Project: IGCC-Cuba

Commissioned to Life ltd by SHERRIT International Corp.

Integrated Gasification Combined Cycle Preliminary Design.

A case study for Varadero Crude

Extract from Project Final Report IGCC-Cuba/oct.98

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- October 1998 -

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Acknowledgements

The present work is an extract of the final report of the project commissioned to Life Ltd in 1998 by Sherrit International Corporation, for the development of an **environment friendly** power generation scheme for Varadero oil consumption in Cuba. The authors wish to acknowledge the Sherrit International Corporation for the permission to pubblication. The authors are also deeply indebted to Prof. O. A. Barra and Dr. P. Ionta (Life Ltd) for the opportunity they have given with the participation to the design team.

1. Introduction

The general description of combined-cycle configuration together with a technical and technological justification of main design criteria is given in **2.**. In **3.** is reported the performance analysis of optimized cycle (designed for ISO conditions) and is discussed the combined cycle environmental de-rating. Furthermore are presented the functional schemes of proposed combined cycle for the discussed operating conditions. In par. **4.** the plant operating strategy is specified. The paragraph **5.** contains the preliminary information concerning P&I, control and logic diagrams. In **6.** a synthetic list of requirements and performance (ISO and de-rated) for each components.

2. IGCC - General description and design criteria

The present work origins from the need of developing in Cuba an **environment friendly** power generation scheme for Varadero oil consumption.

The combined-cycle power plant will bring to fruition the forward-looking concept of combined gas and steam turbine drive with fully integrated oil gasification system. Referring to the actual use of Varadero oil as liquid fuel, the project has, therefore, twofold critical meanings.

From the *energetic* viewpoint, the proposed power plant scheme allows:

- the achievement of the highest efficiencies (tipically around *50%*) in the framework of thermo-mechanical conversion process fed by fossil fuel,
- to carry out great installed power plant (for electric network supply) preserving some of the simplicity that characterized the topper-turbogas cycle (e.g.: plant lay-out), which shares the larger contribution to total installed power.

Furthermore from the *environment* viewpoint, it's possible to envisaged several benefits:

- the increased conversion efficiency leads to a reduction of carbon dioxide emission for unit of produced power,
- the gasification process is able to supply a clean fuel, thus inherently control sulphur oxides emission,
- the role of topper-turbogas plant (in terms of its relative share to total installed power) leads to a substantial reduction of thermal impact due to cooling water supply for unit of produced power.

The combined-cycle is carried out by the matching of *gas turbine* plant (*GT*) as topper and a *steam turbine* (*ST*) plant as bottomer, thermally linked through an *heat recovery steam generator* (*HRSG*). The heat recovery combined cycle consists of several standardized components which can be configured in building blocks to form power plants of required capability. The design of power generation systems should therefore result from pre-engineered combined cycle configuration, optimized in order to achieve the best fitting between each building blocks.

The lay-out proposed for the *Integrated Gasification Combined Cycle* (*IGCC*), is a Joule-Hirn two pressure level combined-cycle. The plant set-up is here described. The main functional interfaces between power generation unit, gasificaton and air separation units are also defined.

Topper – Gas Turbine plant, heavy duty class is arranged in high efficiengy open cycle, single-shaft configuration. The operating condition refers to base load service, with dual firing (including oils).

The adopted *topper* power group configuration consists of parallel operating medium size gas turbine. Such solution is in order to guarantee both power plant service reliability (low probability of contemporary GT failure) and plant operating flexibility during mantainance periods. The exhaust gas flows from turbines are collected to supply plant *Heat Recovery Steam Generator* (*HRSG*).

Topper – Gas Turbine plant fluidic interfaces (Fig. 2.1), the GT power group has two main interfaces with the gasification plant, and a third interface with the *HRSG*. From the *Oil Gasification Unit* (*OGU*) it receives the syngas fuel flow rate, while the fluidic links with *Air Separation Unit* (*ASU*) could comprehend the input flow rate of molecular nitrogen, and as output compressed air flow rate to *ASU*.

The *integration* of standard heavy duty GT power unit with the gasification plant is mainly influenced by the following governing parameter: the use of fuel (*syngas*) with particularly low heating value *LHV = 15.1* (*MJ/kg*). The consequences on GT power group operating conditions are:

- the syngas fuel flow rate, required to obtain the GT nominal thermal output, must be *4 ÷ 5* times greater than in case of natural gas supply,
- the expanding mass flow rate in the turbine is increased (*10%* rising),
- the GT electrical power output is increased.

Furthermore, such modified nominal operating condition affects partially the compressor (whose compression ratio is limited by stall inception), while the turbine inlet temperature (TIT) limit remain approximately constant as a consequence of high quality syngas composition.

Bottomer – Steam Turbine plant, one feed water heater simple super-heated cycle, with a two level vaporization phase supplying dual body steam turbine (back-pressure turbine HP and condensing turbine LP). The condenser is fed by sea-water in open circuit.

Bottomer – Steam Turbine plant fluidic interfaces (Fig. 2.1), the ST power group has two main interfaces with the gasification plant, and a third interface with the GT power group. The first interface comprehends the following links: the *OGU* (steam supplied for the gasification process), the *Syngas Cooler* (supply of saturated water and feedback of saturated steam), the *Sulphur Removal Unit* (*SRU*) (steam supplied for the sulphur removal process). The second interface concerns with the *Air Separation Unit* (*ASU*) that could input the ST power group with treated water.

The *integration* of the ST power group with the gasification plant, uses the bottomer steam cycle as main source of water/steam flows for combined plant auxiliary services and the gasification process. In particular the main water/steam flows, described in Fig. 2.1, concerns with:

- the saturated water flow from high pressure drum for syngas cooling,
- the superheated steam flow for the gