of years

$p$	bar	pressure
$p_s$	bar	water (steam) pressure
$P$	kW(MW)	power
$\dot{Q}$	kW	exchanged heat (duty)
$R$	kJ/kgK	gas constant
$s$	kJ/kgK	specific entropy
$t$	°C	temperature
$t_g$	°C	flue gas temperature
$t_s$	°C	water (steam) temperature
$V_L$	m <sup>3</sup> /s	air volume flow
$\eta_{CC}$	%	combined cycle efficiency
$\eta_{GT}$	%	gas turbine cycle efficiency
$\eta_{HR}$	%	heat recovery efficiency
$\eta_{SC}$	%	steam turbine cycle efficiency
$\eta_{ST}$	%	steam turbine isentropic efficiency
$\eta_{sf}$	%	duct burner efficiency
$\eta_r$	%	radiation heat transfer efficiency
$\dot{Q}_{GT}$	kW	heat induced into gas turbine cycle
$\dot{Q}_{EXH}$	KW	heat released from gas turbine cycle
$\dot{Q}_{SF}$	kW	heat induced via supplementary firing
$\lambda$		excess air factor
$x$	%	relative water content
$W_{GT}$	kW(MW)	work generated in the gas turbine cycle
$W_{ST}$	kW(MW)	work generated in the steam turbine cycle

Abbreviations	
AP	approach point
CC	combined cycle
CNG	compressed natural gas
CONDHT	condensate heater
CNDPUMP	condensate pump
EM	electromotor
EPC	engineering, procurement and construction cost
FW	feed water
FG	flue gases
GT	gas turbine
HRSG	heat recovery steam generator
HTC	heat transfer coefficient
HP	high pressure
HPAP	high pressure approach point
HPECO	high pressure economizer
HPEVA	high pressure evaporator
HPPP	high pressure pinch point
HPPUMP	high pressure pump
HPSH	high pressure superheater
IPECO	intermediate pressure economizer
IPEVA	intermediate pressure evaporator
IPPUMP	intermediate pressure pump
IPSH	intermediate pressure superheater
ICE	internal combustion engine
LNG	liquefied natural gas
LPG	liquid petroleum gas
LDC	load duration curve
LP	low pressure
LPAP	low pressure approach point
LPECO	low pressure economizer
LPEVA	low pressure evaporator
LPPP	low pressure pinch point
LPPUMP	low pressure pump
LPSH	low pressure superheater
MDCC	mechanical drive combined cycle
NG	natural gas
PP	pinch point
dp	pressure drop
SC	steam cycle
ST	steam turbine
SF	supplementary firing
TMIX	temperature mixer
1P	single pressure
2P	double pressure
3P	triple pressure

## List of Figures

Figure 2.1: Annual carbon emissions by region [13].....	18
Figure 2.2: Reserves to production ratio of fossil fuels for various regions[12] .....	19
Figure 2.3: Evolution of NOx emission from gas turbines (2002) [14] .....	21
Figure 2.4: Natural gas production and price trend comparison in the U.S.[16] .....	24
Figure 2.5: Natural gas prices for industry in EU25 [17] .....	24
Figure 2.6: Eurasian gas transportation system [18] .....	25
Figure 2.7: Gas demand in OECD countries [19] .....	27
Figure 2.8: Cost of electricity (\$cents/kWh) (1998) [12].....	28
Figure 2.9: Cost of electricity (\$/MWh) (2005) [20] .....	28
Figure 3.1: Pipeline compressor characteristic [22].....	29
Figure 3.2: Pipeline compressor station [1] .....	30
Figure 3.3: Natural gas pipeline system [24] .....	31
Figure 3.4: Sectional drawing of a 25MW GT for driving pipeline compressors [23].....	32
Figure 3.5: Pipeline compressor [23].....	32
Figure 3.6: THM1304-11 gas turbine for pipeline compressor station in Bolivia [23] .....	33
Figure 3.7: Centrifugal compressor for pipeline compressor station in Bolivia [23].....	33
Figure 3.8: Planned waste heat recovery plant at Weitendorf gas compressor station [28]..	35
Figure 3.9: Waste heat recovery unit at the compressor station Ruswil [29].....	35
Figure 4.1: TGS pipelines [30].....	36
Figure 4.2: Compressor station schematic [23] .....	37
Figure 4.3: Annual demand at pipeline compressor shafts [23].....	37
Figure 4.4: Load duration curve [23].....	38
Figure 4.5: Ambient temperature annual profile (2005) [31] .....	39
Figure 4.6: Flue gases mass flow vs. part load fraction [23].....	39
Figure 4.7: Flue gases temperature vs. part load fraction [23] .....	40
Figure 4.8: GT efficiency vs. part load fraction [23] .....	40
Figure 5.1: Efficiencies of various types of plants (2002) [14] .....	42
Figure 5.2: Simplified flow scheme of CC plant .....	43
Figure 5.3: T-s diagram of CC process without (a) and with supplementary firing (b).....	43
Figure 5.4: GT and HRSG layout [11].....	45
Figure 5.5: Supplementary-fired HRSG [32].....	46
Figure 5.6: Exhaust-fired HRSG [32].....	47
Figure 5.7: Steam generator with higher level of supplementary firing [22].....	48
Figure 5.8: Single pressure waste heat recovery plant [4].....	49
Figure 5.9: Double pressure waste heat recovery plant [4] .....	49
Figure 5.10: In-line duct burner for supplementary firing of GT exhaust [34] .....	50
Figure 5.11: Efficiency of large scale CC plants as a function of the flue gas temperature after SF and gas turbine inlet temperature [6] .....	52
Figure 5.12: Efficiency of small scale CC plants compared with GT and ST plants [33] .....	52
Figure 5.13: Calculated efficiency of waste heat recovery system based on THM1203 for 34.8 kg/s flue gas mass flow and steam parameters of 30 bar and 390°C .....	53
Figure 5.14: T-Q diagrams of waste heat recovery system based on THM1203 for 34.8 kg/s flue gas mass flow and steam parameters of 30 bar and 390°C .....	53
Figure 5.15: Steam quantity and AP of waste heat recovery system based on THM1203 for 34.8 kg/s flue gas mass flow and steam parameters of 30 bar and 390°C .....	54
Figure 5.16: Effect of the temperature after SF on CC efficiency for 1P and 2P system (large scale plants), referent plant without supplementary firing $t_{GT,exh}=525^{\circ}C$ [6].....	55
Figure 5.17: Calculated efficiency of waste heat recovery system based on THM1203 for 34.8 kg/s flue gas mass flow and steam parameters of 30bar/390°C and 5bar/182°C .....	55
Figure 5.18: T-Q diagram showing pinch and approach points .....	56
Figure 5.19: Hierarchic levels of automation [14] .....	59
Figure 5.20: Power generation costs vs. fuel price (1997), discount rate=11% [6].....	61
Figure 5.21: Power generation costs vs. equivalent utilization time (1997), discount rate=11%, fuel price=10\$/MWh [6].....	62
Figure 5.22: Simplified diagram of a MDCC and power generation plant [4].....	63
Figure 5.23: Simplified diagram of MDCC and power generation plant with 3P HRSG [4] ...	64

Figure 5.24: Injection cooler between two superheater stages [25]	66
Figure 6.1: Schematic of SH2	69
Figure 6.2: HPSH2 temperature profile	69
Figure 6.3: System off-design algorithm	76
Figure 6.4: Steam turbine off-design isentropic efficiency vs. part load fraction [23]	77
Figure 6.5: CO <sub>2</sub> emissions as a function of electrical efficiency and fuel [22]	77
Figure 7.1: Flow scheme 1P configuration	80
Figure 7.2: Flow scheme 2P1 configuration	81
Figure 7.3: Flow scheme 2P2 configuration	82
Figure 7.4: Flow scheme 2P3 configuration	83
Figure 7.5: Flow scheme 3P1 configuration	84
Figure 7.6: Flow scheme 3P2 configuration	85
Figure 7.7: Flow scheme 3P3 configuration	86
Figure 7.8: Objective function for initial assumptions	87
Figure 7.9: Objective function for realistic assumptions	88
Figure 7.10: Optimization progress of objective function and its components	88
Figure 7.11: T-Q diagram configuration 1P	90
Figure 7.12: Optimization progress of the objective function for configuration 2P1	92
Figure 7.13: T-Q diagram configuration 2P1	93
Figure 7.14: T-Q diagram configuration 2P2	94
Figure 7.15: T-Q diagram configuration 2P3	96
Figure 7.16: Optimization progress of the objective function for configuration 3P1	97
Figure 7.17: T-Q diagram configuration 3P1	98
Figure 7.18: T-Q diagram configuration 3P2	100
Figure 7.19: T-Q diagram configuration 3P3	101
Figure 7.20: Value of optimal objective function and its components	102
Figure 7.21: CC, SC and HRSG efficiency	104
Figure 7.22: ST power, HRSG surface cost	104
Figure 7.23 Cost of electricity as a function of CC net efficiency	105
Figure 7.24 Objective function and its components vs. exhaust gas mass flow and temperature	105
Figure 7.25 Optimal fresh steam pressure vs. exhaust gas mass flow and temperature	106
Figure 7.26 Optimal fresh steam pressure vs. exhaust gas mass flow and temperature	106
Figure 7.27: T-Q diagram for different HRSG surface cost (1P)	107
Figure 7.28: Optimal operative parameters for different HRSG surface cost (1P)	107
Figure 7.29: HRSG surfaces as a function of surface cost (1P)	108
Figure 7.30: Objective function for reference surface cost	108
Figure 7.31: Objective function for 2x reference surface cost	109
Figure 7.32: Objective function for 4x reference surface cost	109
Figure 7.33: HRSG cost vs. system complexity factor	110
Figure 7.34: Net CC efficiency vs. system complexity factor	111
Figure 8.1: Flue gas enthalpy at GT outlet in compressor station operating regimes	113
Figure 8.2: Net CC efficiency in operating regimes	115
Figure 8.3: Fuel consumed by supplementary firing in operating regimes	115
Figure 8.4: ST generator output power in operating regimes	116
Figure 8.5: Reduced CO <sub>2</sub> emissions in operating regimes	116
Figure 8.6: Influence of fix and sliding pressure mode on CC efficiency (1P)	117
Figure 8.7: Steam inlet pressures in fix and sliding pressure mode (1P)	118
Figure 8.8: Influence of cooling water temperature change on CC and ST heat rate (1P)	119
Figure 8.9: Influence of cooling water quantity change on CC and ST heat rate (1P)	119
Figure 9.1: Typical first cost of various types of plants [14]	121
Figure 9.2: Prices comparison for various fuels [14]	121
Figure 9.3: Share of various costs in total plant life cycle cost [14]	121
Figure 9.4: Cash flow diagram based on end-of-period convention [12]	122
Figure 9.5: Cash flow diagram based on mid-period convention [12]	123
Figure 9.6: Cash flow diagram based on continuous flow convention [12]	123
Figure 9.7: Cumulative cash flow configuration 1P	126

Figure 10.1: Natural gas prices in Europe (2007) [21].....	127
Figure 10.2: World reserves of natural gas (2006) [21] .....	127
Figure 10.3: Electricity production cost vs. natural gas price with CO <sub>2</sub> price as parameter.	128
Figure 10.4: Influence of fuel escalation rate on project viability .....	129

## List of Tables

Table 2-1: Typical natural gas chemical composition .....	21
Table 3-1: Technical data of pipeline compressor [23] .....	32
Table 3-2: Technical data natural gas compressor unit for pipeline in Bolivia [23] .....	34
Table 4-1: Compressor station operating data .....	38
Table 4-2: Exhaust gas Composition .....	41
Table 4-3: Natural gas Composition .....	41
Table 4-4: Design data gas turbines .....	41
Table 5-1: Ways of superheated steam temperature control .....	65
Table 5-2: Boundary conditions for setting up the new MDCC system configurations .....	67
Table 6-1: Values of assumed input data .....	68
Table 6-2: Input data for exergo-economic HRSG calculation .....	74
Table 6-3: Exponents for off design performance calculations of HTC .....	75
Table 6-4: Exponents for off design performance calculations of dp .....	75
Table 7-1: Configuration features .....	79
Table 7-2: 1P configuration operative parameters .....	89
Table 7-3: 1P configuration heat exchanger surface areas .....	91
Table 7-4: Condenser size and cooling water requirement .....	91
Table 7-5: 2P1 configuration operative parameters .....	92
Table 7-6: Heat exchanger surface areas 2P1 configuration .....	93
Table 7-7: Condenser size and cooling water requirement 2P1 configuration .....	93
Table 7-8: 2P2 configuration operative parameters .....	94
Table 7-9: Heat exchanger surface areas 2P2 configuration .....	95
Table 7-10: Condenser size and cooling water requirement 2P2 configuration .....	95
Table 7-11: 2P3 configuration operative parameters .....	95
Table 7-12: Heat exchanger surface areas 2P3 configuration .....	96
Table 7-13: Condenser size and cooling water requirement 2P3 configuration .....	96
Table 7-14: 3P1 configuration operative parameters .....	98
Table 7-15: Heat exchanger surface areas for 3P1 configuration .....	99
Table 7-16: Condenser size and cooling water requirement 3P1 configuration .....	99
Table 7-17: 3P2 configuration operative parameters .....	99
Table 7-18: Heat exchanger surface areas for 3P2 configuration .....	100
Table 7-19: Condenser size and cooling water requirement 3P2 configuration .....	100
Table 7-20: 3P3 configuration operative parameters .....	101
Table 7-21: Heat exchanger surface areas for 3P3 configuration .....	102
Table 7-22: Condenser size and cooling water requirement 3P3 configuration .....	102
Table 7-23: Comparison of 1P, 2P and 3P configurations .....	103
Table 7-24: HRSG operative parameters for fired HRSG .....	111
Table 9-1: Structure of the investment cost .....	125
Table 9-2: Reference values for economic calculation .....	125
Table 9-3: Cost of electricity .....	125
Table 9-4: Internal rates of return .....	126
Table 9-5: Payback period .....	126
Table 10-1: Cost of electricity for fired HRSG .....	129
Table 10-2: Internal rates of return for fired HRSG .....	129
Table 10-3: Payback period for fired HRSG .....	130

## List of Enclosures

- Enclosure 1: Mathematical model – thermodynamic calculation design point
- Enclosure 2: Mathematical model – economic calculation

## Contents

1.	Introduction.....	15
2.	Global energy and environmental situation .....	18
2.1	Sustainable development.....	18
2.1.1	Fossil fuel reserves .....	19
2.1.2	Environmental impact of firing the fossil fuels .....	20
2.2	Natural gas.....	21
2.2.1	Chemical composition and energy content .....	21
2.2.2	Natural gas demand.....	22
2.2.3	Natural gas supply .....	23
2.2.4	Storage and transport .....	25
2.2.4.1	<i>Storage</i> .....	25
2.2.4.2	<i>Gas pipelines</i> .....	25
2.2.4.3	<i>Liquefied natural gas (LNG)</i> .....	26
2.2.4.4	<i>Compressed natural gas (CNG)</i> .....	26
2.2.4.5	<i>Liquid petroleum gas (LPG)</i> .....	26
2.2.5	Use.....	27
3.	Pipeline compressor stations.....	29
3.1	Requirements on pipeline compressor stations .....	29
3.2	Technical features of pipeline compressor stations .....	30
3.2.1	Gas turbines as prime movers for pipeline compressor stations .....	31
3.2.2	Pipeline compressors.....	32
3.3	Pipeline compressor station example .....	33
3.4	Potential for utilization of waste heat .....	34
3.4.1	Waste heat utilization application examples .....	35
4.	Technical features of analyzed pipeline compressor station .....	36
4.1	Function .....	36
4.2	Configuration.....	36
4.3	Demand analysis .....	37
4.4	Gas Turbine output data .....	39
5.	Waste heat recovery via gas and steam combined cycle.....	42
5.1	CC thermodynamic principles .....	42
5.2	Heat recovery steam generator .....	45
5.2.1	Single pressure and double pressure systems .....	48
5.2.2	Influence of supplementary firing on CC efficiency .....	50
5.3	Parameters affecting dimensioning of HRSG .....	56
5.3.1	Pinch point .....	56
5.3.2	Approach point.....	56
5.3.3	Flue gases acid dew point and water dew point .....	57
5.4	Factors affecting CC plant performance .....	58
5.5	Operation and control .....	58
5.5.1	CC operation and control .....	58
5.5.2	MDCC operation and control.....	60
5.6	Economics of waste heat recovery .....	60
5.7	Criteria for configuring the new MDCC system.....	62
5.7.1	Possible application of ST power produced via waste heat utilization .....	62
5.7.1.1	<i>Electricity power production</i> .....	62
5.7.1.2	<i>Natural gas compressing by means of additional compressor</i> .....	64
5.7.1.3	<i>Natural gas compressing by means of electromotor</i> .....	64
5.7.1.4	<i>Electricity production via ORC (organic rankin cycle)</i> .....	65
5.7.2	Viability of using the natural gas cooler for condensate preheating.....	65
5.7.3	Superheated steam temperature control.....	65
5.7.4	Boundary conditions for new MDCC system.....	67
6.	Mathematical model .....	68
6.1	Assumptions .....	68
6.2	Design point model .....	68
6.3	HRSG optimization model.....	71

6.4	Off-design modeling .....	74
6.5	CO <sub>2</sub> emissions reduction potential .....	77
7.	HRSG optimization of various system configurations.....	79
7.1	1P Configuration .....	87
7.1.1	Objective function.....	87
7.1.2	Exergo-economic optimization progress .....	88
7.1.3	System description.....	89
7.1.4	Design point results.....	90
7.2	2P1 Configuration .....	91
7.2.1	Exergo-economic optimization results .....	92
7.2.2	Design point results.....	93
7.3	2P2 Configuration .....	94
7.3.1	Exergo-economic optimization results .....	94
7.3.2	Design point results.....	94
7.4	2P3 Configuration .....	95
7.4.1	Exergo-economic optimization results .....	95
7.4.2	Design point results.....	96
7.5	3P1 Configuration .....	97
7.5.1	Exergo-economic optimization results .....	97
7.5.2	Design point results.....	98
7.6	3P2 Configuration .....	99
7.6.1	Exergo-economic optimization results .....	99
7.6.2	Design point results.....	100
7.7	3P3 Configuration .....	101
7.7.1	Exergo-economic optimization results .....	101
7.7.2	Design point results.....	101
7.8	Comparison and selection .....	102
7.9	Optimization sensitivity analysis .....	105
7.9.1	Influence of different exhaust gas parameters on optimization results .....	105
7.9.2	Influence of HRSG heat exchanger surface cost on optimization results .....	107
7.9.3	Influence of system complexity factor on optimization results .....	110
7.9.4	Influence of supplementary firing on optimization results .....	111
8.	System off-design behavior .....	113
8.1	System design and operating strategy at GT part load.....	113
8.1.1	Constant ST power in all regimes .....	114
8.1.2	Constant ST power in regimes 1-5, ST out of operation in regimes 6-10 .....	114
8.1.3	Maintaining constant flue gas temperature .....	114
8.2	Off-design results for constant flue gas temperature of 500°C.....	115
8.2.1	CC net efficiency.....	115
8.2.2	Fuel consumption.....	115
8.2.3	ST generator output power .....	116
8.2.4	Reduced CO <sub>2</sub> emissions .....	116
8.3	Off-design sensitivity analysis .....	117
8.3.1	Influence of ST operation mode on CC efficiency.....	117
8.3.2	Influence of change in cooling water parameters on overall heat rate.....	118
9.	Economic evaluation .....	120
9.1	Economic considerations of energy projects .....	120
9.1.1	General characteristics of capital investments.....	120
9.1.2	Time value of money concepts .....	122
9.1.3	Cash flow .....	122
9.1.3.1	<i>End-of-period convention</i> .....	122
9.1.3.2	<i>Mid-period convention</i> .....	122
9.1.3.3	<i>Continuous flow convention</i> .....	123
9.1.4	Evaluation of proposals.....	123
9.1.4.1	<i>Payback method</i> .....	123
9.1.4.2	<i>Average rate of return method</i> .....	123
9.1.4.3	<i>Internal rate of return method (IRR)</i> .....	124

9.1.4.4	<i>Net present value method</i> .....	124
9.1.4.5	<i>Profitability index</i> .....	124
9.2	Economic evaluation of selected configurations in demand scenario.....	124
9.2.1	Cost of electricity.....	125
9.2.2	Internal rate of return.....	126
9.2.3	Payback period .....	126
9.2.4	Comparison and selection.....	126
10.	Economic sensitivity analysis.....	127
10.1	Influence of variable fuel and CO <sub>2</sub> emission certificate prices .....	128
10.2	Influence of supplementary firing on project viability .....	129
10.2.1	Cost of electricity.....	129
10.2.2	Internal rate of return.....	129
10.2.3	Payback period .....	130
11.	Conclusion .....	131
12.	References.....	134

## 1. Introduction

Major share in global production of electricity and heat nowadays belongs to technologies based on firing the fossil fuels i.e. coal, oil and natural gas. Despite its greatest reserves among other fossil fuels usage of coal has a falling trend since its firing causes high CO<sub>2</sub> and NO<sub>x</sub> emissions wherefore its usage in thermal power plants in some countries forbidden. Necessity of application of complex and costly flue gas cleaning facilities puts it in unfavorable position in comparison with oil or gas applications. Limited crude oil reserves and its increased demand followed by global energy and political conditions cause unpredictable oscillations of crude oil prices making oil less and less competitive for electrical power generation. In such circumstances natural gas obtrudes since its reserves are larger than oil and its firing produces smaller amount of green house gases than other fossil fuels. Last but not least natural gas prices are so far more acceptable than those of oil despite their interdependency. From this reasons natural gas demand has a rising trend since beginning of last decade. [1] In transport and supply of natural gas an important role belongs to gas pipelines where are compressor stations used for maintaining the operating pressure.

Generally centrifugal compressors in these stations are driven by gas turbine (GT), steam turbine (ST), electromotor (EM) or internal combustion engine (ICE).[2] In US and Canada ICE are represented with ca. 70% and are followed by gas turbines whose share is ca. 25%. The rest belongs to steam turbines and electromotors whose share is minor. In Europe is the share of gas turbines somewhat higher. [3][4]

Since up to now replacement for fossil fuels has not been found it is an imperative to evaluate utilization of waste energy from any kind of energy process so as to increase process efficiency respectively decrease fuel consumption and emissions. This suggests investigation of waste heat utilization possibilities from gas turbine driven pipeline compressor stations which depend mainly on the following criteria:

- site location
- demand (mechanical power, electricity or heat) and its distribution
- availability of electrical grid
- gas turbine type (flue gas mass flow and temperature)

Thereof one of the possible technologies can be considered:

- electricity production
- electricity and heat production
- electricity, heat and cooling production
- additional natural gas compressing
- hydrogen production
- direct waste heat utilization for natural gas inlet or outlet cooling in appropriate chillers

Best available and most acceptable technology for electricity production nowadays with respect to above aspects is combined gas and steam turbine process[5][6] or commonly combined cycle (CC).

Compared with separate gas or steam plants their main advantages are:

- thermodynamic efficiency
- flexibility
- lower cooling water equipment cost

- acceptable influence on environment
- high reliability and relative short delivery time
- possibility of application of broad fuel spectrum

However there are also at least two disadvantages which can mainly be emphasized into:

- lower thermodynamic parameters of a steam process

Gas turbine exhaust gas has lower temperature than typical firing temperature in a classic steam boiler. However flue gases temperature at boiler inlet can be increased by means of supplementary firing what brings to efficiency increase only in certain cases described in 5.2.2.

- inability of a fast startup

Combined cycle plants have longer startup time than gas turbine only plants.

Aiming to deepen former investigations this work focuses on analysis of waste heat utilization possibilities for electricity production via combined gas and steam turbine process. Since the topping cycle already exists and the main function of the plant is compression of natural gas such processes are called mechanical drive combined cycle (MDCC). Natural gas as such is burned either in a gas turbine process or in a combined gas and steam process. Since last decade popularity of the latter increased significantly thanks to its high efficiencies which in case of electricity power production exceed 60% nowadays [7]. In case of simultaneous production of electricity power and heat (cogeneration) overall process efficiency can reach up to 90% [8]. Unlike conventional gas and steam processes with or without cogeneration where the main function of the plant is electricity production for meeting the base load or peak load, main function of considered pipeline compressor stations is mechanical power production for pumping the natural gas. Therefore attention has to be paid to configuration of an upgraded system in the terms of proper selection of operative parameters as also operational behavior in particular at the part load. In this work different single pressure (1P), double pressure (2P) and triple pressure (3P) CC system configurations are developed with respect to parameters of available flue gases followed by site conditions and demand characteristic from one realistic project.

Heat balance calculation of each configuration at design point and in operating regimes is done by means of GateCycle.[9] Exergo-economic HRSG optimization of each configuration is done in order to find an optimum between exergy losses and HRSG heat exchanger surface costs. After optimizing the operative parameters for each configuration a preferable 1P, 2P and 3P configuration is selected. Viability of selected configurations is examined in demand scenario in order to evaluate viability of investment and operation with respect to assumed fuel cost, electricity sellback price and CO<sub>2</sub> emission certificate prices.

Chapter 2 describes global energy and environmental situation followed by aspects of production, transport and consume of natural gas. Chapter 3 describes pipeline compressor stations; requirements on them, their technical features generally and potential for utilization of waste heat. Chapter 4 describes technical features of analyzed pipeline compressor station. Chapter 5 deals with conditions and characteristics of waste heat recovery via gas and steam combined cycle. Chapter 6 describes mathematical model of a system part to be calculated namely: bottoming cycle comprising HRSG and steam turbine, HRSG optimization model and off-design performance algorithm. Chapter 7 presents results of HRSG optimization for different

configurations. Chapter 8 presents results of operation of selected configurations in demand scenario. Chapter 9 describes economic evaluation of selected configurations in demand scenario. Chapter 10 provides economic sensitivity analysis. Chapter 11 provides final considerations based on obtained results outlining guidelines for efficient waste heat utilization and suggesting possibilities for further research and development of MDCC systems used for utilization of waste heat from pipeline compressor stations. Chapter 12 contains list of references used in the text.

## 2. Global energy and environmental situation

### 2.1 Sustainable development

Aspects of energy resources exploitation evolve during centuries, decades and years following development of human civilization. While aiming to fulfill human needs social and industrial development influences planetary balance. First significant environmental problem known as the Ozone gap was identified in the second half of the last century. Despite relative fast reaction of global politic resulting in ratification of Montreal Protocol (1987) it took 20 years to eliminate majority of chlorofluorocarbons for preventing attenuation of ozone layer. However ca. 60 years more shall be required for protection from ultraviolet rays. Further environmental problem known as the global warming alarmed publicity due to unusual hot summer announcing possible climate changes. After 10 years of negotiations agreement known as Kyoto Protocol (1997) was reached. Despite setting emission limits primarily for industrialized countries followed by the fact that it has been ratified by 183 states up to January 2009 it cannot decrease an exponential trend of CO<sub>2</sub> emissions as shown in Figure 2.1.

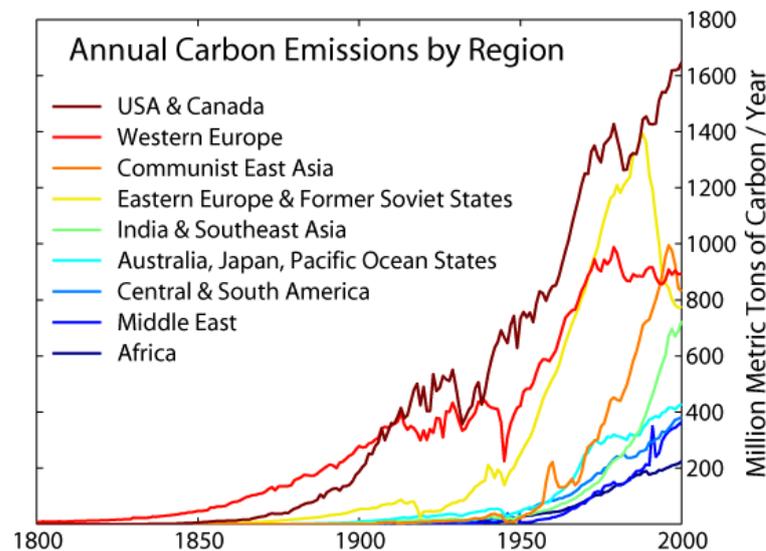


Figure 2.1: Annual carbon emissions by region [13]

Global energy scenarios [10] predict the world CO<sub>2</sub> emissions to reach the level of 1990 increased by 40-45%. That necessitates much drastic interventions into energy production and consume in the following decades also respecting the fact that global warming has a rising trend manifesting in phenomena of tripling the annual melting of icebergs on Greenland in couple of years only.

In the terms of electricity and heat production up to first oil crisis 1973. preference belonged to large systems whereby there was not much attention paid on tradeoff between economic aspects of plant and generated unit power. After the crisis and in particular during the last decade of former century the trend has changed. Focus, particularly in developed countries, was on efforts in research and development of energy efficiency issues striving to energy savings. Thereby highly efficient systems with possibilities of utilizing the waste heat became more important in circumstance of rising primary energy prices oscillations in security of supply and in particular due to ever more environmental requirements.

Priorities and instruments of energy policy in 70s and 80s of the last century accentuated security of supply as the main target. However main targets of European Commission energy policy emphasized in the White Paper from 1995 are:

- Overall competitiveness
- Security of supply
- Environmental protection

Thereof it may be concluded that the control of energy resources and energy trade is considered as too important to be left to the principles of the free market. Such approach brought to development of new technologies that were recently considered as economically and technically unjustifiable.

So further development of human civilization has to ensure a balance between fulfilling the energy needs and thereof resulting environmental impact. Such development is named sustainable development and may be described as phenomena which limit further development of our civilization based on existing technologies. The concept sustainable development came into general usage following publication of the 1987 report of the Brundtland Commission which defines the concept of sustainable development as a progress that meets the needs of today without compromising the ability of future generations to meet their own needs. [11]

### 2.1.1 Fossil fuel reserves

Existing technologies for power production are mainly based on firing the fossil fuels: coal, oil and natural gas whose reserve estimation is shown in Figure 2.2

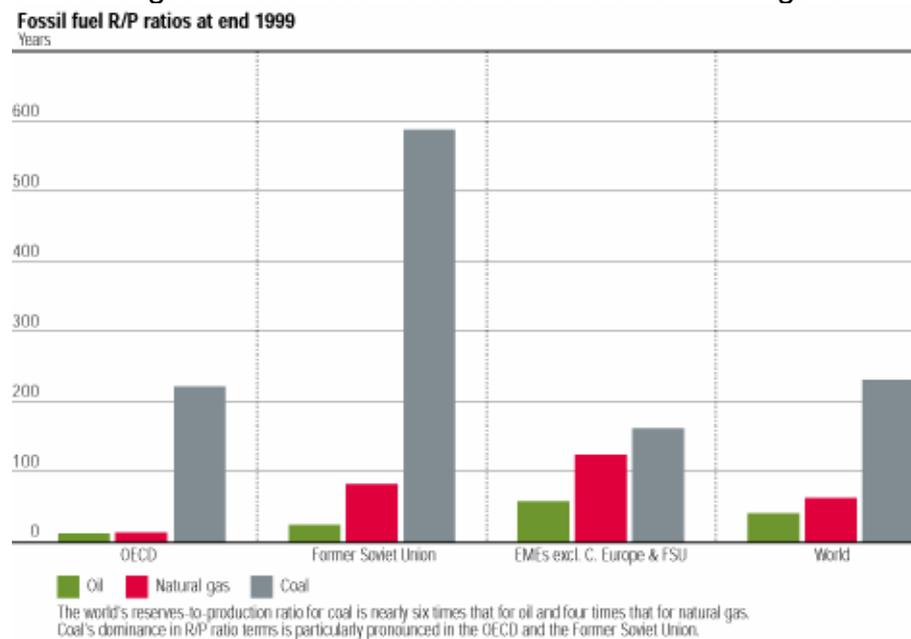


Figure 2.2: Reserves to production ratio of fossil fuels for various regions[12]

It is apparent that on a world scale oil and natural gas reserves are limited on 50 years or less. Firing coal causes high CO<sub>2</sub> and NO<sub>x</sub> emissions wherefore is its usage in thermal power plants in some countries forbidden.

### 2.1.2 Environmental impact of firing the fossil fuels

Firing of fossil fuels in any kind of thermal power plant has an impact on environment which has to be kept as low as possible. Since stringent environmental requirements generally increase the cost of the system, it should be attained a reasonable relationship between the cost and the obtained results.

Following emissions from a power station generally affect the environment:

- products of combustion (exhausts and ash)
- waste heat
- noise

The exhausts can include the following components: NO, NO<sub>2</sub>, CO<sub>2</sub>, CO, C<sub>n</sub>H<sub>n</sub> (unburned hydrocarbons), SO<sub>2</sub>, SO<sub>3</sub>, dust, fly ash, heavy metals, chlorides and others. [6]

The largest part of the flue gas from most fossil fuel combustion processes is uncombusted nitrogen, which is contained in the air with about 79 volume %. The next largest part of the flue gas is carbon dioxide (CO<sub>2</sub>) with 10 to 15 volume % or more of the flue gas. This is closely followed in volume by water vapor (H<sub>2</sub>O) created by the combustion of the hydrogen in the fuel with atmospheric oxygen. Flue gases constituent which can be seen pouring from flue gas stacks is actually the water vapor forming a cloud as it contacts cool air.

A typical flue gas from the combustion of fossil fuels will also contain some very small amounts of nitrogen oxides (NO<sub>x</sub>), sulfur dioxide (SO<sub>2</sub>) and particulate matter. The nitrogen oxides are derived from the nitrogen in the ambient air as well as from any nitrogen-containing compounds in the fossil fuel. The sulfur dioxide is derived from any sulfur-containing compounds in the fuels. The particulate matter is composed of very small particles of solid materials and very small liquid droplets which give flue gases their smoky appearance. [13]

H<sub>2</sub>O, N<sub>2</sub> and O<sub>2</sub> are not harmful for environment, whilst other constituents can have negative impact. Their concentration in the exhaust gas depends upon the composition of the fuel and the type of installation.

CC plant is a popular technology nowadays thanks to its high thermodynamic efficiency and high excess air coefficients in GT which enable complete combustion respectively a very low concentration of unburned elements such as CO or unburned hydrocarbons. Further advantage of large air flow is strong dilution of pollutants.

In case of burning the natural gas, the only toxic emissions contained in the exhaust gas are NO and NO<sub>2</sub>. The NO<sub>x</sub> (NO and NO<sub>2</sub>) level is the most important environmental problem with gas turbines because NO<sub>x</sub> generates nitric acid (H<sub>2</sub>NO<sub>3</sub>) in the atmosphere which is together with sulphuric acid (H<sub>2</sub>SO<sub>4</sub>) one of the factors responsible for acid rain. [6] Furthermore if mixture of nitrogen oxides and evaporable organic compounds is exposed to solar radiation it leads to formation of photochemical smog.[13]

With permanent development of gas turbines which are main source of NO<sub>x</sub> emissions within a GT based CC, these emissions are constantly being decreased as shown in Figure 2.3. [14]

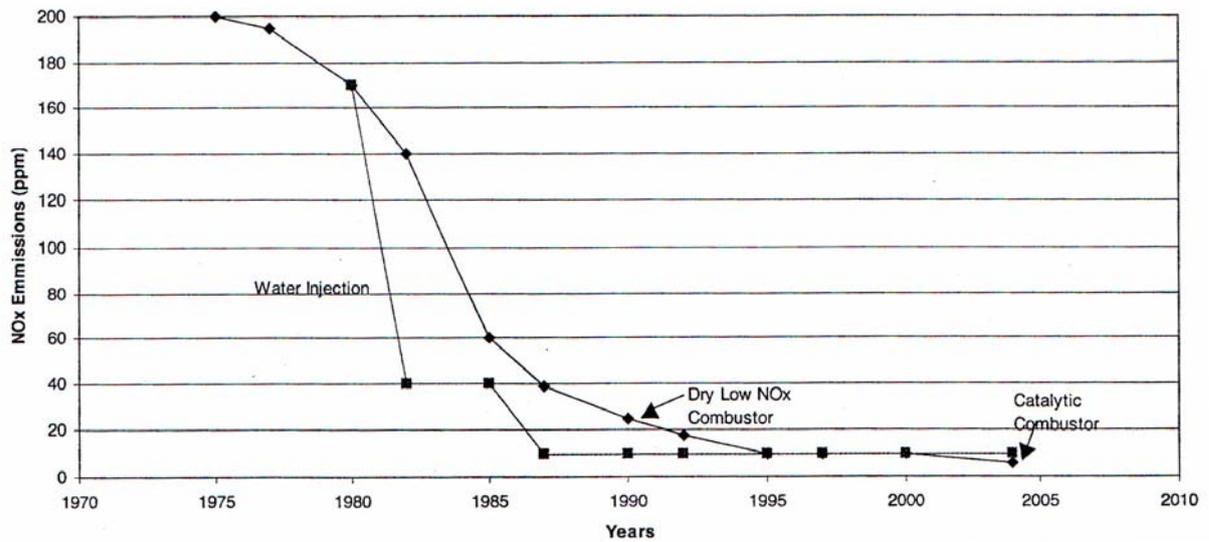


Figure 2.3: Evolution of NOx emission from gas turbines (2002) [14]

Water injection into the combustor reduces NOx emissions by lowering the temperature peaks in the combustor primary zone and hence the tendency for atmospheric nitrogen to dissociate. Side effects include increased power, but often increased emissions of carbon monoxide (CO) due to lower temperatures, especially for low pressure ratios and short residence times. Furthermore specific fuel consumption is worse due to the significant amount of heat required to evaporate the water. Thereof steam injection superseded water injection for installations where steam plant could be used as it overcame the disadvantages associated with the chilling effect of liquid water. However further development of GT combustion systems was focused on dry low emission technologies where no water or steam is injected into the combustor to lower flame temperature and hence NOx. From Figure 2.3 it is apparent that latest dry low NOx combustors provide NOx emissions around 10ppm (parts per million) what is ca. 25 times less than in the case of conventional combustor. Even lower values are obtainable in case of catalytic combustion. Therein the fuel and oxygen react on the surface of a catalyst leading to complete oxidation of compound. As the process takes place without a flame and at much lower temperatures (than in case of conventional combustion) it produces quite lower NOx emissions than conventional combustion.

## 2.2 Natural gas

### 2.2.1 Chemical composition and energy content

Natural gas is a gaseous fossil fuel found mostly in oil fields and natural gas fields like also in coal beds (as coal bed methane). [13]

Table 2-1: Typical natural gas chemical composition

Component	weight (%)
Methane (CH <sub>4</sub> )	70-90
Ethane (C <sub>2</sub> H <sub>6</sub> )	5-15
Propane (C <sub>3</sub> H <sub>8</sub> ) and Butane (C <sub>4</sub> H <sub>10</sub> )	< 5
CO <sub>2</sub> , N <sub>2</sub> , H <sub>2</sub> S, etc.	balance

As shown in Table 2-1, it consists primarily of methane but includes significant quantities of ethane, butane, propane and helium [1] followed by inorganic compounds such as nitrogen, carbon dioxide, and hydrogen sulfide which are not

desirable because some of them are not combustible and cause corrosion and other problems in gas production and processing systems. [15] Natural gas is often informally named simply gas, especially when compared to other energy sources such as electricity. The gross heat of combustion of one normal cubic meter of commercial quality natural gas is around 39 mega joules ( $\approx 10.8$  kWh), but this can vary by several percentages. [16]

### 2.2.2 Natural gas demand

Global demand for natural gas is mainly driven by the following factors: [16]

- Weather
- Demographics
- Economic Growth
- Fuel Competition
- Storage
- Exports

#### Weather

Natural gas demand usually peaks during the colder months of the year (November-March) and decreases during the warmer months. During the warmest summer months, the demand increases due to electricity generated by gas fired power plants. Any unfavorable weather conditions will increase the demand for natural gas. The colder the weather is during the winter, the more shall be pronounced the winter peak. On the other hand, a warmer winter usually results in a less noticeable winter peak. The opposite applies for the summer. If the summer season is extremely hot, it can result in greater cooling demands, which in turn may result in increased summer demand for natural gas.

#### Demographics

Changing demographics also affects the demand for natural gas, especially for core residential customers. As electricity currently supplies most of the cooling energy requirements, and natural gas supplies most of the energy used for heating, population movement influences the natural gas demand depending on the respective climate.

#### Economic Growth

The state of the economy can have a considerable effect on the demand for natural gas in the short term. This is particularly true for industrial and to a lesser extent the commercial customers. When the economy is booming, output from the industrial sectors generally increases. On the other hand, when the economy is experiencing a recession output from industrial sectors drops. These fluctuations in industrial output accompanying the economy affect the amount of natural gas needed by these industrial users.

#### Fuel Competition

Basically, supply and demand dynamics in the marketplace determine the short-term price for natural gas. However, this can work in reverse as well. The price of natural gas can, for certain consumers, affect its demand. This is particularly true for those consumers who have the ability to switch the fuel which they consume. In general the core customers (residential and commercial) do not have this ability, however, a number of industrial and electric generation consumers have the capacity to switch between fuels. For instance, when natural gas prices are extremely high, electric generators may switch from using natural gas to using cheaper coal or fuel oil. This

fuel switching then leads to a decrease for the demand of natural gas, which usually tends to drop its price.

#### Storage

Natural gas storage levels have a significant impact on the commodity's price. When the storage levels are low, a signal is being sent to the market indicating that there is a smaller supply cushion and prices will be rising. On the other hand, when storage levels are high, this sends a signal to the market that there is greater supply flexibility and prices will tend to drop.

#### Exports

Exports are another source of demand and are driven by economic and political factors. In Eurasian largest exporter is Russia.

### 2.2.3 Natural gas supply

The supply for Natural Gas is mainly driven by the following factors:

- Pipeline Capacity
- Storage
- Gas Drilling Rates
- Natural Phenomena
- Technical Issues
- Imports

#### Pipeline Capacity

The ability to transport natural gas from the well heads of the producing regions to the consuming regions affects the availability of supply in the marketplace. The interstate and intrastate pipeline infrastructure has limited capacity and can only transport so much natural gas at any one time. This has the effect of limiting the maximum amount of natural gas that can reach the market. Natural gas pipeline companies continue to expand the pipeline infrastructure in order to meet growing future demand.

#### Storage

As natural gas injections (positive) represent additional demand, withdrawals (negative) represent an additional source of supply, which can be accessed quickly.

#### Gas Drilling Rates

The amount of natural gas produced both from associated and non-associated sources can be controlled to some extent by the producers. The drilling rates and gas prices form a feedback loop. When supply is low, demand and thus prices are high; this gives a market signal to the producer to increase the number of rigs being drilled for natural gas. The increased supply will then lead to decrease the pricing.

#### Natural Phenomena

Natural phenomena can have a significant impact on natural gas production and thus supply. Hurricanes, for example, can have an impact on the offshore production and exploitation of natural gas. This is because safety requirements may mandate the temporary shut down of offshore production platforms. Tornadoes can have a similar effect on onshore production facilities.

### Technical Issues

Equipment malfunction, although not frequent, could temporarily disrupt the flow across a given pipeline at an important market center. This would ultimately decrease the supply available in that market. On the other hand, technical developments in engineering methods can lead to more abundant supply.

### Imports

Imports are a source of supply. In Europe gas is imported from different countries; mainly from Russia but also in the form of LNG from countries such as Trinidad, Algeria and Nigeria.

### Trends in natural gas prices

Figure 2.4 shows a 75-year history of annual United States natural gas production and average wellhead prices from 1930 through 2005. Prices paid by consumers were increased above those levels by processing and distribution costs. Production is shown in billions of cubic meters per year, and average wellhead pricing is shown in US dollars per thousand cubic meters adjusted to spring, 2006.

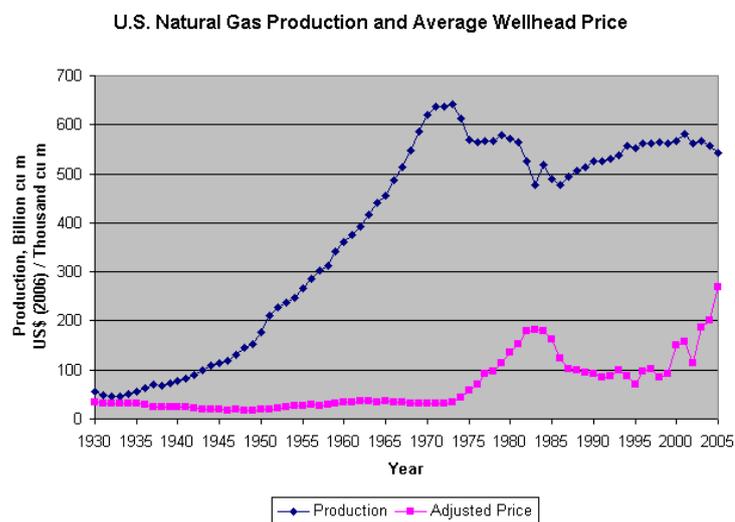


Figure 2.4: Natural gas production and price trend comparison in the U.S.[16]

Trend of natural gas prices for EU25 and its several countries is given in Figure 2.5.

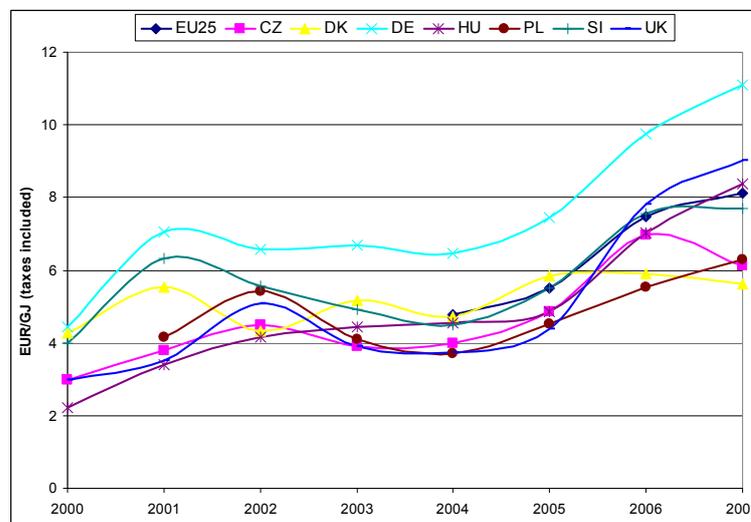


Figure 2.5: Natural gas prices for industry in EU25 [17]

Apparently the highest prices are met in Germany whilst the lowest prices are met in Czechoslovakia and Poland.

## 2.2.4 Storage and transport

The major difficulty in the use of natural gas is transportation and storage because of its low density.

### 2.2.4.1 Storage

Natural gas is often stored in underground caverns formed inside depleted gas reservoirs from previous gas wells, salt domes, or in tanks as liquefied natural gas. The gas is injected during periods of low demand and extracted during periods of higher demand. Storage near the ultimate end-users helps to best meet volatile demands, but this may not always be practicable. In the past, the natural gas which was recovered in the course of recovering petroleum could not be profitably sold, and was simply burned at the oil field (known as flaring). This wasteful practice is now illegal in many countries. Additionally, companies now recognize that value for the gas may be achieved with LNG, CNG, or other transportation methods to end-users in the future. The gas is now re-injected back into the formation for later recovery. This also assists oil pumping by keeping underground pressures higher. In Saudi Arabia, in the late 1970s, a "Master Gas System" was created, ending the need for flaring. The natural gas is used to generate electricity and heat for desalinization. Similarly, some landfills that also discharge methane gases have been set up to capture the methane and generate electricity.

### 2.2.4.2 Gas pipelines

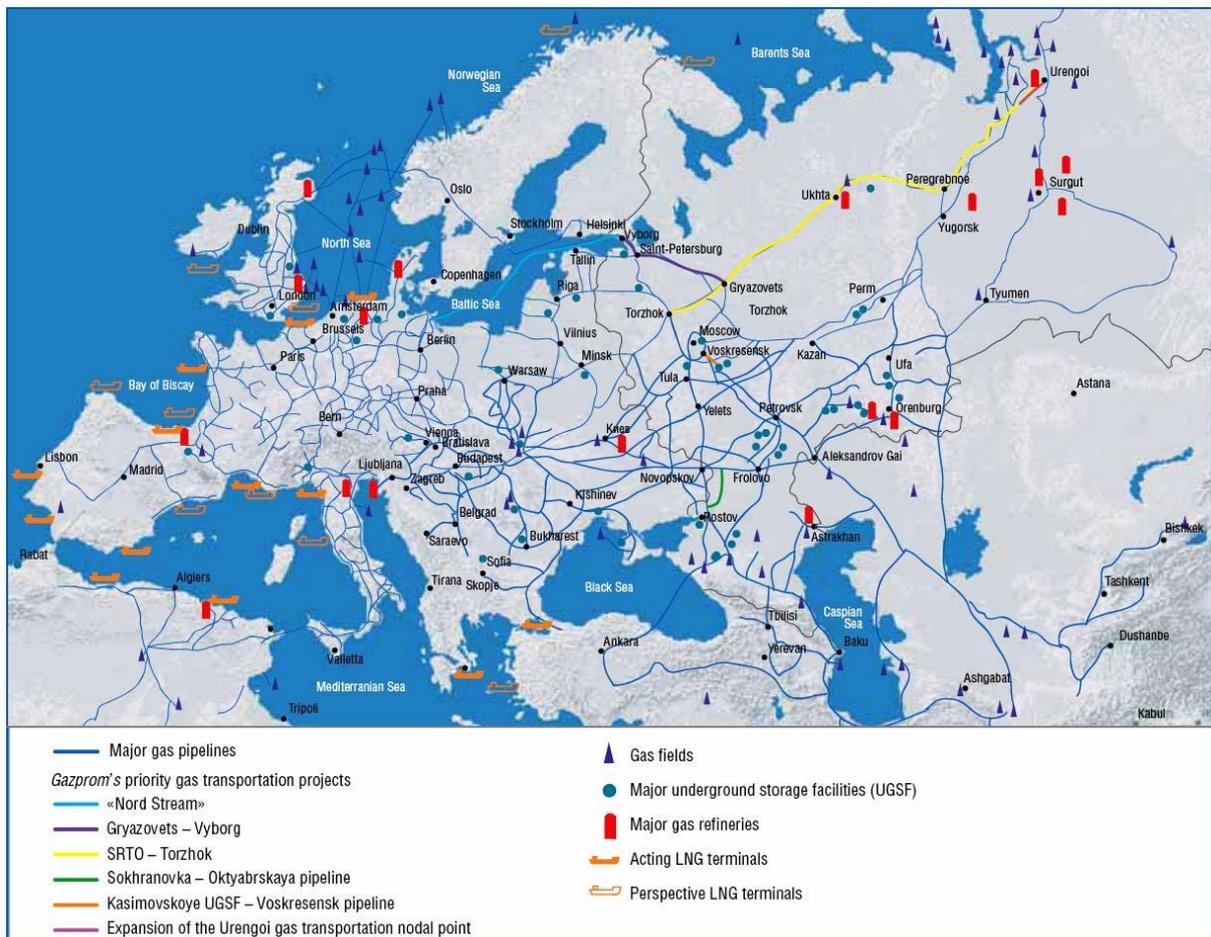


Figure 2.6: Eurasian gas transportation system [18]

Natural gas pipelines are economical, but are impractical across oceans. However in Eurasian they present essential form of gas transport to consumers. Figure 2.6 presents Eurasian gas transportation system with gas pipelines, gas fields, storage facilities, refineries and LNG terminals. Gas pipelines are used for supplying the natural gas from the reserves owing countries to the consuming industrial and residential centers.

#### 2.2.4.3 *Liquefied natural gas (LNG)*

When natural gas is cooled to a temperature of approximately  $-160\text{ }^{\circ}\text{C}$  at atmospheric pressure it condenses to a liquid called liquefied natural gas (LNG). One volume of this liquid takes up about 1/600th the volume of natural gas at a stove burner tip. LNG is only about 45% the density of water. LNG is odourless, colourless, non-corrosive, and non-toxic. When vaporized it burns only in concentrations of 5% to 15% when mixed with air. Neither LNG, nor its vapour, can explode in an unconfined environment. The process can also be designed to purify the LNG to almost 100% methane. Generally, LNG carriers are used to transport LNG across oceans. LNG has far more mass than oil to transport and most gas is transported by pipelines. LNG is in its gaseous state rather bulky. Gas travels much faster than oil though a high-pressure pipeline can transmit only about a fifth of the amount of energy per day. As well as pipelines, LNG is transported using both road/rail truck and ship. LNG will be sometimes taken to cryogenic temperatures to reduce the mass.

#### 2.2.4.4 *Compressed natural gas (CNG)*

Compressed natural gas (CNG) is natural gas pressurized and stored in welding bottle-like tanks at pressures up to 25 MPa. Typically it is same composition of the local "pipeline" gas, with some of the water removed. CNG and LNG are both delivered to gas engines as low pressure vapor (up to 2.1 MPa). CNG is often misrepresented as the only form natural gas can be used as vehicle fuel. LNG can be used to make CNG. This process requires much less capital intensive equipment and about 15% of the operating and maintenance costs.

Tank trucks are usually used to carry liquefied or compressed natural gas (CNG) over shorter distances. They may transport natural gas directly to end-users, or to distribution points such as pipelines for further transport. These may have a higher cost, requiring additional facilities for liquefaction or compression at the production point, and then gasification or decompression at end-use facilities or into a pipeline.

#### 2.2.4.5 *Liquid petroleum gas (LPG)*

Liquid petroleum gas (LPG, and sometimes called propane) is often confused with LNG and vice versa. They are not the same and the differences are significant. Varieties of LPG bought and sold include mixes that are primarily propane, mixes that are primarily butane, and mixes including propane, propylene, n-butane, butylene and iso-butane. Depending on the season—in winter more propane, in summer more butane. Vapour pressures, at  $30\text{ }^{\circ}\text{C}$ , are for commercial propane in the range 10-12 barg (1 to 1.2 MPa), for commercial butane, 2-4 barg (0.2 to 0.4 MPa). In some countries LPG is composed primarily of propane (upwards to 95%) and smaller quantities of butane. The vapour pressure of commercial butane is generally too low to release it from the top vapour space. Pumps and (hot water, steam, electricity or direct-fired) vaporizers are frequently used. An alternative to using neat butane vapour which overcomes the need for pipework heating is to use a gas-air mixture (well outside flammability limits). Air depresses the vapour dew-point temperature. Another advantage is that the mixture can be made to "simulate" natural gas or town

gas to produce the same heat release through a burner under equal supply pressures, characterized by a term known as Wobbe number or Wobbe index.

LPG compared to natural gas has a significantly higher heating value and requires a different air-to-gas mixture (propane: 24:1, butane: 30:1) for good combustion. LPG can be stored as a liquid in tanks by applying pressure alone. While the distribution of LNG requires heavy infrastructure investments (pipelines, etc.), LPG is portable. This fact makes LPG very interesting for developing countries and rural areas. LPG (sometimes called autogas) has also been used as fuel in light duty vehicles for many years. An increasing number of petrol stations around the world offer LPG pumps as well. A final example that should not be forgotten is that the "bottled gas" can often be found under barbecue grills.

### 2.2.5 Use

Natural gas, because of its clean burning nature, has become a very popular fuel for the generation of electricity. In the 1970's and 80's, the choices for most electric utility generators were large coal or nuclear powered plants; but due to economic, environmental, and technological changes, natural gas has become the fuel of choice for new power plants. Furthermore it is widely used in residential and commercial sector as also for transport and industry. As shown in Figure 2.7, natural gas demand is expected to increase significantly over the next 10 years. [1][19]

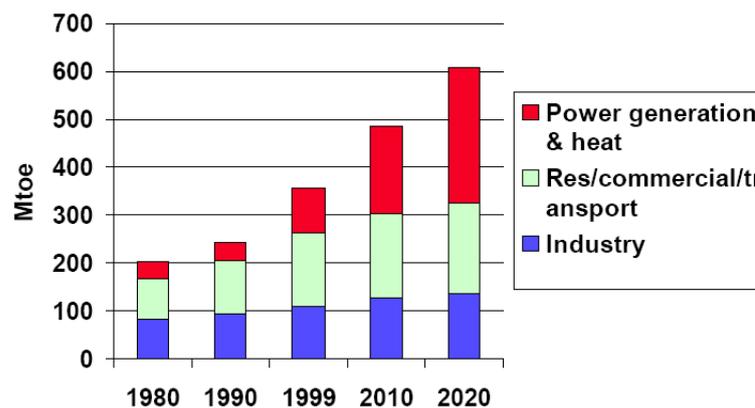


Figure 2.7: Gas demand in OECD countries [19]

There are many reasons for increased reliance on natural gas in electricity production. While coal is the cheapest fossil fuel for generating electricity, it is also the dirtiest, releasing the highest levels of pollutants into the air. [1]

CC power generation using natural gas is the cleanest available source of power based on fossil fuels and this technology is widely used wherever gas can be obtained at a reasonable cost. Fuel cell technology may eventually provide cleaner options for converting natural gas into electricity, but at the time being it is not price-competitive. Also, the natural gas supply is expected to peak around the year 2030, 20 years after the peak of oil. It is also projected that the world's supply of natural gas could be exhausted around the year 2085. [16] Cost of electricity generation from CC compared with other technologies is shown in Figure 2.8. for the year 1998 [12].

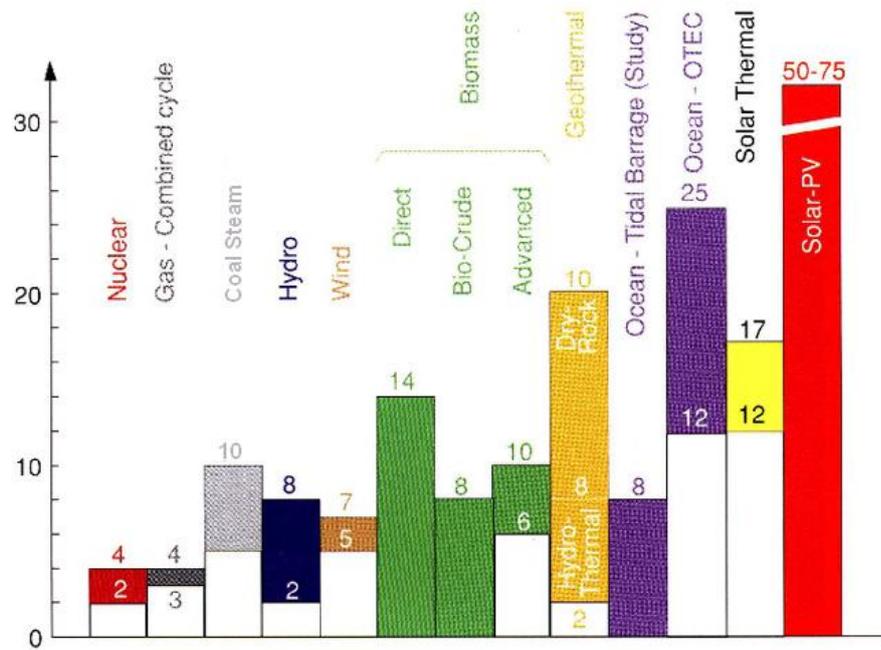


Figure 2.8: Cost of electricity (\$cents/kWh) (1998) [12]

Figure 2.9 shows cost of electricity generation in 2005 indicating slight increase for natural gas fired CC. [20]

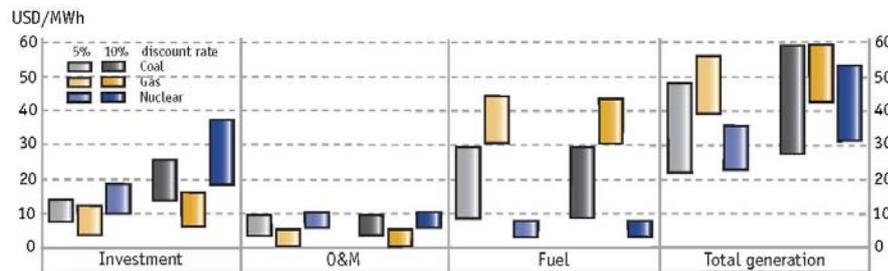


Figure 2.9: Cost of electricity (\$/MWh) (2005) [20]

However natural gas fired CC plant, which is studied out in this work, still presents quite attractive technology for electricity production.

Among mentioned natural gas can also be used for production of hydrogen, for use in fertilizer production, like also in the manufacture of fabrics, glass, steel, plastics, paint and other products.

### 3. Pipeline compressor stations

#### 3.1 Requirements on pipeline compressor stations

Because of the pressure loss which occurs during gas transmission through pipelines due to friction and gas off-take, the gas pressure has to be boosted every 150km in order to maintain operating pressure in the pipeline at approximately 80bar. That is done by means of compressor station comprising several compressor units. [4] These compressor units may be connected in series or parallel. Large pressure ratios require a number of compressors to operate in series whereas high natural gas volume flows at low pressure ratios necessitate that the compressor be arranged in parallel. [22]

Since the supply pressure to the pipeline station can fluctuate significantly (due to variations in mass flow) the pressure ratio generated in the compressor has to vary accordingly. Because of economic reasons, operation at optimum efficiency is constantly required also under such variable conditions, wherefore compressor must be capable to operate over as wide a speed range as possible. It is important to provide a compressor operation control so that on reduction of the compressor load it is possible to approach the performance map limits. With respect to the curves shown in Figure 3.1 compressor speed regulation has to provide operation at highest possible polytrophic efficiency aiming to minimize the energy input respectively fuel consumption. [22]

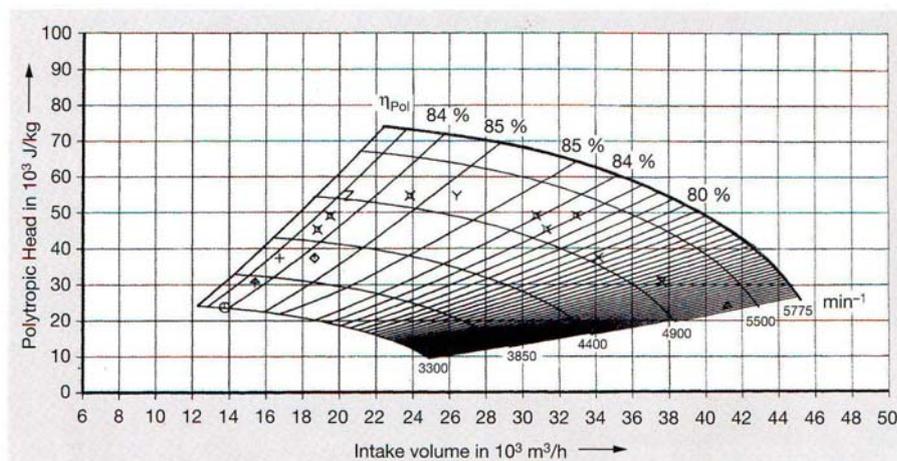


Figure 3.1: Pipeline compressor characteristic [22]

Except in gas pipelines, compressor stations may be employed in underground storage facilities in order to raise the gas pressure being injected into storage, or to compress the natural gas as it leaves storage to be fed into the pipeline. [4] Pipeline compressors are also used in gas collection and gas processing plants as export compressors. [23]

### 3.2 Technical features of pipeline compressor stations



*Figure 3.2: Pipeline compressor station [1]*

Gas turbines driving the pipeline compressors gain their energy by using up a small proportion of the natural gas that they compress. The turbine itself serves to operate a centrifugal compressor, which contains a fan that compresses and pumps the natural gas through the pipeline. Some compressor stations are operated by using an electric motor to turn the same type of centrifugal compressor. This type of compression does not require the use of natural gas from the pipe; however it does require a reliable source of electricity nearby. Reciprocating natural gas engines are also used to power some compressor stations. These engines resemble a very large automobile engine, and are powered by natural gas from the pipeline. The expansion of combusted gas powers pistons on the outside of the engine, which serves to compress the natural gas.

In addition to compressing natural gas, compressor stations also usually contain some type of liquid separator, much like the ones used to dehydrate natural gas during its processing. Usually, these separators consist of scrubbers and filters that capture any liquids or other undesirable particles from the natural gas in the pipeline. Although natural gas in pipelines is considered 'dry' gas, it is not uncommon for a certain amount of water and hydrocarbons to condense out of the gas stream while in transit. The liquid separators at compressor stations ensure that the natural gas in the pipeline is as pure as possible, and usually filters the gas prior to compression. [1]

Figure 3.3 shows configuration of a natural gas pipeline system, in which gas is pumped from a well head to industrial and domestic consumers. Pipelines have diameters of typically 915-1420 mm and are usually underground. A notable exception is in permafrost areas where they must be raised to avoid melting the permafrost. These systems may extend over thousands of kilometers, with compression stations approximately every 150-200 km. [24]

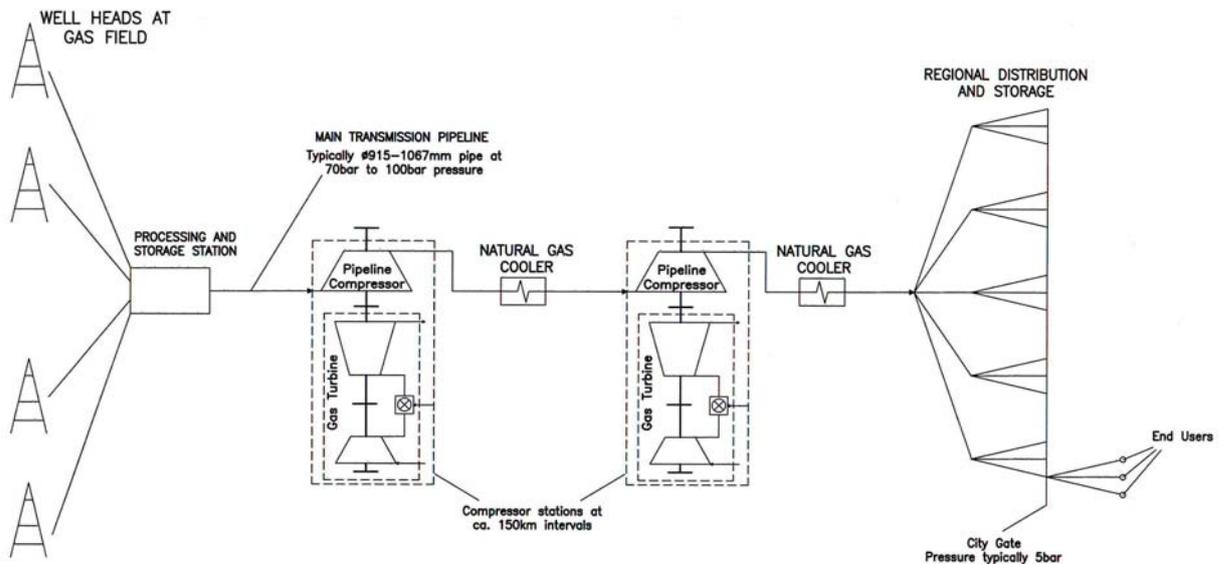


Figure 3.3: Natural gas pipeline system [24]

In order to maximize reliability of compressor stations particular attention is paid to the arrangement and selection of machine sets. Common pipeline compressor station consists of a gas turbine as prime mover and a pipeline compressor as a driven engine, which are described in following paragraphs 3.2.1 and 3.2.2.

### 3.2.1 Gas turbines as prime movers for pipeline compressor stations

Gas turbine is favourable prime mover for pipeline compressors in compressor stations thanks to its ability to be fired by various fuels including oil and natural gas. Another advantage is ability of a variable speed depending on varying mass flows of supplied medium. In most of these cases, fuel fired in the gas turbine is the same as the one being transported by means of driven compressor, what is also the case in the system analyzed in this work. [25] The major power blocks required for running pipeline compressors are around 6–10MW, 15MW and 25–30MW. These power levels generally exceed the practical size for a high-speed diesel engine given the requirements outlined below, and hence are gas turbines used almost exclusively. In order of importance these requirements are sorted by:

- low weight, as the engines often have to be transported to remote locations, where it also may be difficult and costly to build substantial foundations
- good base load thermal efficiency, since utilization is as near 100% as possible
- reasonable part power torque, to respond to load changes on the gas compressor. However a fast start time is not essential, and a loading rate of 2 minutes from idle to full power is typical

For offshore well heads the turbine must be located on a platform, hence the importance of low weight is amplified and low volume essential. While the gas or oil may have a high pressure as it comes out of the ground it invariably needs further pressurization to pipe it back onshore. [24]

Generally, multi-shaft gas turbines are more suitable for driving pipeline compressors and comprise gas generator and a freewheeling power turbine. Example of such GT design is shown in Figure 3.4.

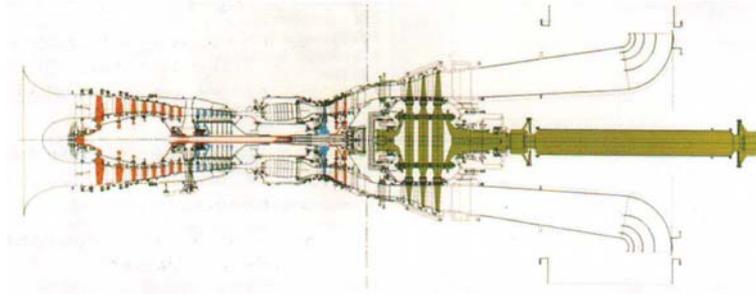


Figure 3.4: Sectional drawing of a 25MW GT for driving pipeline compressors [23]

Gas turbine types used in compressor stations may also be referred to as medium-range gas turbines. [14] Their efficiency is achieved by letting the compressor section operate at maximum efficiency while the power turbine operates over a great range of speeds. It is interesting to note that 35% of produced industrial gas turbines (power up to 50 MW) are delivered to the oil & gas sector. [26]

### 3.2.2 Pipeline compressors

Generally, centrifugal compressors use a vaned rotating disk or impeller in a shaped housing to force the gas to the rim of the impeller, increasing the velocity of the gas. A diffuser (divergent duct) section converts the velocity energy to pressure energy. They are primarily used for continuous, stationary service in industries such as oil refineries, chemical and petrochemical plants and natural gas processing plants. With multiple staging, they can achieve extremely high output pressures greater than 690 bar. [13] Example of one as-built pipeline compressor is shown in Figure 3.5. [23]



Figure 3.5: Pipeline compressor [23]

Technical features of such pipeline compressor station using gas turbine as prime mover are given in Table 3-1. [23]

Table 3-1: Technical data of pipeline compressor [23]

Data/Characteristic	Unit	Value
Casing		Vertically split external casing
Discharge pressure	[bar]	up to 130
Suction volume flow, eff.	[m <sup>3</sup> /h]	from 2000 to 85000
Gas flow transported	[Nm <sup>3</sup> /h]	from 100000 to 3000000

Centrifugal type pipeline compressors shown in Figure 3.5 are designed with a vertically split casing with connecting nozzles fitted horizontally on opposite sides of the casing. They are used for discharge pressures of up to 130 bar and power inputs of up to 35 MW. The effective intake flow is between approx. 3000m<sup>3</sup>/hr and approx. 60,000m<sup>3</sup>/h. Depending on the level of inlet pressure in the pipeline, gas flows of approx. 100,000 Nm<sup>3</sup>/hr to approx. 3,000,000 Nm<sup>3</sup>/hr can be transported through the pipeline. The connecting nozzles are normally fitted horizontally on opposite sides of the casing as shown in Figure 3.5. This design, together with the compact installation on the base frame enables absorption of the greater forces that arise from the connected piping in this application. The vertical design of the casing with the laterally fitted flanges enables the inspection and replacement of all internals without separating the connections to the gas line. If required, the operating characteristics can be very quickly adopted to e.g. differences in seasonal demand or longer-term capacity increases by replacing the inner bundle as a whole.

### 3.3 Pipeline compressor station example

An example of a pipeline compressor station, using THM1304 gas turbine as prime mover, is project used as terminal station for the Rio Grande pipeline in Bolivia. THM1304 is a newer version of THM1203.[23]

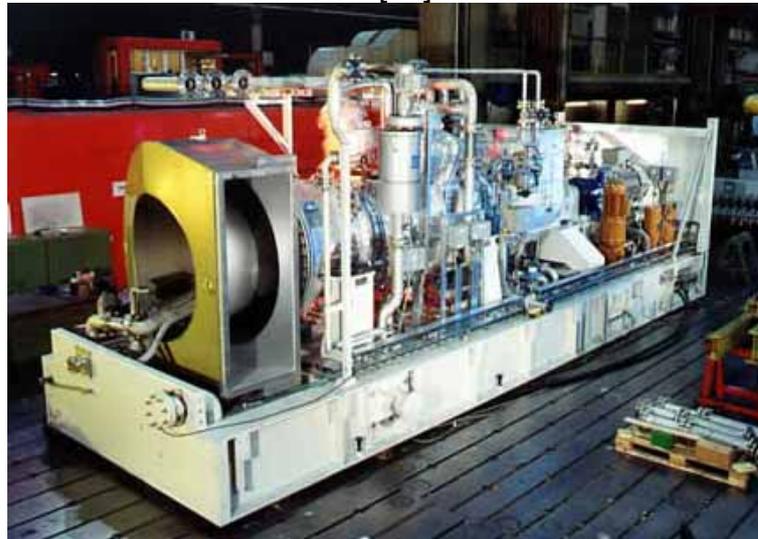


Figure 3.6: THM1304-11 gas turbine for pipeline compressor station in Bolivia [23]



Figure 3.7: Centrifugal compressor for pipeline compressor station in Bolivia [23]

Upstream of this compressor station is a natural gas processing plant, where the gas obtained from reserves in the region of Santa Cruz is processed by removing the heavy hydrocarbons and cleaning it. The type THM 1304-11 gas turbine shown in Figure 3.6 drives a centrifugal pipeline compressor type RV063/04 shown in Figure 3.7. The industrial design of this gas turbine enables maintenance of machines in situ. [23] Technical data of this pipeline compressor station are given in Table 3-2.

Table 3-2: Technical data natural gas compressor unit for pipeline in Bolivia [23]

Gas turbine Type THM1304-11		
ISO output	[kW]	11200
Compressor Type RV050 / 04		
Volumetric flow	[m <sup>3</sup> /h]	8000
Speed	[rpm]	9000
Inlet pressure	[bar]	60
Outlet pressure	[bar]	100

### 3.4 Potential for utilization of waste heat

Currently, heat embraced in the flue gases leaving the gas turbine of pipeline compressor stations explained in 3.1 and 3.2, is not utilized at all or is only partially recovered. Potential of this heat in the pipeline compressor stations which are part of gas mains of Russia and member-countries of the Commonwealth of Independent States (CIS) is estimated at 1260 million GJ/year. By utilizing this heat, up to 39400 GWh of electricity can be produced. [27]

Most compressors in pipeline compressor stations are driven by gas turbines of 6 to 25MW capacity. [27] Typically, exhaust gas temperatures are from 450-550°C and present valuable clean energy in the form of exhaust heat, which may be recovered. According to assessment done in [27] waste heat utilization can be done either by means of 1P or 2P HRSG, wherein the latter gives ca. 4% higher thermal efficiency.

Physical properties of water (steam) and necessity of non-zero value of pinch point (PP) limit the waste heat recovery of 1P HRSG. Therefore it is expected that employment of additional pressure levels leads to certain efficiency increase. Since more pressure levels make the system more complicated and more expensive system selection is based on compromise using exergo-economic criteria respectively economic evaluation with assumed hourly demand of the plant.

### 3.4.1 Waste heat utilization application examples

Figure 3.8 presents a waste heat recovery plant at the site of the Weitendorf gas compressor station in Austria. The project is being developed by OMV Gas & Power and is actually one of the frontrunners of this technology in Europe.



Figure 3.8: Planned waste heat recovery plant at Weitendorf gas compressor station [28]

The plant is characterized by a compressor power of 40 MW and an installed electrical power of 16 MW producing 100 GWh able to satisfy the annual demand of approximately 28,500 households. Through the recovery of waste-heat ca. 90000 t/a CO<sub>2</sub> can be substituted compared to conventional electrical power generation using coal. Furthermore it is possible to equip the waste-heat recovery plant with low temperature heat extraction and thus raise the overall efficiency even further if demand for district heating by the municipality or commercial operators emerges.[28] Figure 3.9 presents the waste heat recovery units at the compressor station Ruswil belonging to Transitgas AG.



Figure 3.9: Waste heat recovery unit at the compressor station Ruswil [29]

From the waste heat of the gas turbines two employed waste heat recovery units produced 21.74 GWh of electrical energy in 2008. This energy was fed into the mains of the local electricity distributor. Besides producing electricity, the waste heat recovery units generate hot water for the internal use of the station and for a nearby greenhouse. It is interesting to note that the laws of the canton of Lucerne mandatory require the use of waste heat.[29]

## 4. Technical features of analyzed pipeline compressor station

### 4.1 Function

Considered pipeline compressor station Pico Truncado belongs to TGS gas pipeline company whose transportation system connects Argentina's southern and western gas reserves to the main consumption centers. It has a contracted capacity of 75,9 mil. m<sup>3</sup>/d transporting 62% of the gas consumed in Argentina. The pipeline system consists of three pipelines: Neuba I, Neuba II and San Martin as shown in Figure 4.1. Considered compressor station is part of the pipeline San Martin.



Figure 4.1: TGS pipelines [30]

Complete TGS pipeline system has a total length of 7471km and comprises 30 compressor stations. The pipeline San Martin is 3756km long, has a nominal pipe size of 30" and is equipped with 16 compressor stations with a total power of ca. 486MW. [30]

### 4.2 Configuration

Concerned pipeline compressor station, as shown in Figure 4.2 consists of 4 pipeline compressors each being driven by THM1203 gas turbine, whereby 2 or 3 gas turbines are in operation whilst 1 is used for backup for the case that any of GT would be out of operation. Load duration curve given per Figure 4.4 presents the load of the pipeline compressors that has to be covered by the gas turbines.

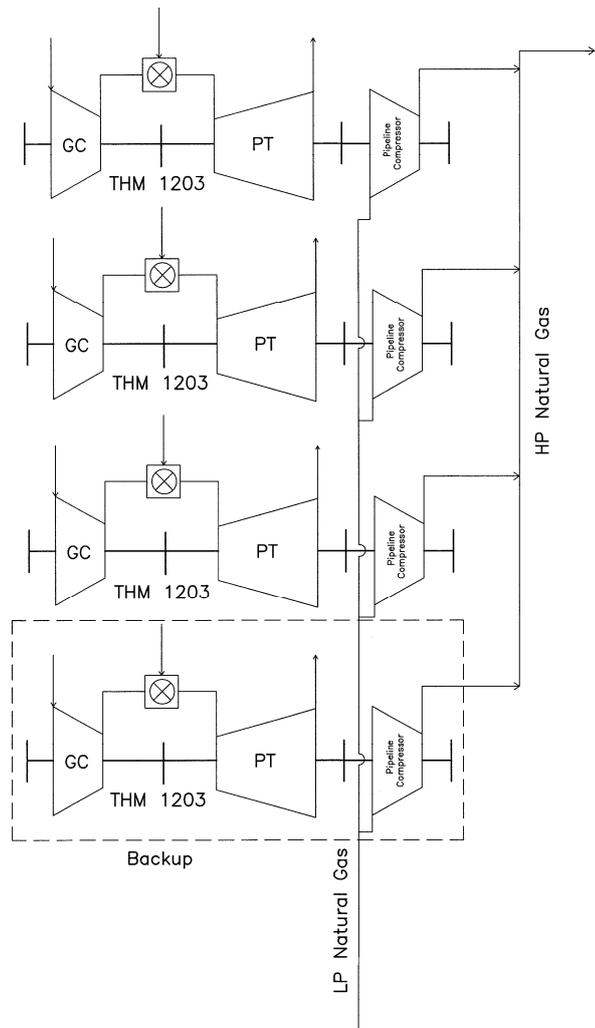


Figure 4.2: Compressor station schematic [23]

### 4.3 Demand analysis

Annual demand at pipeline compressor shafts on hourly basis is given in Figure 4.3.

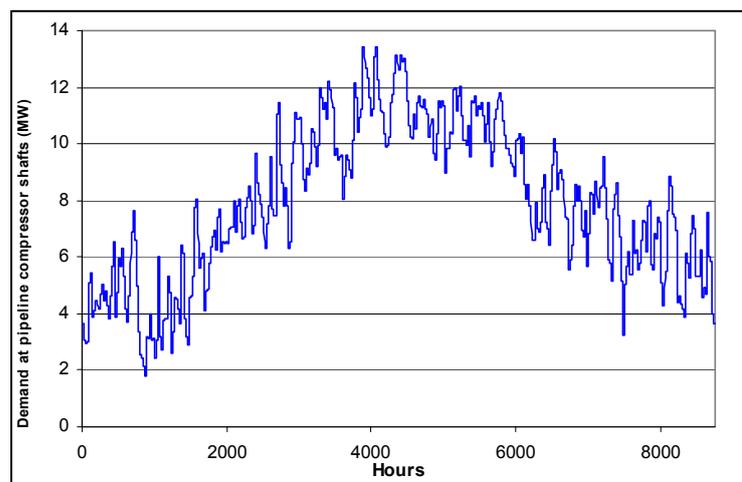


Figure 4.3: Annual demand at pipeline compressor shafts [23]

From this demand pattern a load duration curve is constructed and is shown in Figure 4.4. For simplification of calculations this annual system characteristic is divided into

10 operating regimes whose power demand is obtained as an average power over each time period as shown in Figure 4.4.

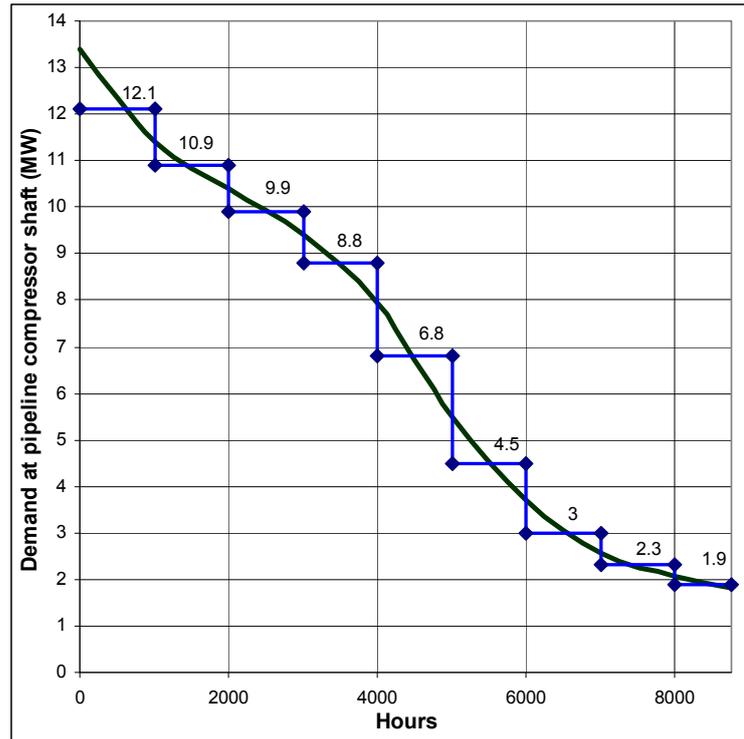


Figure 4.4: Load duration curve [23]

Each operating regime is characterized by the number of gas turbines in operation and their part load fraction. That defines the mass flow and temperature of flue gases at GT outlet as shown in Table 4-1.

Table 4-1: Compressor station operating data

Regime	1	2	3	4	5	6	7	8	9	10
$P_{\text{mech}}$ (MW) GT Total	12.1	10.9	9.9	8.8	6.8	6.8	4.5	3	2.3	1.9
No. of GT	3	3	3	3	3	2	2	2	2	2
$P_{\text{mech}}$ (MW) per 1 GT	4.03	3.63	3.3	2.93	2.27	3.4	2.25	1.5	1.15	0.95
$\eta_{\text{GT}}$ (%)	20.4	19.4	18.5	17.4	15.9	19.2	15.6	13	11.4	10
$\dot{m}_{\text{flue gases}}$ (kg/s)	31.8	31	30	29	27.4	31.5	28	24.5	22	21
$T_{\text{flue gases}}$ ( $^{\circ}\text{C}$ )	468	460	455	450	414.7	427	415	382	381	381
GT part load (%)	76%	68%	62%	55%	43%	64%	42%	28%	22%	18%

It is interesting to note that the natural gas demand scenario shows dependency on ambient temperature at location whose annual profile for 2005 is given in Figure 4.5.

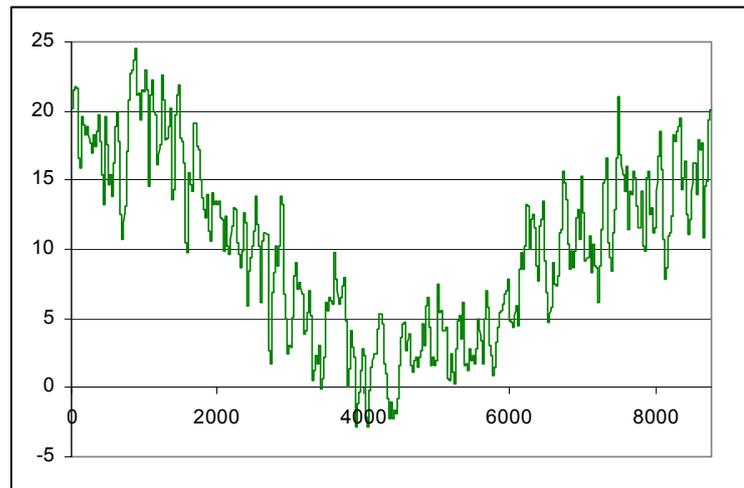


Figure 4.5: Ambient temperature annual profile (2005) [31]

#### 4.4 Gas Turbine output data

Gas turbines running pipeline compressors in different operating regimes have an output power of 12.1 MW down to 1.9 MW. Each of 10 operating regimes characterizes different flue gas parameters, gas turbine power, efficiency and ambient temperature, as shown in Table 4-1 [23]. Flue gases mass flow and temperature as a function of gas turbine load are shown graphically in Figure 4.6 and Figure 4.7. These parameters are the boundary conditions for further heat balance calculations of the HRSG. Dependence of GT el. efficiency of GT load is given in Figure 4.8

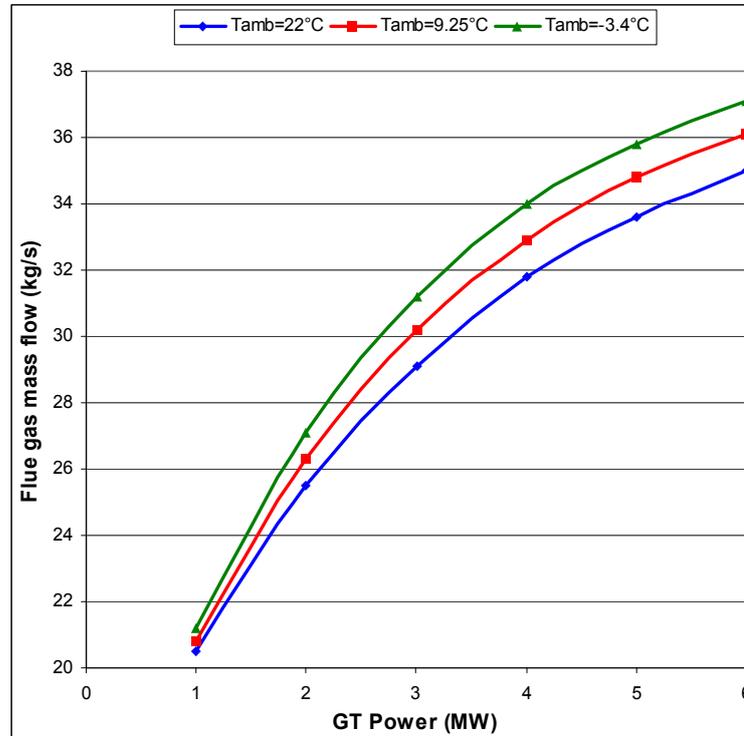


Figure 4.6: Flue gases mass flow vs. part load fraction [23]

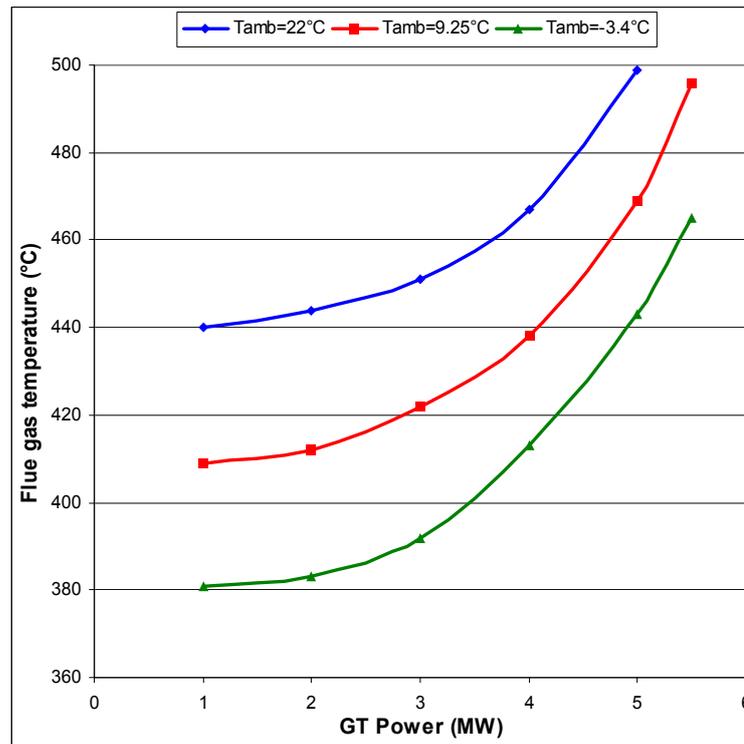


Figure 4.7: Flue gases temperature vs. part load fraction [23]

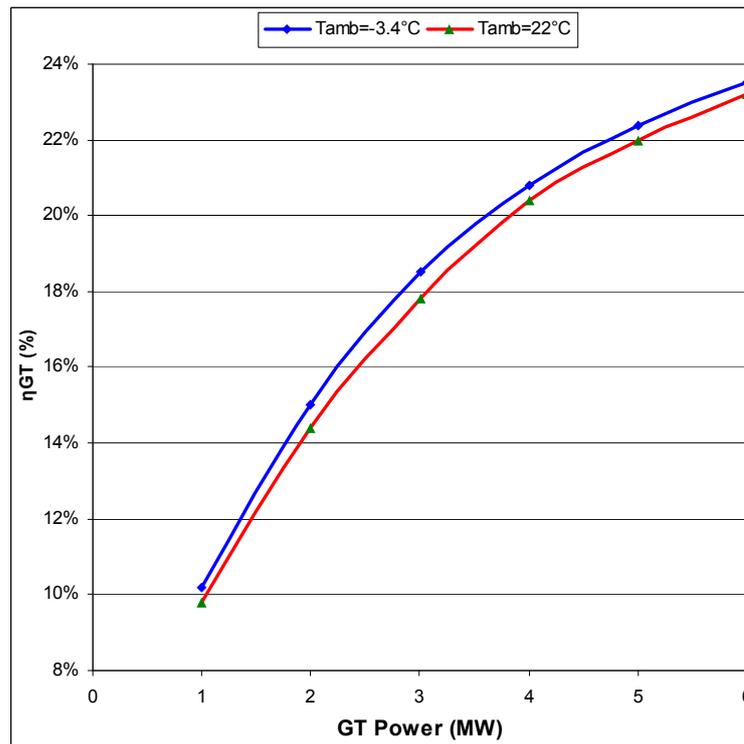


Figure 4.8: GT efficiency vs. part load fraction [23]

Composition of exhaust gas at the GT outlet is given in Table 4-2.[23]

*Table 4-2: Exhaust gas Composition*

Gas	vol. %
Nitrogen	75
Oxygen	15,3
Carbon Dioxide	2,4
Argon	0,9
Water	6,3

Composition of natural gas burned in gas turbines is given in Table 4-3 [23] and has LHV of 48000 kJ/kg.

*Table 4-3: Natural gas Composition*

Gas	vol. %
CH <sub>4</sub>	96,5
C <sub>2</sub> H <sub>6</sub>	1,0
C <sub>3</sub> H <sub>8</sub>	0,4
N <sub>2</sub>	1,8
O <sub>2</sub>	0,3

Design point data valid for THM1203 gas turbines employed at TGS Pico Truncado pipeline compressor station are given in Table 4-4 [23].

*Table 4-4: Design data gas turbines*

Data	Value
P <sub>mech</sub> (MW) GT Total	15.6
No. of GT	3
P <sub>mech</sub> (MW) per 1 GT	5.3
η <sub>GT</sub> (%)	22.4
$\dot{m}_{\text{flue gases}}$ (kg/s)	34.8
T <sub>flue gases</sub> (°C)	496

## 5. Waste heat recovery via gas and steam combined cycle

In this chapter thermodynamic aspects of a combined cycle are discussed focusing on application of this cycle for utilizing the waste heat from pipeline compressor stations.

### 5.1 CC thermodynamic principles

Conventional CC power plant consists of the combination of Brayton and Rankine cycle forming one of the most efficient cycles used for power generation today. Therein the Brayton Cycle is the gas turbine cycle, also called the topping cycle and the Rankine Cycle is the steam turbine cycle also described as bottoming cycle. Thermal efficiency of the CC plants given in Figure 5.1 is today somewhat higher and exceeds 60% [7]. Higher efficiency can be gained with CC plants employing advanced gas turbine cycles as: compressed air energy storage cycle, intercooled simple cycle, reheat cycle, intercooled regenerative reheat cycle, steam injection cycle or evaporative regenerative cycle.[14] Even higher efficiency is gained with hybrid power plants coupling e.g. solar thermal technology with combined cycle. [13]

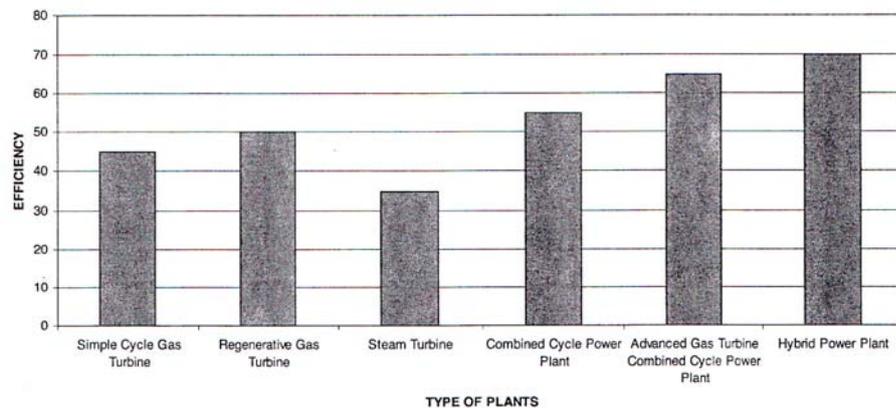


Figure 5.1: Efficiencies of various types of plants (2002) [14]

In the typical combination, the gas turbine produces about 60% of the power and the steam turbine about 40%. Individual unit thermal efficiencies of the gas turbine and the steam turbine are between 30% and 40%. The steam turbine utilizes the energy of the gas turbine exhaust gas as its input energy. The energy transferred from the gas turbine to the HRSG is usually equivalent to about the rated output of the gas turbine at design conditions. At gas turbine off-design conditions, the inlet guide vanes (IGV) are used to regulate the air flow so as to maintain high temperatures at GT outlet respectively at HRSG inlet. [14]

Simplified schematic of CC process is given in Figure 5.2 followed by process T-s diagram given in Figure 5.3

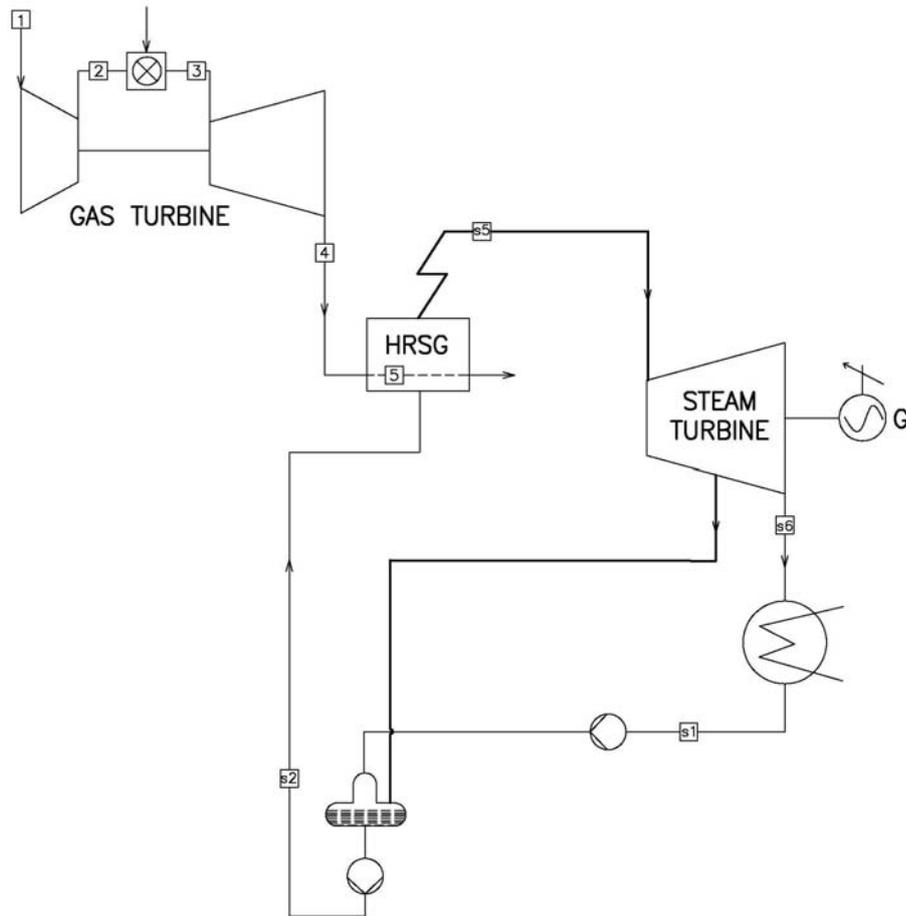


Figure 5.2: Simplified flow scheme of CC plant

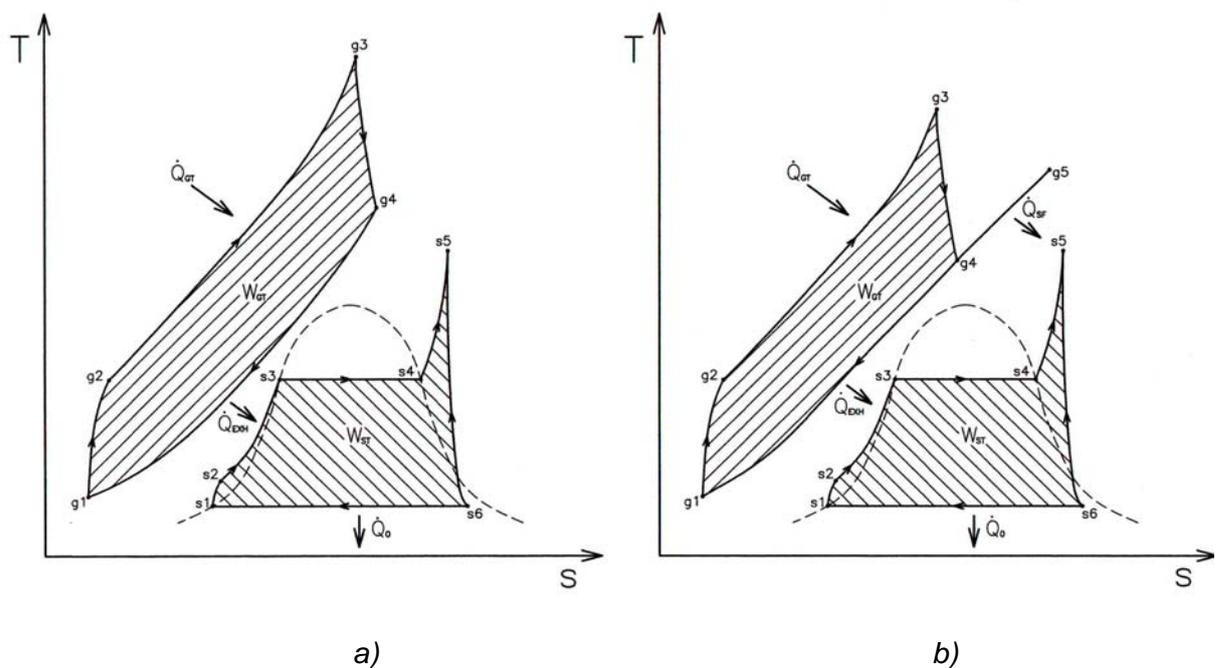


Figure 5.3: T-s diagram of CC process without (a) and with supplementary firing (b)

The gas turbine cycle in its ideal form consists of two isobaric processes and two isentropic processes. The two isobaric processes consist of the combustion system of the gas turbine and the gas side of the HRSG. The two isentropic processes

represent the compression which is done in compressor (gas generator [23]) and the expansion process which is done in the turbine expander (power turbine [23]).

In a steam turbine cycle water enters the feed water pump and is pumped isentropically into the HRSG. Heat released from the gas turbine ( $Q_{EXH}$ ) is used for heating up the compressed water until saturation temperature, its evaporation and superheating. The steam leaving the HRSG expands isentropically in the steam turbine and passes to the condenser. Cooling water condenses the steam to a saturated liquid from which state the cycle repeats itself.

Heat induced into the gas turbine cycle ( $Q_{GT}$ ) is actually a surface under the curve g1-g2-g3-g4 in the Figure 5.2. Heat released from the gas turbine cycle ( $Q_{EXH}$ ) is actually a surface under the curve g1-g4. This amount of heat is induced into the steam turbine cycle and is also represented as a surface under the curve s2-s5. In case of using the supplementary firing amount of heat induced into the steam cycle is the sum of heat  $Q_{EXH}$  and  $Q_{SF}$ .  $Q_{SF}$  refers to the amount of heat induced through burning additional fuel in the duct burner and is actually a surface under the curve g4-g5.

Work obtained in the gas turbine cycle ( $W_{GT}$ ) is presented as a surface within the curve g1-g2-g3-g4 whilst the work obtained in the steam turbine cycle ( $W_{ST}$ ) is presented as a surface within the curve s1-s2-s3-s4-s5-s6.  $Q_0$  refers to the amount of heat transferred from the cooling water to the condensing steam in condenser.

Thermal efficiency of gas respectively steam cycle can be expressed per (5.1-1) and (5.1-2):

$$\eta_{GT} = \frac{W_{GT}}{\dot{Q}_{GT}} \quad (5.1-1)$$

$$\eta_{SC} = \frac{W_{ST}}{\dot{Q}_{EXH}} \quad (5.1-2a),$$

$$\eta_{SC} = \frac{W_{ST}}{\dot{Q}_{EXH} + \dot{Q}_{SF}} \quad (5.1-2b)$$

Thereby equation (5.1-2a) refers to the cycle without supplementary firing and (5.1-2b) to the cycle with supplementary firing. By combining these two equations, thermal efficiency of CC can be expressed per (5.1-3):

$$\eta_{CC} = \frac{W_{GT} + W_{ST}}{\dot{Q}_{GT}} \quad (5.1-3a),$$

$$\eta_{CC} = \frac{W_{GT} + W_{ST}}{\dot{Q}_{GT} + \dot{Q}_{SF}} \quad (5.1-3b)$$

In slightly different form, CC thermal efficiency can be expressed per (5.1-4):

$$\eta_{CC} = \eta_{ST} + \eta_{GT} \cdot (1 - \eta_{ST}) \quad (5.1-4a),$$

$$\eta_{CC} = \eta_{ST} + \eta_{GT} \cdot (1 - \eta_{ST}) \cdot \frac{\dot{Q}_{GT}}{\dot{Q}_{GT} + \dot{Q}_{SF}} \quad (5.1-4b)$$

Analogically equations (5.1-3a) and (5.1-4a) refer to the cycle without supplementary firing whilst equations (5.1-3b) and (5.1-4b) refer to the cycle with supplementary firing. Since the last term in expressions (5.1-4a) and (5.1-4b) can not be negative, it means that the total efficiency of the combined cycle must exceed the efficiency of the steam turbine cycle.

## 5.2 Heat recovery steam generator

Heat recovery steam generator (HRSG) may be defined as a heat exchanger that recovers heat from a hot gas stream. It produces steam that can be used in a process or used to drive a steam turbine. A common application for a HRSG is in a CC power station, where hot exhaust from a gas turbine is fed to HRSG to generate steam which in turn drives a steam turbine. This combination produces electricity more efficiently than either the gas turbine or steam turbine alone. Example of GT and HRSG layout is given in Figure 5.4.

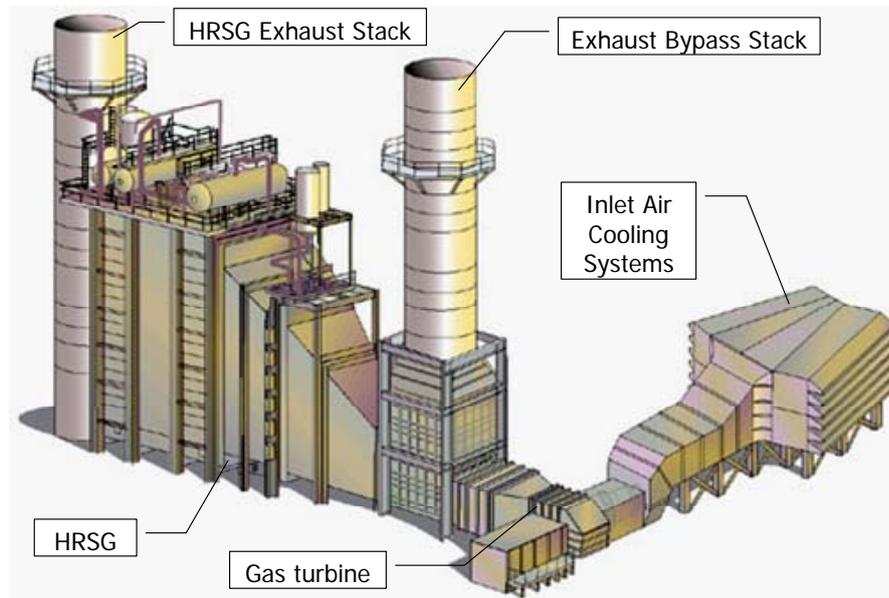


Figure 5.4: GT and HRSG layout [11]

Some HRSG include supplemental, or duct firing. These additional burners provide additional energy to the HRSG, which produces more steam and hence increases the output of the steam turbine. Generally, duct firing provides electrical output at lower capital cost but at lower efficiency compared with combined cycle generation, as further explained in 5.2.2. Emissions controls may also be located in the HRSG. Some may contain a Selective Catalytic Reduction system to reduce nitrogen oxides (a large contributor to the formation of smog and acid rain) and/or a catalyst to remove carbon monoxide. [13]

There are many different configurations of the HRSG units. Most HRSG units are divided into the same amount of sections as the steam turbine. In most cases, each section of the HRSG has air pre-heater, an economizer, evaporator, and superheater. The steam entering the steam turbine is superheated.

In the past, the HRSG was viewed as a separate item of CC system. This view is being changed with the realization that good performance, both thermodynamically and in terms of reliability, grows out of designing the heat recovery system as an integral part of the overall system. The concept that the CC power plant is always base loaded has also been changed and with that, some of the design philosophy behind the HRSG.

Depending on the steam requirements, HRSG can have one of the three forms:

- Unfired
- Supplementary-fired
- Exhaust-fired

In unfired HRSG, the energy from the exhaust is used as such, while in supplementary-fired and exhaust-fired HRSG, additional fuel is inputted to the exhaust gas to increase the steam production. [14]

Unfired and supplementary-fired HRSGs are similar in appearance and construction, both being convective designs. Roughly 2/3 of the HRSGs purchased today are unfired since their investment costs are lower than in the case of fired HRSGs. [32]

A supplementary-fired HRSG, shown in Figure 5.5, has a duct burner located upstream. A duct burner typically has a rectangular cross-section and fits into the ductwork carrying the exhaust gases. It consists of vertical or horizontal grids with holes that provide combustion of injected fuel (such as natural gas or distillate oil) in the exhaust gas stream. Generally, no additional air is used, except when the exhaust gas is injected with large quantities of steam, which reduces the amount of oxygen available for combustion. [32]

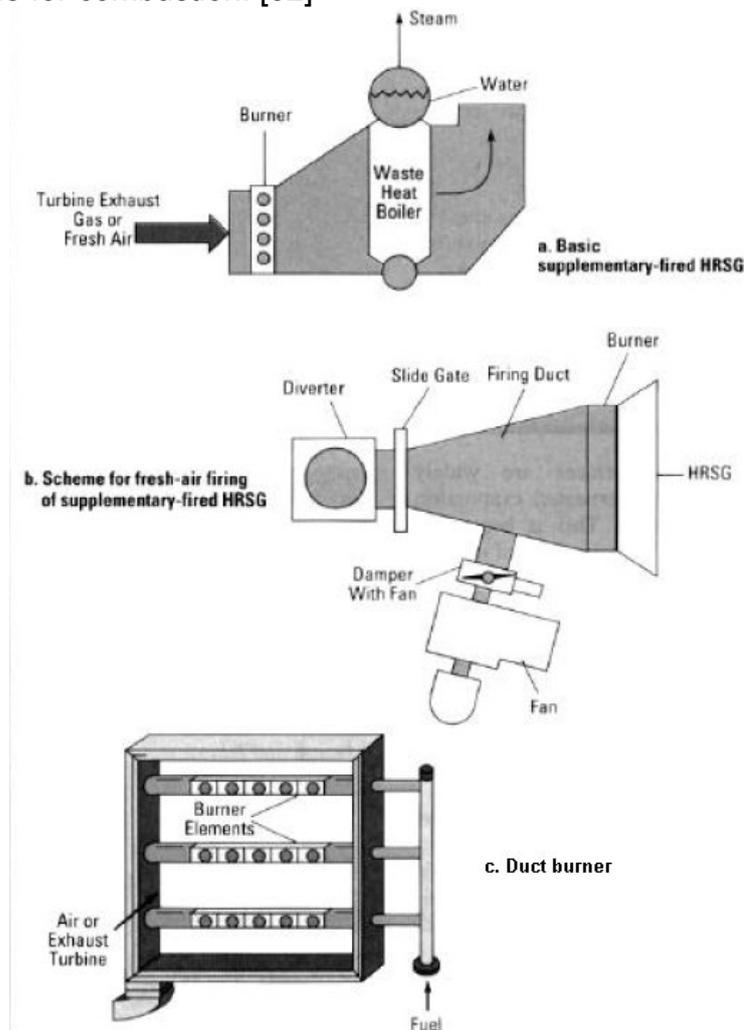


Figure 5.5: Supplementary-fired HRSG [32]

In the exhaust-fired HRSG shown in Figure 5.6 the firing temperature ranges from 900°C to 1600°C and it uses a completely water-cooled furnace to contain the flame, since the temperature could approach the adiabatic combustion temperature. The burner used is typically a register burner with a windbox, although a duct burner may be used up to 1300°C. The gas turbine exhaust is used as hot air for combustion. In certain plants, even solid fuels such as coal have been fired in these HRSGs using register burners.

The gas pressure drop across the register burner is high (about 10-15 mbar). The HRSG is typically a 1P unit in these systems, as the exit gas temperature can be brought down to ca. 150°C, unlike in an unfired or supplementary-fired HRSG, which has a high exit gas temperature if a 1P system is used.

Due to the high gas temperature entering the HRSG, the design is likely to consist of more bare tubes than finned tubes. A radiant furnace is required to cool the gases before they enter the superheater or convective sections. [32]

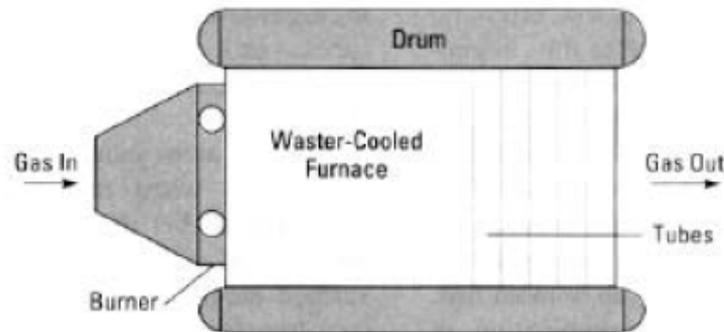


Figure 5.6: Exhaust-fired HRSG [32]

HRSG's are key components in combined cycle plants. On one side, they have to be designed for the quick start characteristics of the gas turbine and, on the other; they have to maintain selected respectively required steam parameters. In CC processes used for waste heat utilization mostly HRSG's with maximum possible supplementary firing are used. Figure 5.7 shows the construction of a steam generator of this type, whose pressure part can be freely suspended expanding downwards underneath and which was based on the 1½ pass design. The high adiabatic combustion temperatures require water-cooled membrane walls which are designed as a natural circulation evaporator. Two injection coolers are arranged between the superheaters to control the superheated steam temperature. The convection heating surfaces consist of a convection evaporator and the economizer in addition to the three-stage superheater. [33]

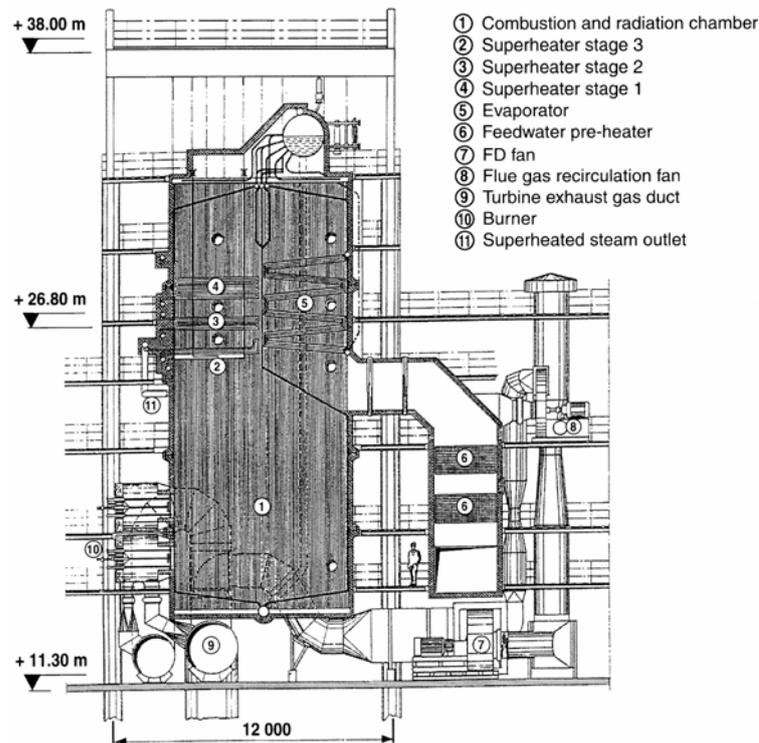


Figure 5.7: Steam generator with higher level of supplementary firing [22]

### 5.2.1 Single pressure and double pressure systems

Heat recovery with single-pressure HRSG becomes limited at higher pressures due to higher saturation temperatures. In some cases, multiple-pressure HRSG may be used to serve process loads or gas turbine steam injection requirements, improving overall efficiency. One to four separate pressure sections may be used, each with superheater, evaporator, and economizer sections. Multiple-pressure operation is also effective for increasing CC system efficiency where admission steam turbines include high pressure and low pressure inlets.[34]

In [27][22] application of 1P and 2P HRSGs for waste heat utilization from pipeline compressor stations is considered. In both cases supplementary firing plays important role at GT part-load operation for maintaining the required steam parameters. In consideration of various systems configurations in chapter 7, 3P configuration is also considered.

Flow schemes of typical single pressure and double pressure heat recovery plants are given in Figure 5.8 respectively Figure 5.9. [27]

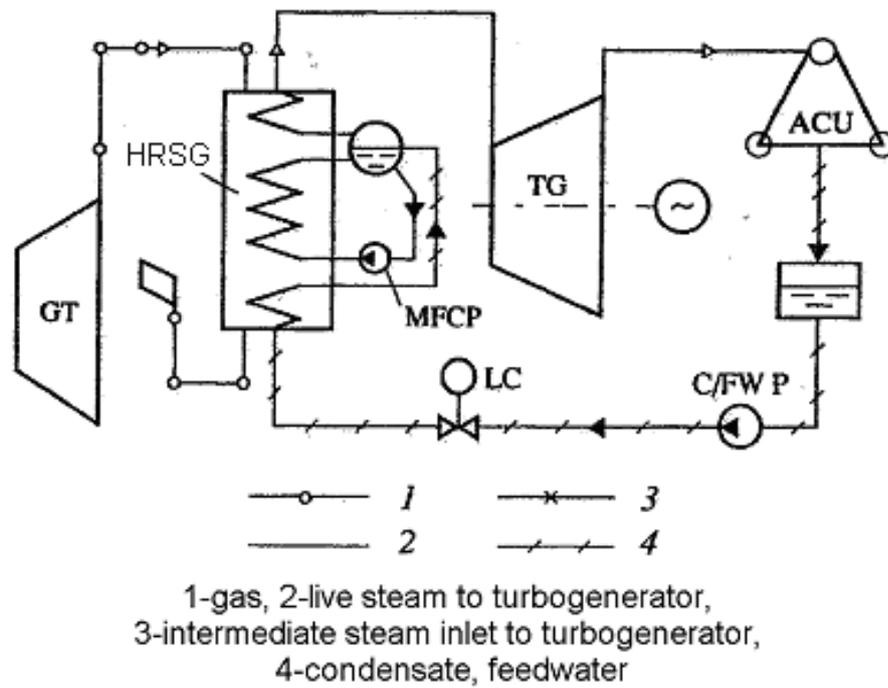


Figure 5.8: Single pressure waste heat recovery plant [4]

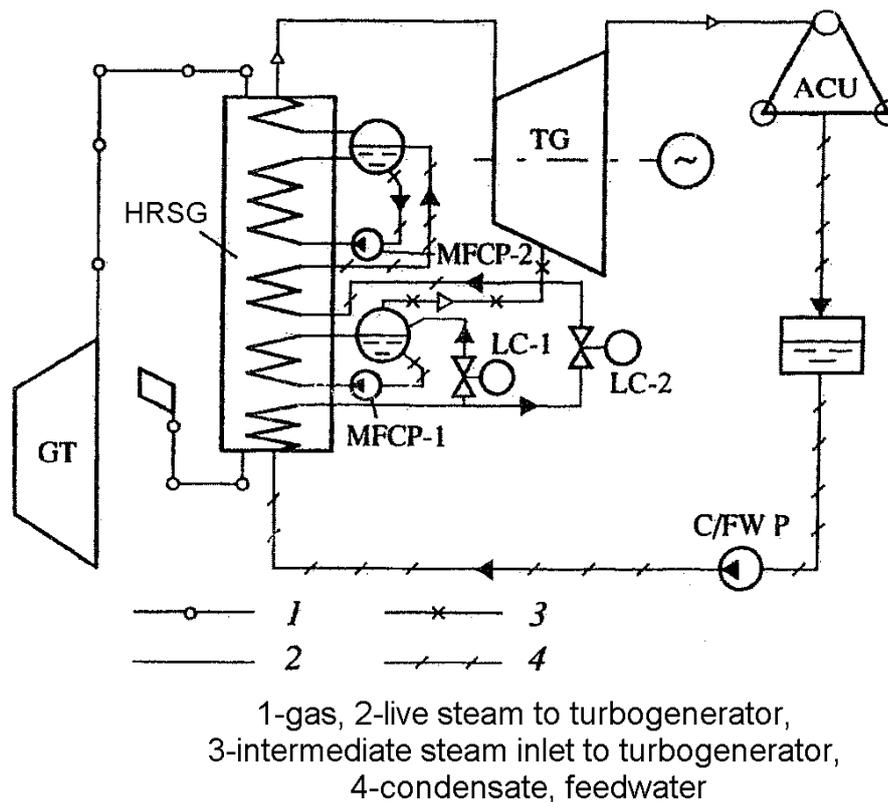


Figure 5.9: Double pressure waste heat recovery plant [4]

If these two layouts using ST of 20MW power are compared in the terms of thermal efficiency, it is apparent from [27] that double-pressure system offers 4% higher thermal efficiency. On the other hand weight of HRSG in such system is 20-25% greater and the control circuits are much more complicated what makes manufacturing and installation time of such unit longer and also requires more effort to service under the conditions at the gas compressor station.

### 5.2.2 Influence of supplementary firing on CC efficiency

Generally, supplementary firing is the use of a burner in the flue gas path upstream of the HRSG to raise the temperature of the entering gas stream. This is most commonly applied in gas turbine applications where the oxygen-rich (15 to 18%) exhaust can provide an efficient combustion. The typical exhaust temperature from a turbine is in the range of 470 to 616°C [34] [35]. By means of supplementary firing it is possible to compensate the lower flue gas enthalpy appearing at GT part-load respectively it enables maintaining the steam parameters constant independently of gas turbine exhaust. One example of a duct burner arrangement is shown in Figure 5.10.



Figure 5.10: In-line duct burner for supplementary firing of GT exhaust [34]

The increased temperature difference across the HRSG results in more heat recovered per kg of exhaust gas. With duct firing, the exhaust gas entering the HRSG can be elevated to 830°C [23], given the typical operating limits of the ducting materials. Increased gas stream temperatures ranging up to nearly 1,650°C can be achieved by adding a radiant heat section to the system in the form of conventional burner system. This section transfer radiant energy from the burner flame to water contained in a membrane on the outer wall of the gas duct. Elevated steam temperatures increase the efficiency of the steam cycle. However, steam temperatures exceeding 540 to 590°C may require an upgrade in material quality, which will increase the cost of the equipment considerably. [34]

Upon introducing time as variable equations from 5.1 can be expressed as:

$$\eta_{CC} = \frac{P_{GT} + P_{ST}}{\dot{Q}_{GT} + \dot{Q}_{SF}} \quad (5.2-1)$$

What would in the case of unfired HRSG be:

$$\eta_{CC} = \frac{P_{GT} + P_{ST}}{\dot{Q}_{GT}} \quad (5.2-2)$$

The efficiencies of the single gas and steam cycles can be expressed per

$$\eta_{GT} = \frac{P_{GT}}{\dot{Q}_{GT}} = \frac{\dot{Q}_{GT} - \dot{Q}_{EXH}}{\dot{Q}_{GT}} = 1 - \frac{\dot{Q}_{EXH}}{\dot{Q}_{GT}} \quad (5.2.-3)$$

$$\eta_{ST} = \frac{P_{ST}}{\dot{Q}_{SF} + \dot{Q}_{EXH}} = \frac{P_{ST}}{\dot{Q}_{SF} + \dot{Q}_{GT}(1 - \eta_{GT})} \quad (5.2-4)$$

Substituting equations (5.2-3) and (5.2-4) into (5.2-1) gives:

$$\eta_{CC} = \frac{\eta_{GT} \dot{Q}_{GT} + \eta_{ST} (\dot{Q}_{SF} + \dot{Q}_{GT} [1 - \eta_{GT}])}{\dot{Q}_{GT} + \dot{Q}_{SF}} \quad (5.2-5)$$

Additional firing in the HRSG improves the CC efficiency whenever:

$$\frac{\partial \eta_{CC}}{\partial \dot{Q}_{SF}} > 0 \quad (5.2-6)$$

Differentiation of equation (5.2-6) produces the inequality:

$$\frac{\partial \eta_{CC}}{\partial \dot{Q}_{SF}} = \frac{1}{(\dot{Q}_{GT} + \dot{Q}_{SF})^2} \left\{ \eta_{GT} \cdot \dot{Q}_{GT} \left( \frac{\partial \eta_{ST}}{\partial \dot{Q}_{SF}} \dot{Q}_{SF} + \eta_{ST} \right) \cdot (\dot{Q}_{GT} + \dot{Q}_{SF}) - \eta_{ST} \dot{Q}_{SF} + \left[ \frac{\partial \eta_{ST}}{\partial \dot{Q}_{SF}} \dot{Q}_{GT} (1 - \eta_{GT}) \right] \cdot (\dot{Q}_{GT} + \dot{Q}_{SF}) - \eta_{ST} \dot{Q}_{GT} (1 - \eta_{GT}) \right\} > 0 \quad (5.2-7)$$

That gives:

$$\frac{\partial \eta_{ST}}{\partial \dot{Q}_{SF}} \left[ \dot{Q}_{SF} + \dot{Q}_{GT} (1 - \eta_{GT}) \right] + \eta_{ST} > \frac{\eta_{GT} \dot{Q}_{GT} + \eta_{ST} [\dot{Q}_{SF} + \dot{Q}_{GT} (1 - \eta_{GT})]}{\dot{Q}_{GT} + \dot{Q}_{SF}} \quad (5.2-8)$$

Since the second term of the inequality is equal to  $\eta_{CC}$  per (5.2-5), the inequality reduces to:

$$\frac{\partial \eta_{ST}}{\partial \dot{Q}_{SF}} \left[ \dot{Q}_{SF} + \dot{Q}_{GT} (1 - \eta_{GT}) \right] > \eta_{CC} - \eta_{ST} \quad (5.2-9)$$

The term  $\left[ \dot{Q}_{SF} + \dot{Q}_{GT} (1 - \eta_{GT}) \right]$  is actually the heat induced into the steam cycle, resulting

in:

$$\frac{\partial \eta_{ST}}{\partial \dot{Q}_{SF}} \cdot \frac{P_{ST}}{\eta_{ST}} > \eta_{CC} - \eta_{ST} \quad (5.2-10)$$

Equation (5.2-10) means that increasing the additional firing improves the CC efficiency only if it improves the efficiency of the steam process. The greater the difference is between the efficiencies of the CC and ST process, and the lower the temperature is of the heat input to the ST process, the more effective that improvement will be.

In practical considerations, supplementary firing brings to CC efficiency increase only in the case of relative older generation of gas turbines characterized by combustion temperatures below 1000°C as shown in Figure 5.11. Should the combustion temperatures be higher the additional firing in HRSG does not increase the CC efficiency.

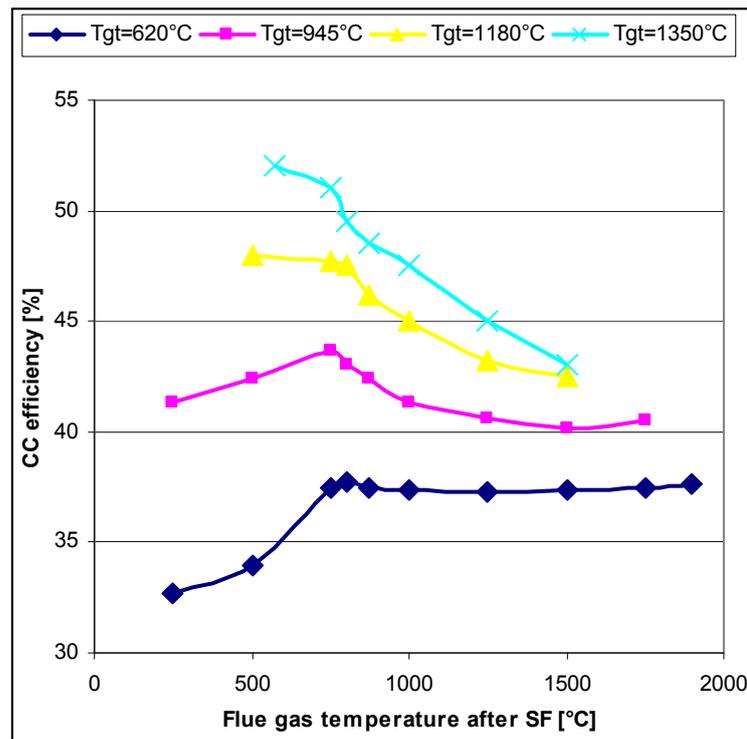
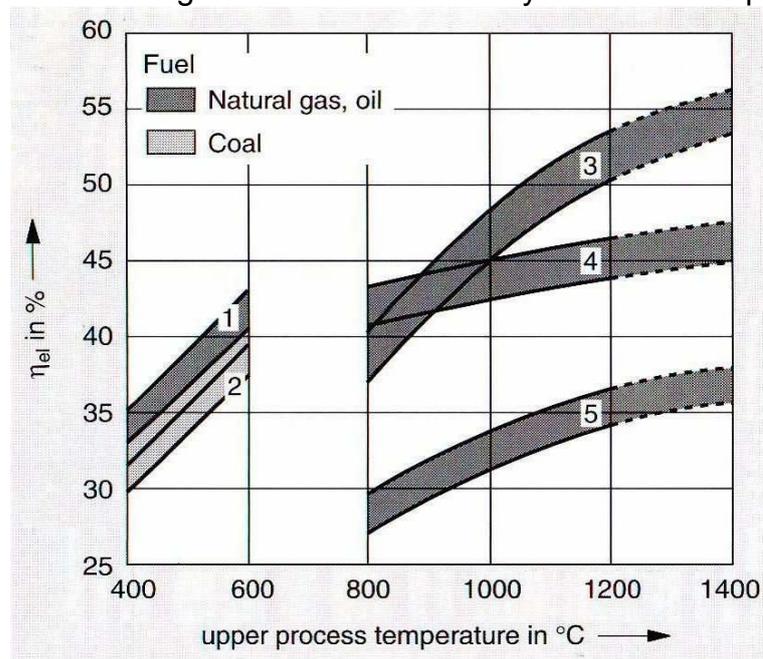


Figure 5.11: Efficiency of large scale CC plants as a function of the flue gas temperature after SF and gas turbine inlet temperature [6]

Figure 5.12 shows the efficiency curves of CC plants compared with conventional GT and ST plants in industrial sector. Obviously, efficiency of fired CC plants with firing temperatures above 950°C gets lower than efficiency of unfired CC plant.



- 1 conventional steam turbine process
- 2 conventional steam turbine process with FGD
- 3 combined cycle process with pure waste heat boiler
- 4 combined cycle process with fully fired steam boiler
- 5 pure gas turbine process

Figure 5.12: Efficiency of small scale CC plants compared with GT and ST plants [33]

Figure 5.13 depicts dependence of combined cycle efficiency in case of industrial gas turbine for different firing temperatures. Apparently efficiency increase is obtained only for exhaust gas temperatures below  $490^{\circ}\text{C}$ . Change in gradient of all CC efficiency curves at temperature of ca.  $750^{\circ}\text{C}$  is notable.

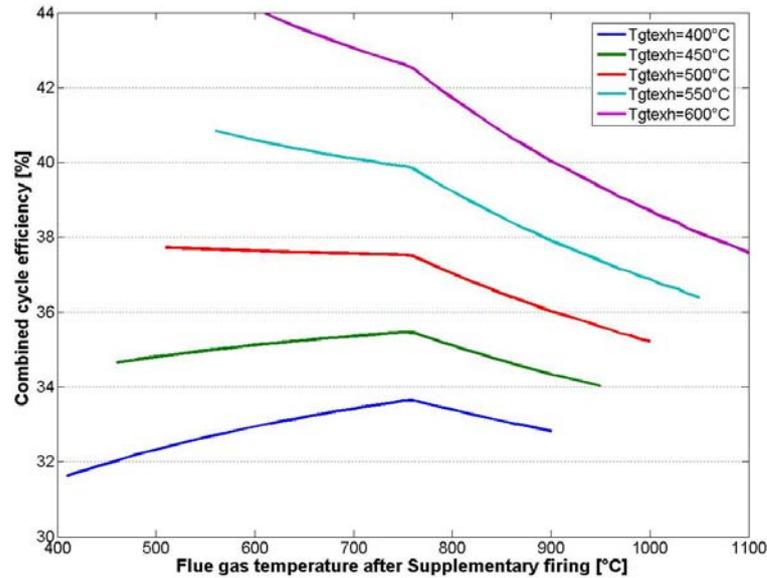


Figure 5.13: Calculated efficiency of waste heat recovery system based on THM1203 for  $34.8\text{ kg/s}$  flue gas mass flow and steam parameters of  $30\text{ bar}$  and  $390^{\circ}\text{C}$

So as to determine the thermodynamic reasons for such shape of the CC efficiency curves T-Q diagrams for the same application are constructed and are depicted in Figure 5.14.

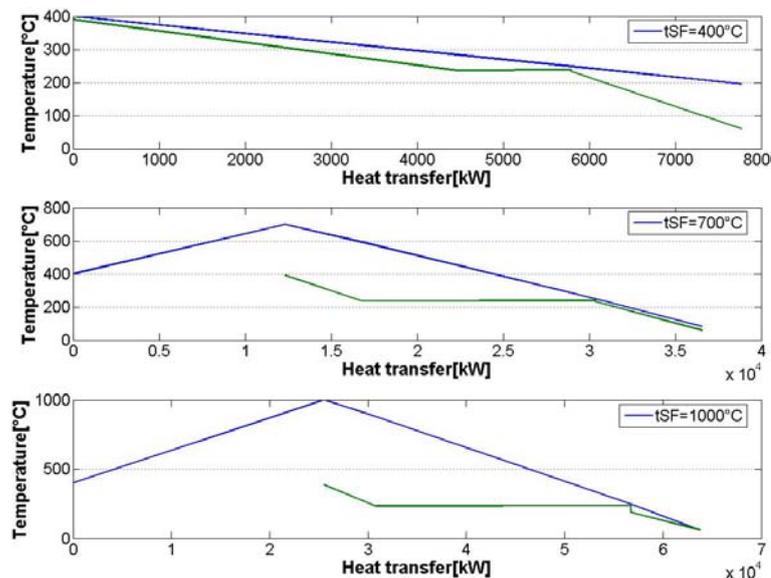


Figure 5.14: T-Q diagrams of waste heat recovery system based on THM1203 for  $34.8\text{ kg/s}$  flue gas mass flow and steam parameters of  $30\text{ bar}$  and  $390^{\circ}\text{C}$

It is notable that up to firing temperatures of ca.  $700\text{--}750^{\circ}\text{C}$  approach point at evaporator inlet remains the same. At higher firing temperatures inclination of the flue gas temperature line gets higher than the inclination of the feed water temperature line at economizer. Furthermore the approach point at evaporator inlet increases. Thereof it is interesting to examine behavior of generated steam quantity in dependence on firing temperature what is shown in Figure 5.15.

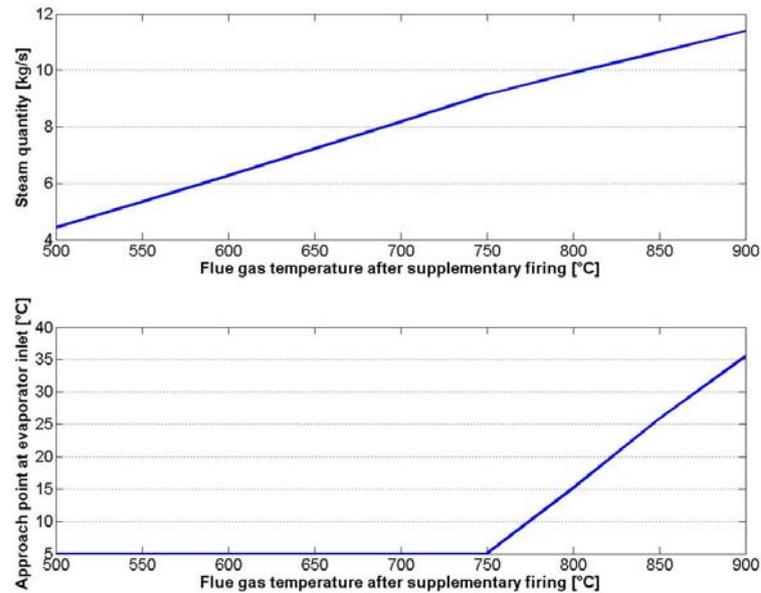


Figure 5.15: Steam quantity and AP of waste heat recovery system based on THM1203 for 34.8 kg/s flue gas mass flow and steam parameters of 30 bar and 390°C

From Figure 5.15 it is obvious that with increase of firing temperature approach point remains constant up to temperature of ca. 750°C whereupon the approach point rises. This phenomenon can be explained based on the second law of thermodynamics and equation for heat exchanger mean logarithmic temperature difference given in (5.2-11).

$$dt_m = \frac{dt_1 - dt_2}{\ln \frac{dt_1}{dt_2}} \quad (5.2-11)$$

As the flue gas temperature at HRSG exit reaches the feed water inlet temperature the smaller temperature difference  $dt_2$  tends to zero. If the firing temperature increases further the amount of heat contained in flue gases increases as well. However more heat can be transferred to the feed water in economizer only if the greater temperature difference  $dt_1$  increases. Increase of  $dt_1$  leads to lower feed water temperature at economizer exit. Heat required for heating up the feed water to the saturation temperature has to be transferred at evaporator wherefore the steam production decreases as shown in Figure 5.15.

Furthermore it is interesting to examine influence of supplementary firing on double pressure HRSG and compare it with influence on single pressure HRSG. From Figure 5.16 it is apparent that supplementary firing temperatures up to ca. 750°C increase the CC efficiency only in the case of 1P system whilst in case of 2P system increase of SF leads to decrease of CC efficiency.

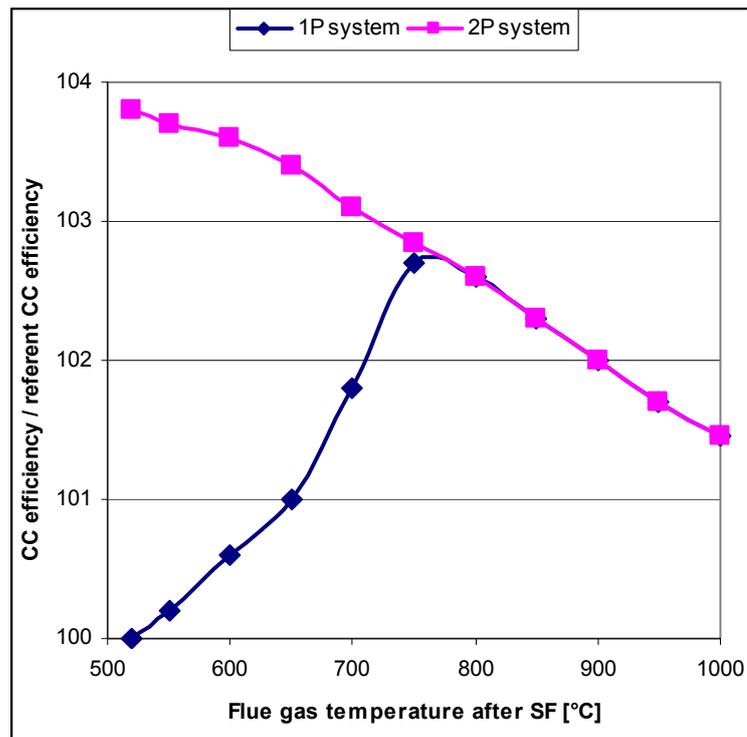


Figure 5.16: Effect of the temperature after SF on CC efficiency for 1P and 2P system (large scale plants), referent plant without supplementary firing  $t_{GT,exh}=525^{\circ}\text{C}$  [6]

Figure 5.17 depicts CC efficiency (same flue gas parameters as in case of 1P configuration) for various GT exhaust temperatures and firing temperatures. Decrease of CC efficiency for all temperatures is notable.

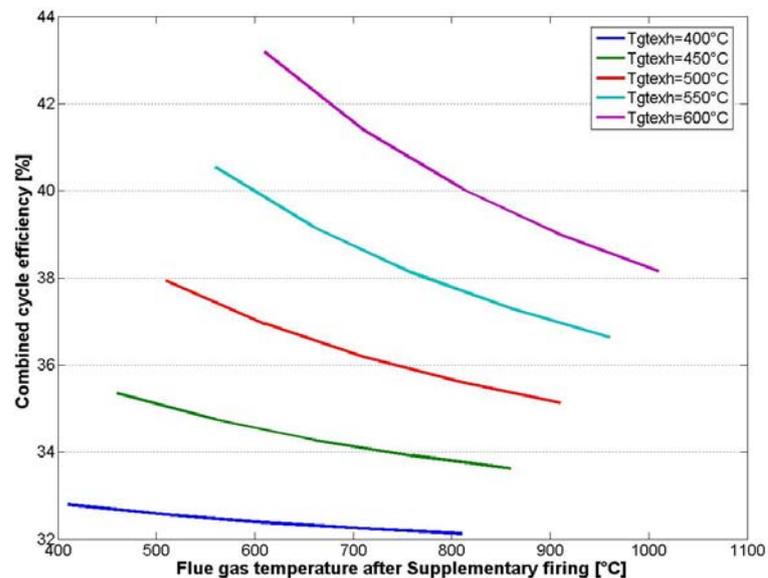


Figure 5.17: Calculated efficiency of waste heat recovery system based on THM1203 for 34.8 kg/s flue gas mass flow and steam parameters of 30bar/390°C and 5bar/182°C

Because of cited reasons, supplementary firing is in development terms of CC power plants becoming less and less interesting because it is generally better to burn the fuel in a modern gas turbine since the heat is thereby supplied to the process at a higher temperature level resulting in higher temperature difference. [6]

Thereof it is important to analyze the waste potential for each particular case with respect to available GT inlet respectively outlet temperatures and influence of supplementary firing on CC efficiency.

### 5.3 Parameters affecting dimensioning of HRSG

The two variables directly affecting steam production and the gas and steam temperature profiles are the pinch (PP) and the approach (AP) point. They have to be selected for the design case (typically no or minimal supplementary firing [32]) in order to size the HRSG surfaces. Furthermore HRSG performance has to be examined in other cases of different ambient conditions and GT part load. Target is finding the optimum between the sizes of the surface areas respectively their cost and obtained CC efficiency.

#### 5.3.1 Pinch point

Pinch point is defined as difference between the exhaust gas temperature leaving the evaporator section and the saturation temperature of the steam. [14]

In case of lower pinch point more heat is recovered but that result in larger heat transfer surfaces and higher capital cost. Typical pinch point differentials range from 8 to 33°C. Generally, an HRSG with a pinch point in the range of 8 to 14°C will have about 50% more surface in the evaporating section than a unit with a pinch point in the range of 22 to 28°C. [34]

Also, excessively low pinch points can mean inadequate steam production if the exhaust gas is low in energy i.e. low mass flow or low exhaust gas temperature. [14] Final choice of pinch point is based on exergo-economic considerations.

#### 5.3.2 Approach point

Approach temperature is defined as the difference between the saturation temperatures of the steam and the inlet temperature of the water into the drum. Lowering the approach temperature can result in increased steam production. On the other hand high approach temperatures ensure that no steam generation takes place in the economizer. Typically, approach temperatures are in the 11°C to 27°C range.

Typical temperature profile showing the pinch and approach point (AP) is shown in Figure 5.18.

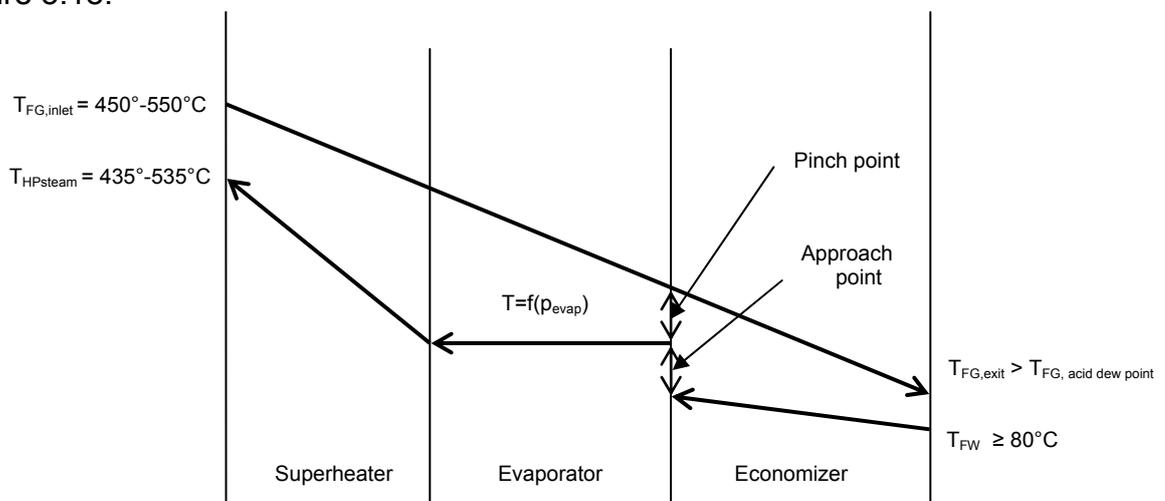


Figure 5.18: T-Q diagram showing pinch and approach points

During the initial design stages, these two factors may be selected. In unfired units, the pinch point and approach point lie in the range of 16°C to 27°C. Low values of pinch and approach point mean that the mean logarithmic temperature differential (defined in 6.2 per equation (6.2-8)) in the evaporator and economizer are reduced, thereby increasing the surface area requirements, and adding to the gas pressure drop and cost of the HRSG. However, the exit gas temperature for the HRSG can be lower, leading to more steam production. An optimum combination of HRSG surface (reflecting the initial cost), gas pressure drop and steam production (reflecting the operating cost and savings) can be determined using exergo-economic considerations.

Once the temperature profile is selected in the unfired mode, the HRSG design is nearly fixed. Its performance at other operating conditions, namely different ambient temperatures and firing modes is calculated following the off-design algorithm. [14]

### 5.3.3 Flue gases acid dew point and water dew point

So as to prevent low temperature corrosion in HRSG tubes (in particular in the last economizer in the flue gas path) inlet feed water temperature must not drop below the flue gases water dew point (if the fuel is sulphur-free) or the acid dew point (if it contains sulphur).

The acid dew point is defined as the temperature at which acids in the exhaust gas will begin to condense and is a function of the amount of sulphur in the fuel and subsequently water content in the exhaust gas. Exhaust streams from natural gas firing have a lower dew point than exhaust from fuels containing sulphur, allowing improved heat recovery efficiency.

Similarly is the water dew point defined as the temperature at which water vapour contained in exhaust gas will begin to condense. Although existence of water is not so aggressive for the economizer tubes as sulphuric acid ( $H_2SO_4$ ) it is still undesirable still its permanent contact with steel tubes would lead to corrosion. For the fuel content defined per Table 4-3 the water dew point equals ca. 50°C wherefore feed water temperature at last economizer inlet has to be in the range of 50-60°C. [6]

Final HRSG design has to be such that it does not induce a gas side pressure drop higher than 25 mbar. [6][36] Main influence on gas side pressure drop belongs to:

- Positioning and dimensioning of heat exchangers [36]
- Shape and pitch of the tube finning, followed by employment of different materials[36]
- Design and position of distribution grids within HRSG [37]

Distribution grids are used by some manufacturers for controlling the undesirable velocities, temperatures, spin angles and pulsations generated by certain gas turbines

- Usage of guide vanes in HRSG [37]

Excess pressure drop in HRSG can impose a high backpressure on prime mover systems, resulting in loss of power and efficiency, as well as overheating. In relation to gas turbines, back-pressure reduces the pressure ratio across the turbine section and, therefore, the power output of the turbine. The turbine can lose about 0.1% of power per mbar of back pressure. [34]

Exhaust gas temperature and resulting heat recovery efficiency are function of selected steam parameters, number of pressure levels as also pinch and approach point.

#### 5.4 Factors affecting CC plant performance

CC power plant provides certain power output and efficiency valid at full load and at design ambient conditions. However this performance changes with change of following parameters:

- GT load
- ST load
- Air temperature
- Air pressure
- Relative humidity
- Cooling water temperature and quantity

As indicated in Figure 4.8 efficiency of GT changes significantly depending on induced load. In the terms of heat recovery this results also in reduced mass flow and temperature of flue gases leaving the GT as shown in Figure 4.6 and Figure 4.7. Decrease of ST load leads to lower steam cycle efficiency due to decrease of internal (isentropic) ST efficiency as it will be shown in Figure 6.4. Lower isentropic efficiency leads to lower enthalpy drop in the ST resulting in lower ST power. On the other hand in case of fix pressure mode there is further efficiency decrease due to throttling losses appearing in steam control valves. Throttling losses decrease the steam turbine inlet pressure resulting in lower ST power. So as to minimize these losses there are commonly several control valves installed. If there is no additional air supply into HRSG what is the case in most CC plants, air temperature, pressure and relative humidity influence only GT performance which is described by correction curves provided by GT manufacturer. If air is supplied also to HRSG than state of the air also influences heat exchange between flue gases and water (steam). Changes in cooling water temperature and quantity affect steam cycle efficiency.

#### 5.5 Operation and control

Aim of this paragraph is consideration of MDCC system operation with respect to operating regimes of pipeline compressor station, defined per 4.3 and assumed ST load demand.

Steam turbine together with an AC generator forms a turbo-generator whose purpose is electricity supply to consumers. This supply is done via public grid in which the generator is connected in parallel with other utility power plants (hydro, thermal, wind and other). Likewise private, industrial and public consumers buy the electricity power from the grid according conditions at electricity market.

For understanding the difference in operation and control of MDCC system proposed in this work from CC systems applied in thermal power plants, firstly aspects of operation and control of the latter shall be discussed briefly.

##### 5.5.1 CC operation and control

Generally, CC system includes single-shaft and multi-shaft configurations. The single-shaft system consists of one gas turbine, one steam turbine, one generator and one HRSG, with the gas turbine and steam turbine coupled to the single generator in a tandem arrangement on a single shaft. Multi-shaft systems have one or more gas turbine-generators and HRSGs that supply steam through a common header to a separate single steam turbine-generator. In terms of overall investment a

multi-shaft system is about 5% higher in costs. [38] Despite that, most CC systems applied in thermal power plants comprise several gas turbines, one HRSG and one steam turbine driving the AC generator.

Generator in such CC systems is connected to an electrical grid and has to cover the demand given by the grid operator. Therefore is each CC plant equipped with a control system whose function is to regulate the plant power output. Furthermore it has to ensure safe start-up and shutdown of the plant as also its proper dynamic behaviour.

In CC power plants mostly closed loop control systems are used.[6] Such a system receives one or more measured process variables and then uses it to move the manipulated variable to control a certain device.

CC power plant closed loop control system comprises three control loops:

- Main control loop, which controls the gas and steam turbines
- Secondary control loop, which controls the important process parameters such as the gas turbine firing temperature, the steam turbine inlet temperature, pressure and flow.
- Auxiliary control loops, which maintain the fuel injection pressure, the lubrication oil pressure and temperature followed by feed water and condensate pumps.

Automation of these control loops is done on hierarchic levels as shown in Figure 5.19.

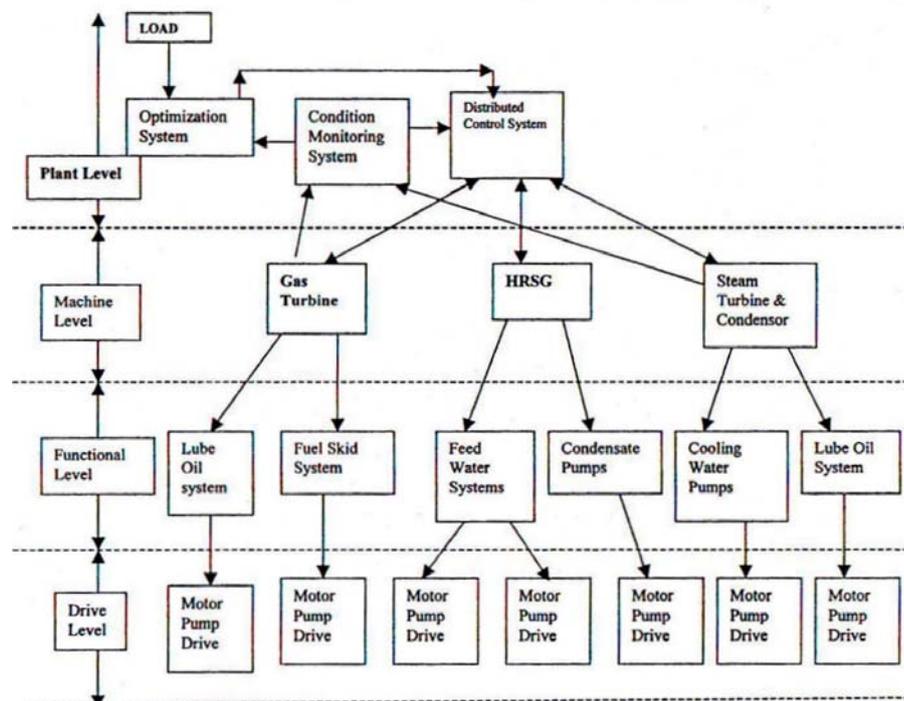


Figure 5.19: Hierarchic levels of automation [14]

The optimization system at the plant level receives the load and sends a signal to the Distributed Control System (DC-S) which sends the signal to the gas turbine and steam turbine so as to achieve the best settings of the steam and gas turbine to meet the load. Condition monitoring system receives all its inputs from the DC-s system and from the steam and gas turbine controllers. The gas and steam turbine control loops are the most important in a CC power plant. [14] In case of CC plants without supplementary firing it is only the power output of gas turbines adjusted because

varying the steam turbine load produces only a temporary effect. If supplementary firing is provided, the steam turbine is usually equipped with a load control whereof outputs of steam and gas turbines are adjusted.

Essential closed control loops in a CC power plant are: [6]

- Drum level control
- Live steam pressure control
- Feed water temperature
- Live steam pressure
- Level in feed water tank and hot well

### 5.5.2 MDCC operation and control

MDCC system that is to be proposed in this work comprises four gas turbines each driving one pipeline compressor whereof, depending on the induced load, simultaneously are only two or three gas turbines in operation. Furthermore its upgraded part comprises one HRSG and one steam turbine driving the AC generator.

Main difference to the CC thermal power plant systems is that only ST is driving the generator whilst gas turbines have a function of meeting the pipeline compressor load. Since the generator is connected to the grid it has to be capable of responding to the load induced by the grid operator. However for simplification of consideration ST load is assumed constant in all operating regimes of the pipeline compressor station.

In the terms of design and sizing the MDCC system that suggest to size the ST for the base load of gas turbines.

## 5.6 Economics of waste heat recovery

Economic analysis techniques used for analyzing investment potential for waste-heat-recovery systems are not different than those used for the analysis of any other industrial capital project. Therein the capital cost is proportional to system size and complexity. Except on capital cost, capital recovery depends also on fuel cost, revision cost as also on operation and maintenance cost. Furthermore level of discount rate has also an influence on the capital recovery.

Thereby the capital recovery refers to amount which plant has to gain annually for paying of mentioned costs. Income that the heat recovery plant gains depends on the plant layout. Basically, as it is further elaborated in next paragraph, the steam turbine can either be used for compressing additional amount of natural gas or it can be used for running a generator respectively production of electricity. In the former case the plant income depends on compressed natural gas volume and contracted rate typically defined in  $\$/\text{m}^3$  or by some operators in  $\$/\text{m}^3/100\text{km}$ . In the latter case the plant income depends on the amount of electricity produced annually as also on the electricity rate available on the free market. Furthermore it depends on the fact whether the utility company treats the plant operator as a privileged electricity producer in which case utility company guarantees sellback of all produced electricity into the grid. In opposite case only amount of electricity required by utility company is sold back into the grid.

Exact values can hardly be determined as they depend on a number of factors, such as the time distribution of waste heat source availability, the time distribution of heat load availability (in case of cogeneration), the availability of waste heat recovery equipment that can perform at the specified thermal conditions, and the current and future utility rates and prices of fuel. The inability to accurately predict these factors

can make the normal investment decision-making process ineffectual. There is another important distinction to be made about waste-heat recovery investment. When capital projects involve production-related equipment, the rate of capital recovery can be adjusted by manipulating product-selling prices. When dealing with waste-heat recovery systems, this is generally not an option. The rate of capital recovery is heavily influenced by utility rates and fuel prices, which are beyond the influence of the investor. That points out the importance of gathering solid data, and using the best available predictions in preparing analyses for the investment decision process.

The load factor is defined as the ratio of average annual load to rated capacity and the use factor as the fractional part of a year that the equipment is in use. It is clear that the capital recovery rate is directly proportional to these factors. [39]

For illustration of change in various parameters on power generation cost on example of 50MW plant is given in Figure 5.20 and Figure 5.21. [6]

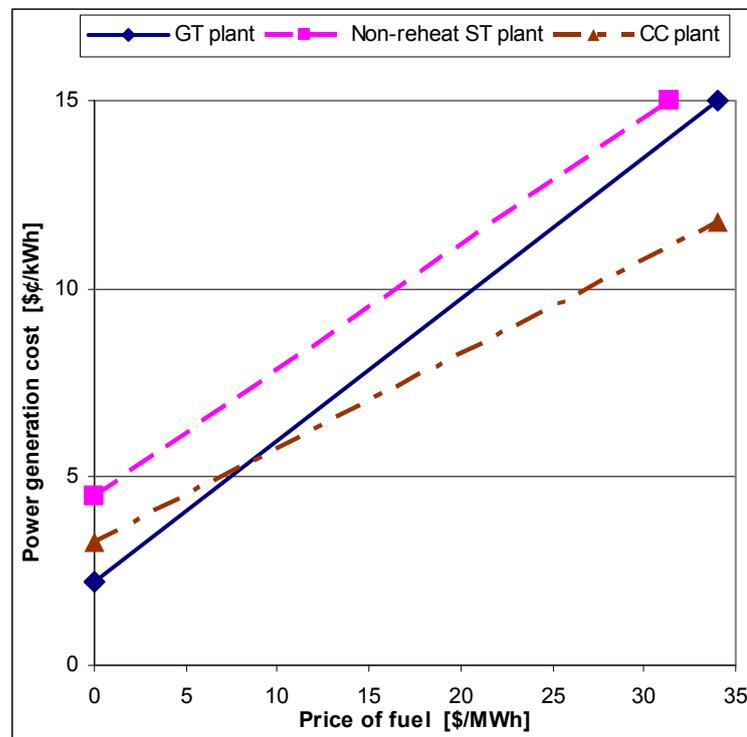


Figure 5.20: Power generation costs vs. fuel price (1997), discount rate=11% [6]

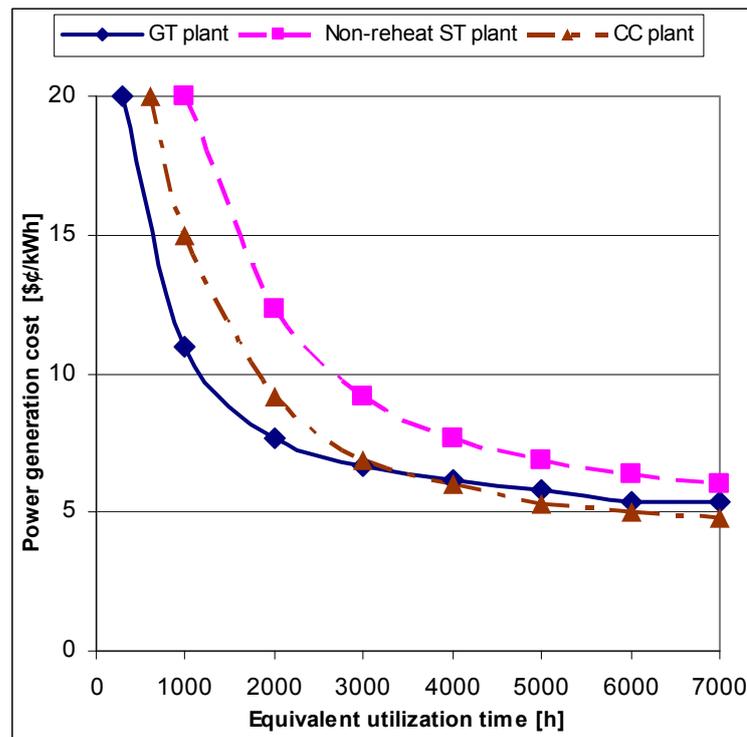


Figure 5.21: Power generation costs vs. equivalent utilization time (1997), discount rate=11%, fuel price=10\$/MWh [6]

## 5.7 Criteria for configuring the new MDCC system

There are many possible configurations that might be used for utilizing the waste heat from pipeline compressor stations. Intention of this paragraph is determination of boundary conditions required for setting up the various configurations which shall be examined and compared by means of thermo-economic evaluation.

Furthermore it is necessary to conduct economic evaluation with respect to annual characteristic of natural gas demand taking into account site conditions, fuel price and electricity sellback price.

Before setting up the flow schemes of various configurations to be examined, it makes sense to investigate present considerations outlined in [4].

### 5.7.1 Possible application of ST power produced via waste heat utilization

Independent from GT flue gas parameters and quantity which are depending on project and pipeline compressor operating regime on one side and different HRSG configuration on the other side it is clear (from previous paragraphs in this chapter) that each waste heat utilization configuration is set up in a way that produced steam in HRSG is producing rotational energy in ST. This energy can further be transferred into different forms of energy depending on economical and technical conditions valid for the certain project in consideration.

#### 5.7.1.1 Electricity power production

In case of electricity demand, steam turbine is used for running the AC generator, which is connected in parallel with national grid. Thereby the steam cycle has to be configured in a way to ensure ability of electrical load following independently of a gas turbine operating regime. Electricity produced at generator is sold on the free market under conditions, which may vary significantly depending on various

parameters as: shortage or excess of power in the grid. CC systems employing the HRSG for waste heat utilization from GT and ST for running the AC generator can be quite different employing one or more HRSGs and one or more steam turbines. Furthermore HRSG can be single-, double-, or triple pressure followed by balance of plant equipment which can be with or without condensate preheaters. Here are only two possible systems briefly described.

In the system shown in Figure 5.22 waste heat from GT is passed through a HRSG, one for each GT and the steam generated is used to produce rotational energy in a ST which drives an AC generator.

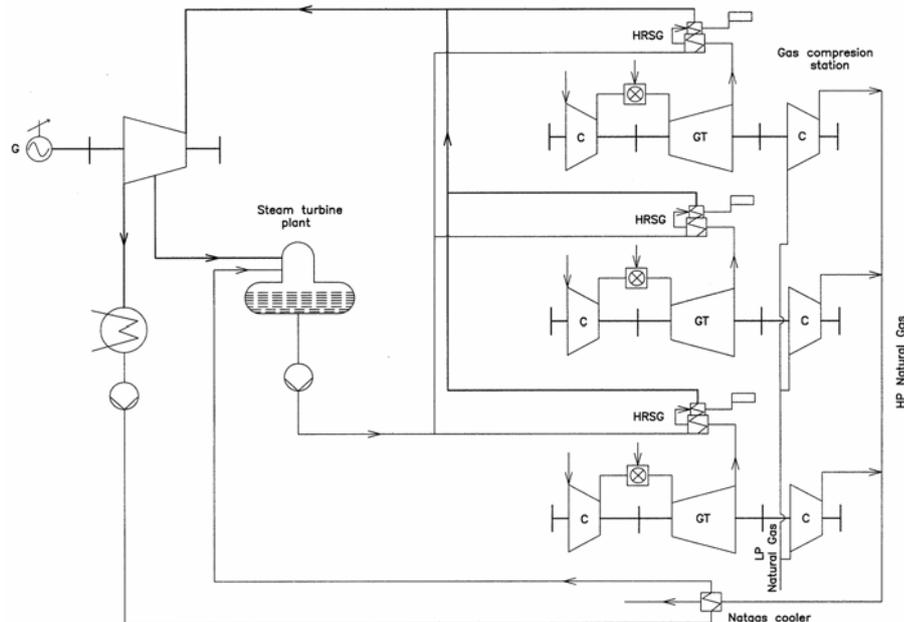


Figure 5.22: Simplified diagram of a MDCC and power generation plant [4]

Generally ST can be supplied with the steam produced from two or more waste heat recovery units (parallel powered) or there might be one ST per waste heat recovery unit (single powered). Greatest advantage of parallel powered MDCC lies in its versatility in respect to design, capital costs and operation.[4]

Furthermore, HRSG can be single loaded from each GT as shown in Figure 5.22 or parallel loaded from several GT as shown in Figure 5.23.

System shown in Figure 5.23 implements single parallel loaded triple pressure HRSG. The hot exhaust gases from all GTs are fed into single HRSG through a collecting line equipped with a fan. Although in such configurations it has to be counted with pressure and temperature losses in the flue gas supply pipes, they offer efficiency increase for several percentages if compared with the former configuration shown in Figure 5.22. [4]

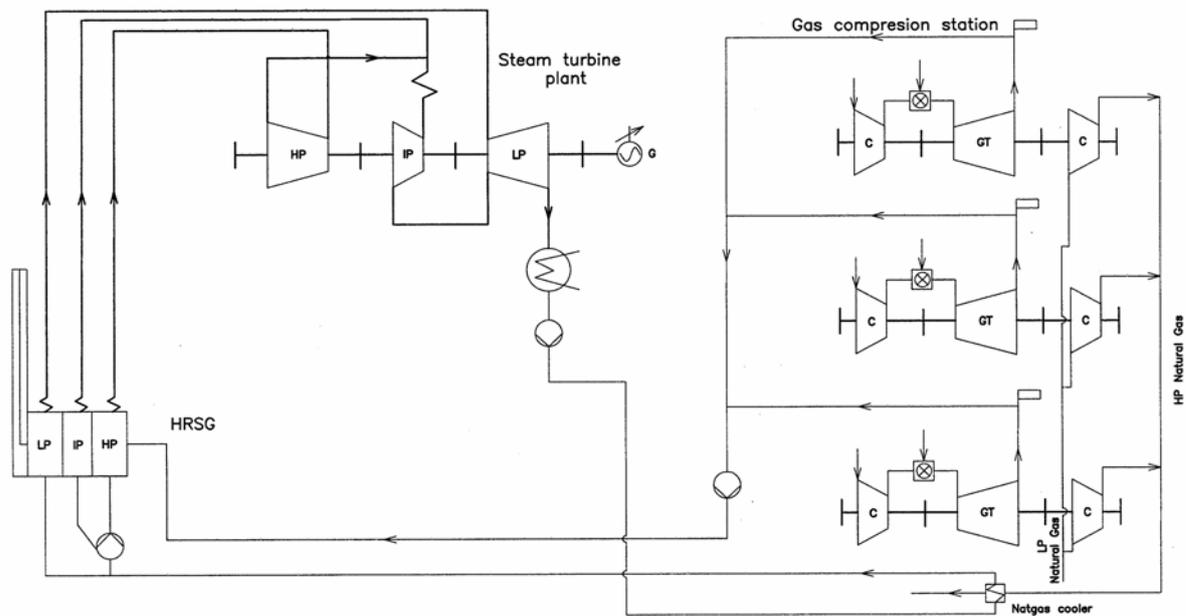


Figure 5.23: Simplified diagram of MDCC and power generation plant with 3P HRSG [4]

During compression of natural gas in the pipeline compressor stations its temperature increases. Since temperature within the gas pipelines has to be maintained within allowable limits for protecting the piping and its coating it makes sense to incorporate a natural gas cooler for preheating the condensate. That provides better heat utilization and minimizes pressure losses appearing during gas transport in pipeline. Both systems shown in Figure 5.22 and in Figure 5.23 are equipped with such natural gas cooler. Its influence on ST efficiency for the boundary conditions is considered in 5.7.2.

#### 5.7.1.2 Natural gas compressing by means of additional compressor

In this case the steam turbine is directly coupled to the pipeline compressor. Thereof it enables the pipeline compressor station to pump higher gas quantity than it has been designed for. Such configuration is used in the case of unavailability of electric grid followed by increased demand of natural gas. Disadvantage of such configuration in comparison with gas turbine driven stations is longer startup time of steam turbine and slower respond to load oscillations. Some steam turbine manufacturers have developed for this purpose special system of load following via injection of variable quantities of feed water into superheated steam. Through injection of feed water the fresh steam temperature decreases resulting in decrease of inlet enthalpy of steam. Thus available enthalpy drop decreases resulting in lower ST power. Should the power be increased the procedure goes in opposite direction. Since portion of injected water into superheated steam can in such processes be higher than in conventional steam cycle or combined gas and steam cycle particular attention has to be paid to allowable water content in the last ST stages.

#### 5.7.1.3 Natural gas compressing by means of electromotor

In this configuration produced electricity by AC generator is used for running the electromotor which drives the pipeline compressor. Such configuration is used in the case of unavailability of electrical grid. Further argument for this configuration might be the fact that considered electromotor has preferable characteristic of part load following than a steam turbine system in previous case. Disadvantage of this system

is increased cost due to the fact that there it comprises four components: steam turbine, generator, electromotor and pipeline compressor.

#### 5.7.1.4 Electricity production via ORC (organic rankin cycle)

LDC of compressor station shown in Figure 4.4 shows that during certain periods its part load is quite low i.e. 12% and less. However flue gas temperatures at the lowest considered operating regime is 380°C. Compressor stations in which the part load is quite low and appears over a longer time period exhaust gas temperatures can be below 300°C. If that is followed by the fact that the gas turbine efficiency at part load decreases significantly exhaust gas temperatures below 250°C can be met. In such cases it makes sense to consider installation of ORC (organic Rankin cycle) which is using Ammonia (or isopentane or silicon oil) as a working medium instead of water. Ammonia has lower evaporating temperature than water making this cycle favorable in the case of lower temperatures in a topping cycle. Practical applications have shown that usage of ORC cycle in waste heat utilization from pipeline compressor stations is viable if the exhaust gas temperatures are between 200-250°C. Should the temperatures be higher the preference belongs to a Rankin cycle.[23]

#### 5.7.2 Viability of using the natural gas cooler for condensate preheating

Due to the fact that operating temperature in gas pipeline is ca. 50°C[23] and condensate temperature is ca. 45°C it is not viable to install a natural gas cooler to preheat the condensate for the assumed condensing pressure (0.1 bar) respectively environmental conditions for the cooling water. Furthermore, a bypass piping with control valve should be installed in this case so as to avoid pressure drop occurring in a gas pipeline over installed heat exchanger thus making the natural gas cooler utilization in case of considered work inadequate. In case of colder sites providing lower condensing pressure and thus temperature it can be examined if the installation of natural gas cooler is viable. Also in this case a bypass piping with control valve is required.

#### 5.7.3 Superheated steam temperature control

Although superheated steam temperature control belongs to secondary control loop of a CC plant, way how it shall be performed has to be selected prior designing system flow scheme so as to enable proper calculation of system mass and heat balance.

Generally, there are several ways of controlling the superheated steam temperature as shown in Table 5-1. [25]

Table 5-1: Ways of superheated steam temperature control

Location of injection	Equipment used	Injection media
upstream the SH	by means of feed water injection cooler	
	by means of surface heat exchanger	
downstream the SH	by means of feed water injection cooler	
	by means of surface heat exchanger	with feed water with boiler water
between two SH stages	by means of feed water injection cooler	
	by means of surface heat exchanger	with feed water with boiler water

During steam plant operation in various modes such as startup, normal operation and shutdown as also during steam turbine operation at various loads the steam flow changes. With increase of the steam flow, the temperature of the steam in the banks of tubes that are directly influenced by the radiant heat of combustion starts to decrease as the increasing flow of fluid takes away more of the heat that falls on the metal. On the other hand, the temperature of the steam in the banks of tubes in the convection passes tends to increase because of the higher heat transfer brought about by the increased flow of gases. No matter how successfully the HRSG is designed and operated it cannot achieve an absolutely flat temperature/flow characteristic. Without any additional control, the temperature of the steam leaving the final superheater of HRSG would vary with the rate of steam flow what is known as the 'natural characteristic' of the HRSG.

Theoretical approach suggests arranging the HRSG natural characteristic so as to attain the correct steam temperature when the rate of steam flow is that at which the HRSG will normally operate. If this is possible, it means that spray water is used only during startup and at off-design conditions. However in practice that is not possible because the HRSG natural characteristic changes with time due to factors such as fouling of the metal surfaces, which affects the heat transfer. Therefore it is common to operate with continuous spraying, which has the advantage of allowing the steam temperature to be adjusted both upwards and downwards. Component used for spraying feed water into the superheated steam is referred to as desuperheater although it may also be described as attemperator.

If HRSG is not equipped with supplementary firing no control of the live steam temperature is necessary because of the low flue gas temperatures. If it is though installed it serves more as a limiter than as an actual control and is used in case of extreme operating conditions. However in case of supplementary fired HRSGs flue gas temperatures can reach  $830^{\circ}\text{C}$ , as mentioned in 5.2.2., wherefore it is necessary to maintain the superheated steam temperature within safe limits so as to protect the superheater tubes, steam piping and turbine blades from overheating. In these cases the superheater is divided into two or three sections with water injected between them.[6]

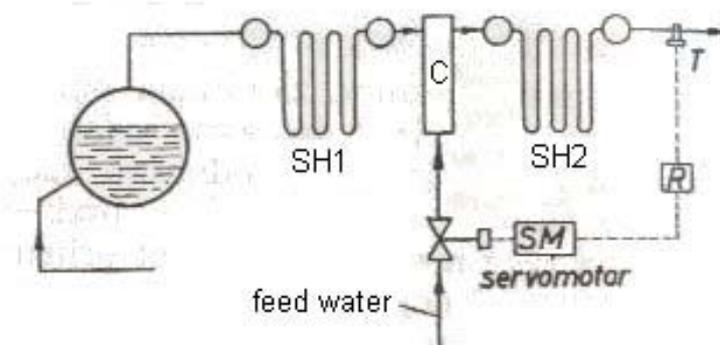


Figure 5.24: Injection cooler between two superheater stages [25]

Concept of such steam temperature control is shown in Figure 5.24. It has an advantage of low inertia respectively possibility of fast load change followed simplicity and lower cost than in the case of other options. [25]

#### 5.7.4 Boundary conditions for new MDCC system

Conclusions made previously regarding configuration features and boundary conditions for certain operating parameters are summarized in the Table 5-2.

*Table 5-2: Boundary conditions for setting up the new MDCC system configurations*

Parameter or Data	Value	Unit	Reference	Comment
Number of ST	1		[4]	
Number of HRSG	1		[4]	
Min. feed water temperature	60	°C	5.3.3	
Min. exit gas temperature	70	°C	5.3.3	Water dew point+10°C
Natural gas cooler	n/a		5.7.2	
Feed water preheating	Extracted steam from ST		[6] and [5]	
Superheated steam temperature regulation	Injection cooler between two superheater stages		[25]	

## 6. Mathematical model

System is described by mathematical equations used for thermodynamic calculation at the design point and at off-design regimes. Mathematical model is based on algorithms given per [11] and contains equation systems of HRSG, supplementary firing system, steam turbine, balance of plant equipment comprising deaerator, condenser and pumps.

Equations are based on:

Law of mass balance

Law of energy balance

In addition to that, equations for calculating properties of water (steam) and flue gases, followed by other thermodynamic calculations are also used.

### 6.1 Assumptions

Assumptions used in calculation are given in Table 6-1.

Table 6-1: Values of assumed input data

Data	Designation	Value	Unit	Reference
Pressure drop HP Economizer	dp HPECO	7.5	%	[5][9]
Pressure drop HP Evaporator	dp HPEVA	2.5	%	[5][9]
Pressure drop HP Superheater 1	dp HPSH1	5	%	[5][9]
Pressure drop HP Superheater 2	dp HPSH2	5	%	[5][9]
Condenser pump isentropic efficiency	$\eta_{i, \text{CNDPUMP}}$	80	%	[9]
HP/IP/LP pump isentropic efficiency	$\eta_{i, \text{PUMP}}$	80	%	[9]
Pressure drop steam control valve	dp CVLV	2	%	[9]
Minimal pinch point	HP PP	5	°C	[51]
Minimal approach point	HP AP	5	°C	[51]
Overall HTC HPSH2	$k_{\text{HPSH2}}$	58	W/m <sup>2</sup> K	[23]
Overall HTC HPSH1	$k_{\text{HPSH1}}$	58	W/m <sup>2</sup> K	[23]
Overall HTC HPEVA	$k_{\text{HPEVA}}$	27	W/m <sup>2</sup> K	[23]
Overall HTC HPECO	$k_{\text{HPECO}}$	27	W/m <sup>2</sup> K	[23]
Overall HTC condenser	$k_{\text{COND}}$	3000	W/m <sup>2</sup> K	[23]
Radiation heat transfer efficiency	$\eta_r$	0,99		[11]

### 6.2 Design point model

Typical calculating algorithm used for thermodynamic calculation shall be explained on example of 1P configuration whose flow scheme is given in Figure 7.1. Algorithm for each other configuration type is analogical.

Since flue gas condition at GT outlet is predefined per Table 4-2, calculation begins with supplementary firing.

$$\dot{m}_{\text{GTTotal}} = 3 \dot{m}_{\text{THM}} \quad (6.2-1)$$

$$\dot{m}_{\text{GTTotal}} + \dot{m}_{\text{sf}} = \dot{m}_{\text{gd}} \quad (6.2-2)$$

$$\dot{Q}_{\text{sf}} = \text{BH}_{\text{low}} \eta_{\text{sf}} \quad (6.2-3)$$

$$\dot{m}_{\text{GTTotal}} \cdot h_{\text{GTTotal}} + \dot{Q}_{\text{sf}} = \dot{m}_{\text{gd}} \cdot h_{\text{g2}} \quad (6.2-4)$$

Calculation of each heat exchanger within HRSG (in American literature described as coil) shall be explained on example of the first heat exchanger in the flue gas path namely superheater 2 (SH2).

Superheater 2:

$$\dot{Q}_{SH2} = \eta_r \dot{m}_{gd} (h_{g2} - h_{g3}) \quad (6.2-5)$$

$$\dot{Q}_{SH2} = \dot{m}_{HP} (h_{s1} - h_{s2}) \quad (6.2-6)$$

$$\dot{Q}_{SH2} = k_{HPSH2} A_{HPSH2} dt_m \quad (6.2-7)$$

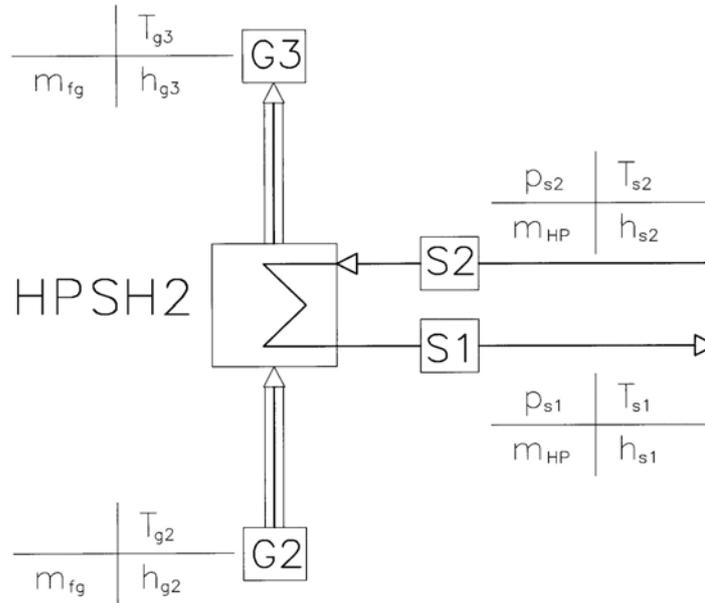


Figure 6.1: Schematic of SH2

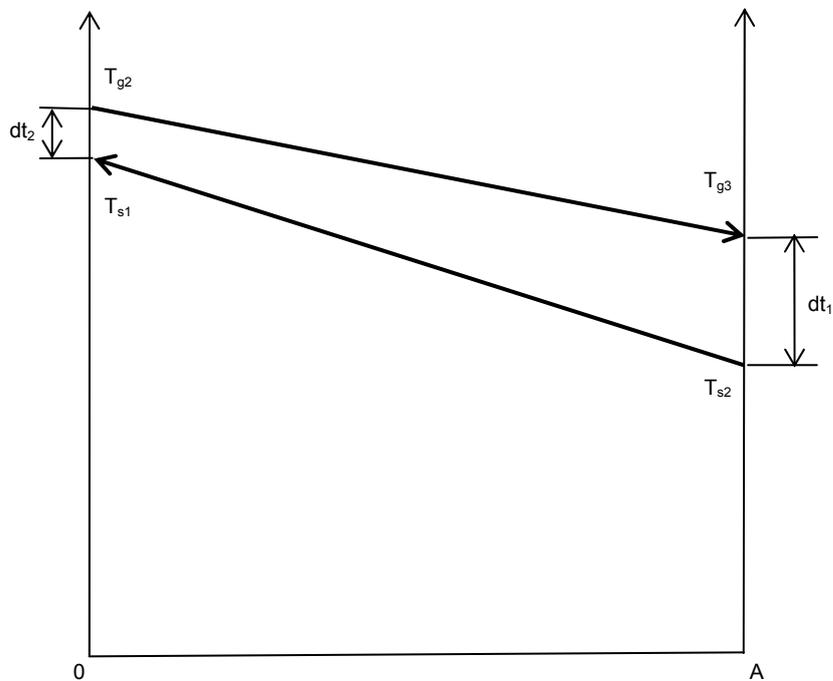


Figure 6.2: HPSH2 temperature profile

System of equations (6.2-5)-(6.2-7) is valid for each heat exchanger in HRSG. It comprises equation of the enthalpy drop of the flue gases (6.2-5), equation of the enthalpy rise of water (steam) (6.2-6) followed by a common heat exchanger equation (6.2-7). Equation (6.2-5) and (6.2-6) is composed of the product of mass flow and specific enthalpy difference. Equation (6.2-7) is composed of product of overall heat transfer coefficient, mean logarithmic temperature difference and heat exchanger surface area.

Thereby is the mean logarithmic temperature difference calculated from (6.2-8).

$$dt_m = \frac{dt_1 - dt_2}{\ln \frac{dt_1}{dt_2}}; dt_1 = T_{g2} - T_{s1}; dt_2 = T_{g3} - T_{s2} \quad (6.2-8)$$

Water (steam) enthalpy is calculated from pressure in the saturated region and from pressure and temperature in the liquid respectively superheated region.

Steam pressure at superheater inlet is calculated from the fresh stem pressure and pressure drop per Table 6-1 according (6.2-9).

$$p_{s2} = (1 + dp_{HPSH2}) \cdot p_{s1} \quad (6.2-9)$$

Analogical algorithm is used for other heat exchangers. Output values from the design point calculation are the surfaces and both temperature profiles for gas and water (steam). These values are further used for off-design calculations further explained in 6.4. Continuation of the calculating routine at the design point is given in Enclosure 1.

Generally steam generator efficiency is calculated from:

$$\eta_{SG} = 100 - (g_{out} + g_k + g_m + g_o + g_{slag}) \quad (6.2-10)$$

Wherein  $g_{out}$  means loss of flue gases sensitive heat as their temperature is above environmental. This is the greatest steam generator heat loss [11].  $g_o$  means loss due to heat transfer from boiler surfaces to environment.  $g_k$  means heat loss due to incomplete combustion caused by chemical reasons and  $g_m$  means heat loss due to incomplete combustion caused by mechanical reasons.  $g_{slag}$  means heat loss due to exiting physical heat of slag.

In case of gaseous fuel equation (6.2-10) becomes:

$$\eta_{SG} = 100 - (g_{out} + g_o) \quad (6.2-11)$$

Values for  $g_o$  are typically taken from diagrams since its accurate calculation would be too complex whilst  $g_{out}$  is calculated from the term (6.2-12).

$$g_{out} = \frac{H_{gas, SG-out} - H_{gas, amb}}{H_d + V_L (h_L - h_{L0})} \quad (6.2-12)$$

Therein numerator refers to difference between enthalpy of exit flue gases and their enthalpy at ambient conditions. The first member in denominator refers to fuel lower heating value and the second to the enthalpy difference due to preheating the air prior to its inlet into the steam generator.

The term (6.2-12) is valid for classic steam generators with furnace whilst in case of HRSG the term in denominator substitutes the flues gas enthalpy at HRSG inlet.

$$g_{\text{out}} = \frac{H_{\text{gas, HRSG-out}} - H_{\text{gas, amb}}}{H_{\text{gas, HRSG, inlet}}} \quad (6.2-13)$$

From the terms (6.2-11) and (6.2-13) it is apparent that for the available flue gases at GT outlet HRSG efficiency depends only on the flue gas temperature at HRSG outlet meaning that the steam generator providing lower exit gas temperature obtains higher efficiency.

### 6.3 HRSG optimization model

In the recent history there was not much attention paid on HRSG and its efficiency since it was understood as another kind of a steam boiler. [36]

Since at the present time CC power plants, due to the reasons cited in 2.1.2 and 2.2.5, enable meeting the growing energy demand with the least fuel consumption it is of great interest to get greater performances and efficiencies from them. From this reason world manufacturers involved in this sector are working on improving the GT inlet temperatures, what directly increases the CC efficiency as explained in 5.1 and 5.2.2.

Further efficiency improvement can be done by optimizing the HRSG operating parameters so as to maximize the work obtained in the steam cycle. With pre-selected number of pressure levels the operating parameters to be optimized are the pressures and temperature differences between flue gas and water (steam) known as the pinch point.

There are different methods which can be used for HRSG optimization. In selection of proper method following criteria have been observed:

- Practical and reliable for computations with many iterative runs
- Considering economizer, evaporator and superheater sections of HRSG in the terms of heat transfer and in the terms of cost
- Considering economic aspects of the plant

A general heat exchanger optimization method known as the “pinch point method” is based on optimizing the min. temperature difference between the two streams without taking into account any economic aspects. [40] [41]

Another approach is based on the gas side pressure drop. This parameter affects the performance of the gas turbine and the size of the heat recovery unit. Method used for such optimization [42] necessitates detail designing or knowing the detailed HRSG design respectively tube size, arrangement, length and type of finning.

There are number of HRSG optimization methods based on maximization of exergy transfer respectively on second law analysis of HRSG. Therein [43] introduces a general equation for entropy generation number.

Method for design and operation of HRSG with minimum irreversibility is described in [44]. It suggests a thermodynamic optimum based on estimation of non-dimensional saturation temperature with a proper evaluation of the number of evaporator transfer units. Method disadvantage is not taking into account the superheater. Furthermore it does not take into account neither surface cost nor economic aspects of the plant (e.g. electricity sellback price).

Approach based on exergetic HRSG optimization striving to maximize the exergy transfer from flue gases to water (steam) circuit is explained in [45]. Method drawback is assumption that the total heat transfer area of HRSG is constant. Thereby it does not consider separately surface costs of economizer, evaporator and superheater respectively their influence on overall cost and efficiency.

A general method for the optimum design of HRSG is explained in [46]. Thereby optimization is organized at two levels: first one obtains the main HRSG operating parameters, while the second involves the detailed design of the component concerning the geometric variables of the heat transfer sections. The second level of the optimization is split in two different steps. The first step aims to minimize the pressure drop for a given heat flow. The second step leads to a reduction of the overall dimensions, maintaining the imposed performance of the HRSG in terms of heat flow and pressure drop. Method seems to be complex for computations with many iterative runs.

Broader combined cycle optimization interconnecting a conceptual design, performance, cost and financial analysis is given in [47]. Methodology is taking into account several input variables; gas turbine type, fuel type (cost), efficiency, capital cost, return on investment, electricity sellback price followed by emission limits. Method limitation is not taking into account detailed HRSG optimization. Furthermore it is applicable for projects where complete CC plant including gas turbine is to be proposed.

Methodology used in this work is based on minimizing the objective function comprising both the thermodynamic and thermoeconomic component.[48][49] Thereby the thermodynamic component tends to minimize the exergy losses and thermoeconomic component tends to minimize the total HRSG cost by introducing the reduction to a common monetary base of the costs of exergy losses and of HRSG surface costs. Final solution expresses compromise between the two criteria.

Generally, exergy is defined as a maximal work that can be done from internal energy of a substance with regard to the state of environment. By considering HRSG as an open system, its exergy flow balance can be expressed per (6.3-1).

$$E_{\text{gas, in}} + \sum_{\text{in}} E_{\text{w, in}} = E_{\text{gas, out}} + \sum_{\text{out}} E_{\text{w, out}} + E_Q + I \quad (6.3-1)$$

The term  $\sum_{\text{in}} E_{\text{w, in}}$  is negligible since the inlet temperature of water is similar to environmental. The term  $E_{\text{gas, out}}$  is negligible since the exhaust gas leaves the HRSG and is not more used. Furthermore the exergy losses  $E_Q$  related to the heat exchange with the environment are considered negligible.

Approach of neglecting the terms  $\sum_{\text{in}} E_{\text{w, in}}$ ,  $E_{\text{gas, out}}$  and  $E_Q$  is overtaken from [48][49].

Upon rearranging equation (6.3-1) the exergy loss can be expressed per (6.3-2).

$$I = E_{\text{gas, in}} - \sum_{\text{out}} E_{\text{w, out}} \quad (6.3-2)$$

Wherein  $E_{\text{gas, in}}$  means exergy of the flue gases at HRSG inlet and  $\sum_{\text{out}} E_{\text{w, out}}$  means exergy of outlet steam generated in HRSG.

Exergy of inlet gas respectively outlet steam is calculated per (6.3-3).

$$E = H_2 - H_1 - T_1(S_2 - S_1) \quad (6.3-3)$$

Wherein  $H_2$  and  $S_2$  mean enthalpy and entropy of inlet gas respectively outlet steam and  $H_1$ ,  $T_1$  and  $S_1$  mean the enthalpy, temperature and entropy at the state of environment. Thereby is the entropy difference of the outlet steam obtained from GateCycle [9] whilst entropy difference of the flue gases is calculated per (6.3-4) assuming flue gases as an ideal gas.

$$s_2 - s_1 = c_p \ln \frac{T_2}{T_1} - R \ln \frac{p_2}{p_1} \quad (6.3-4)$$

Since equation (6.3-4) defines specific entropy difference, entropy is calculated from (6.3-5).

$$S_1 = m_{\text{gas}} \cdot s_1; S_2 = m_{\text{gas}} \cdot s_2 \quad (6.3-5)$$

Cost of exergy losses is expressed per:

$$K_I = k_I \cdot H \cdot I \quad (6.3-6)$$

In equation (6.3-6)  $k_I$  represents the specific cost of exergy losses, which is considered equal to an average value of the selling price of the electrical energy, and  $H$  is the functioning duration of the plant.

HRSG surface cost can be expressed as the sum of the costs of various sections; economizers, evaporators and superheaters multiplied by system complexity factor which accounts for additional equipment costs in systems with more pressure levels.

$$K_{\text{HRSG}} = \left( \sum_{\text{eco}} C_{\text{eco}} + \sum_{\text{eva}} C_{\text{eva}} + \sum_{\text{sh}} C_{\text{sh}} \right) \cdot f_c \quad (6.3-7)$$

Therein is the cost of each section calculated from:

$$C_{\text{eco}} = c_{\text{eco}} A_{\text{eco}}^{\text{exp}}; C_{\text{eva}} = c_{\text{eva}} A_{\text{eva}}^{\text{exp}}; C_{\text{sh}} = c_{\text{sh}} A_{\text{sh}}^{\text{exp}} \quad (6.3-8)$$

In equation (6.3-8)  $\text{exp}$  means a surface exponent used for assigning proper weighting factor to the exergetic component of the objective function.

Total annualized cost of the HRSG to be minimized is expressed per (6.3-9).

$$K_{\text{total}} = K_I + K_{\text{HRSG}} = k_I \cdot H \cdot I + \frac{1}{D} \left( \sum_{\text{eco}} C_{\text{eco}} + \sum_{\text{eva}} C_{\text{eva}} + \sum_{\text{sh}} C_{\text{sh}} \right) \cdot f_c \quad (6.3-9)$$

Where  $D$  means the economic life of the plant. Input data used in calculation are given in Table 6-2. [48][49]

Table 6-2: Input data for exergo-economic HRSG calculation

Variable	Value	Unit	Reference
D	20	years	[48]
H	7884	hours	[23]
$k_l$	0.03	\$/kWh	[50]
$C_{hpsh}$	382.5	\$/m <sup>2</sup>	[51]
$C_{hpeva}$	66	\$/m <sup>2</sup>	[51]
$C_{hpeco}$	39	\$/m <sup>2</sup>	[51]
$C_{ipsh}$	75	\$/m <sup>2</sup>	[51]
$C_{ipeva}$	69	\$/m <sup>2</sup>	[51]
$C_{ipeco}$	39	\$/m <sup>2</sup>	[51]
$C_{ipsh}$	75	\$/m <sup>2</sup>	[51]
$C_{ipeva}$	69	\$/m <sup>2</sup>	[51]
$C_{ipeco}$	39	\$/m <sup>2</sup>	[51]
$f_c$ (1P system)	1	-	-
$f_c$ (2P system)	1.1	-	-
$f_c$ (3P system)	1.2	-	-

The objective function given in (6.3-9) is to be minimized by means of Nelder-Mead method. [13] The method is also known as downhill simplex method or amoeba method and is a numerical method for minimizing an objective function in a many-dimensional space. It uses the concept of a simplex, which is a polytope of  $N+1$  vertices in  $N$  dimensions. Examples of simplexes include a line segment on a line, a triangle on a plane, a tetrahedron in three-dimensional space and so on. The method approximately finds a locally optimal solution to a problem with  $N$  variables when the objective function varies smoothly.

For proper convergence and determination of independent variables by means of Nelder-Mead method it is desirable to approximately define a local minimum. Actually that means that order of magnitude of parameters being optimized; pressure and pinch point has to be compared with relevant literature so as to verify results. Better possibility is to verify the results with another program respectively routine.

In this work optimization is conducted by means of Nelder-Mead optimization method using Matlab whereupon are the optimization results verified by means of GateCycle respectively its subprogram CycleLink.

#### 6.4 Off-design modeling

Intention of the off-design calculations is predicting the behavior of MDCC system, when GT output power decreases due to decreased power demand on GT shaft. Thereby the flue gases mass flow and temperature respectively their enthalpy decrease.

For HRSG that means that it is designed for a set of gas and steam conditions but has to operate under different parameters due to plant constraints, steam demand and different ambient conditions, such as ambient and cooling water temperature. Because of that, gas and steam temperature profiles in the HRSG and steam production differ from the design conditions, affecting the entire plant performance and economics. [52]

Off-design calculations for different operating regimes are performed by means of GateCycle which conducts iteration loops on different levels until reaching convergence of the whole MDCC system. For off-design calculation of the each operating regime the temperature profiles on gas and water (steam) side followed by surface areas calculated in 6.2 are used as input data.

The HTC in off-design changes as the flue gas velocity and temperature change influencing the rate of heat transfer from the flue gases to water or steam. This change of heat transfer rate is mathematically expressed by off-design performance calculations for HTC and pressure drop.

Off-design correction of the overall HTC is calculated from (6.4-1) [9] with exponents given in Table 6-3. [23]

$$k = k_{\text{design}} \cdot \left( \frac{\dot{m}}{\dot{m}_{\text{design}}} \right)^x \cdot \left( \frac{T}{T_{\text{design}}} \right)^y \cdot \left( \frac{p}{p_{\text{design}}} \right)^z \quad [\text{W/m}^2\text{K}] \quad (6.4-1)$$

Table 6-3: Exponents for off design performance calculations of HTC

	finned tubes (superheater)	finned tubes (economizer, evaporator)
x	0,6	0,625
y	0,505	0,4425
z	0	0

On the water/ steam side off-design pressure drop is corrected according (6.4-2) [9] using values for exponents given in Table 6-4 [23] whereby exponent x accounts for change in velocity whilst exponent y accounts for change in temperature.

$$dp = dp_{\text{design}} \cdot \left( \frac{\dot{m}}{\dot{m}_{\text{design}}} \right)^x \cdot \left( \frac{T}{T_{\text{design}}} \right)^y \cdot \left( \frac{p}{p_{\text{design}}} \right)^z \quad (6.4-2)$$

Table 6-4: Exponents for off design performance calculations of dp

	Steam side
x	1,98
y	0,2
z	0

Changes in pressure drop on the flue gas side are minor and therefore can be neglected so it is assumed that the gas side pressure through the HRSG is mostly constant.

Algorithm for off-design modeling used in this work is based on assumptions for HTC and dp at the design point given per Table 6-1 and calculated heat exchanger surfaces. Analog as in design-point calculation, for making the heat balance of each heat exchanger section in off-design there is an equation system with three equations available namely enthalpy drop of the flue gases, enthalpy rise of the water or steam and common heat exchanger equation. System off-design algorithm is depicted in Figure 6.3.

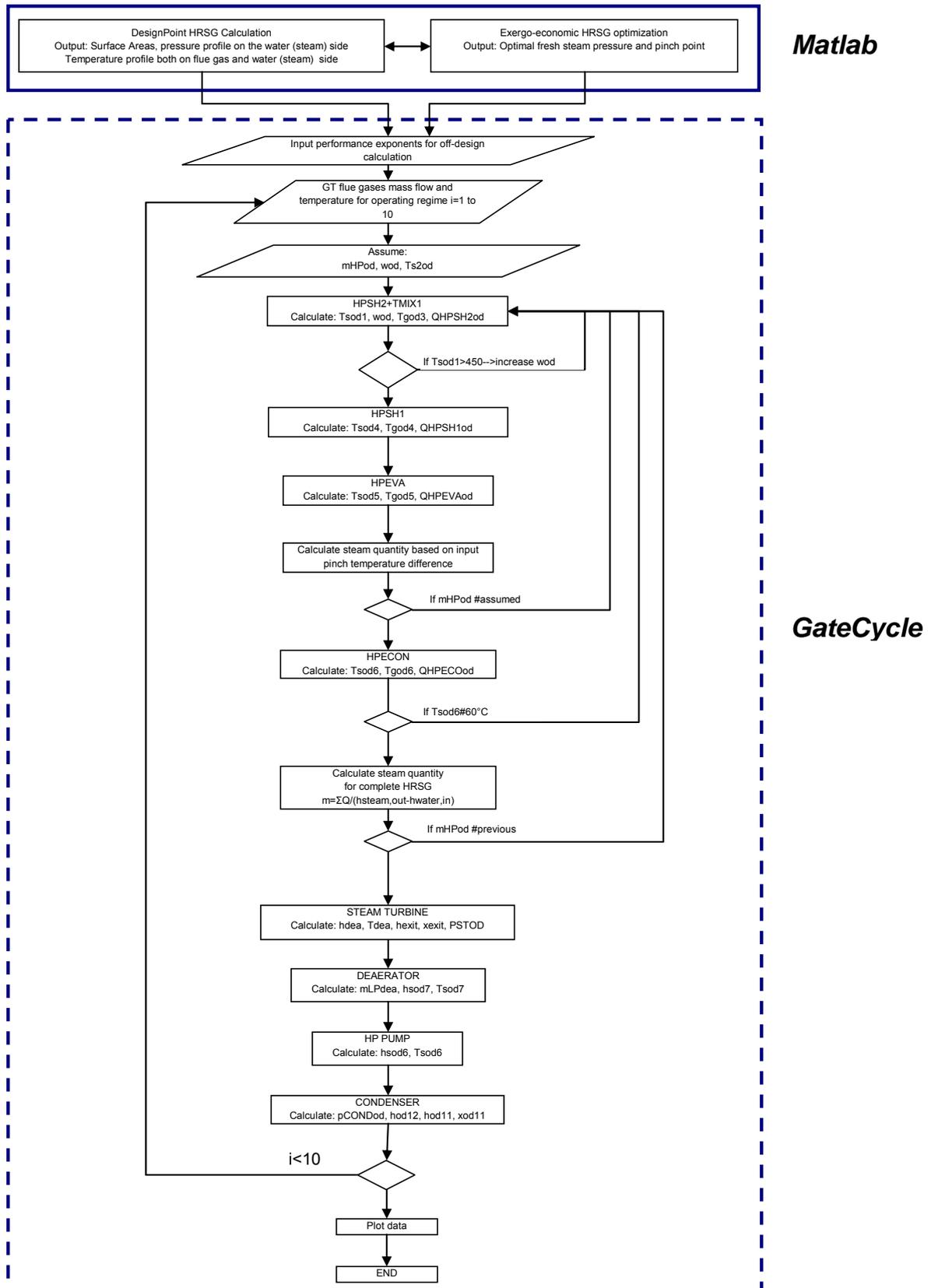


Figure 6.3: System off-design algorithm

Therefore it is apparent that the design point HRSG calculation and exergo-economic HRSG optimization are conducted by means of Matlab whilst off-design CC calculation is conducted by means of GateCycle.

ST power in off-design regimes is calculated as in the design point calculation as product of steam mass flow and real enthalpy drop of steam. Thereby it is important to take into account variable ST isentropic efficiency, which depends on ST part load fraction. Dependency of isentropic efficiency at off-design and part load fraction for the chosen ST is given in Figure 6.4.

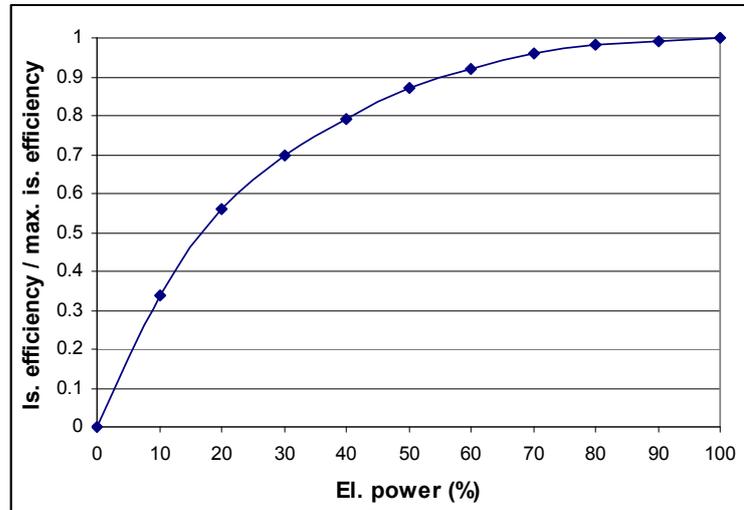


Figure 6.4: Steam turbine off-design isentropic efficiency vs. part load fraction [23]

## 6.5 CO<sub>2</sub> emissions reduction potential

Main target of Kyoto protocol [53][54] and its belonging mechanisms is reduction of a greenhouse gas emissions. Among different emissions for the power plant area most important are CO<sub>2</sub> emissions. Different technologies for electrical power production emit different CO<sub>2</sub> quantities depending on its efficiency and the fuel employed, as shown in Figure 6.5.

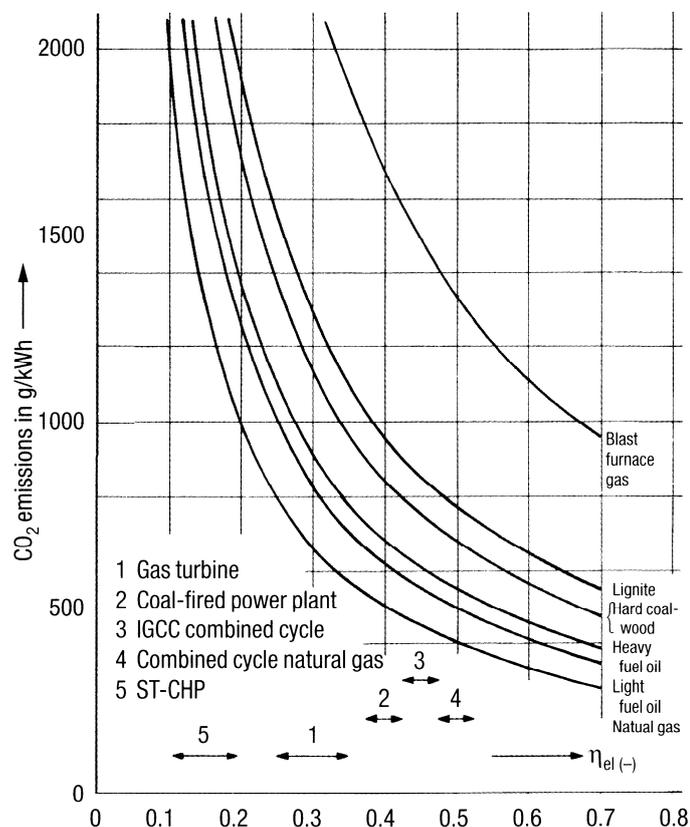


Figure 6.5: CO<sub>2</sub> emissions as a function of electrical efficiency and fuel [22]

So as to limit and monitor CO<sub>2</sub> emissions from any power plant or industry each country gets an annual emission limit which may not be exceeded. This emission limit is divided between the sectors as e.g. thermal power plants, pipeline compressor stations and other industrial sectors. Thereof each individual plant or factory receives its annual emission limit in tons of CO<sub>2</sub> what is defined by the local regulation of each respective country. In case that the annual emitted CO<sub>2</sub> quantity of an individual plant or factory is above its emission limit, there are two possibilities. Either the emission certificates for the excess CO<sub>2</sub> quantity are purchased on the market or CO<sub>2</sub> taxes are paid for the excess CO<sub>2</sub> quantity. In case that the annual emitted CO<sub>2</sub> quantity is below its emission limit, individual plant or factory can sell the difference in emission certificates on the market. Pipeline compressor station considered in this work emits certain amount of CO<sub>2</sub> annually which is either below or above its annual emission limit. Exact data of emission limit for the considered plant is not relevant for this work since much important information is whether the system upgrade shall increase or decrease the CO<sub>2</sub> emissions. If the CO<sub>2</sub> emissions are decreased in comparison with the actual CO<sub>2</sub> emissions then this is revenue for the owner independently on how the current CO<sub>2</sub> emissions are treated. So actually for evaluating the impact of system upgrade on CO<sub>2</sub> emission quantity it is necessary to calculate the difference between the CO<sub>2</sub> quantities emitted from the upgraded MDCC system and from the existing system. Thereby is the CO<sub>2</sub> quantity emitted from an exiting system given per (6.5-1). CO<sub>2</sub> quantity emitted from MDCC system is given per (6.5-2) with the difference in emitted CO<sub>2</sub> quantity given per (6.5-3).

$$\dot{m}_{\text{CO}_2, \text{GT}} = \frac{r_{\text{CO}_2} \dot{m}_{\text{fluegas, GT}}}{P_{\text{GT}}} \left[ \frac{\text{g}}{\text{kWh}} \right] \quad (6.5-1)$$

$$\dot{m}_{\text{CO}_2, \text{MDCC}} = \frac{r_{\text{CO}_2} \dot{m}_{\text{fluegas, HRSG}}}{P_{\text{GT}} + P_{\text{ST}}} \left[ \frac{\text{g}}{\text{kWh}} \right] \quad (6.5-2)$$

$$\Delta \dot{m}_{\text{CO}_2} = \dot{m}_{\text{CO}_2, \text{MDCC}} - \dot{m}_{\text{CO}_2, \text{GT}} \left[ \frac{\text{g}}{\text{kWh}} \right] \quad (6.5-3)$$

Obviously, if the result of term (6.5-3) is positive emissions are reduced.

## 7. HRSG optimization of various system configurations

Optimal CC configuration in the terms of the lowest total annualized HRSG cost is determined. So as to simplify the optimization procedure number of pressure levels and arrangement of heat exchangers in HRSG are predefined. Thereof seven configurations are proposed with main features listed in Table 7-1 followed by flow schemes given in Figure 7.1 to Figure 7.7.

*Table 7-1: Configuration features*

Configuration	Main features
1P	single pressure
2P1	double pressure, HPECO arranged after LPECO
2P2	double pressure, HPECO arranged after HPEVA
2P3	double pressure, HPECO1 arranged after HPEVA, HPECO2 arranged after LPECO
3P1	triple pressure, one HPECO arranged after HPEVA
3P2	triple pressure, one HPECO arranged after LPECO
3P3	triple pressure, three HP economizers

All configurations are characterized by following common features:

- condensate degassing with extracted LP steam from the steam turbine
- two high pressure superheaters (HPSH) and one intermediate injection cooler
- natural circulation drum

Each configuration is optimized by means of exergo-economic algorithm explained in 6.3 and compared with respect to total annualized HRSG cost. Within exergo-economic optimization the fresh steam pressures and pinch points obtaining the lowest total annualized HRSG cost are calculated. Reference values for each operative parameter are obtained from [6][14].

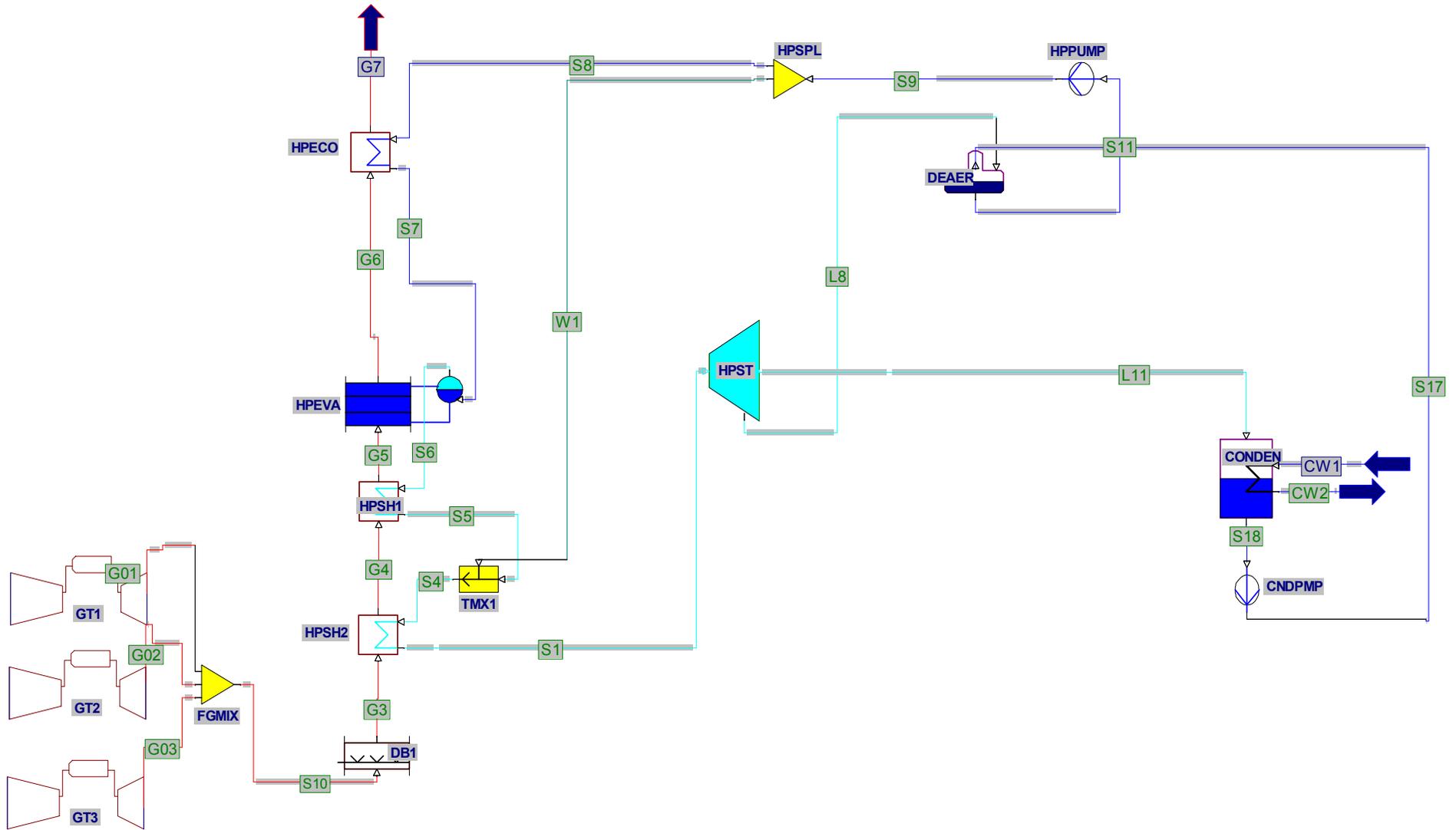


Figure 7.1: Flow scheme 1P configuration

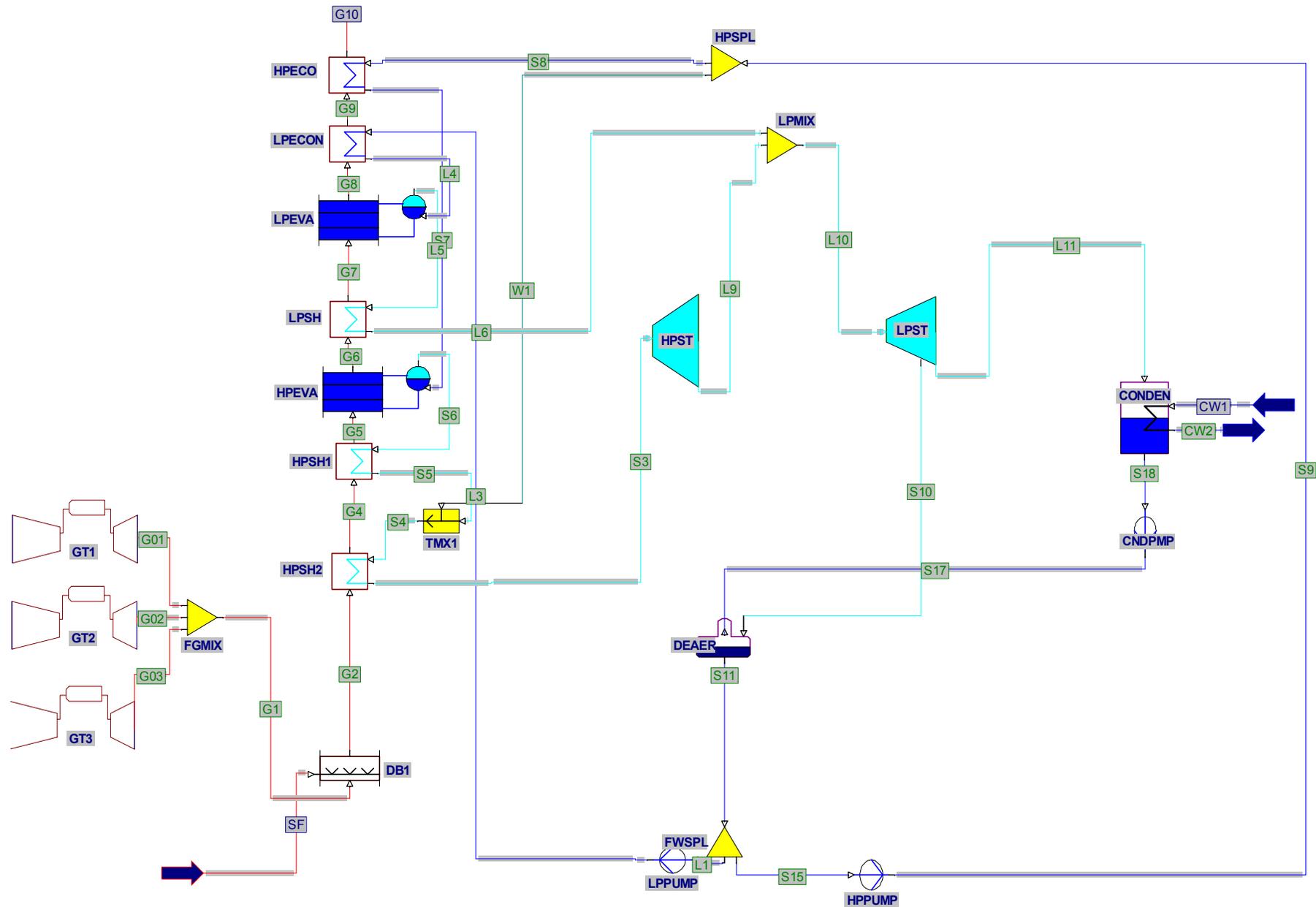


Figure 7.2: Flow scheme 2P1 configuration

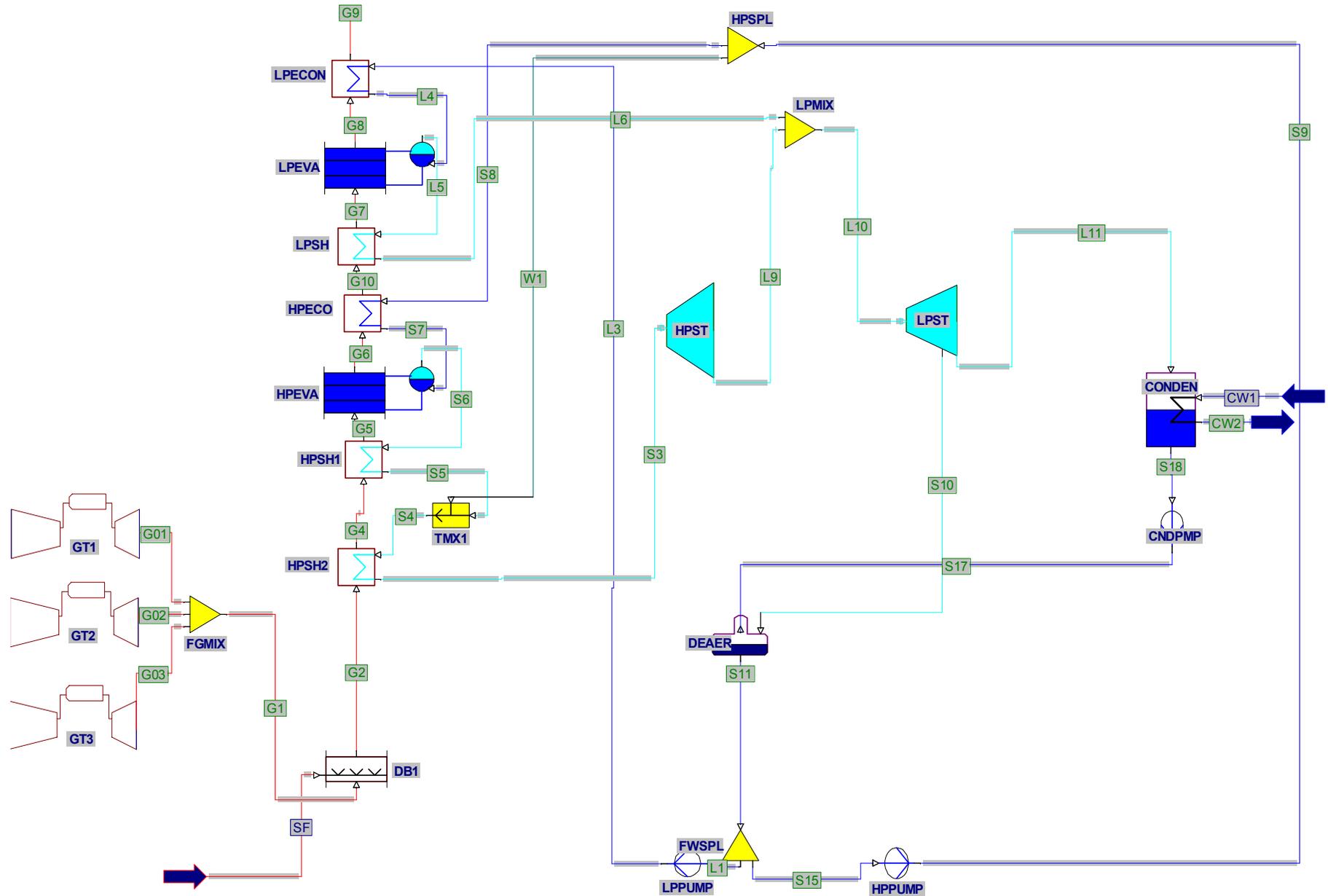


Figure 7.3: Flow scheme 2P2 configuration

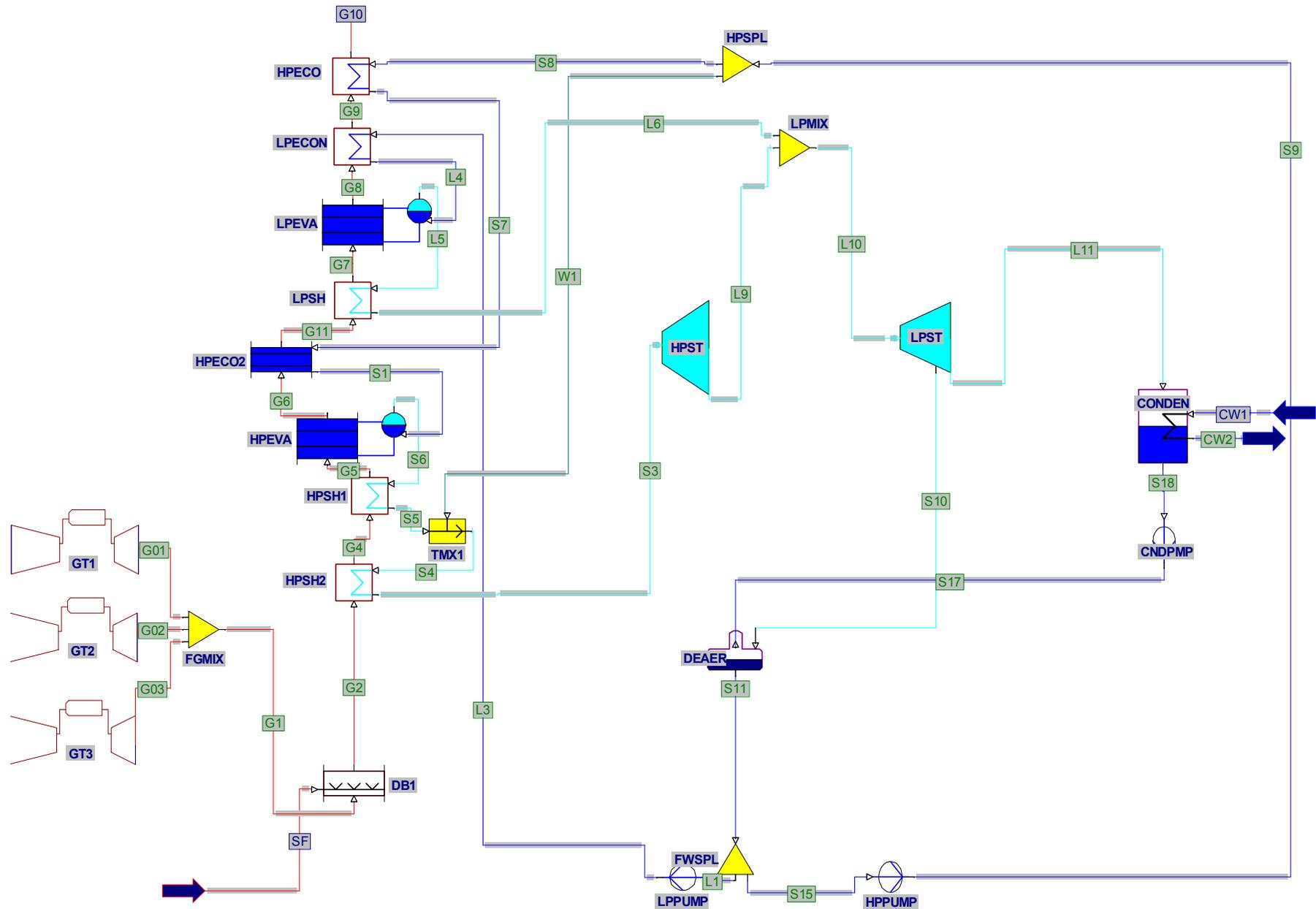


Figure 7.4: Flow scheme 2P3 configuration

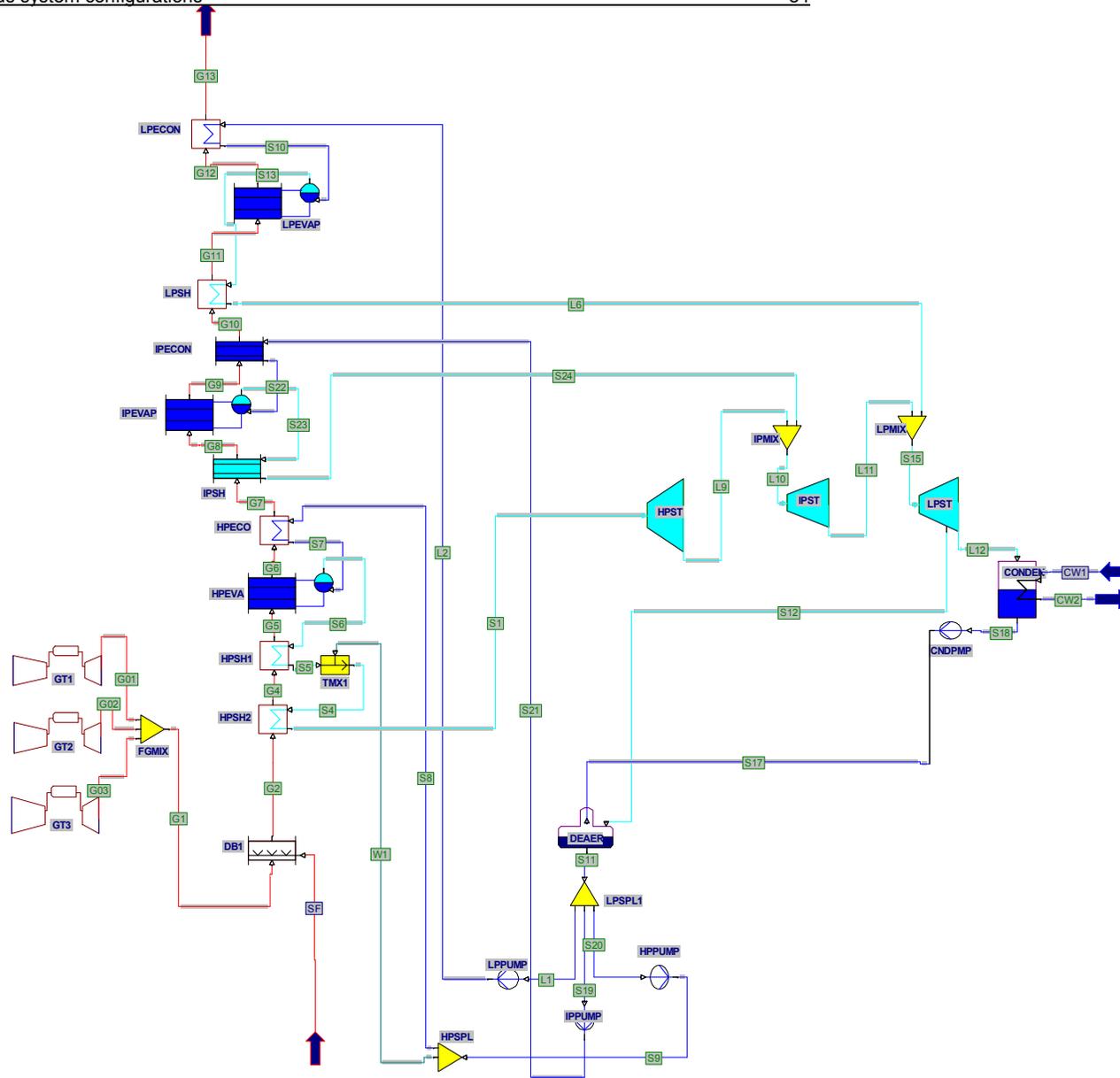


Figure 7.5: Flow scheme 3P1 configuration

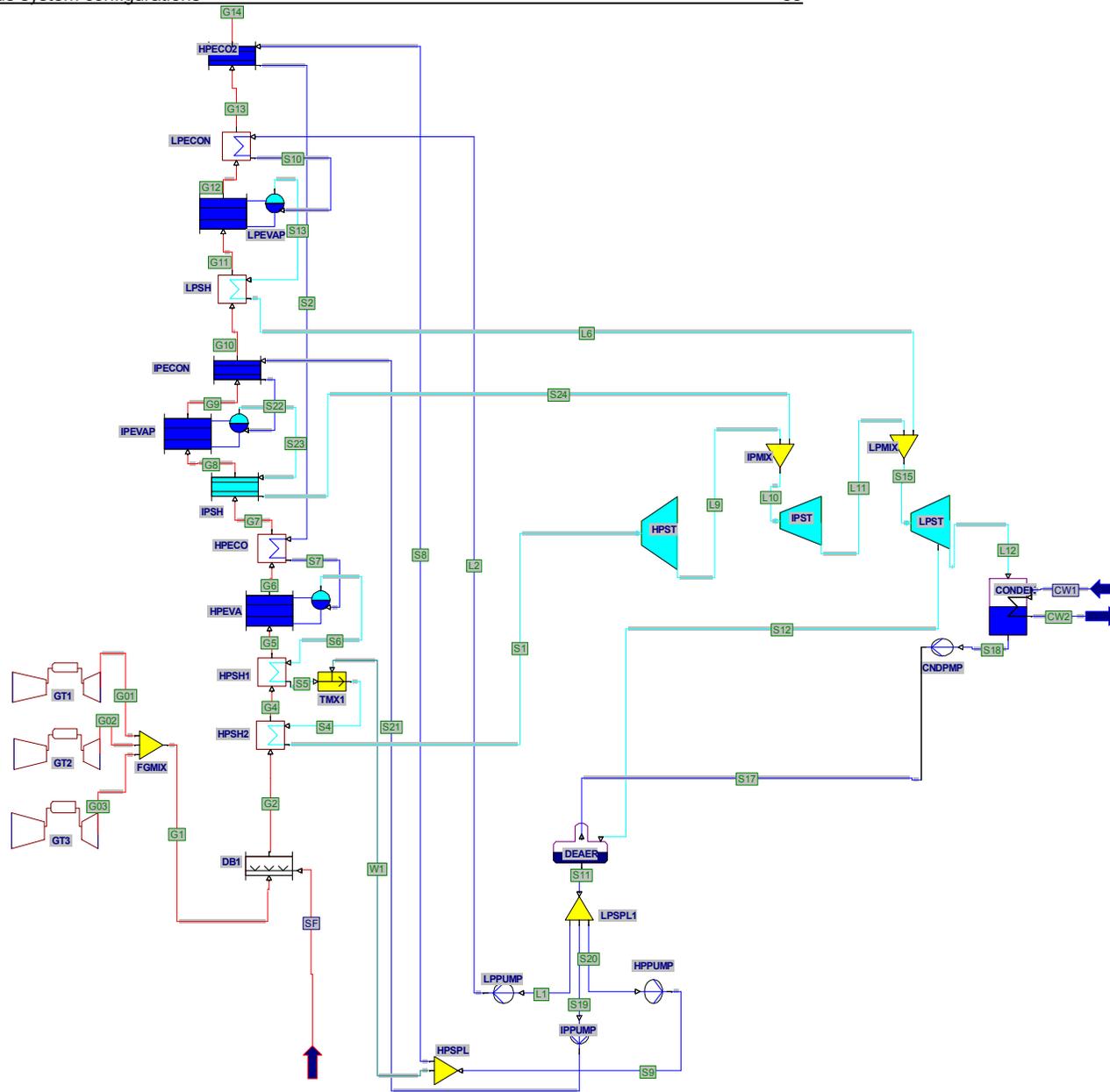


Figure 7.6: Flow scheme 3P2 configuration

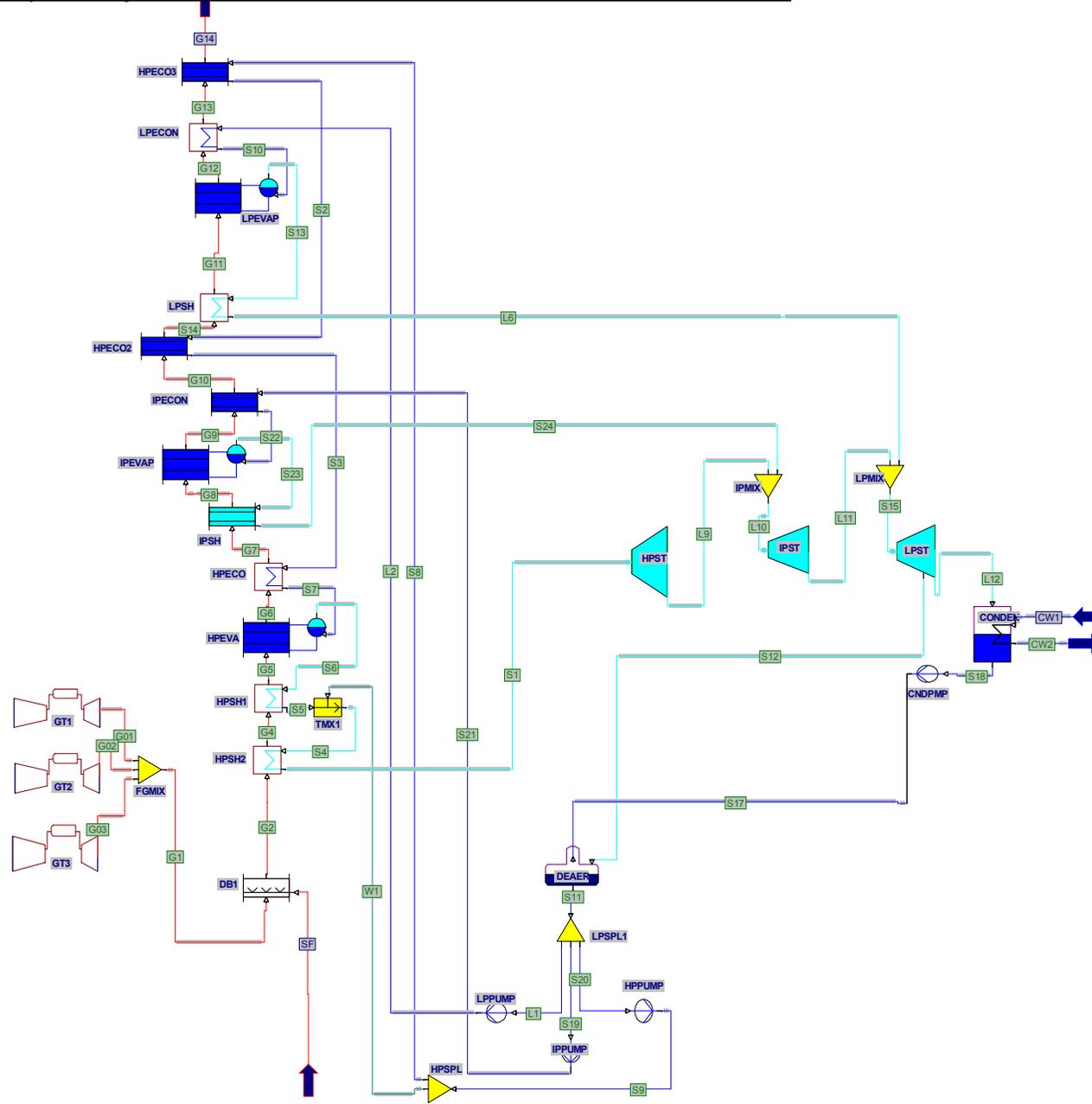


Figure 7.7: Flow scheme 3P3 configuration

In following paragraphs, technical features of each configuration are described. Thereby is the functional description of complete system explained on configuration 1P and other configurations are described with respect to difference in HRSG configuration and its impact on system behavior. Furthermore profile of a heat recovery is given in a T-Q diagram for each configuration.

## 7.1 1P Configuration

Basic configuration 1P is equipped with a single pressure HRSG and uses common way of degassing the feed water by means of extracted low quality steam from the steam turbine. Configuration 1P flow scheme is given in Figure 7.1. Flue gases leaving the gas turbines mix together and enter the HRSG. Type of HRSG arrangement is vertical. HRSG comprises two HPSH, HPEVA and HPECO. Fresh steam with 30.6bar and 470°C is entering the ST governing stage via shutoff valve.

### 7.1.1 Objective function

Upon introduction of assumed values for surface costs into optimization algorithm per 6.3 objective function in Figure 7.8 is obtained.

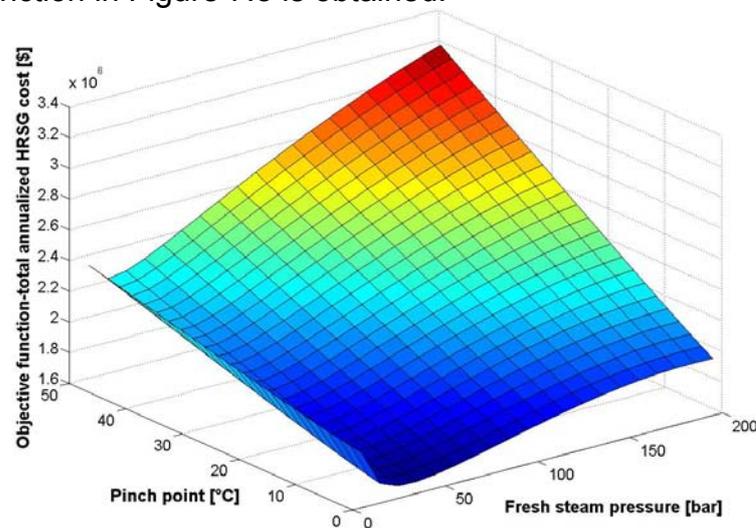


Figure 7.8: Objective function for initial assumptions

Apparently objective function reaches its minimum at fresh steam pressure value around 30 bar and pinch point of 0°C. However it is not possible to design a steam generator with zero pinch point as that would necessitate infinite large surfaces. Furthermore according to the second law of thermodynamics for heat transfer between two media existence of finite temperature difference is essential. Therefore is surface exponent according equation (6.3-8) introduced. Idea is to find the value of exp providing positive value of pinch point. To define the criteria for optimization of all configurations minimal feasible pinch point with a value of 5°C for unfired HRSG is obtained [51]. Optimization procedure for 1P configuration is repeated until pinch point of 5°C is reached whereby surface exponent attains value of 1.18. With this value of surface exponent objective function gets the shape as shown in Figure 7.9.

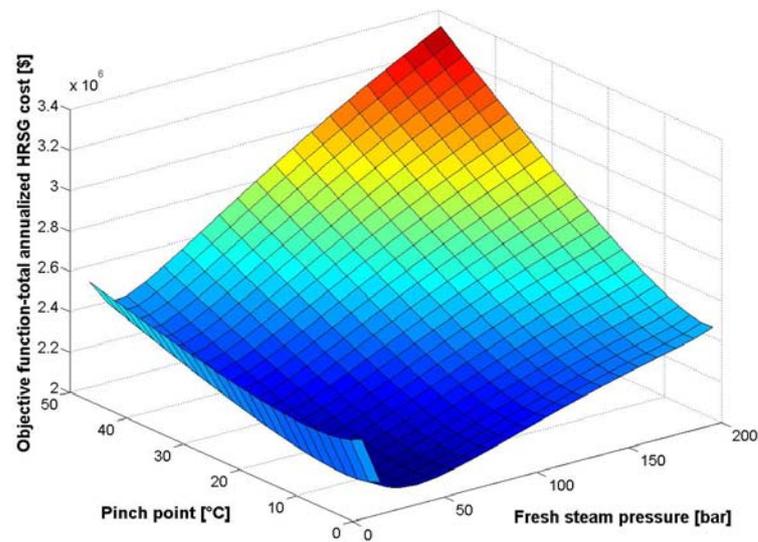


Figure 7.9: Objective function for realistic assumptions

Apparently objective function comprising surface exponent with a value of 1.18 reaches its minimum at fresh steam pressure value of 30.6 bar and pinch point of 4.98°C. Obtained surface exponent value of 1.18 is further used for optimization of other configurations.

### 7.1.2 Exergo-economic optimization progress

Figure 7.10 presents dependence of overall objective function and its components on fresh steam pressure and pinch point temperature difference.

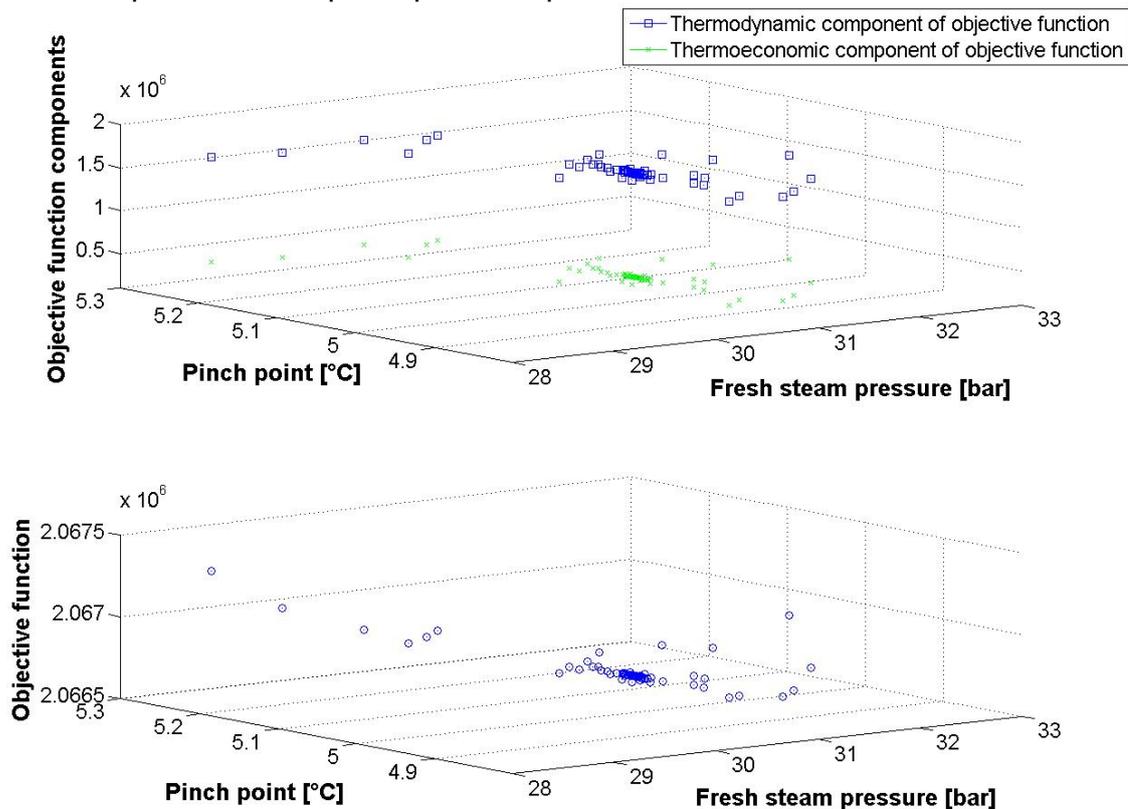


Figure 7.10: Optimization progress of objective function and its components

Apparently increase of both the fresh steam pressure and pinch point lead to increase of thermodynamic and decrease of thermoeconomic component of the objective function. Furthermore it can be noted that the thermodynamic component has an order of magnitude of  $10^6$  whilst thermoeconomic component has an order of magnitude of  $10^5$ . Thereof it may be concluded that the cost of exergy losses has greater influence on optimization results than the cost of HRSG surfaces. For other configurations optimization progress of thermodynamic and thermoeconomic component of the objective function are not shown but only optimization progress of the overall objective function.

The lowest value of total annualized HRSG cost is obtained for the operative parameters given in Table 7-2.

Table 7-2: 1P configuration operative parameters

Parameter	Abbreviation	Unit	Value
Total annualized HRSG cost	$K_{total, min}$	\$	2066746
Fresh steam pressure	$p_{HP}$	bar	30.6
HP pinch	HP PP	°C	5.0
Steam quantity	$m_{HP}$	kg/s	12.0
Heat recovery efficiency	$\eta_{HR}$	%	71.4
Steam cycle efficiency	$\eta_{SC}$	%	19.2
Net cycle CC efficiency	$\eta_{CC}$	%	37.6
Steam turbine el. power	$P_{ST,el}$	kW	10989

### 7.1.3 System description

Selected ST operating mode is fix pressure. From equation for ST power results that in this case regulation of power is done by regulation of steam quantity. Steam is expanding in the ST to the condenser pressure of 0.1bar. ST is equipped with one unregulated extraction for condensate deaeration. Since considered fuel is natural gas containing no or very small amount of sulphur selected feed water temperature at HRSG inlet is 60°C. Saturation pressure at this temperature appearing in deaerator is 0.2bar. Since this pressure is below atmospheric the deaerator has to be designed as vacuum deaerator. At the same pressure of 0.2bar the steam is extracted from the ST via unregulated extraction.

Extracted steam leaves the ST and is entering condenser at condensing pressure of 0.1bar. Condenser is water cooled by means of river water with an assumed cooling water inlet temperature of 20°C and pressure of 6bar. Chosen cooling water temperature rise is 20°C.[23] Condensate pump is used for increasing the condensate pressure to the deaerator operating pressure of 0.2bar. Condensate is entering the deaerator where the condensate droplets fall over the tubes through which flows the steam extracted from the ST. Condensate is thereby completely degassed and is entering the HP pump used for pressurizing the HRSG feed water to the chosen fresh steam pressure increased by sum of pressure drops through the water-steam side of HRSG. HP splitter is used for dividing small amount of HP feed water flow to the injection cooler TMX1. HP economizer is used for preheating the feed water up to the saturation temperature diminished by an approach temperature of 5°C. Steam evaporation appears in HP evaporator. Since obtained fresh steam pressure in all configurations is not above 100 bar natural circulation drum is assumed.

As to design of evaporators and economizers, finned tubes are used in order to improve the heat transfer from the flue gases to water respectively steam. Main reason for that is in poor HTC on the flue gas side. So as to improve the heat transfer intention is to increase the surface area of the heat exchanger keeping thereby the volume of tube package nearly the same. That is done by adding the additional layer called finning (or extended surfaces) what typically reduces the HRSG size [36][23]. Finning materials need to have high conductivity. Fin shape and pitch have to be selected properly to prevent excessive pressure drops in HRSG, respectively too high gas side backpressure to the gas turbine. [36]

Two superheaters HPSH1 and HPSH2 with an intermediate injection cooler TMX1 have a function of superheating the steam to a temperature of 470°C. Thereby is each superheater sized in a way to provide approximately half of the temperature rise from the saturation temperature to the selected fresh steam temperature of 470°C. SH tubes are also equipped with finning which is of different material than in economizer and evaporator due to higher temperatures. Furthermore finning layer at superheater tubes is thinner so as to avoid oxidation of the fining [23][51]. In the terms of flow direction all heat exchangers are designed as counter flow heat exchangers. Since the gas turbines operate over a quite wide load range, it is the function of supplementary firing to maintain a constant pressure of fresh steam independently on enthalpy of flue gases from the gas turbines.

#### 7.1.4 Design point results

HRSG temperature profile on the gas respectively steam and water side is shown in a T-Q diagram given in Figure 7.11.

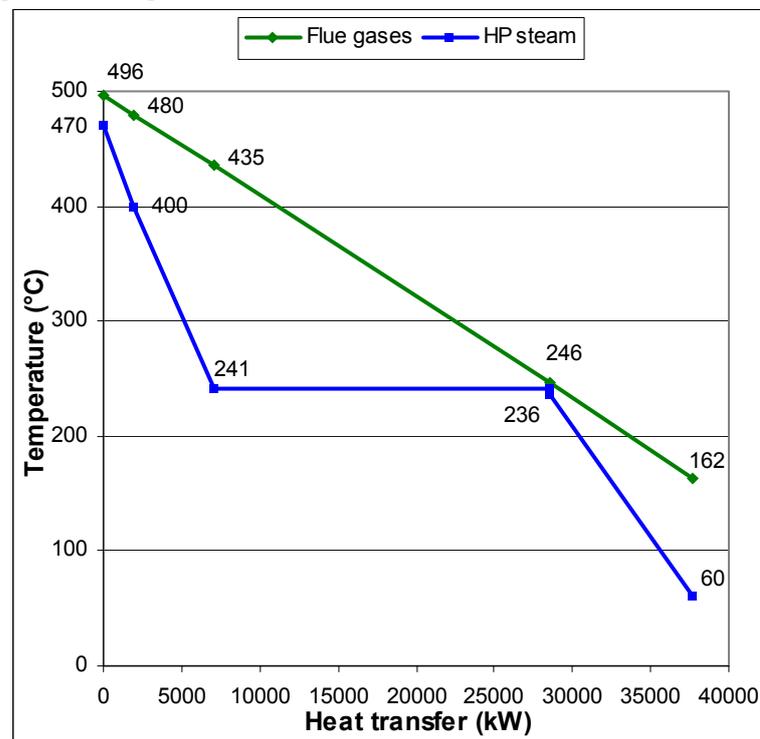


Figure 7.11: T-Q diagram configuration 1P

It is apparent from Figure 7.11 that flue gases enter the HRSG with the state according Table 4-4 respectively that there is no additional firing used. Surfaces of HRSG sections calculated at the design point following procedure given in 6.2 are shown in Table 7-3.

*Table 7-3: 1P configuration heat exchanger surface areas*

Component	Abbreviation	Unit	Value
HP Superheater 2	AHPSH2	m <sup>2</sup>	716
HP Superheater 1	AHPSH1	m <sup>2</sup>	786
HP Evaporator	AHPEVA	m <sup>2</sup>	15631
HP Economizer	AHPECO	m <sup>2</sup>	8951

Size of condenser and required cooling water mass flow are given in Table 7-4.

*Table 7-4: Condenser size and cooling water requirement*

Condenser surface area	ACONDEN	m <sup>2</sup>	661
Cooling water mass flow	mcw	kg/s	319

## 7.2 2P1 Configuration

2P1 configuration is shown in Figure 7.2. In direction of flue gas path it comprises two HPSH, HPEVA followed by LP loop consisting of LPSH, LPEVA and LPECO. HPECO is arranged after the LP loop at the end of the flue gas path.

Feed water is entering both LPECO and HPECO with the same temperature as in the case of 1P configuration with a value of 60°C. HPECO is sized in a way to provide temperature difference between the inlet gas and feed water outlet temperature of 5°C. LPECO is sized in a way to provide approach temperature at LPEVA inlet of 5°C. Low pressure pinch point (LPPP), high pressure pinch point (HPPP) and fresh steam pressures are being optimized. LP fresh steam temperature is not controlled since its temperature rise in operating regimes does not exceed 5°C and therefore does not jeopardize the ST operation. The ST is equipped with one admission for LP steam and one uncontrolled extraction for deaerator steam. Analogically as in the case of 1P configuration extraction pressure is 0.2bar what is also the operating pressure of the deaerator.

### 7.2.1 Exergo-economic optimization results

Optimization progress of the objective function as a function of independent variables is shown graphically in Figure 7.12 whereby are the optimization results given in Table 7-5.

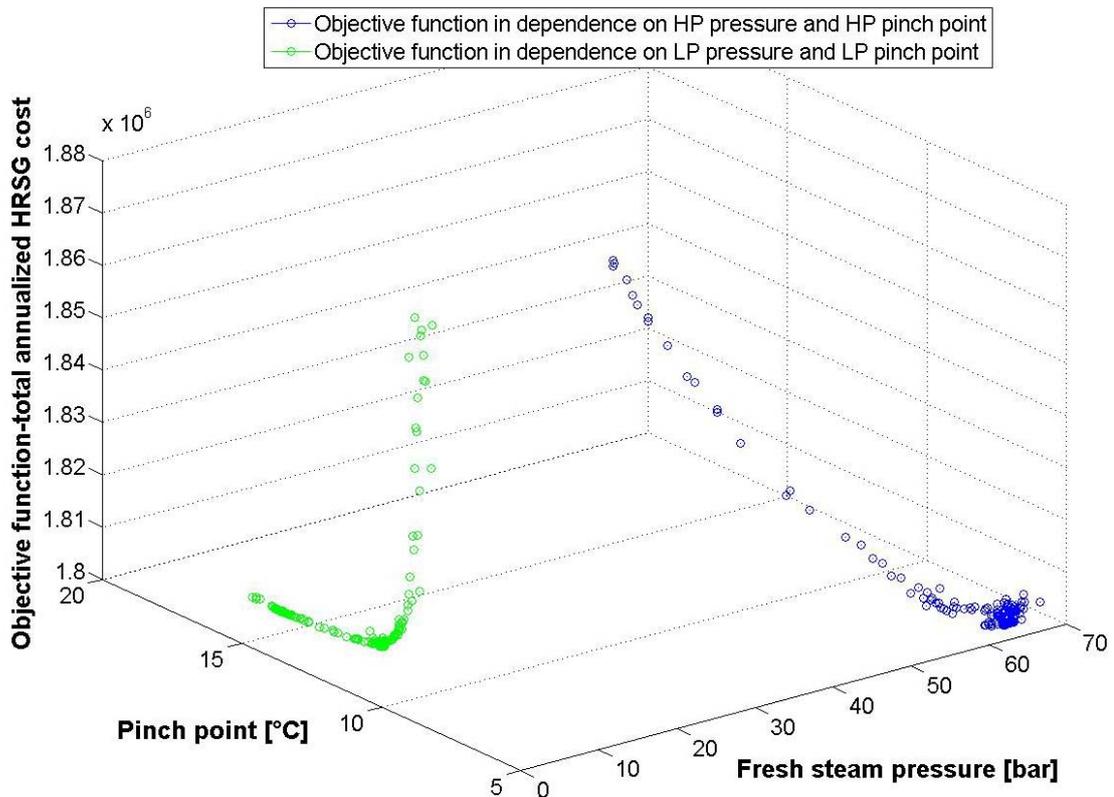


Figure 7.12: Optimization progress of the objective function for configuration 2P1

Table 7-5: 2P1 configuration operative parameters

Parameter	Abbreviation	Unit	Value
Total annualized HRSG cost	$K_{total, min}$	\$	1802987
Fresh steam pressure	pHP	bar	61.7
Fresh steam pressure	pLP	bar	6.0
HP pinch	HP PP	°C	5.0
LP pinch	LP PP	°C	15.2
Steam quantity	mHP	kg/s	8.8
Steam quantity	mLP	kg/s	5.3
Heat recovery efficiency	$\eta_{HR}$	%	79.6
Steam cycle efficiency	$\eta_{SC}$	%	20.7
Net cycle CC efficiency	$\eta_{CC}$	%	38.8
Steam turbine el. Power	PST,el	kW	11820

## 7.2.2 Design point results

HRSG temperature profile on the gas respectively steam and water side is shown in a T-Q diagram given in Figure 7.13.

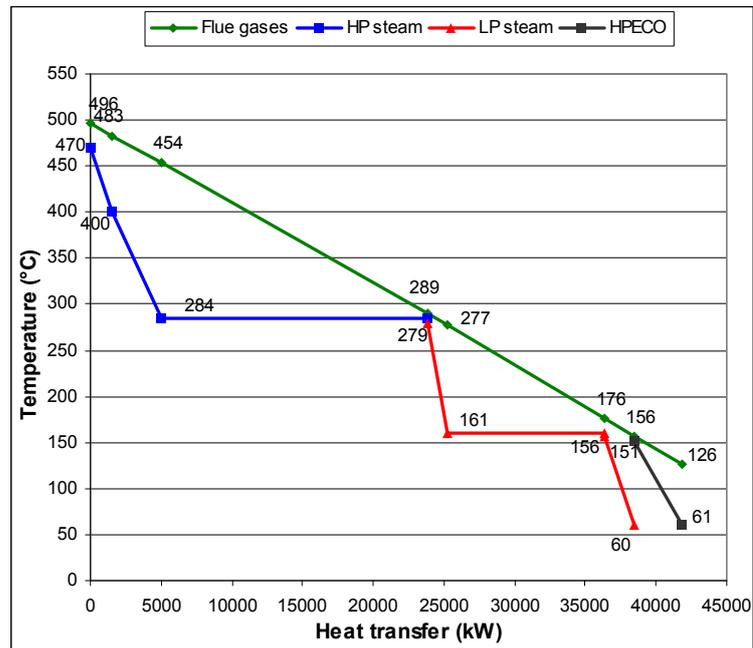


Figure 7.13: T-Q diagram configuration 2P1

Quite lower stack temperature than in the case of 1P configuration is notable. Surfaces of HRSG sections calculated at the design point are shown in Table 7-6.

Table 7-6: Heat exchanger surface areas 2P1 configuration

Component	Abbreviation	Unit	Value
HP superheater 2	AHPSH2	m <sup>2</sup>	584
HP superheater 1	AHPSH1	m <sup>2</sup>	630
HP evaporator	AHPEVA	m <sup>2</sup>	15126
HP economizer	AHPECO	m <sup>2</sup>	5659
LP superheater	ALPSH	m <sup>2</sup>	1276
LP evaporator	ALPEVA	m <sup>2</sup>	8443
LP economizer	ALPECO	m <sup>2</sup>	1675

Size of condenser and required cooling water mass flow is given in Table 7-7.

Table 7-7: Condenser size and cooling water requirement 2P1 configuration

Condenser surface area	ACONDEN	m <sup>2</sup>	743
Cooling water mass flow	m <sub>cw</sub>	kg/s	360

### 7.3 2P2 Configuration

2P2 configuration shown in Figure 7.3 is similar to configuration 2P1 with HPECO arranged after the HPEVA in the flue gas path so as to provide lower approach point at HPEVA inlet. Thereof preheating of feed water to a saturation temperature is done via economizer unlike configuration 2P1 where great portion of heat required for preheating the feed water is transferred by means of evaporator.

#### 7.3.1 Exergo-economic optimization results

The lowest value of total annualized HRSG cost is obtained for the operative parameters given in Table 7-8.

Table 7-8: 2P2 configuration operative parameters

Parameter	Abbreviation	Unit	Value
Total annualized HRSG cost	$K_{total, min}$	\$	1899096
Fresh steam pressure	pHP	bar	110.2
Fresh steam pressure	pLP	bar	4.0
HP pinch	HP PP	°C	15.8
LP pinch	LP PP	°C	7.2
Steam quantity	mHP	kg/s	9.9
Steam quantity	mLP	kg/s	3.9
Heat recovery efficiency	$\eta_{HR}$	%	76.6
Steam cycle efficiency	$\eta_{SC}$	%	21.1
Net cycle CC efficiency	$\eta_{CC}$	%	39.0
Steam turbine el. Power	$P_{ST,el}$	kW	12077

#### 7.3.2 Design point results

Temperature profile of flue gases respectively water and steam is given in Figure 7.14.

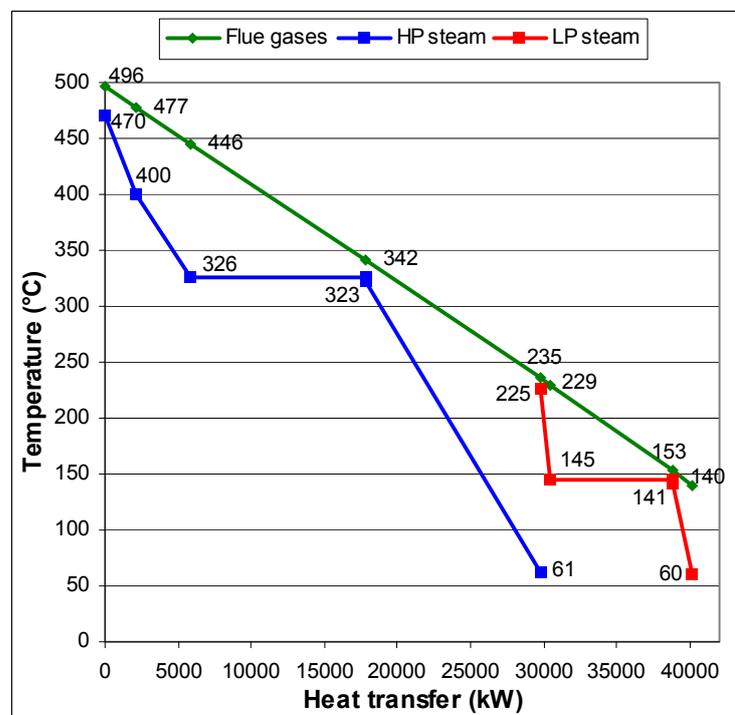


Figure 7.14: T-Q diagram configuration 2P2

From Table 7-8 and Figure 7.14 it is apparent that configuration 2P2 provides higher net cycle efficiency but with higher stack temperature resulting in lower heat recovery

efficiency. Thereof it makes sense to split the HPECO in two parts and install the second one at the end of the flue gas path.

Surfaces of HRSG sections calculated at the design point are shown in Table 7-9.

Table 7-9: Heat exchanger surface areas 2P2 configuration

Component	Abbreviation	Unit	Value
HP superheater 2	AHPSH2	m <sup>2</sup>	884
HP superheater 1	AHPSH1	m <sup>2</sup>	1031
HP evaporator	AHPEVA	m <sup>2</sup>	8744
HP economizer	AHPECO	m <sup>2</sup>	6298
LP superheater	ALPSH	m <sup>2</sup>	769
LP evaporator	ALPEVA	m <sup>2</sup>	10099
LP economizer	ALPECO	m <sup>2</sup>	1457

Size of condenser and required cooling water mass flow is given in Table 7-10.

Table 7-10: Condenser size and cooling water requirement 2P2 configuration

Condenser surface area	ACONDEN	m <sup>2</sup>	698
Cooling water mass flow	m <sub>cw</sub>	kg/s	337

## 7.4 2P3 Configuration

Configuration 2P3 is similar to 2P2 configuration with addition of second HP economizer installed after the LPECO in the flue gas path. Intention is further increase of heat recovery efficiency through closing the water-steam temperature profile to the temperature profile of the flue gases.

### 7.4.1 Exergo-economic optimization results

The lowest value of total annualized HRSG cost is obtained for the operative parameters given in Table 7-11.

Table 7-11: 2P3 configuration operative parameters

Parameter	Abbreviation	Unit	Value
Total annualized HRSG cost	K <sub>total, min</sub>	\$	1740566
Fresh steam pressure	p <sub>HP</sub>	bar	105.2
Fresh steam pressure	p <sub>LP</sub>	bar	4.7
HP pinch	HP PP	°C	15.7
LP pinch	LP PP	°C	13.1
Steam quantity	m <sub>HP</sub>	kg/s	9.9
Steam quantity	m <sub>LP</sub>	kg/s	4.5
Heat recovery efficiency	η <sub>HR</sub>	%	80.4
Steam cycle efficiency	η <sub>SC</sub>	%	21.9
Net cycle CC efficiency	η <sub>CC</sub>	%	39.6
Steam turbine el. Power	P <sub>ST,el</sub>	kW	12514

### 7.4.2 Design point results

Temperature profile of flue gases respectively water and steam is given in Figure 7.15.

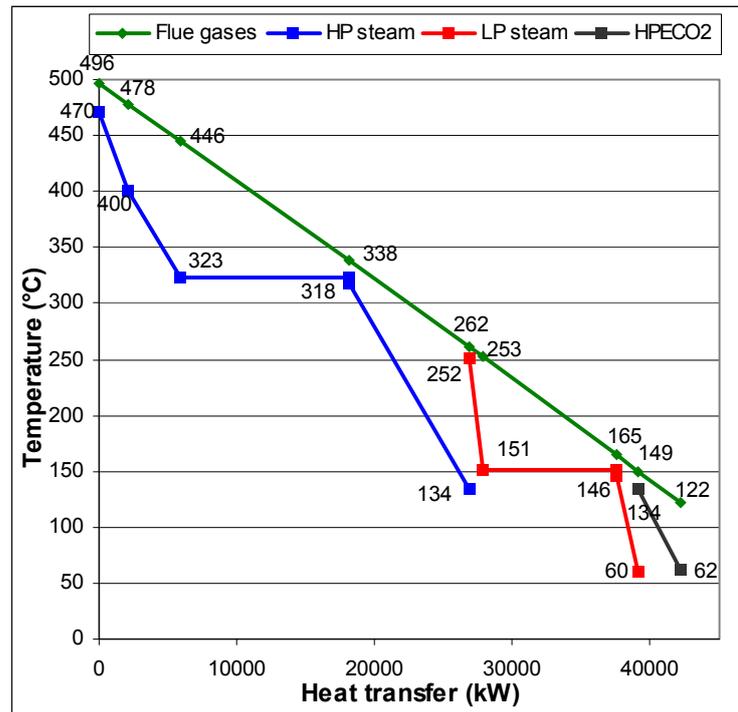


Figure 7.15: T-Q diagram configuration 2P3

In comparison with former 2P1 and 2P2 configurations here is lower stack temperature obtained resulting in higher heat recovery efficiency followed by also higher net cycle CC efficiency.

Surfaces of HRSG sections calculated at the design point are given in Table 7-12.

Table 7-12: Heat exchanger surface areas 2P3 configuration

Component	Abbreviation	Unit	Value
HP superheater 2	AHPSH2	m <sup>2</sup>	861
HP superheater 1	AHPSH1	m <sup>2</sup>	991
HP evaporator	AHPEVA	m <sup>2</sup>	8899
HP economizer 1	AHPECO1	m <sup>2</sup>	3479
HP economizer 2	AHPECO2	m <sup>2</sup>	5334
LP superheater	ALPSH	m <sup>2</sup>	996
LP evaporator	ALPEVA	m <sup>2</sup>	8450
LP economizer	ALPECO	m <sup>2</sup>	1412

Size of condenser and required cooling water mass flow is given in Table 7-13.

Table 7-13: Condenser size and cooling water requirement 2P3 configuration

Condenser surface area	ACONDEN	m <sup>2</sup>	736
Cooling water mass flow	m <sub>cw</sub>	kg/s	355

## 7.5 3P1 Configuration

Additional to 2P1 configuration here is an IP loop installed between the LP and HP loop comprising IP economizer, IP evaporator and IP superheater. Feed water is entering LPECO, IPECO and HPECO with the same temperature of 60°C as in the case of 2P1 configuration. Selected approach temperatures at LPEVA, IPEVA and HPEVA inlet are 5°C. HP/IP/LP fresh steam pressures and pinch points are being optimized with results given in Table 7-14. LP and IP fresh steam temperatures are not controlled since a temperature rise in operating regimes does not exceed 5°C and therefore does not jeopardize the ST operation. The ST is of single casing design and is equipped with two admissions; one for IP steam comprising one stop valve and two control valves and one for LP steam comprising one stop valve and two control valves. Furthermore the ST is equipped with one uncontrolled extraction for deaerator steam. Analogically as in the case of 2P1 configuration extraction pressure is 0.2bar what is also the operating pressure of the deaerator.

### 7.5.1 Exergo-economic optimization results

Optimization progress of the objective function as a function of independent variables is shown graphically in Figure 7.16 with optimization results given in Table 7-14.

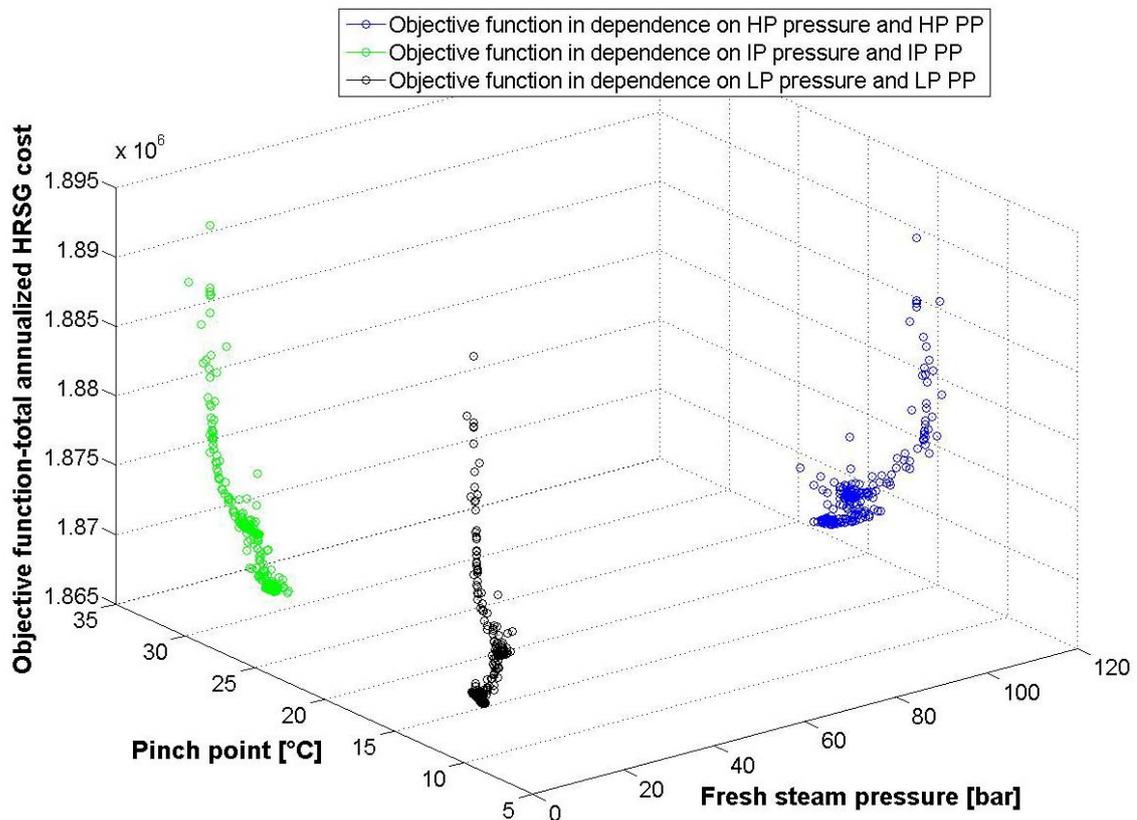


Figure 7.16: Optimization progress of the objective function for configuration 3P1

Table 7-14: 3P1 configuration operative parameters

Parameter	Abbreviation	Unit	Value
Total annualized HRSG cost	$K_{\text{total, min}}$	\$	1869441
Fresh steam pressure	pHP	bar	105.5
Fresh steam pressure	pIP	bar	7.0
Fresh steam pressure	pLP	bar	2.0
HP pinch	HP PP	°C	18.3
IP pinch	IP PP	°C	26.5
LP pinch	LP PP	°C	9.3
Steam quantity	mHP	kg/s	9.7
Steam quantity	mIP	kg/s	2.3
Steam quantity	mLP	kg/s	2.5
Heat recovery efficiency	$\eta_{\text{HR}}$	%	79.7
Steam cycle efficiency	$\eta_{\text{SC}}$	%	21.4
Net cycle CC efficiency	$\eta_{\text{CC}}$	%	39.3
Steam turbine el. Power	$P_{\text{ST,el}}$	kW	12253

### 7.5.2 Design point results

Temperature profile of flue gases respectively water and steam is given in Figure 7.17.

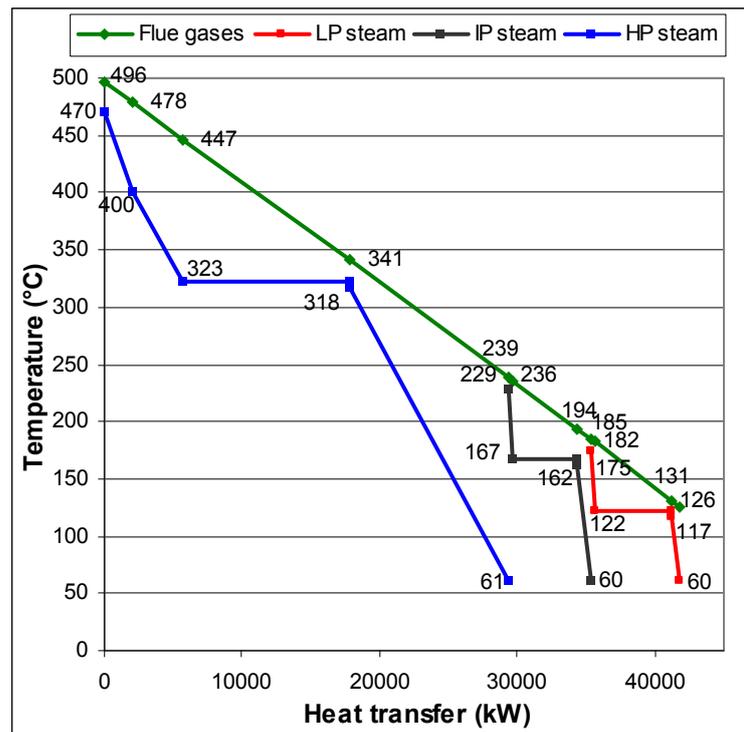


Figure 7.17: T-Q diagram configuration 3P1

Apparently configuration 3P1 provides slightly lower heat recovery efficiency and CC net efficiency than the most efficient 2P configuration. Furthermore high amount of heat transferred to HPECO is notable. This suggests splitting the HPECO in more parts.

Surfaces of HRSG sections calculated at the design point following procedure given in 6.2 are shown in Table 7-15.

Table 7-15: Heat exchanger surface areas for 3P1 configuration

Component	Abbreviation	Unit	Value
HP superheater 2	AHPSH2	m <sup>2</sup>	840
HP superheater 1	AHPSH1	m <sup>2</sup>	970
HP evaporator	AHPEVA	m <sup>2</sup>	8269
HP economizer	AHPECO	m <sup>2</sup>	5538
IP superheater	AIPSH	m <sup>2</sup>	421
IP evaporator	AIPEVA	m <sup>2</sup>	3960
IP economizer	AIPECO	m <sup>2</sup>	542
LP superheater	ALPSH	m <sup>2</sup>	383
LP evaporator	ALPEVA	m <sup>2</sup>	7642
LP economizer	ALPECO	m <sup>2</sup>	674

Size of condenser and required cooling water mass flow is given in Table 7-16.

Table 7-16: Condenser size and cooling water requirement 3P1 configuration

Condenser surface area	ACONDEN	m <sup>2</sup>	733
Cooling water mass flow	m <sub>cw</sub>	kg/s	354

## 7.6 3P2 Configuration

Configuration 3P2 is similar to 3P1 configuration with addition of the second HP economizer installed after LPECO in the flue gas path. Intention is further increase of heat recovery efficiency through closing the water-steam temperature profile to the temperature profile of the flue gases.

### 7.6.1 Exergo-economic optimization results

The lowest value of total annualized HRSG cost is obtained for the operative parameters given in Table 7-17.

Table 7-17: 3P2 configuration operative parameters

Parameter	Abbreviation	Unit	Value
Total annualized HRSG cost	K <sub>total, min</sub>	\$	1746602
Fresh steam pressure	pHP	bar	106.5
Fresh steam pressure	pIP	bar	7.0
Fresh steam pressure	pLP	bar	2.0
HP pinch	HP PP	°C	18.5
IP pinch	IP PP	°C	25.1
LP pinch	LP PP	°C	18.0
Steam quantity	mHP	kg/s	9.7
Steam quantity	mIP	kg/s	3.5
Steam quantity	mLP	kg/s	1.8
Heat recovery efficiency	η <sub>HR</sub>	%	82.9
Steam cycle efficiency	η <sub>SC</sub>	%	22.3
Net cycle CC efficiency	η <sub>CC</sub>	%	39.9
Steam turbine el. Power	P <sub>ST,el</sub>	kW	12726

## 7.6.2 Design point results

Temperature profile of flue gases respectively water and steam is given in Figure 7.18.

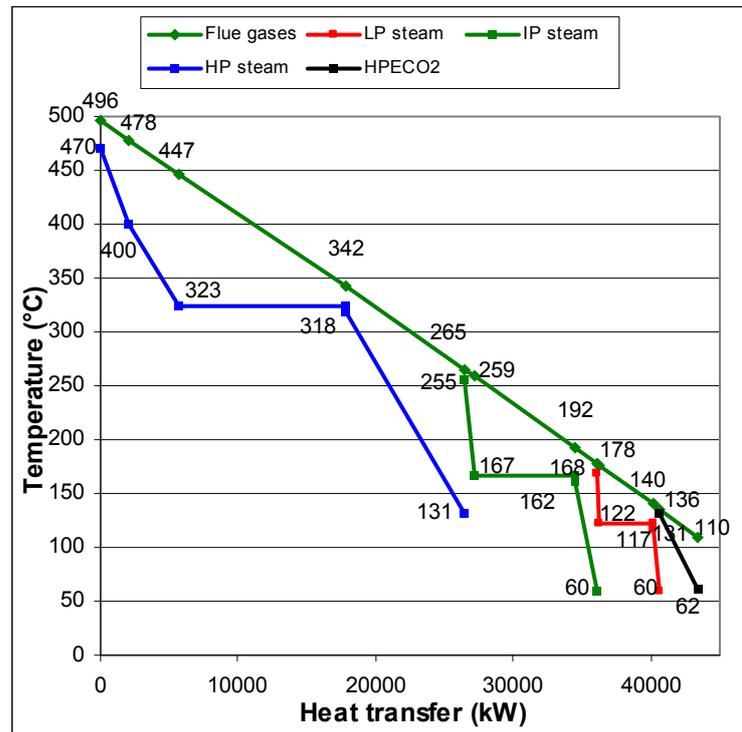


Figure 7.18: T-Q diagram configuration 3P2

Minor increase of heat recovery efficiency and net CC efficiency compared to former 3P1 configuration is notable. Surfaces of HRSG sections calculated at the design point are shown in Table 7-18.

Table 7-18: Heat exchanger surface areas for 3P2 configuration

Component	Abbreviation	Unit	Value
HP superheater 2	AHPSH2	m <sup>2</sup>	842
HP superheater 1	AHPSH1	m <sup>2</sup>	974
HP evaporator	AHPEVA	m <sup>2</sup>	8199
HP economizer 1	AHPECO1	m <sup>2</sup>	4944
HP economizer 2	AHPECO2	m <sup>2</sup>	5711
IP superheater	AIPSH	m <sup>2</sup>	767
IP evaporator	AIPEVA	m <sup>2</sup>	5385
IP economizer	AIPECO	m <sup>2</sup>	898
LP superheater	ALPSH	m <sup>2</sup>	258
LP evaporator	ALPEVA	m <sup>2</sup>	4547
LP economizer	ALPECO	m <sup>2</sup>	367

Size of condenser and required cooling water mass flow is given in Table 7-19.

Table 7-19: Condenser size and cooling water requirement 3P2 configuration

Condenser surface area	ACONDEN	m <sup>2</sup>	764
Cooling water mass flow	m <sub>cw</sub>	kg/s	368

## 7.7 3P3 Configuration

Configuration 3P3 is similar to 3P2 configuration with addition of the third HP economizer installed between IPECO and LPSH in the flue gas path. Intention is further increase of heat recovery efficiency through closing the water-steam temperature profile to the temperature profile of the flue gases.

### 7.7.1 Exergo-economic optimization results

The lowest value of total annualized HRSG cost is obtained for the operative parameters given in Table 7-20.

Table 7-20: 3P3 configuration operative parameters

Parameter	Abbreviation	Unit	Value
Total annualized HRSG cost	$K_{\text{total, min}}$	\$	1763105
Fresh steam pressure	pHP	bar	107.8
Fresh steam pressure	pIP	bar	6.9
Fresh steam pressure	pLP	bar	2.0
HP pinch	HP PP	°C	20.7
IP pinch	IP PP	°C	27.3
LP pinch	LP PP	°C	14.4
Steam quantity	mHP	kg/s	9.6
Steam quantity	mIP	kg/s	4.3
Steam quantity	mLP	kg/s	1.2
Heat recovery efficiency	$\eta_{\text{HR}}$	%	83.3
Steam cycle efficiency	$\eta_{\text{SC}}$	%	22.5
Net cycle CC efficiency	$\eta_{\text{CC}}$	%	40.0
Steam turbine el. Power	$P_{\text{ST,el}}$	kW	12844

### 7.7.2 Design point results

Temperature profile of flue gases respectively water and steam is given in Figure 7.19.

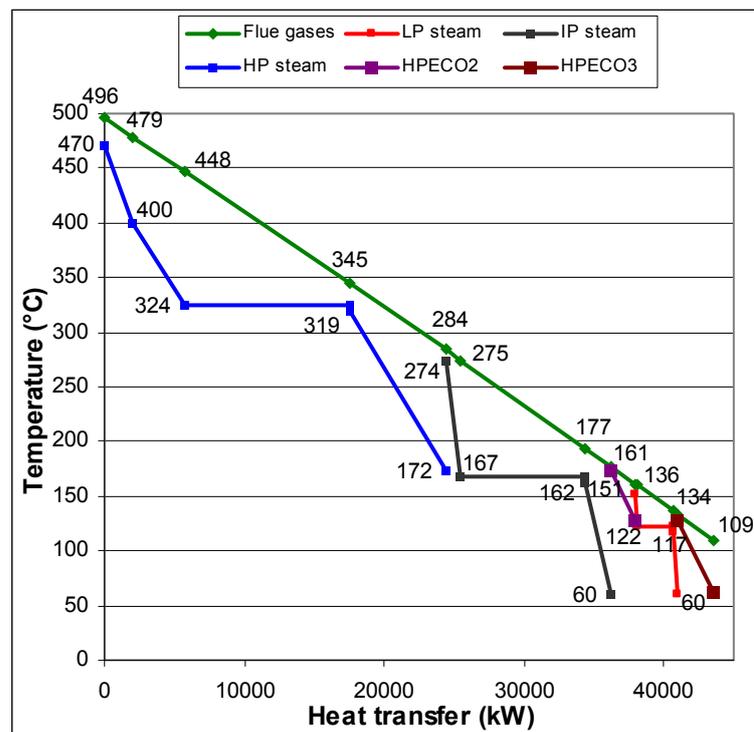


Figure 7.19: T-Q diagram configuration 3P3

Surfaces of HRSG sections calculated at the design point are shown in Table 7-21.

Table 7-21: Heat exchanger surface areas for 3P3 configuration

Component	Abbreviation	Unit	Value
HP superheater 2	AHPSH2	m <sup>2</sup>	837
HP superheater 1	AHPSH1	m <sup>2</sup>	965
HP evaporator	AHPEVA	m <sup>2</sup>	7730
HP economizer 1	AHPECO1	m <sup>2</sup>	4272
HP economizer 2	AHPECO2	m <sup>2</sup>	4662
HP economizer 3	AHPECO3	m <sup>2</sup>	5528
IP superheater	AIPSH	m <sup>2</sup>	997
IP evaporator	AIPEVA	m <sup>2</sup>	5711
IP economizer	AIPECO	m <sup>2</sup>	1057
LP superheater	ALPSH	m <sup>2</sup>	135
LP evaporator	ALPEVA	m <sup>2</sup>	4023
LP economizer	ALPECO	m <sup>2</sup>	264

Size of condenser and required cooling water mass flow is given in Table 7-22.

Table 7-22: Condenser size and cooling water requirement 3P3 configuration

Condenser surface area	ACONDEN	m <sup>2</sup>	765
Cooling water mass flow	m <sub>cw</sub>	kg/s	370

## 7.8 Comparison and selection

Comparison of minimal values of objective function and its components for all configurations is given in Figure 7.20.

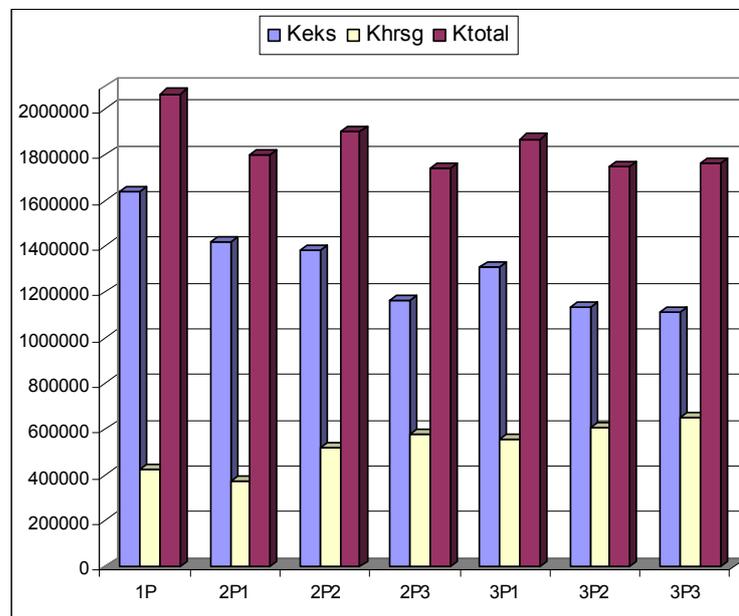


Figure 7.20: Value of optimal objective function and its components

Falling trend of thermodynamic component and rising trend of thermoeconomic component is notable. It is also apparent that the objective function provides minimal value in case of configuration 2P3.

Table 7-23: Comparison of 1P, 2P and 3P configurations

Configuration	1P	2P1	2P2	2P3
Steam turbine power [kW]	10989	11820	12077	12514
Net CC efficiency [%]	37.6	38.8	39.0	39.6
Steam cycle efficiency [%]	19.2	20.7	21.1	21.9
Heat recovery efficiency [%]	71.4	79.6	76.6	80.4
HRSG surface cost [\$]	1955398	2427076	2366647	2352113
$K_{total, min}$ [\$]	2066746	1802987	1899096	1740566

Configuration	3P1	3P2	3P3
Steam turbine power [kW]	12253	12726	12844
Net CC efficiency [%]	39.3	39.9	40.0
Steam cycle efficiency [%]	21.4	22.3	22.5
Heat recovery efficiency [%]	79.7	82.9	83.3
HRSG surface cost [\$]	2362172	2463025	2571792
$K_{total, min}$ [\$]	1869441	1746602	1763105

From Table 7-23 it is apparent that the configuration 3P3 offers the highest CC efficiency, steam cycle efficiency and heat recovery efficiency due to the lowest stack temperature. Configuration 2P3 provides the lowest total annualized HRSG cost. Calculation results prove that configuration with the lowest total annualized HRSG cost does not obtain the highest thermodynamic efficiency (with respect to optimized pressure and pinch point values).

Comparison of net CC efficiency, steam cycle efficiency and heat recovery efficiency is given in Figure 7.21. Apparently each configuration provides further increase of net CC efficiency and steam cycle efficiency with exception of 3P1 compared with 2P3 configuration. Furthermore heat recovery efficiency has different values due to the fact that it mostly depends on the flue gas temperature obtained at HRSG exit.

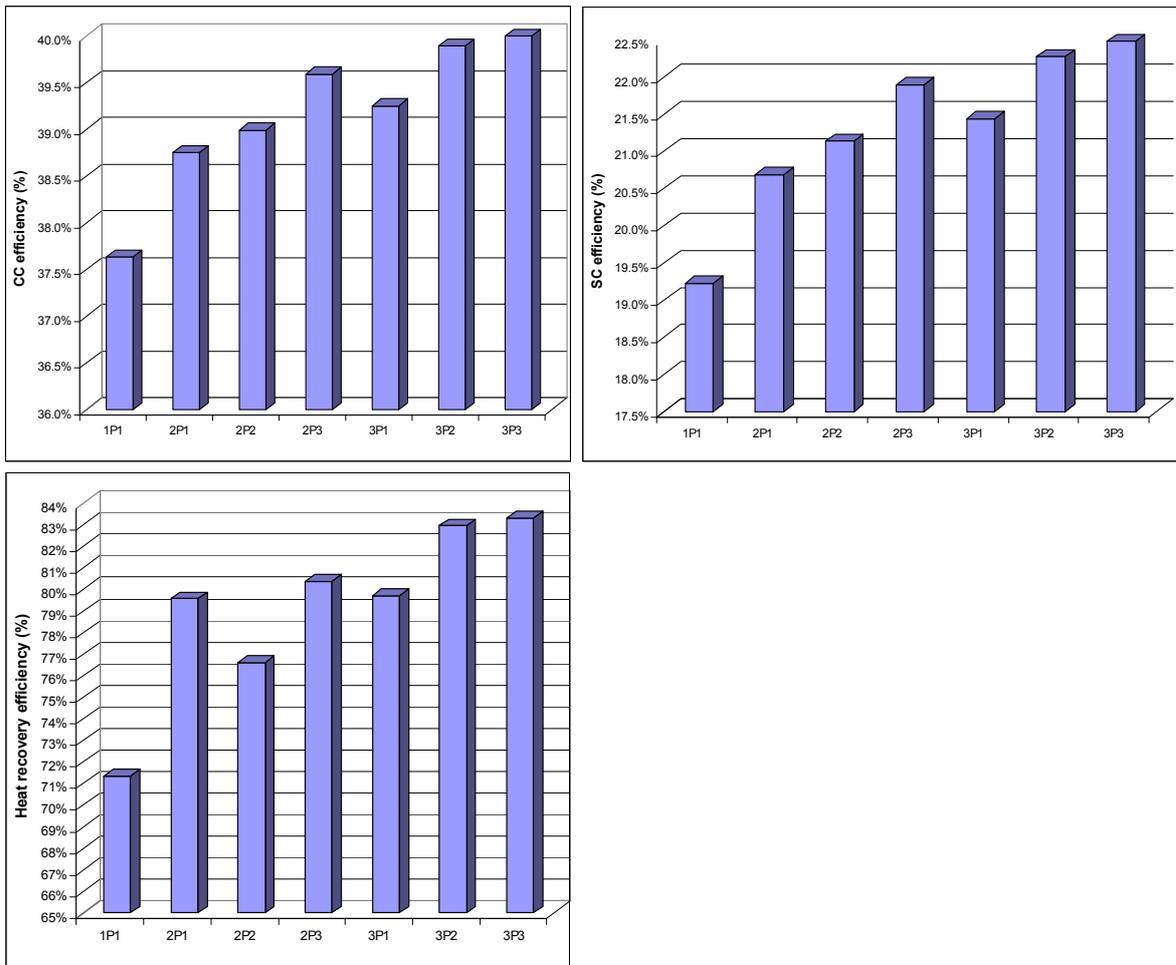


Figure 7.21: CC, SC and HRSG efficiency

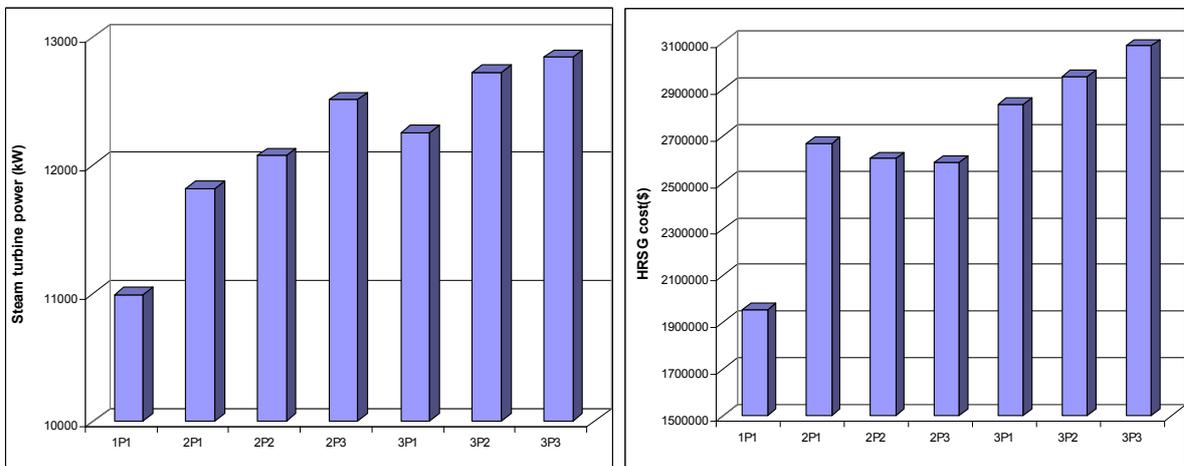


Figure 7.22: ST power, HRSG surface cost

Figure 7.22 shows comparison of steam turbine power and HRSG cost. It is apparent that steam turbine power series follows the shape of the series of net CC efficiency. HRSG cost has a maximal value for the 3P3 configuration. At 2P configurations HRSG cost slightly decreases due to surface size resulting from optimization procedure. In the terms of costs steam turbine cost is proportional to the steam turbine power. As the steam turbine and HRSG form majority of the complete heat recovery system cost it is apparent that the 3P3 configuration is the most expensive.

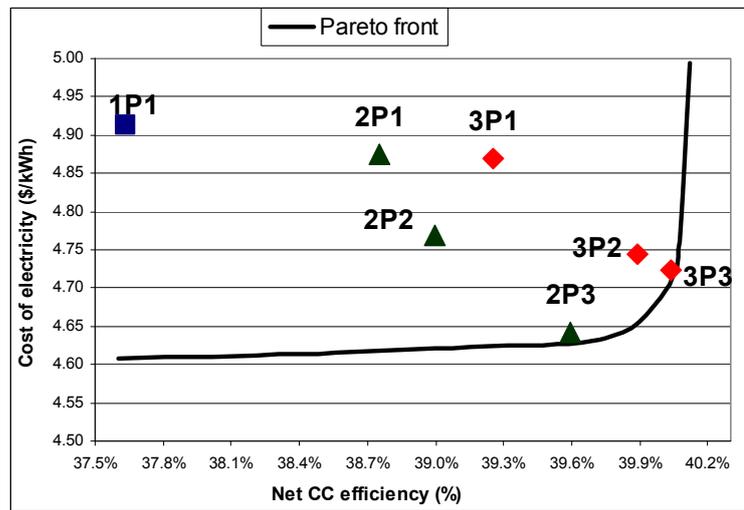


Figure 7.23 Cost of electricity as a function of CC net efficiency

Figure 7.23 shows dependency of cost of electricity on net CC efficiency for all configurations. So as to compare configurations based on criteria of maximal efficiency and the lowest cost of electricity Pareto front is used. It is constructed by approximation of polynomial which is passing through the points with maximal efficiency and minimal cost of electricity. Furthermore the polynomial is extended in both x and y directions so as to illustrate vicinity of other configurations to the Pareto front. [13][55] Apparently 2P3 and 3P3 configurations are the closest to the Pareto front whereby the first is characterized by the lowest cost of electricity and the latter by the highest system efficiency.

## 7.9 Optimization sensitivity analysis

### 7.9.1 Influence of different exhaust gas parameters on optimization results

So as to examine behavior of objective function on a wide range of exhaust gas mass flow and temperature, optimal fresh steam pressure and pinch point are calculated. Thereby are exhaust gas temperatures varied from 480 to 540°C and exhaust gas mass flow from 30 to 80 kg/s. The values are usual for industrial gas turbines in the range of 5-25MW [23].

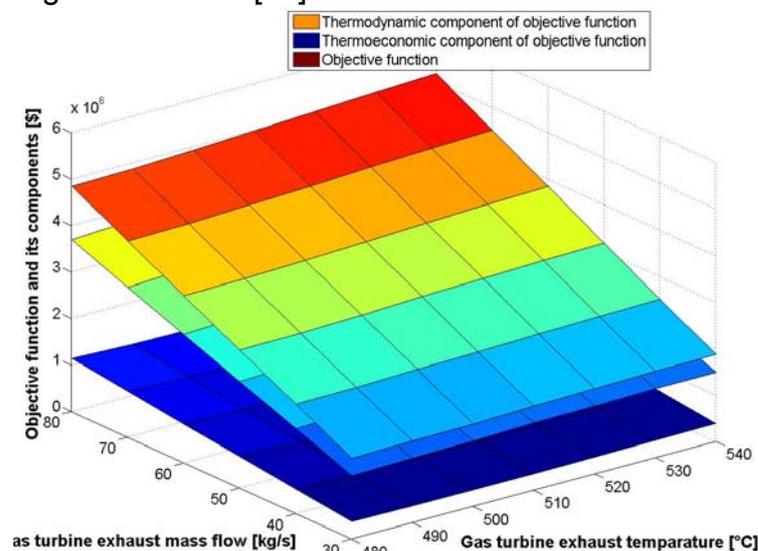


Figure 7.24 Objective function and its components vs. exhaust gas mass flow and temperature

Figure 7.24 shows the shape of the objective function and its components over the given range of exhaust gas parameters. Bottom surface presents the thermoeconomic component of the objective function and the middle surface presents the thermodynamic component of the objective function. The top surface presents the objective function and is actually the sum of the bottom and the middle surface. Sensitivity of both components and objective function itself on the gas mass flow is notable. Objective function reaches its minimal values at the lowest gas mass flows. On the other hand influence of exhaust gas temperature on objective function is negligible. It is apparent that objective function and its components are continuous on the given range of exhaust gas parameters.

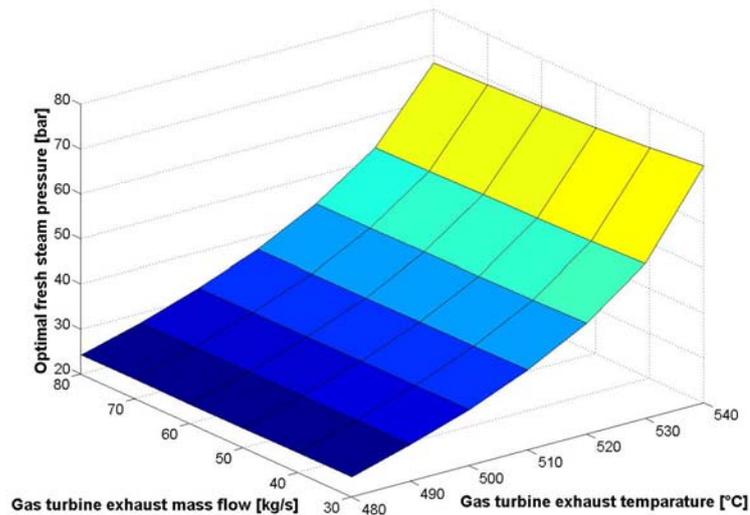


Figure 7.25 Optimal fresh steam pressure vs. exhaust gas mass flow and temperature

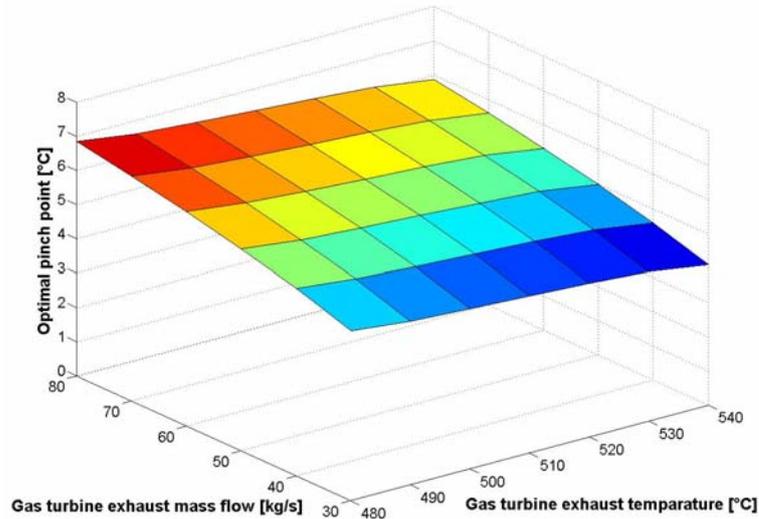


Figure 7.26 Optimal fresh steam pressure vs. exhaust gas mass flow and temperature

It is interesting to note that relatively low temperature of considered GT results in a quite low pressure (ca. 30bar) as shown in Figure 7.25. Higher values of optimal pressure are achieved at higher exhaust gas temperatures what corresponds to reality. Minor influence of gas mass flow is notable. However lower gas mass flows (for the same temperature) result in slightly higher fresh steam pressure.

Figure 7.26 shows dependence of PP on exhaust gas parameters. Apparently optimal PP reaches its minimum at ca. 540°C. For lower exhaust gas temperatures optimal pinch point increases.

In the context of GT performance both graphs have to be understood qualitatively as arbitrary combinations of temperatures and gas mass flows in reality are not possible because thermodynamic features of GT process depend on the compression ratio, the ratio of fuel and air, turbine geometry and other factors.

### 7.9.2 Influence of HRSG heat exchanger surface cost on optimization results

It is interesting to examine influence of increased HRSG surface cost on optimization results. Sensitivity analysis is conducted for configuration 1P for the reference cost given in Table 6-2 followed by 2 and 4 times increased surface cost valid for economizer, evaporator and superheater.

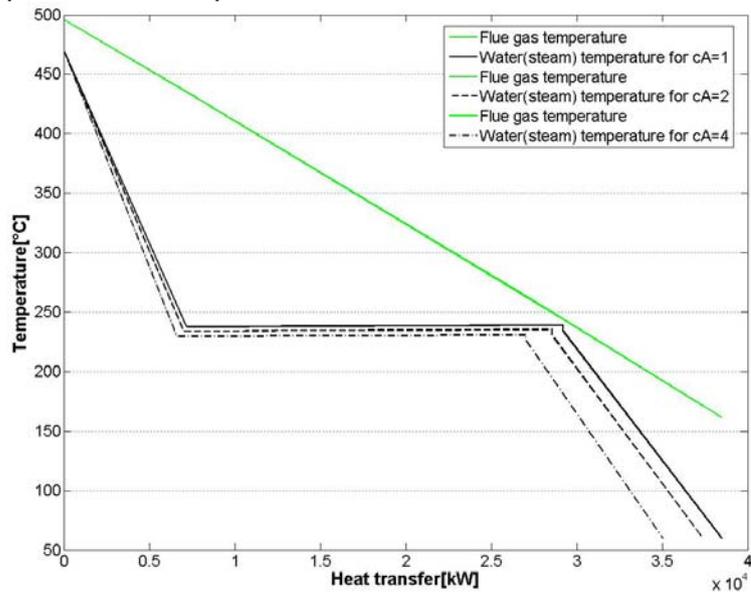


Figure 7.27: T-Q diagram for different HRSG surface cost (1P)

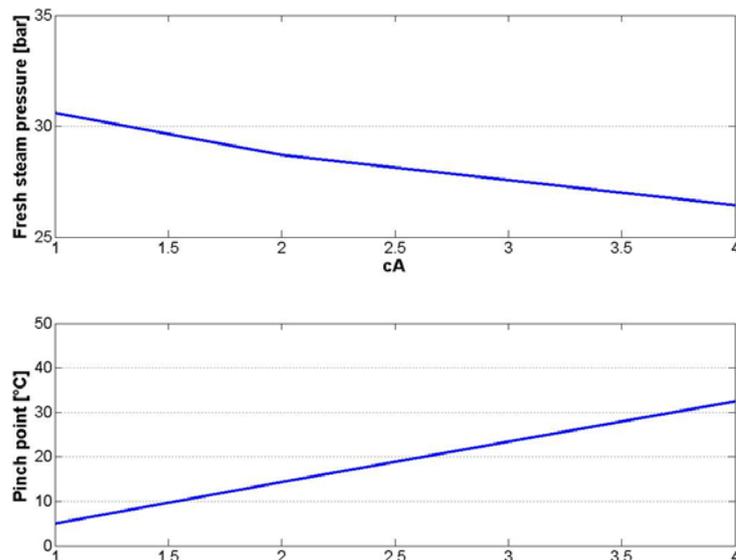


Figure 7.28: Optimal operative parameters for different HRSG surface cost (1P)

From Figure 7.27 and Figure 7.28 it is apparent that increased HRSG surface cost (for the available flue gas conditions) leads to:

- less heat transfer from flue gases to water
- lower pressure respectively higher pinch difference at evaporator inlet

Reason for such phenomena is in the fact that in the case of more expensive HRSG surfaces it is not any more viable to have small temperature difference between the flue gases and water (steam). Since this temperature difference (for constant flue gas conditions) is dictated by steam pressure and pinch difference thereof increased surface cost leads to lower steam pressure and higher pinch difference. Dependence of economizer, evaporator and superheater surface size on increased surface cost is shown in Figure 7.29.

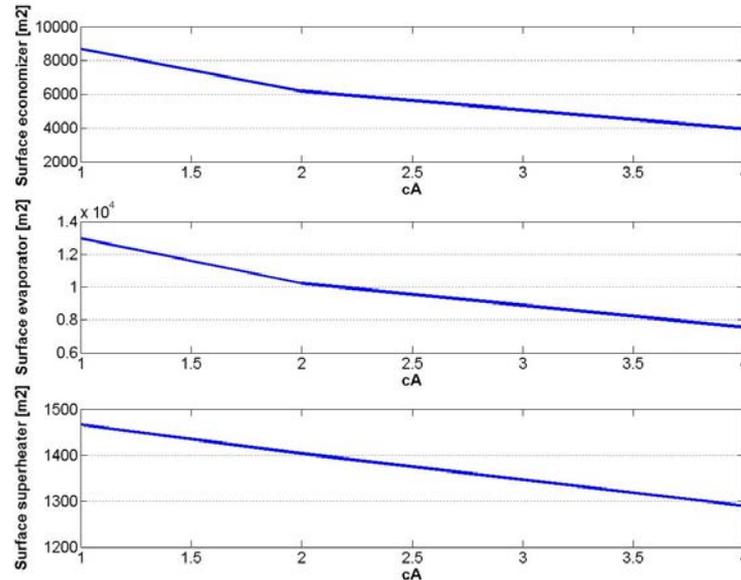


Figure 7.29: HRSG surfaces as a function of surface cost (1P)

Apparently selection of more expensive surface for the respective heat exchanger results in its smaller surface size. Thereby it is notable that influence of increased surface cost on the size of economizer and evaporator is slightly higher than in the case of superheater. Figure 7.30 to Figure 7.32 show dependence of objective function on pressure and pinch point in case of inducing the surface cost factor.

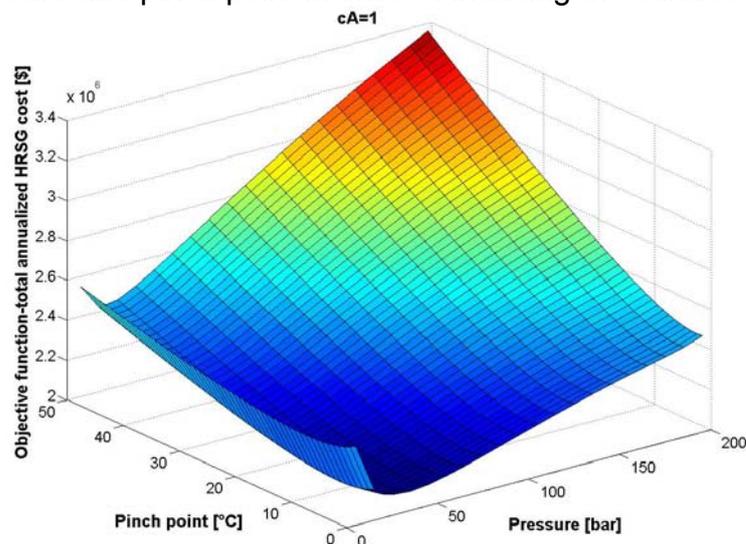


Figure 7.30: Objective function for reference surface cost

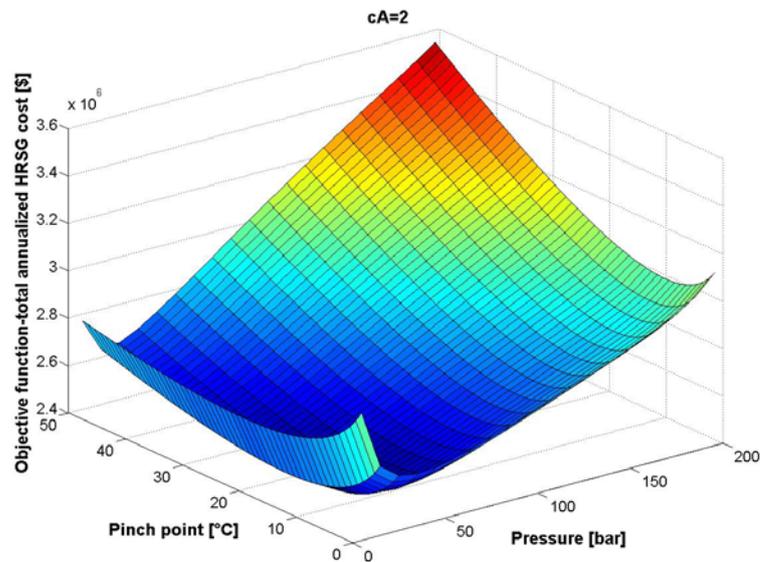


Figure 7.31: Objective function for 2x reference surface cost

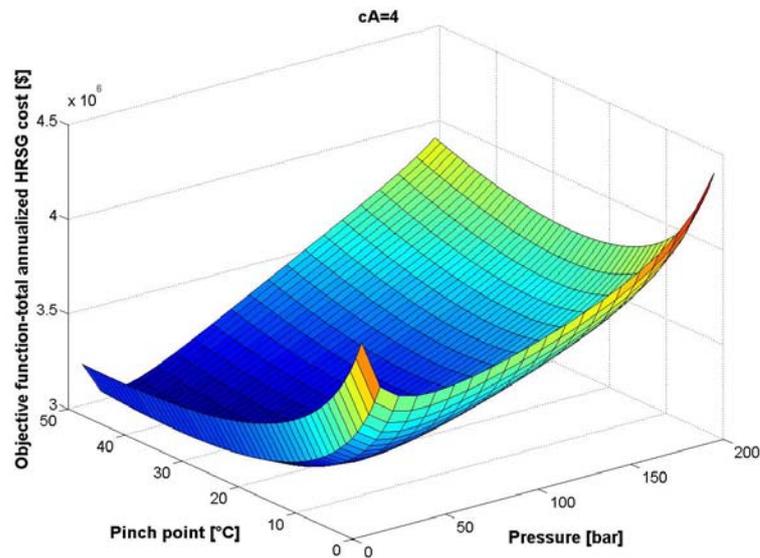


Figure 7.32: Objective function for 4x reference surface cost

It is apparent that pinch point value of 5°C obtained for the reference cost increases as the surfaces get more expensive. Thus for the 2x reference cost pinch point of 13°C is obtained with slight decrease of optimal pressure to a value of 30 bar. For the case of 4x reference cost pinch point of 31°C is obtained followed by optimal pressure of 30bar. Thereof it is obvious that more expensive HRSG surfaces lead to higher pinch point and lower fresh steam pressure.

### 7.9.3 Influence of system complexity factor on optimization results

Origin optimization methodology [48][49] takes into account only HRSG surfaces respectively their cost. However inducing of two or three pressure level HRSG necessitates installation of additional piping, instrumentation, fittings, supports and insulation which increase the HRSG cost. This equipment is in some literature referred to as passive part of HRSG. Unlike HRSG surfaces, HRSG passive part does not depend on heat and mass balance however it depends mainly on number of pressure levels and type of circulation. So as to incorporate the system complexity factor into mathematic model for optimization, values of 1.1 for 2P and 1.2 for 3P configurations are introduced as given in the Table 6-2.

Figure 7.33 and Figure 7.34 show dependence of HRSG cost and net CC efficiency on different values of system complexity factor  $f_c$  resulting from optimization procedure. Apparently higher values of  $f_c$  result in more expensive HRSG and in lower net CC efficiency.

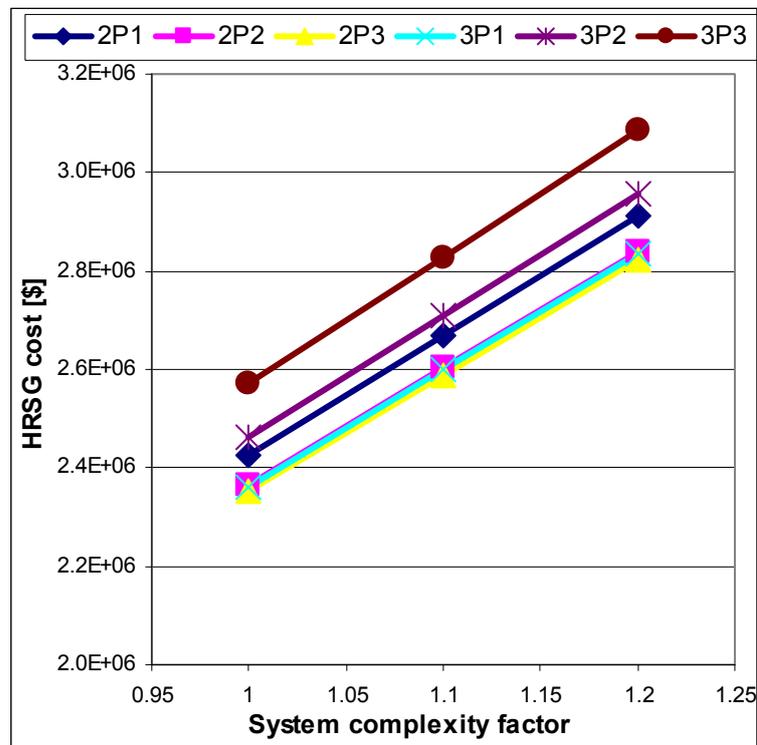


Figure 7.33: HRSG cost vs. system complexity factor

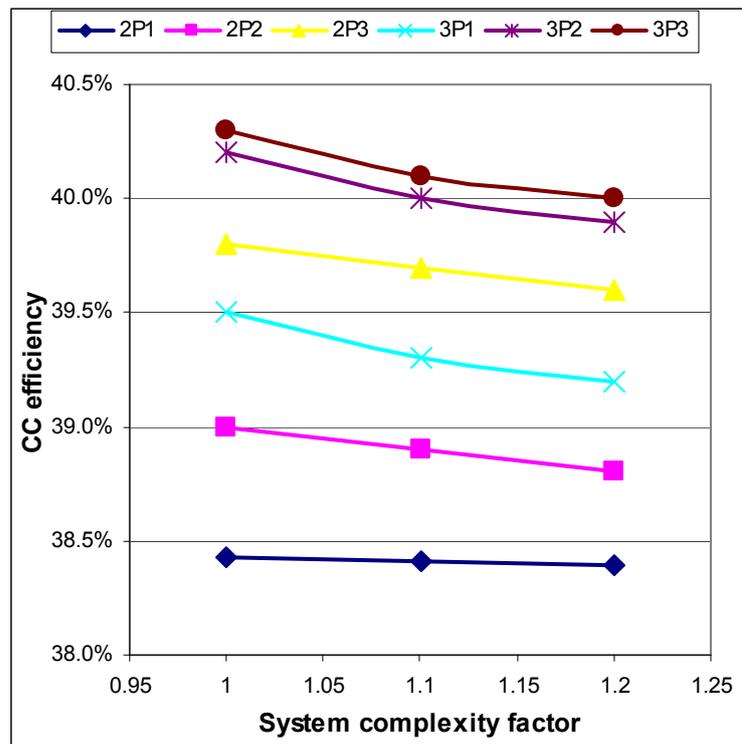


Figure 7.34: Net CC efficiency vs. system complexity factor

#### 7.9.4 Influence of supplementary firing on optimization results

It is interesting to examine influence of supplementary firing on exergo-economic optimization results. Following algorithm explained in 6.3 beside steam pressures and pinch points there is also flue gas temperature after the supplementary firing being optimized so as to find minimal total annualized HRSG cost.

Table 7-24: HRSG operative parameters for fired HRSG

Configuration	1P	2P1	3P1
Ktot min [\$]	3002444	2618357	1930520
$\eta_{CC}$ [%]	40.31	38.4	40.0
Tg2 [°C]	700	510	515
pHP [bar]	120	70	110
pIP [bar]			6
pLP [bar]		4	1.5
HP pinch [°C]	10	10	10
IP pinch [°C]			10
LP pinch [°C]		10	10

In comparison with optimization results for unfired case, from Table 7-24 it is apparent that in the case of fired HRSG all configurations except 3P1 offer lower total annualized HRSG cost. On the other hand 1P and 3P configurations provide slightly higher CC efficiency in case of fired HRSG. However both 2P configurations provide slightly higher CC efficiency in case of unfired HRSG.

It is interesting to note that optimal flue gas temperature after the supplementary firing in case of fired HRSG for the single pressure configuration equals 700°C due to the reasons explained in 5.2.2. Furthermore one should note higher steam pressures in case of fired HRSG.

Obviously it is not possible to select the preferable configuration based only on exergo-economic criteria wherefore additional indexes have to be obtained by means of economic evaluation.

## 8. System off-design behavior

This chapter deals with aspects of system design and operation in pipeline compressor station regimes wherefore are several strategies considered. In order to be able to offer on the free market a constant generator output power in all operating regimes of pipeline compressor station it is interesting to examine sizing of the system in a way to provide constant ST power. With respect to quite low enthalpy of available flue gases in operating regimes with two gas turbines wherein is regime 10 characterized by three times lower enthalpy than the regime 1 it makes sense to consider sizing of the system in a way to run the ST only in regimes 1-5. So as to obtain electricity sellback in all operating regimes but with inducing lower rates of supplementary firing it is interesting to examine the strategy of maintaining the constant flue gas temperature at HRSG inlet. For the chosen strategy CC efficiency, ST power, SF fuel consumption and quantity of CO<sub>2</sub> emissions are calculated. Finally off-design sensitivity analysis is conducted.

### 8.1 System design and operating strategy at GT part load

So as to determine appropriate system design and operating strategy at GT part load appearing in operating regimes of compressor station enthalpy of flue gases is calculated according (8.1-1) with values given in Figure 8.1.

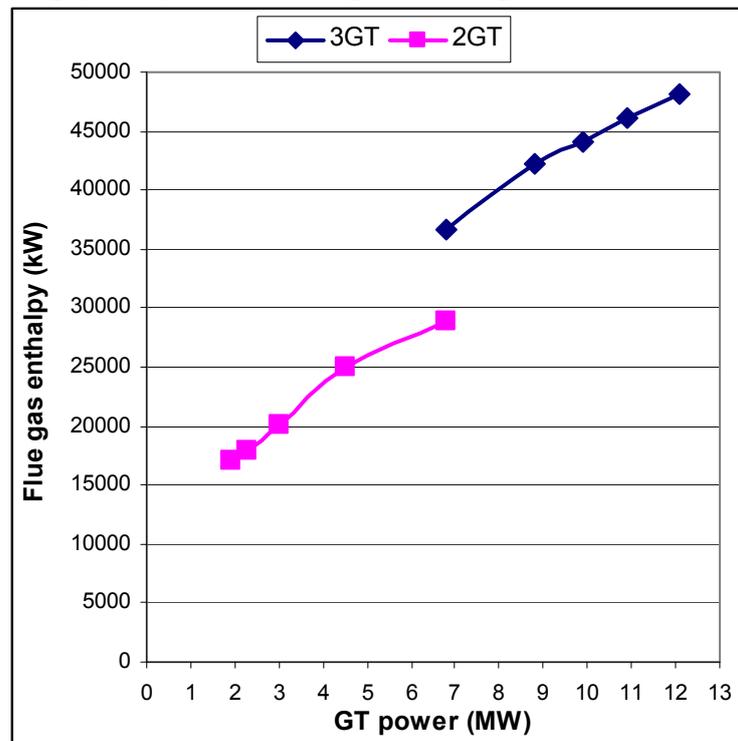


Figure 8.1: Flue gas enthalpy at GT outlet in compressor station operating regimes

$$H = n_{GT} m_{fg} h_{fg} \quad (8.1-1)$$

Therein  $n_{GT}$  means number of gas turbines in operation and  $m_{fg}$  means flue gas mass flow overtaken from Table 4-1.  $h_{fg}$  means specific enthalpy of flue gases calculated for the respective temperature from the h-t polynomial (8.1-2) which is constructed from exhaust gas composition given in Table 4-2 and specific enthalpies of constituent gases. Flue gas temperatures are obtained from Table 4-1.

$$h = 1.05E-13t^5 - 3.54E-10t^4 + 4.03E-07t^3 - 6.09E-05t^2 + 1.05E+00t - 1.3217E-02 \quad (8.1-2)$$

Upon consideration of available flue gas enthalpies per Figure 8.1 and thermodynamic principles of CC system outlined in chapter 5 following main design and operating strategies obtrude:

- Constant ST power in all regimes
- Constant ST power in regimes 1-5, ST out of operation in regimes 6-10
- Maintaining constant flue gas temperature

### **8.1.1 Constant ST power in all regimes**

In this case ST is sized for available flue gas enthalpy at design point. As GT load in operating regimes decreases and as a result of that flue gas enthalpy, supplementary firing is used for maintaining design ST power constant. However calculation of mass and energy balance [9] for such prediction shows that steam production at operating regimes with two gas turbines would not be possible due to too small temperature differences in economizer. Second restriction is quite high firing temperatures reaching 900°C necessitating employment of water cooled enclosure making the system more expensive. Third restriction is high fresh steam temperature which can not be held within the limits despite employment of three HPSH and two injection coolers.

### **8.1.2 Constant ST power in regimes 1-5, ST out of operation in regimes 6-10**

So as to overcome drawbacks of strategy explained in 8.1.1 it may be considered to install a flue gas bypass upstream the HRSG. Its function would be to vent flue gases into atmosphere as soon as their enthalpy falls beneath 35000kW. Knowing that compressor station demand is quite stochastic such prediction would place high requirements on control systems. So as to avoid jeopardizing of steam production in the steam drum and overheating of tubes it may be considered to install a steam bypass upstream the steam turbine. In this case steam would be produced in all operating regimes however in case of flue gas enthalpies below 35000kW steam would be vented into atmosphere or eventually utilized for district heating or space heating system.

### **8.1.3 Maintaining constant flue gas temperature**

With respect to drawbacks of strategies explained in 8.1.2 and 8.1.3 it makes sense to design and operate the system so that the flue gas temperature is kept constant at 500°C in all operating regimes. That means that the supplementary firing shall be used only for maintaining the flue gas temperature constant at 500°C.

## 8.2 Off-design results for constant flue gas temperature of 500°C

### 8.2.1 CC net efficiency

CC net efficiencies given in Table 7-23 are valid at the design point. However system CC efficiency in operating strategy explained per 8.1.3 is shown in Figure 8.2.

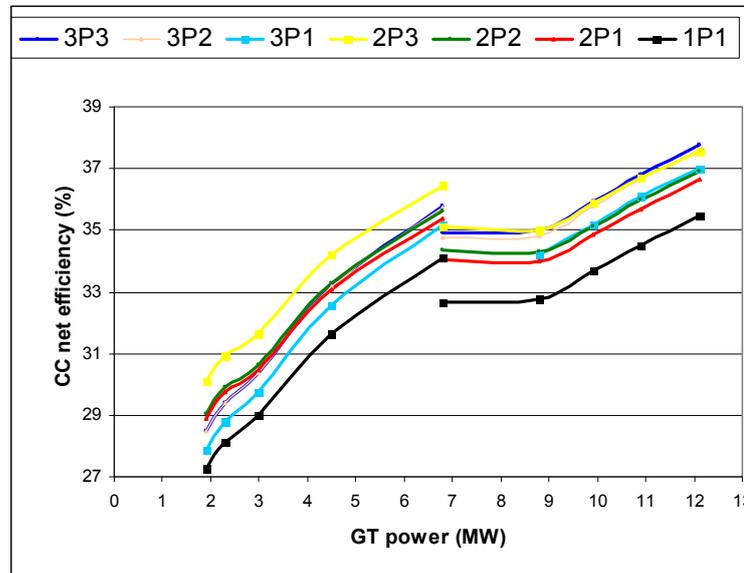


Figure 8.2: Net CC efficiency in operating regimes

Apparently 2P3 configuration has the highest CC efficiency in operating regimes with 2 gas turbines whilst the highest CC efficiency in operating regimes with 3 gas turbines has the 3P3 configuration. 1P configuration is characterized by the lowest CC efficiency.

### 8.2.2 Fuel consumption

Fuel consumption by means of supplementary firing in operating regimes is given in Figure 8.3.

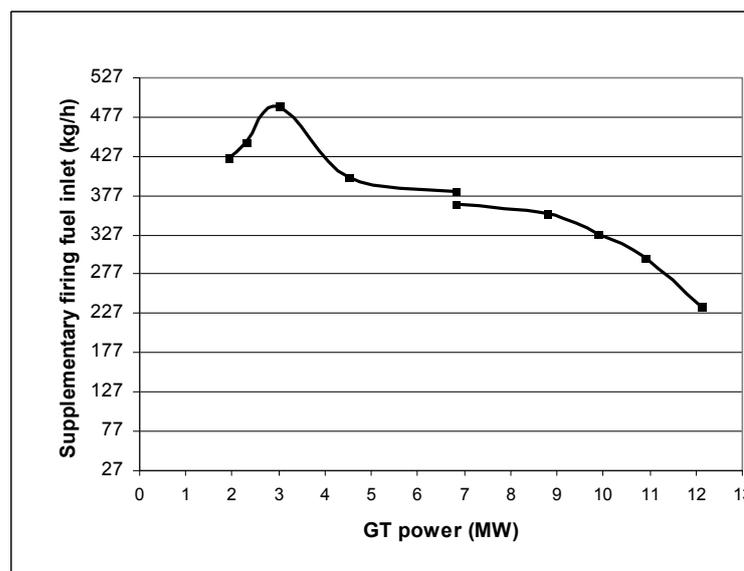


Figure 8.3: Fuel consumed by supplementary firing in operating regimes

Apparently SF fuel consumption of all configurations is the same as it is used for attaining constant flue gas temperature of 496°C and is not depending on HRSG configuration.

### 8.2.3 ST generator output power

Comparison of output power at ST generator in operating regimes is given in Figure 8.4.

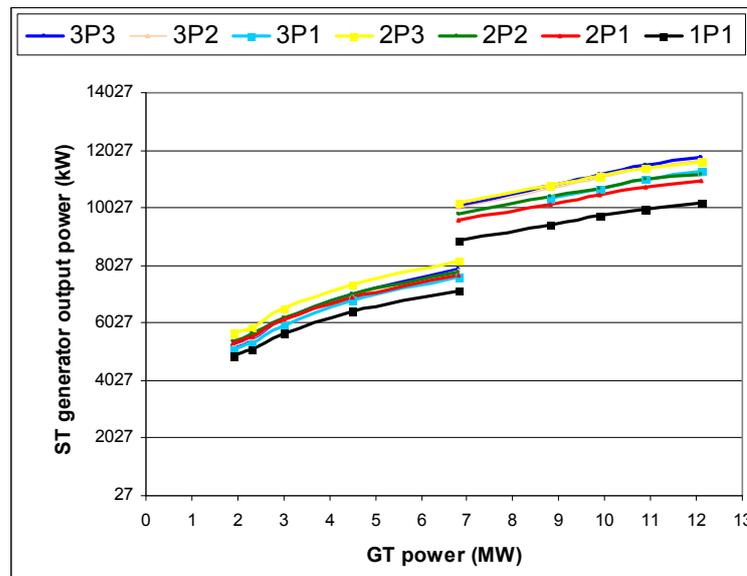


Figure 8.4: ST generator output power in operating regimes

Apparently 2P3 configuration has the highest output power in operating regimes with 2 gas turbines whilst the highest output power in operating regimes with 3 gas turbines has the 3P3 configuration what is consistent with result for the net CC efficiency.

### 8.2.4 Reduced CO<sub>2</sub> emissions

Reduced CO<sub>2</sub> emissions in operating regimes are given in Figure 8.5.

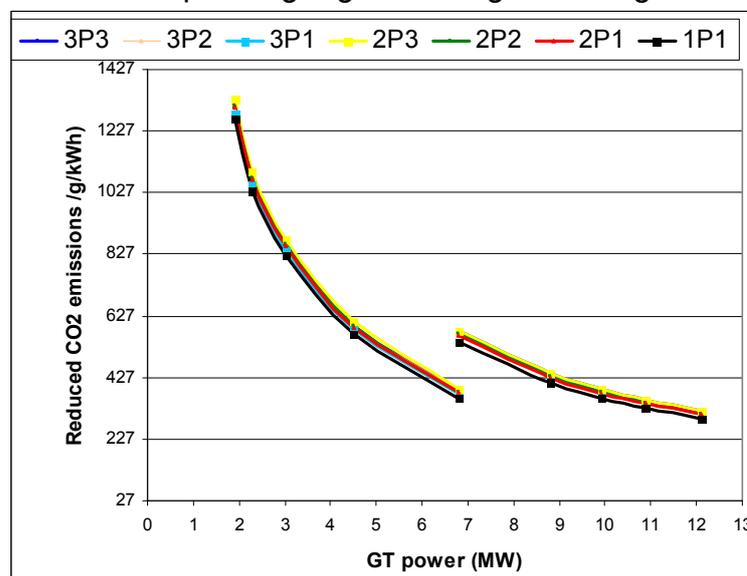


Figure 8.5: Reduced CO<sub>2</sub> emissions in operating regimes

Apparently system configuration has minor influence on the values of decreased CO<sub>2</sub> emissions.

### 8.3 Off-design sensitivity analysis

#### 8.3.1 Influence of ST operation mode on CC efficiency

As it is foreseen in 7.1.3 steam turbine operating mode is fix pressure. That means that there is constant pressure of fresh steam maintained in all ST operating regimes whilst power regulation is done by means of regulation of steam quantity. Since there is not a specific load at ST respectively at generator shaft predefined assumption is that all the steam produced in HRSG is fed into the ST. However it is interesting to examine steam turbine behavior in operating regimes of pipeline compressor station given in Table 4-1 if the turbine would operate in sliding pressure mode. Thereby are two sliding pressure modes considered; one without supplementary firing and the other with supplementary firing for attaining flue gas temperature of 500°C (as in the case of fix pressure). Influence of steam turbine operation in all three modes on CC efficiency is presented in Figure 8.6.

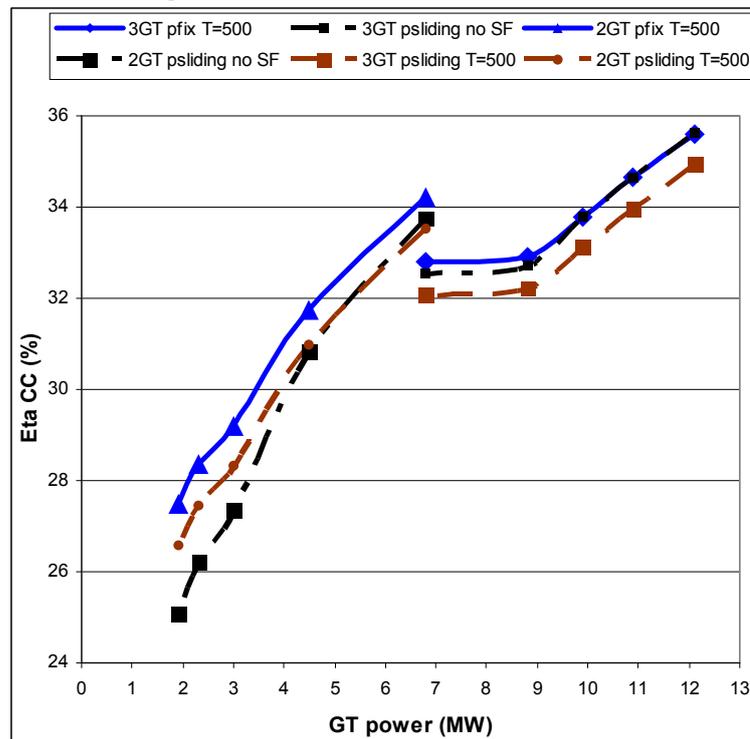


Figure 8.6: Influence of fix and sliding pressure mode on CC efficiency (1P)

Apparently sliding pressure operation is favorable only for the first two regimes providing minor difference in the order of magnitude of 0.1%. For other regimes preference belongs to fix pressure operating mode. Therefore fix pressure mode shall also be applied to other considered configurations. Steam inlet pressures valid for considered operation modes are given in Figure 8.7. It is interesting to note that the sliding pressure operation without supplementary firing causes 4-5 bars lower inlet pressures compared to the sliding pressure operation with flue gas temperature of 500°C.

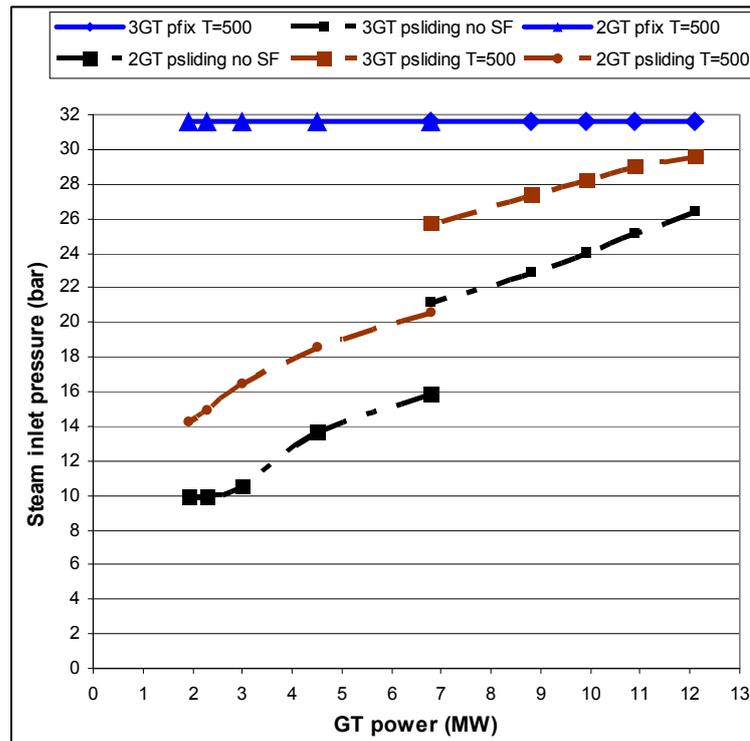


Figure 8.7: Steam inlet pressures in fix and sliding pressure mode (1P)

### 8.3.2 Influence of change in cooling water parameters on overall heat rate

Important information for process with employed condensing steam turbine is influence of the variations in cooling water temperature and quantity on the overall net heat rate, which is calculated from (8.5-1)[56].

$$HR_{\text{net}} = \frac{(\dot{m}_{\text{fuelGT}} + \dot{m}_{\text{fuelSF}}) \cdot \text{LHV}}{(W_{\text{GT1}} + W_{\text{GT2}} + W_{\text{GT3}} + W_{\text{ST}} - W_{\text{AUXILIARY}})} \times 3600 \left[ \frac{\text{kJ}}{\text{kWh}} \right] \quad (8.3-1)$$

Influence of cooling water inlet temperature change for the 1P configuration is shown in Figure 8.8 and Figure 8.9 shows influence of change in cooling water quantity.

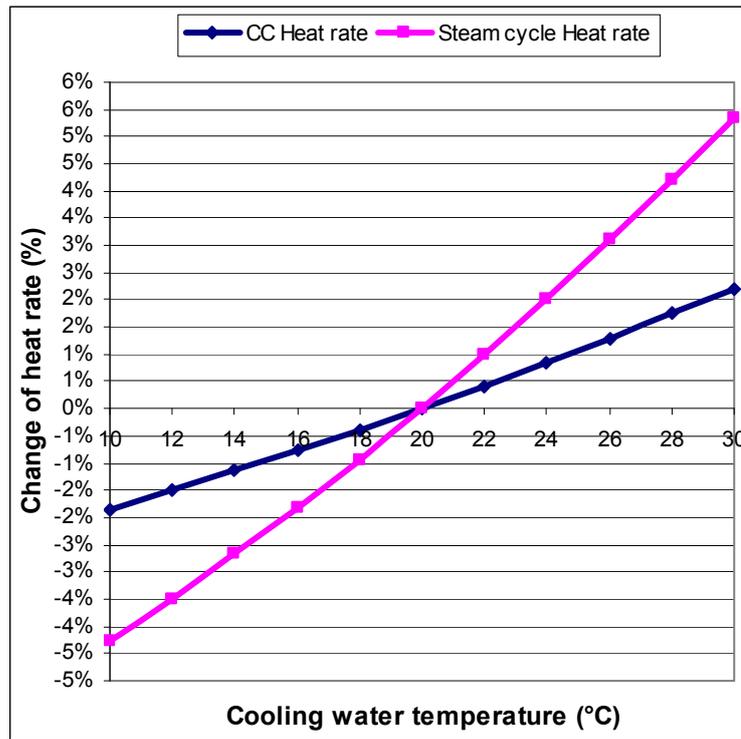


Figure 8.8: Influence of cooling water temperature change on CC and ST heat rate (1P)

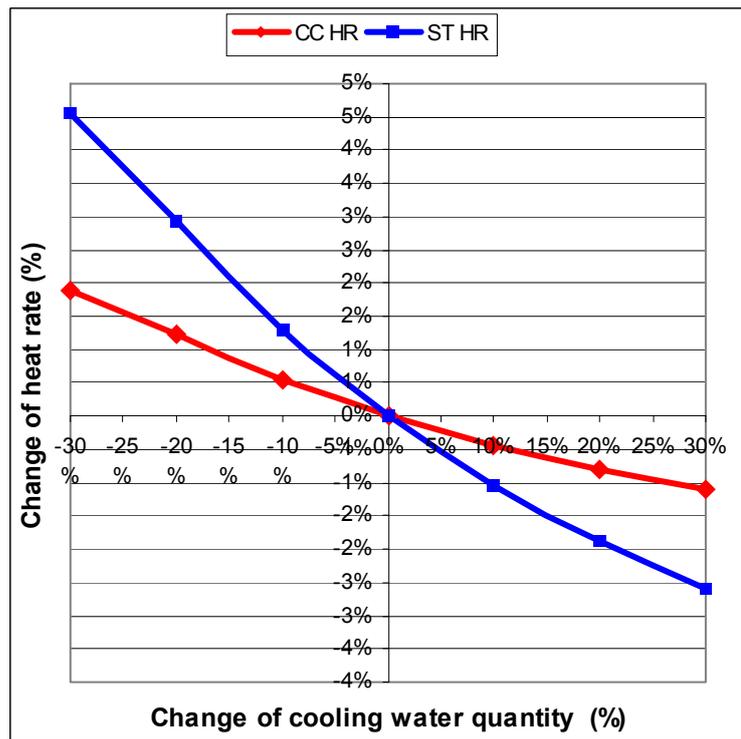


Figure 8.9: Influence of cooling water quantity change on CC and ST heat rate (1P)

## 9. Economic evaluation

### 9.1 Economic considerations of energy projects

#### 9.1.1 General characteristics of capital investments

Capital investment may be defined as strategic cash expenditure for certain object with long term effect. Decisions for such investments are usually made at higher levels within certain organizational hierarchy. [57]

There are three characteristics of capital investments to be concerned when performing life cycle cost analysis. First, capital investments usually require a relatively large initial cost. The initial cost may occur as a single expenditure such as purchasing a new heating system or occur over a period of several years such as designing and constructing a new building. It is not uncommon that the funds available for capital investment projects are limited. In other words, the sum of the initial costs of all the viable and attractive projects exceeds the total available funds. This creates a situation known as capital rationing which imposes special requirements on the investment analysis.

The second important characteristic of a capital investment is that the benefits (revenues or savings) resulting from the initial cost occur in the future, normally over a period of years. The period between the initial cost and the last future cash flow is the life cycle or life of the investment. Because of the fact that cash flows occur over the investment's life it is required to introduce time value of money concept border to properly evaluate investments. In the case of energy project respectively, power plant regular costs appearing each year are usually named operation and maintenance costs (O&M) and include all costs related to service and maintenance of equipment as also paying the personal.

The last important characteristic of capital investments is that they are relatively irreversible. Frequently, after the initial investment has been made, terminating or significantly altering the nature of a capital investment has substantial cost consequences; being usually negative. This is one of the reasons that capital investment decisions are usually evaluated at higher levels of the organizational hierarchy than operating expense decisions. [57]

In almost every case, the costs, which occur over the life of a capital investment, can be classified into one of the following categories:

- Initial Cost
- Annual Expenses and Revenues – O&M costs, fuel costs and periodic replacement and maintenance
- Salvage Value

Typical initial costs for various types of plants are given in Figure 9.1.

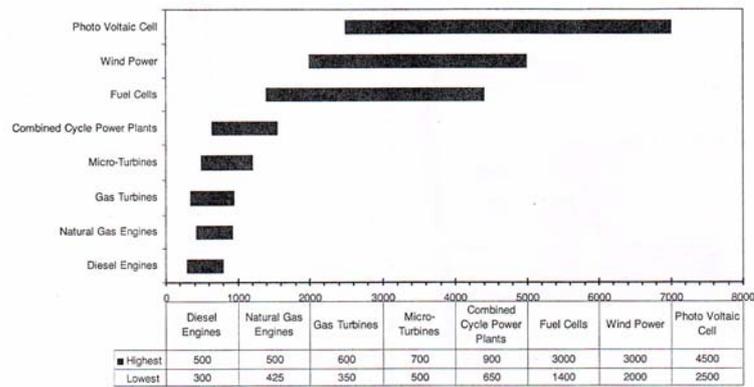


Figure 9.1: Typical first cost of various types of plants [14]

Typical costs for various fuels are given in Figure 9.2 whilst Figure 9.3 gives share of different costs in total life cycle cost of a thermal power plant.

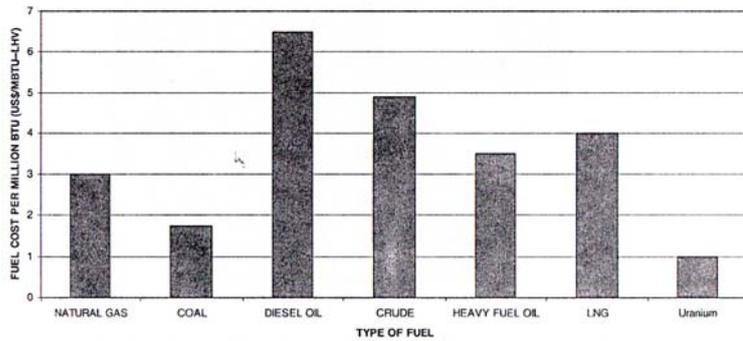


Figure 9.2: Prices comparison for various fuels [14]

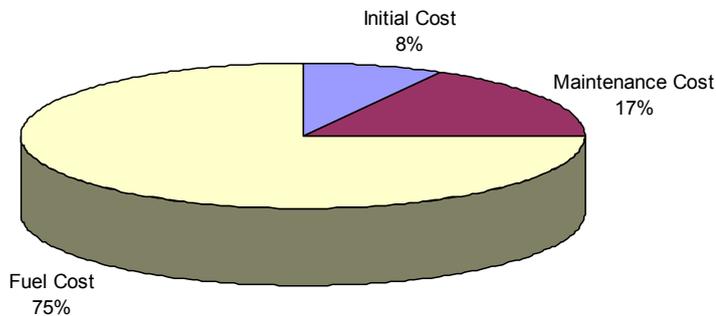


Figure 9.3: Share of various costs in total plant life cycle cost [14]

To simplify assumption, the cash flows, which occur during a year, are generally summed and regarded as a single end-of-year cash flow. While this approach does introduce some inaccuracy in the evaluation, it is generally not regarded as significant relative to the level of estimation associated with projecting future cash flows. Initial costs include all costs associated with preparing the investment for service. This includes purchase cost as well as installation and preparation costs. Initial costs are usually nonrecurring during the life of an investment. Annual expenses and revenues are the recurring costs and benefits generated throughout the life of the investment. Periodic replacement and maintenance costs are similar to annual expenses and revenues except that they do not (or are not expected to) occur annually. The salvage (or residual) value of an investment is the revenue (or expense) attributed to disposing of the investment at the end of its useful life.

### 9.1.2 Time value of money concepts

The time value of money is the premise that an investor prefers to receive a payment of a fixed amount of money today, rather than an equal amount in the future, all else being equal. [13]

In other words, the present value of a certain amount of money is greater than the present value of the right to receive the same amount of money at certain time in the future. This is because the amount a could be deposited in an interest-bearing bank account (or otherwise invested) from now to certain time in future in order to yield interest.

All of the standard calculations are based on the most basic formula, the present value of a future sum, "discounted" to a present value. For example, a sum of future value (FV) to be received in one year is discounted (at the appropriate rate of  $r$ ) to give a sum of present value (PV) at present.

So, as to convert cash from one time into another, any one of the following seven standard interest factors can be used [12]

- Single Payment Compound Amount (SPCA)
- Single Payment Present Worth (SPPW)
- Uniform Series Compound Amount (USCA)
- Sinking Fund Payment (SFP)
- Uniform Series Present Worth (USPW)
- Capital Recovery (CR)
- Gradient Present Worth (GPW)

### 9.1.3 Cash flow

Convenient way to display the benefits and costs associated with an investment is a cash flow diagram. By using it, the timing of the cash flows is more apparent and the chances of properly applying time value of money concepts are increased. [57]

[12] In evaluating energy investment opportunities, the timing of cash receipts and expenditures is essential. One of the following approaches can be used.

#### 9.1.3.1 End-of-period convention

All cash flows, which occur during any time interval, are assumed to occur at the end of the interval. This procedure simplifies computations and will not ordinarily introduce major errors in the results. Figure 9.4 illustrates this over a three-year period. [12]

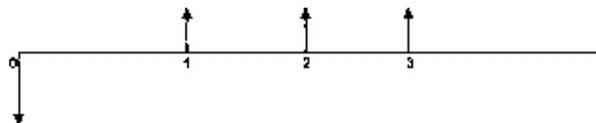


Figure 9.4: Cash flow diagram based on end-of-period convention [12]

#### 9.1.3.2 Mid-period convention

Cash flows can be assumed to occur at mid-period, as shown in the Figure 9.5 over a three-year period. [12]

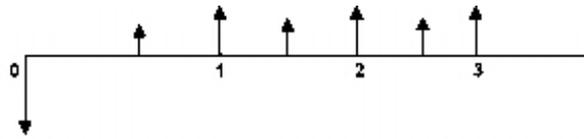


Figure 9.5: Cash flow diagram based on mid-period convention [12]

### 9.1.3.3 Continuous flow convention

Cash flows for any period are received at a continuous rate. Figure 9.6 shows this for eight cash flows per year, again over a three-year period. [12]

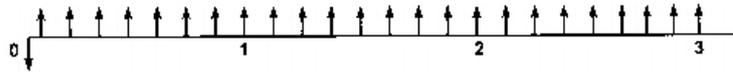


Figure 9.6: Cash flow diagram based on continuous flow convention [12]

## 9.1.4 Evaluation of proposals

Upon calculation or estimation of investment costs and benefits, attractiveness of various investment proposals under consideration can be evaluated. It is presumed that the risk or quality of all investment proposals under consideration does not differ from the risk of existing investment projects of the factory and that the acceptance of any one or a group of investment proposals does not change the relative business risk of the factory. The investment decision will be either to accept or to reject the proposal. [12]

### 9.1.4.1 Payback method

This method is based on a profitability criterion, represented by the time at the end of which the total revenue connected with the operation of an installation, after deduction of all outgoings including taxes, is equal to the amount of investment required for the purchase, construction and commissioning of the installation. It is the ratio of the initial investment over the annual cash inflows for the recovery period, i.e.:

$$\text{payback period} = \frac{\text{initial investment}}{\text{annual savings}} \text{ [years]}$$

The advantage of this method is that it is quite simple; if the payback period calculated is less than the maximum period acceptable by the company, the proposal is accepted; if not, it is rejected. The drawback is that it does not consider cash flows after the payback period; it ignores the magnitude or timing of cash flows during the payback period.

### 9.1.4.2 Average rate of return method

The average rate of return is an accounting method that represents the ratio of the average annual profits after taxes to the average investment in the project, i.e.

$$\text{average rate of return} = \frac{\text{annual profits after taxes}}{\text{average investment}} \text{ [%]}$$

Sometimes original investment is used rather than average investment.

The advantages of this method are that it is simple to use; it makes use of readily available accounting information; calculated figures can be compared readily with a

required or cut-off rate of return; and the proposal is acceptable if its average rate of return is higher than the cut-off rate. The drawbacks are that the method is based upon accounting income rather than upon cash inflows and outflows and the time value of money is ignored.

#### 9.1.4.3 Internal rate of return method (IRR)

This method takes account of both the magnitude and timing of expected cash flows in each period of a project life. The IRR for an investment proposal is the discount rate that equates the present value of the expected cash outflows with the present value of the expected inflows.

$$\text{initial investment} = \frac{\text{cashflow}_1}{(1+i)^1} + \frac{\text{cashflow}_2}{(1+i)^2} + \dots + \frac{\text{cashflow}_n}{(1+i)^n}$$

If IRR is greater than the cut-off or hurdle rate, respectively discount rate, the proposal is accepted; if not, the proposal is rejected.

#### 9.1.4.4 Net present value method

All cash flows are discounted to present value using the required rate of return.

$$\text{NPV} = -\text{initial investment} = \frac{\text{cashflow}_1}{(1+i)^1} + \frac{\text{cashflow}_2}{(1+i)^2} + \dots + \frac{\text{cashflow}_n}{(1+i)^n}$$

If the sum of these discounted cash flows is equal to or greater than zero, the proposal is accepted; if not, it is rejected.

#### 9.1.4.5 Profitability index

The profitability index or benefit/cost ratio of the project is the present value of future net cash flows divided by the initial cash outlay.

$$\text{profitability index} = \frac{\text{present value of net cash flows}}{\text{initial cash outlay}}$$

As long as the profitability index is equal to or greater than 1.00, the investment proposal is acceptable.

## 9.2 Economic evaluation of selected configurations in demand scenario

Exergo-economic optimization of HRSG and its operative parameters enables finding preferable configuration for maximizing flue gases heat recovery from certain size of HRSG respectively its surface areas. So in the terms of costs of new CC system it only provides information about HRSG surface cost.

Total CC investment cost comprises costs of ST with generator and gear, balance of plant equipment and HRSG which except heat exchanger surface costs also includes cost of components such as duct burner, framework, ducting, fans, riser and downcomer tubes, circulating pumps and fans. It is therefore crucial to calculate economic viability of investment into a new system for given pipeline compressor station demand using the investment cost given in Table 9-1 and economic assumptions given in Table 9-2.

Table 9-1: Structure of the investment cost

Configuration	1P	2P1	2P2	2P3	3P1	3P2	3P3	Ref.
ST generator output [kW]	10989	11820	12077	12514	12253	12726	12844	
ST specific cost [\$ /kW]	511							[10][23]
ST cost [\$]	\$5,615,177	\$6,040,203	\$6,171,548	\$6,394,886	\$6,261,243	\$6,503,096	\$6,563,460	
Generator, gear and lubricating oil equipment	\$2,250,000	\$2,250,000	\$2,250,000	\$2,250,000	\$2,250,000	\$2,250,000	\$2,250,000	[23]
Condenser size [m <sup>2</sup> ]	661	743	698	736	733	764	765	
Condenser specific cost [\$ /m <sup>2</sup> ]	2027							[23]
Water cooled condenser cost [\$]	\$1,339,830	\$1,505,167	\$1,414,259	\$1,492,259	\$1,485,694	\$1,549,032	\$1,549,707	
HRSG surfaces[\$]	1955398	2427076	2366647	2352113	2362172	2463025	2571792	
HRSG passiv part[\$]	\$9,431,100	\$10,479,000	\$10,479,000	\$10,479,000	\$11,526,900	\$11,526,900	\$11,526,900	[23][51]
HRSG cost [\$]	\$11,386,498	\$12,906,076	\$12,845,647	\$12,831,113	\$13,889,072	\$13,989,925	\$14,098,692	
Balance of plant equipment [\$]	\$375,000	\$375,000	\$375,000	\$375,000	\$375,000	\$375,000	\$375,000	[23][51]
Demi water preparation equipment [\$]	\$300,000	\$300,000	\$300,000	\$300,000	\$300,000	\$300,000	\$300,000	[23][51]
Electrical and controlling equipment [\$]	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	[23][51]
Civil work [\$]	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	[23][51]
Grid connection [\$]	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	[23][51]
Total EPC cost	\$25,766,505	\$27,876,446	\$27,856,453	\$28,143,258	\$29,061,008	\$29,467,053	\$29,636,859	
Additional (main contractor) cost								
Conceptual engineering (3%)	\$772,995	\$836,293	\$835,694	\$844,298	\$871,830	\$884,012	\$889,106	
Power and fuel for commissioning (0.5%)	\$128,833	\$139,382	\$139,282	\$140,716	\$145,305	\$147,335	\$148,184	
Insurance (0.5%)	\$128,833	\$139,382	\$139,282	\$140,716	\$145,305	\$147,335	\$148,184	
Verifications, risk analysis, environmental studies (1%)	\$257,665	\$278,764	\$278,565	\$281,433	\$290,610	\$294,671	\$296,369	
Total investment cost	\$27,054,830	\$29,270,268	\$29,249,276	\$29,550,421	\$30,514,059	\$30,940,406	\$31,118,701	

Table 9-2: Reference values for economic calculation

Data	Value	Unit	Reference
Project lifetime (n)	20	Years	
Load factor	90	%	
Revision cost (% of investment cost)	5	%	
Revision frequency	10	Years	
O&M cost (% of investment cost)	1	%	
Fuel price	228	\$/t	[58]
Discount rate (i)	10	%	
Electricity sellback price	5	\$/kWh	[50]
CO <sub>2</sub> emission certificate	30	\$/tCO <sub>2</sub>	[23]

So as to compare the proposed configurations in the terms of economic viability according algorithm explained in paragraph 9.1 with assumptions given in Table 9-1 and Table 9-2 cost of electricity, internal rate of return and payback period shall be calculated for demand scenario defined in 4.3.

### 9.2.1 Cost of electricity

Cost of electricity is calculated from (9.2-1).

$$E = \frac{CR}{T} \quad (9.2-1)$$

Wherein CR means capital recovery and T means number of operating hours per year. Thereby is CR calculated from the term (9.2-2).

$$CR = (PV(\text{inv. cost}) + PV(\text{rev. cost}) + PV(\text{O \& M cost}) + PV(\text{fuel cost})) * \frac{i(1+i)^n}{(1+i)^n - 1} \quad (9.2-2)$$

Wherein PV means present value, n is number of years and i means the discount rate. Complete algorithm used for calculating the cost of electricity is given in Enclosure 2.

Table 9-3: Cost of electricity

Configuration	1P	2P1	2P2	2P3	3P1	3P2	3P3
El. production cost [\$/kWh]	4.91	4.87	4.77	4.64	4.87	4.74	4.72
CO <sub>2</sub> reduction [\$/kWh]	2.15	2.01	1.97	1.90	2.00	1.94	1.93
Net electricity cost [\$/kWh]	2.76	2.86	2.79	2.74	2.87	2.80	2.79

As it is apparent form Table 9-3 configuration 2P3 offers the cheapest electricity.

### 9.2.2 Internal rate of return

As it is apparent from Table 9-4 all configurations under assumed values are viable since IRR values are greatest than assumed discount rate of 10%. Most viable is configuration 2P3 as it provides the highest IRR. Furthermore it is interesting to note that viability limit is at the natural gas price of 387\$/t.

Table 9-4: Internal rates of return

Configuration	1P	2P1	2P2	2P3	3P1	3P2	3P3
IRR [%]	12.45	12.20	12.50	12.96	11.49	11.74	11.74

### 9.2.3 Payback period

As it is apparent from Table 9-5 configuration 2P3 also offers the shortest payback period.

Table 9-5: Payback period

Configuration	1P	2P1	2P2	2P3	3P1	3P2	3P3
Payback period [years]	7.12	7.23	7.10	6.91	7.55	7.43	7.43

### 9.2.4 Comparison and selection

Apparently configuration 2P3 is preferable since it provides the cheapest electricity, the lowest IRR and the shortest payback period. Its cash flow diagram is given in Figure 9.7

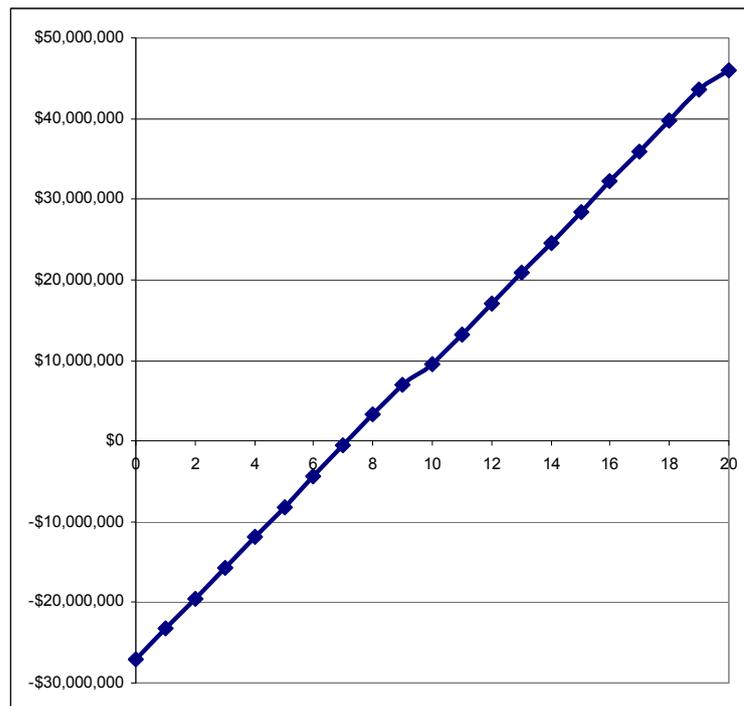


Figure 9.7: Cumulative cash flow configuration 1P

The cash flow show in Figure 9.7 is certainly unfavorable offering relative long payback of 6.91 years. Change of the curve inclination between 9 and 10 years and between 19 and 20 years refers to the revision cost. Causes and guidelines for more effective and more viable waste heat utilization are outlined in conclusion.

## 10. Economic sensitivity analysis

In order to simplify the economic evaluation carried out in this work, certain parameters such as inflation, taxes and depreciation are not taken into account. Among assumptions given per Table 9-2, some of them are certainly exposed to variations either due to global political and economic situation or because of local contracting conditions between the client and corresponding institution (bank, Electricity Company etc.).

One of the parameter that is subject to global and regional political and economic situation is definitely natural gas price, which is already today very much varying between different regions and countries, as shown in Figure 10.1. [21] Its future price can only roughly be estimated since it is a matter of security of supply, global market trends and outstanding natural gas reserves, which are shown in Figure 10.2. [21]

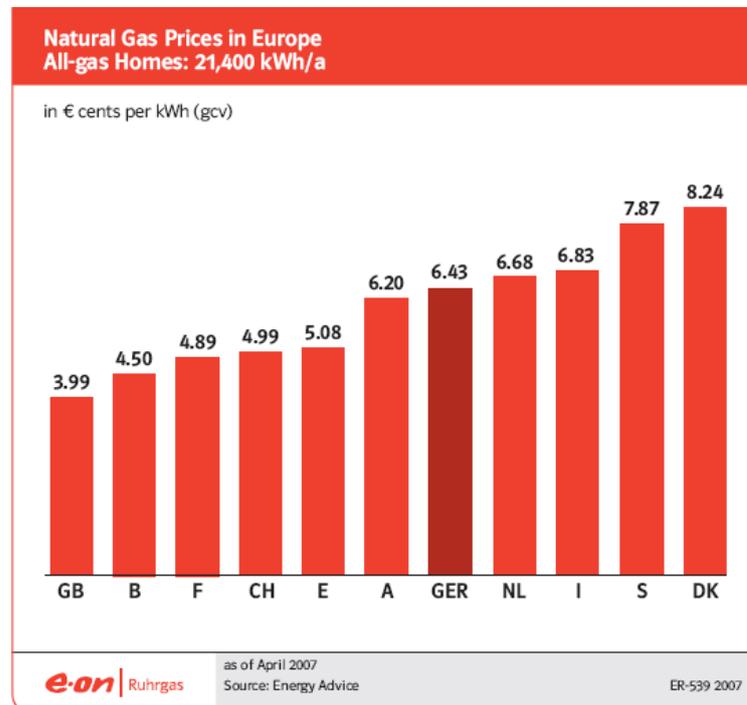


Figure 10.1: Natural gas prices in Europe (2007) [21]



Figure 10.2: World reserves of natural gas (2006) [21]

### 10.1 Influence of variable fuel and CO<sub>2</sub> emission certificate prices

Assuming that the total investment cost, followed by revision cost and O&M cost which are linked to it, shall not vary significantly it is interesting to examine the dependency of net electricity production cost on various fuel prices and various prices of CO<sub>2</sub> emission certificates. Such a diagram is shown for the basic configuration 1P in Figure 10.3.

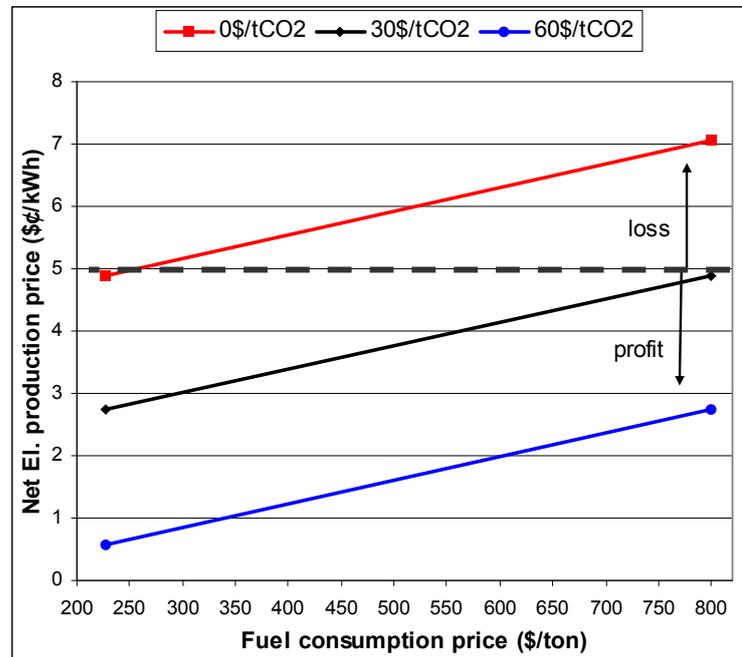


Figure 10.3: Electricity production cost vs. natural gas price with CO<sub>2</sub> price as parameter

Figure 10.3 outlines dependence of electricity production cost on fuel price and CO<sub>2</sub> emission certificate price. Graph has to be understood qualitatively since real viability index is IRR shown in Table 9-4. However it is apparent that the investment in waste heat recovery projects is viable even under increased fuel prices if the emission certificate price would increase.

Furthermore it is interesting to examine influence of fuel annual escalation rate on project viability what is examined for fuel annual escalation rates from 0-10%.

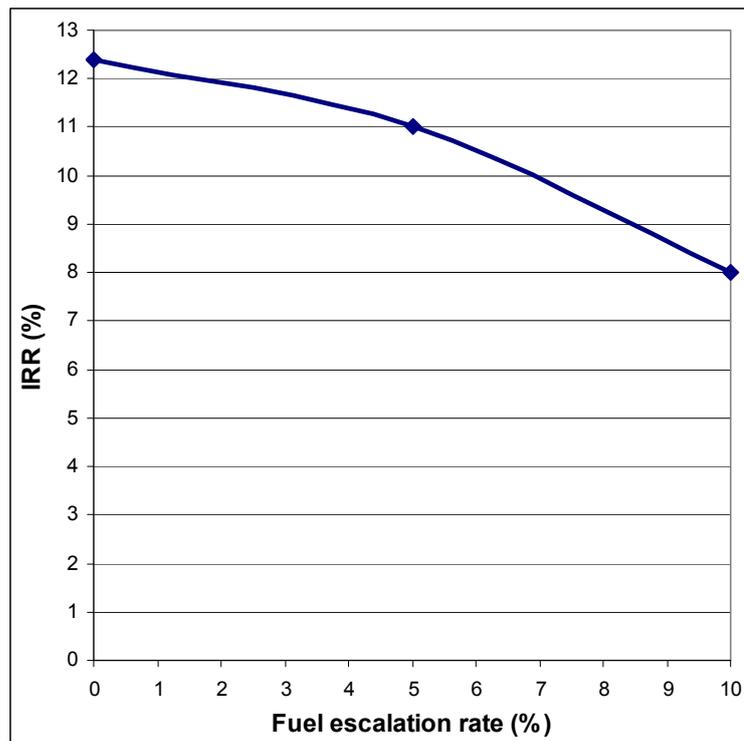


Figure 10.4: Influence of fuel escalation rate on project viability

As it is apparent from Figure 10.4 under annual fuel escalation rates higher than 7.2% considered waste heat recovery project is not any more viable.

## 10.2 Influence of supplementary firing on project viability

### 10.2.1 Cost of electricity

Cost of electricity is calculated according algorithm given in Enclosure 2.

Table 10-1: Cost of electricity for fired HRSG

Configuration	1P	2P1	3P1
Electricity production cost [\$/kWh]	4.9	5.6	5.7
Compensation for CO <sub>2</sub> reduction [\$/kWh]	0.9	1.8	1.8
Net electricity cost [\$/kWh]	4.0	3.8	3.9

As it is apparent from Table 10-1 configuration 2P1 provides the cheapest electricity what is consistent with the result for the unfired mode where the 2P3 configuration provides the cheapest electricity. Reason is in a large system whereby the revenues can not payoff investment and other costs in a viable way.

### 10.2.2 Internal rate of return

As it is apparent from Table 10-2 none of considered configurations are viable since IRR values are quite below the assumed discount rate of 10%.

Table 10-2: Internal rates of return for fired HRSG

Configuration	1P	2P1	3P1
IRR [%]	5	8	7

### 10.2.3 Payback period

As it is apparent from Table 10-3 configuration 2P1 offers the shortest payback period which is however too long to be considered for a potential investor as are also the payback periods for the other two configurations.

*Table 10-3: Payback period for fired HRSG*

Configuration	1P	2P1	3P1
Payback period [years]	12.5	10.2	10.5

## 11. Conclusion

Shortage in reserves of most important fossil fuels of present; oil and natural gas followed by environmental concerns caused by excessive firing of these fuels bring to stronger requirements on energy efficiency issues in all kinds of power production and consume.

Pipeline compressor stations present only one part of turbomachinery whose efficiency might be increased by utilizing the waste heat from the gas turbines that are used as prime movers. There was not much effort made in investigation and assessment of such possibilities in the recent history. Firstly it can be attributed to the fact that due to sufficient reserves and acceptable prices of natural gas it was not an imperative to invest time and money in such attempts. On the other hand, modern methods for CC plant optimization and development of computer programs used for CC plant calculation and optimization took part in a larger extent in the last 10 years.

In closer history thermodynamic calculations of CC power plants have been conducted at base load (also called the design point) for the e.g. summer regime followed by calculation at only one additional operating point (e.g. winter regime) using empiric correction factors for taking into account different ambient conditions and rate of heat transfer in HRSG. Today's higher requirements on efficient CC operation followed by respect to economic aspects make the calculation of the CC system in all operating regimes more complex wherefore is the calculation per hand not any more reliable. Development of computer programs used for heat and mass balance calculations enabled more reliable prediction of system behavior at various part load and ambient conditions. By that it is possible to make more accurate estimations of consumed fuel, produced power and optimally required HRSG surface areas followed by optimal HRSG operative parameters. Usage of such computer programs enables determination of best configuration for certain application and also provides the possibility to test the newly developed equipment for the certain application.

In this work an exergo-economic HRSG optimization based on minimization of the objective function comprising the HRSG cost of exergy losses and HRSG surface cost is conducted. By means of this method several 1P, 2P and 3P configurations are optimized for finding HRSG operative parameters providing the lowest total annualized HRSG cost. Employed optimization method [48][49] and obtained results manifest meaning of the thermo-economic optimization whereby more expensive HRSG surfaces cause less heat transfer from flue gases to water(steam) respectively increase of pinch point and decrease of fresh steam pressure resulting in a smaller surface size. However initial optimization results show insensibility of the objective function on the cost of surfaces providing zero pinch point. Such result demonstrates low surface costs compared to the cost of exergy losses for given parameters of available flue gases and assumed factors of the objective function. Incorporation of exergy of inlet feed water and outlet gas (which are neglected by authors of the selected optimization method) result in max. pinch point and pressure values striving to reach the critical state. Such result of pressure value is however not feasible for design of the industrial steam turbine. Furthermore tendency of reaching upper limit of pinch point shows inadequacy of this approach. Sensitivity analysis of extended objective function shows decrease of pinch point at lower values of exp however the fresh steam pressure remains unchanged wherefore is application of extended objective function not adequate for analyzed waste heat recovery system.

So as to obtain positive but realistic pinch point values surface exponent  $\exp$  is introduced into optimization based on reduced objective function as proposed by authors of [48][49]. So as to take into account higher cost of 2P and 3P HRSG due to more complex HRSG passive part, a system complexity factor  $f_c$  is introduced. Idea is to find the value of the surface exponent obtaining minimal feasible pinch point provided by the heat recovery steam generator manufacturer. Referent minimal feasible pinch point is selected for unfired case. Proper value of the surface exponent is determined on the basic single pressure configuration. Upon defining optimization criteria this value is further used for optimization of other configurations.

Existence of thermoeconomic criteria means that optimal configuration shall not provide maximal thermodynamic efficiency. Calculation results show that the configuration 3P3 offers the highest CC efficiency whilst the configuration 2P3 offers the lowest total annualized HRSG cost. Predefined single pressure, dual pressure and triple pressure system configurations are further examined in operating regimes of the pipeline compressor station. Thereby are cost of electricity, cash flow and internal rate of return calculated. Comparison of configurations is based on diagram presenting electricity cost and net system efficiency. Pareto front with respect to both criteria; maximal system efficiency and minimal cost of electricity is constructed. Thereof 2P3 and 3P3 configurations are the closest to the Pareto front whereby the first is characterized by the lowest cost of electricity and the latter by the highest system efficiency.

Economic evaluation considers annual respectively lifetime operation of a new heat recovery system in operating regimes of pipeline compressor station so as to examine if the configuration costs can be paid off with obtainable revenues and in which payback period. Economic results show that all configurations are viable whereby the preference in the terms of viability belongs to configuration 2P3.

There are three major reasons influencing the relative low efficiency of the heat recovery system followed by unfavorable payoff conditions.

Firstly, gas turbines have relative low efficiency in particular at the part load. THM1203 gas turbines employed in a considered pipeline compressor station are ca. 15 years old and do not offer very high efficiency. New gas turbine of the same turbine family, which was developed recently, has ca. 8% higher efficiency at the base load. It is characterized by slightly smaller exhaust gas temperature but has ca. 40% higher exhaust gas flow at base load. [23]

Secondly, demanded mechanical power at pipeline compressor shafts is quite oscillating. LDC for the actual demand scenario shows that during ca. 4000h pipeline compressor shaft is operating at a load below 50% of nominal. Furthermore, mode of operation in which there are always at least two gas turbines in operation necessitates operation of gas turbines at quite low loads. However such an operating mode is selected for security reasons, respectively to enable one turbine to overtake the demand if the other is out of operation due to any reason. From Table 4-1 and Figure 4.8 results that such demand and operating strategy cause the gas turbines to operate between 18-88% of the nominal power where the gas turbine efficiency reaches values even 45% below nominal in some regimes.

Thirdly, investment cost is quite high for plants of this size. Although we are dealing with a CC system respectively its particular kind MDCC system in the terms of

upgraded equipment it is actually a ST plant with a HRSG instead a conventional fossil boiler.

Last but not least, it is not very viable to produce only electricity from such an upgraded system. However for gaining higher efficiencies a significant heating and/or cooling load should be present at location whereby heating or cooling energy could be produced simultaneously with production of electricity. [59][10][61] Reason for that lies in the fact that the condensing steam turbine employed in proposed configurations characterizes quite high heat loss resulting from heat transfer of condensing steam to the cooling water equaling ca. 45% [59][60]. In case of cogeneration cycle exhaust steam from the ST is supplied to consumers where it is fully utilized. This is accomplished by employing a backpressure steam turbine (or alternatively extraction condensing steam turbine) providing overall thermal efficiencies up to 90%.[8] Furthermore, in case of a cooling demand a trigeneration system is applied. It typically comprises a cogeneration system equipped with an absorption chiller (or other equipment) producing the cooling energy from the LP steam which is used for process or space cooling.

Based on result presented in this work guidelines for efficient waste heat utilization from pipeline compressor stations might be emphasized into:

- Focus should be on compressor stations providing considerable enthalpy of available flue gases
- Annual demand and operating mode should be such that gas turbines in operation do not run below 50% load
- Preference should be given to the sites where exists a significant demand for process heating, district heating or district cooling

Further efforts for research of waste heat utilization from pipeline compressor stations should also be directed in feasibility and viability examination of other technologies for waste heat utilization. Predictable rising demand for natural gas suggests incorporation of a gas pipeline model aiming to assess influence of variations in natural gas supply on optimal gas turbine operation. Prediction and modeling of available heating or cooling demand at location would make the results even more reliable.

## 12. References

- [1] NaturalGas, <http://www.naturalgas.org/>, as accessed 5.7.2010
- [2] The Interstate Natural Gas Association of America (INGAA), <http://www.ingaa.org/>, as accessed 5.7.2010
- [3] A. J. Smalley and others, Technologies to Enhance the Operation of Existing Natural Gas Compression Infrastructure, Final report, SwRI® Project No. 18.06223, DOE Award No. DE-FC26-02NT41646 (2006)
- [4] D.A. Popov, Advanced gas turbine plant cycles for more efficient natural gas transmission, Department of Thermal and Nuclear Power Engineering, Technical University of Sofia, Bulgaria
- [5] Branko Staniša, Characteristics of Modern Combined gas-steam Turbine Power Plants, *Strojarstvo* 36 (3,4) (2004), 169-182
- [6] Rolf Kehlhofer, Combined-cycle gas and steam turbine power plants, PennWell (1997)
- [7] Siemens Power Generation - Automation Systems Combined Cycle Power Facility, [www.siemens.com](http://www.siemens.com), as accessed 5.7.2010
- [8] Š. Hadžiefendić and others: Cogeneration and alternative technologies in electricity production, Bosna-S Oil Services Company, Konzalting Biro
- [9] GE Energy, GateCycle Analysis Program, [http://www.gepower.com/prod\\_serv/products/oc/en/opt\\_diagsw/gatecycle1.htm](http://www.gepower.com/prod_serv/products/oc/en/opt_diagsw/gatecycle1.htm), as accessed 5.7.2010
- [10] Paoli, Višković, Economy and politic of electricity production, Kigen (2007)
- [11] Lecture materials, International MScSEE, [www.fsb.hr/see](http://www.fsb.hr/see), FMENA, University of Zagreb, as accessed 5.7.2010
- [12] N. Duić (2002), Digital Tutorial, <http://powerlab.fsb.hr/OsnoveEnergetike/udzbenik/>, as accessed 5.7.2010
- [13] Wikipedia, the free encyclopedia, <http://en.wikipedia.org>, as accessed 5.7.2010
- [14] Dr. Meherwan P. Boyce , Handbook for cogeneration and combined cycle power plants, ASME press (2002)
- [15] Dr. Boyun Guo and Dr. Ali Ghalambor, Natural gas engineering handbook, University of Louisiana at Lafayette, United States of America (2005)
- [16] Encyclopedia The Free Dictionary, <http://encyclopedia.thefreedictionary.com/>, as accessed 5.7.2010

- [17] John Goerten, Emmanuel Clement, Statistics in focus, Environment and energy 10/2006, Gas prices for EU households and industrial consumers on 1<sup>st</sup> January 2006 (2006)
- [18] Gazprom in Figures 2002-2006, [http://www.gazprom.com/documents/Stat\\_Report\\_Eng.pdf](http://www.gazprom.com/documents/Stat_Report_Eng.pdf), as accessed 5.7.2010
- [19] International Energy Agency, [www.iea.org](http://www.iea.org), as accessed 5.7.2010
- [20] World Energy Council , <http://www.worldenergy.org/>, as accessed 5.7.2010
- [21] RUHRGAS AG, [www.eon-ruhrgas.com/](http://www.eon-ruhrgas.com/), as accessed 5.7.2010
- [22] Bernd Gericke, Integrated waste heat utilization in pipeline compressor stations, Part 1: Gas Turbines from VGB Power Tech 2/2002 page 64-69, Part 2: Steam Turbines and Steam Generator Types from VGB Power Tech 3/2002 page 47-56
- [23] MAN TURBO, <http://www.manturbo.com>, as accessed 5.7.2010
- [24] Philip P. Walsh, Paul Fletcher, Gas Turbine Performance, 2<sup>nd</sup> edition, Blackwell Science (2004)
- [25] Group of authors, Praktičar 3, 1<sup>st</sup> edition, Strojarstvo 2, Školska Knjiga Zagreb (1973)
- [26] Lars Malmrup, The Industrial Gas Turbine, the European Perspective, EuMIGT (2003)
- [27] O.O.Mil'man, I.V. Belousenko, Heat-recovery power units for the GAZPROM Company, Thermal Engineering, vol. 48 , No.3 (2001), 239-243
- [28] OMV Gas & Power GmbH – Power projects, [www.omv.com](http://www.omv.com), as accessed 5.7.2010
- [29] Transitgas AG - Transport System - Waste Heat Recovery Units, [http://www.transitgas.ch/en/transp\\_heat\\_recovery.htm](http://www.transitgas.ch/en/transp_heat_recovery.htm), as accessed 5.7.2010
- [30] Transportadora de Gas del Sur (TGS) home page, [www.tgs.com.ar](http://www.tgs.com.ar), as accessed 5.7.2010
- [31] Surface meteorology and Solar Energy, A renewable energy resource web site (release 6.0) sponsored by NASA's Earth Science Enterprise Program, <http://eosweb.larc.nasa.gov/sse/>, as accessed 5.7.2010
- [32] V. Ganapathy, Heat-Recovery Steam Generators, Understand the Basics, Chemical engineering progress (1996)

- [33] Bernd Gericke, Combined Cycle Plants for Industry and Communities, Part 1: Supplementary Fired Single-Pressure Process from VGB Power Tech 10/1997 page 715-721, Part 2: Single-pressure Process with supplemental Firing and Recuperator and Multi-pressure Process with supplemental Firing and IP-Superheater from VGB Power Tech 11/1997 page 912-919 (1997)
- [34] Neil Petchers, Combined heating, cooling & power handbook: technologies & applications, an integrated approach to energy resource optimization, Fairmont Press (2003)
- [35] Alstom Power home page, [www.alstom.com](http://www.alstom.com), as accessed 5.7.2010
- [36] Fred Starr, Background to the design of HRSG systems and implications for CCGT plant cycling, OMMI, Vol.2, Issue 1 (2003)
- [37] Jeff Oaiber, Fluid dynamics of the HRSG gas side, Power Platt (2006),
- [38] Combined cycle plant arrangements, Ecoling Partner AG, [http://www.ecoling.ch/englisch/thermal\\_power\\_plant.htm](http://www.ecoling.ch/englisch/thermal_power_plant.htm), as accessed 5.7.2010
- [39] Wayne C. Turner and Steve Doty, 6<sup>th</sup> edition, Energy management handbook, 6th Edition, Fairmont Press (2007)
- [40] Truls Gundersen, Pinch point method, Master Degree Program; Energy Efficiency and Environment MSC-6: Energy Management for integrated Energy Systems, Part A: Process Integration Methodology and the Systems Level (2004)
- [41] Burer et al., Multi-criteria optimization of a district cogeneration plant integrating a solid oxide fuel cell-gas turbine combined cycle, heat pumps and Chillers, Fuel and Energy Abstracts, Volume 44, Issue 5 (2003), Page 302
- [42] Yongjun Zhao and others, Modeling and cost optimization of CC HRSG systems, Proceedings of ASME Turbo Expo, Power of Land, Sea, and Air, Atlanta, Georgia, USA (2003)
- [43] B.V. Reddy and others, Second law analysis of a waste heat recovery steam Generator, International Journal of Heat and Mass Transfer 45 (2002), 1807-1814
- [44] P. K. Nag and S. De\*, Design and operation of heat recovery steam generator with minimum irreversibility, Applied Thermal Engineering Vol. 17, No. 4 (1997), 385-391
- [45] Victor-Eduard Cenus and others, Exergetic Optimization of the HRSG by Imposing the Total Heat Transfer Area, Int.J. Thermodynamics, Vol.7, No.3 (2004), 149-156
- [46] Alessandro Franco, Nicola Giannini, A general method for the optimum design of heat recovery steam generators, Energy 31 (2006) 3342-3361

- [47] A. Ragland, W. Stenzel, Combined Cycle Heat Recovery Optimization, Proceedings of 2000 International Joint Power Generation Conference Miami Beach, Florida (2000)
- [48] C. Casarosa, F. Donatini, A. Franco, Thermo-economic optimization of heat recovery steam generators operating parameters for combined plants, *Energy*, Volume 29, Issue 3 (2004), 389-414
- [49] A. Franco, A. Russo, Combined cycle plant efficiency increase based on the optimization of the heat recovery steam generator operating parameters, *International Journal of Thermal Sciences* 41 (2002), 843–859
- [50] Ente regulador de los servicios publicos, Anuario estadístico del sector eléctrico, Precios e ingresos, (2001)
- [51] Josef Bertsch Gesellschaft m.b.H. & Co., Herrengasse 23, 6700 Bludenz, Austria, [www.bertsch.at](http://www.bertsch.at), as accessed 5.7.2010
- [52] V. Ganapathy, Simplify heat recovery steam generator evaluation, *Hydrocarbon processing* (1990)
- [53] United Nations Framework Convention on Climate Change [http://unfccc.int/kyoto\\_protocol/background/items/2879.php](http://unfccc.int/kyoto_protocol/background/items/2879.php), as accessed 5.7.2010
- [54] European Commission, <http://ec.europa.eu/environment/climat/kyoto.htm>, as accessed 5.7.2010
- [55] Flavio Nicolin, Vittorio Verda, Lifetime optimization of a molten carbonate fuel cell power system coupled with hydrogen production, presented at the 5<sup>th</sup> Dubrovnik Conference on Sustainable Development of Energy, Water and Environment Systems (2009)
- [56] F.R.P. Arrieta, E.E.S. Lora, Influence of ambient temperature on combined-cycle power-plant performance, Technical paper, *Applied Energy* 80 (2005)
- [57] Albert Thuman and Eric A. Woodruff, Handbook of financing energy projects, Fairmont Press (2005)
- [58] Energy Information administrator, Argentina – Natural Gas, <http://www.eia.doe.gov/emeu/cabs/Argentina/NaturalGas.html>, as accessed 5.7.2010
- [59] Zvonko Elčić, Steam Turbines , ABB, Karlovac (1995)
- [60] Bojan Kraut, 9<sup>th</sup> edition, Mechanical engineering handbook, Tehnička Knjiga Zagreb, (1987)
- [61] Bernd Gericke (2003), The Use of 10-25 MW Class Topping Cycle Gas Turbines to increase CHP Plants Efficiency, VDI-Berichte No. 1769
- [62] Zvonimir Guzović (1994): Design and strength calculation of gas and steam turbine segments, FMENA Zagreb

Desuperheater (Temperature mixer):

$$(\dot{m}_{\text{HPT}} - w_1)h_{s3} + w_1 h_{s6} = \dot{m}_{\text{HPT}} h_{s1} \quad (\text{E1-1})$$

HP Superheater 1:

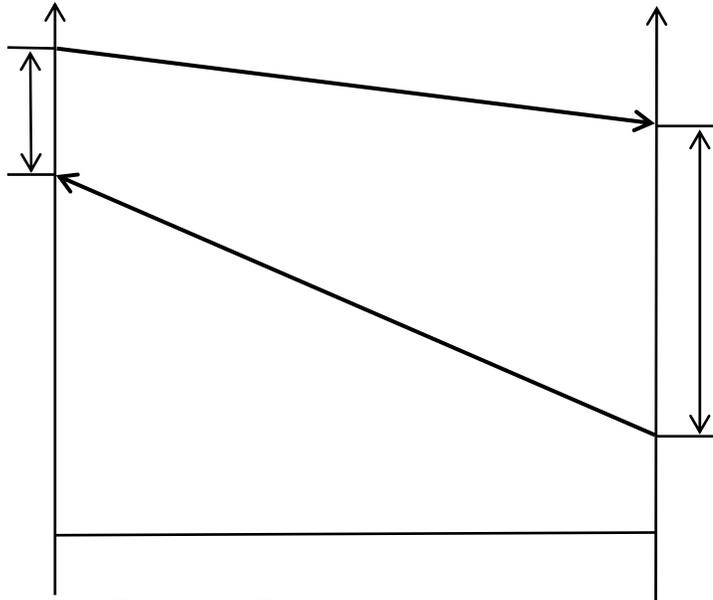


Figure 1: HPSH1 temperature profile

$$\dot{Q}_{\text{HPSH1}} = \eta_r \dot{m}_{\text{fg}} (h_{g3} - h_{g4}) \quad (\text{E1-2})$$

$$\dot{Q}_{\text{HPSH1}} = (\dot{m}_{\text{HPT}} - w_1)(h_{s3} - h_{s4}) \quad (\text{E1-3})$$

$$\dot{Q}_{\text{HPSH1}} = k_{\text{HPSH1}} A_{\text{HPSH1}} dt_m \quad (\text{E1-4})$$

HP Evaporator:

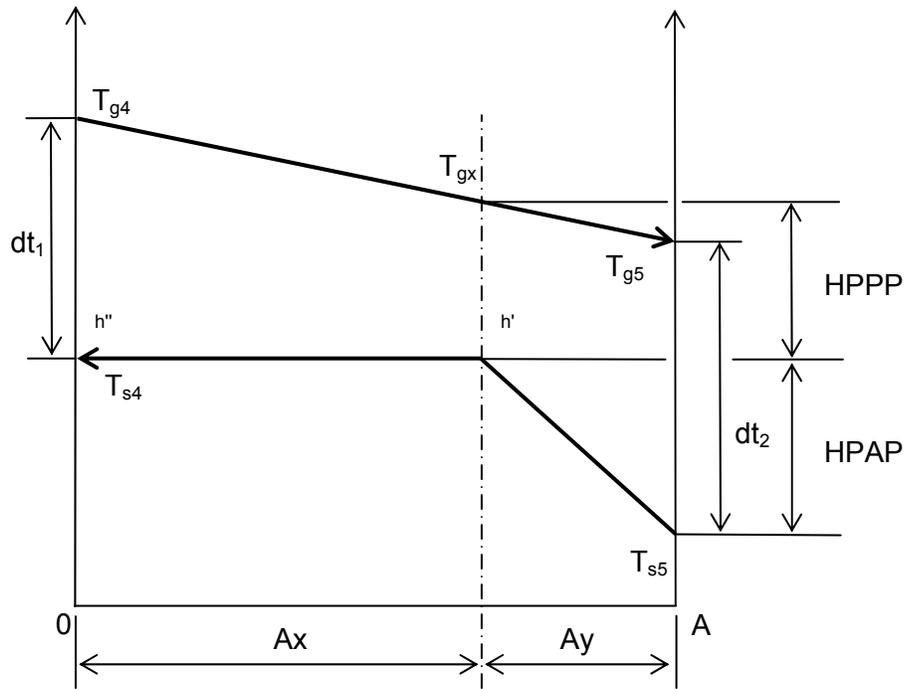


Figure 2: HPEVA temperature profile

$$\dot{Q}_x = (m_{\text{HPT}} - w_1)(h'' - h') \quad (\text{E1-5})$$

$$h_{gx} = h_{g4} - \frac{\dot{Q}_x}{\dot{m}_{gd}} \quad (\text{E1-6})$$

$$T_{gx} = \frac{h_{gx}}{c_p} \quad (\text{E1-7})$$

$$dt_1 = T_{g4} - T_{s4} \quad (\text{E1-8})$$

$$dt_2 = T_{gx} - T_{s4} \quad (\text{E1-9})$$

$$dt_m = \frac{dt_1 - dt_2}{\ln \frac{dt_1}{dt_2}} \quad (\text{E1-10})$$

$$A_x = \frac{\dot{Q}_x}{k_{\text{HPEVA}} dt_m} \quad (\text{E1-11})$$

$$T_{s5} = T_{s4} - \text{HPAP} \quad (\text{E1-12})$$

$$p_{s5} = p_{s4} (1 + d_{\text{HPEVA}}) \quad (\text{E1-13})$$

$$h_{s5} = f(p_{s5}, T_{s5}) \quad (\text{E1-14})$$

$$\dot{Q}_y = (\dot{m}_{\text{HPT}} - w_1)(h' - h_{s5}) \quad (\text{E1-15})$$

$$h_{g5} = h_{gx} - \frac{\dot{Q}_y}{\dot{m}_{gd}} \quad (\text{E1-16})$$

$$T_{g6} = \frac{h_{g6}}{c_p} \quad (\text{E1- 17})$$

$$dt_1 = T_{g5} - T_{s5} \quad (\text{E1- 18})$$

$$dt_2 = T_{gx} - T_{s5} \quad (\text{E1- 19})$$

$$dt_m = \frac{dt_1 - dt_2}{\ln \frac{dt_1}{dt_2}} \quad (\text{E1- 20})$$

$$A_y = \frac{\dot{Q}_y}{k_{\text{HPEVA}} dt_m} \quad (\text{E1- 21})$$

$$\dot{Q}_{\text{HPEVA}} = \dot{Q}_x + \dot{Q}_y \quad (\text{E1- 22})$$

$$A_{\text{HPEVA}} = A_x + A_y \quad (\text{E1- 23})$$

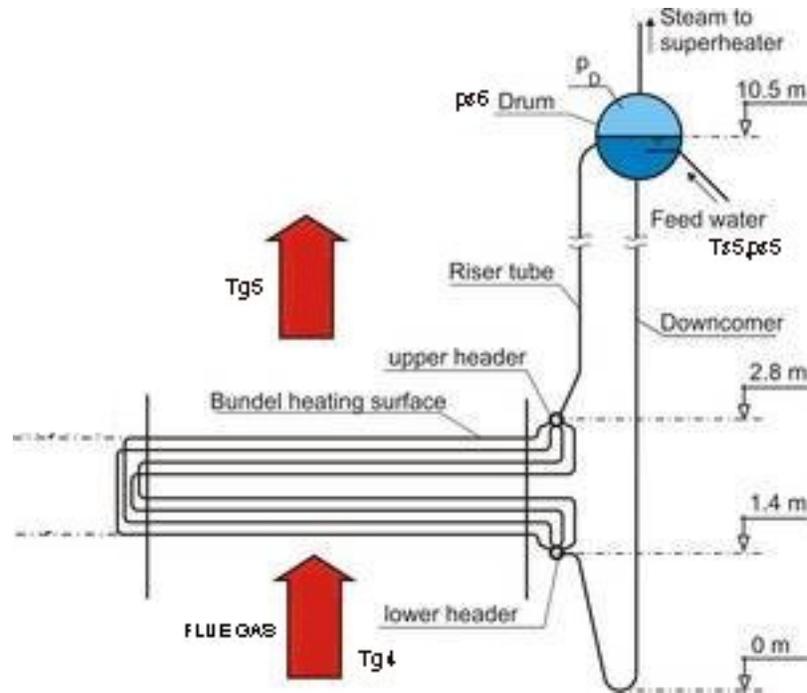


Figure 3: Schematic of HP drum and evaporator

$$Q_{\text{evap}} = D((h'' - h_{s5}) = m(h_{\text{ex}} - h_{\text{in}}) \quad (\text{E1- 24})$$

$$\Delta h = h_{\text{ex}} - h_{\text{in}} = \frac{D}{m}(h'' - h_{s5}) \quad (\text{E1- 25})$$

Wherein  $m/D$  is described as  $O$  and is known as the circulation ratio. It depends on a drum arrangement and drum type which can be natural or forced circulation.

HP Economizer:

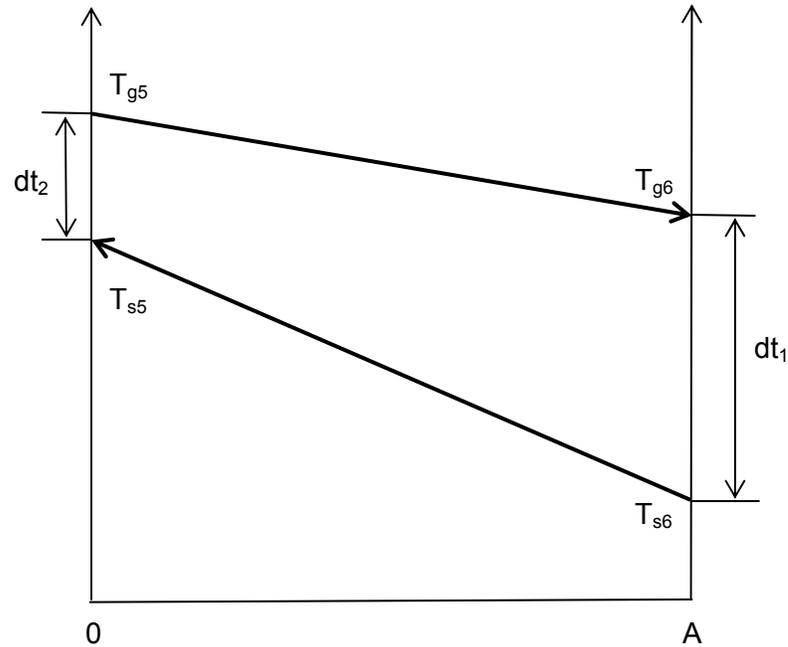


Figure 4: HPECO temperature profile

$$\dot{Q}_{\text{HPECO}} = \eta_r \dot{m}_{fg} (h_{g5} - h_{g6}) \quad (\text{E1- 26})$$

$$\dot{Q}_{\text{HPECO}} = \dot{m}_{\text{HP}} (h_{s5} - h_{s6}) \quad (\text{E1- 27})$$

$$\dot{Q}_{\text{HPECO}} = k_{\text{HPECO}} A_{\text{HPECO}} dt_m \quad (\text{E1- 28})$$

HP Splitter:

$$h_{\text{HP}} = h_{w1} \quad (\text{E1- 29})$$

HP Pump:

$$h_{s6} = h_{s7} + \frac{h_{s6i} - h_{s7}}{\eta_{\text{pump}}} \quad (\text{E1- 30})$$

HP turbine:

Number of control valves: 2

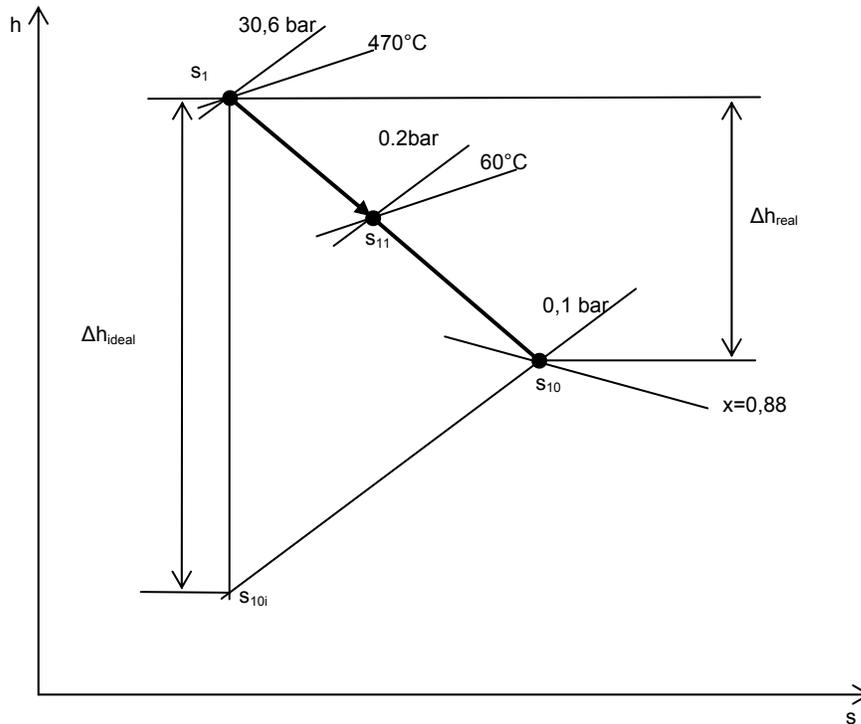


Figure 5: h-s diagram of ST expansion

Expansion process in the steam turbine is determined by following predefined parameters:  
Enthalpy at the HP steam turbine inlet is calculated from inlet pressure and temperature.

$$h_{HPinlet} = f(p, T)$$

Extraction point (for deaeration) lies on the expansion line drawn between the two above points, and its enthalpy is calculated from extraction pressure.

$$h_{LPinlet} = f(p, T)$$

Enthalpy at the steam turbine outlet calculated from condensing pressure:

$$h_{LPoutlet} = f(p=0,1\text{bar})$$

Real enthalpy drop is calculated from ideal enthalpy drop and ST isentropic efficiency:

$$\Delta h_{HPST,real} = \Delta h_{HPST,ideal} \eta_{isentropic} \quad (E1-31)$$

Produced power at ST shaft is calculated from:

$$P_{HPST} = \dot{m}_{HP} \cdot \Delta h_{HPST,real}$$

Condenser:

$$\dot{Q}_{COND} = \dot{m}_{cond} (h_{s10} - h_{s9}) \quad (E1-32)$$

$$\dot{Q}_{COND} = \dot{m}_{cw} c_{cw} (T_{cw2} - T_{cw1}) \quad (E1-33)$$

$$\dot{Q}_{COND} = k_{COND} A_{COND} dt_m \quad (E1-34)$$

Deaerator:

$$\dot{m}_{\text{cond}} + \dot{m}_{\text{DEA}} = \dot{m}_{\text{FW}} \quad (\text{E1- 35})$$

$$\dot{m}_{\text{cond}} h_{\text{cond}} + \dot{m}_{\text{DEA}} h_{\text{dea}} = \dot{m}_{\text{FW}} h_{\text{fw}} \quad (\text{E1- 36})$$

LP steam mass flow needed for deaeration is expressed from:

$$\dot{m}_{\text{DEA}} = \dot{m}_{\text{HPT}} - \dot{m}_{\text{cond}} \quad (\text{E1- 37})$$

Condensate Pump:

$$h_{\text{s8}} = h_{\text{s9}} + \frac{h_{\text{s8i}} - h_{\text{s9}}}{\eta_{\text{CNDpump}}} \quad (\text{E1- 38})$$

Project lifetime	= 20 years
Hours/year	= 8760 h/year
Load factor	= 90%
Production per kW per year	= Hours/year x Load factor = 8760 h/year x 0.9 = 7884 kWh/kW/year
Revision cost [%]	= 5%
Revision frequency	= 10 years
Revision cost	= Revision cost x Investment costs
O&M costs [%]	= 1%
O&M costs	= O&M cost x Investment costs
Fuel price	= 228\$/t
Fuel cost	= Annual fuel consumption x Fuel price

In order to obtain present value of all costs, present value (PV) of each cost is calculated.

Discount rate	= 10 %
PV revision	= Revision cost x SPPW (i,n) = Revision cost x $SPPW = \frac{1}{(1+i)^n}$
PV O&M cost	= O&M cost x USPW (i,n) = O&M cost x $USPW = \frac{(1+i)^n - 1}{i(1+i)^n}$
PV fuel cost	= Fuel cost per kW x USPW (i,n) = Fuel cost per kW x $USPW = \frac{(1+i)^n - 1}{i(1+i)^n}$
PV total costs	= Investment cost + PV revision + PV O&M + PV fuel

By knowing the present value of costs it is possible to calculate the capital recovery, respectively the annual income which is required for paying off the costs.

Capital recovery	= PV total costs x CR (i, n) = PV total costs x $CR = \frac{i(1+i)^n}{(1+i)^n - 1}$
Electricity cost	= CR/ Production per kW per year

Revenue for electricity sellback to the grid

El. sellback price	= 5 \$¢/kWh = 0,05 \$/kWh
El. sellback annual income	= $E_{GEN}$ x electricity sellback price

Revenue for CO<sub>2</sub> reduction

$$\begin{aligned} \text{Annual revenue} \\ \text{for CO}_2 \text{ reduction} &= \text{CO}_2 \text{ reduced} \times \text{emission certificate price} \\ &= [\text{tCO}_2/\text{year}] \times 30 \text{ [$/tCO}_2\text{]} \end{aligned}$$

$$\begin{aligned} \text{Revenue for CO}_2 \\ \text{reduction per kWh} &= \frac{\text{Annual revenue for CO}_2 \text{ reduction}}{E_{\text{ST, GEN}}} \end{aligned}$$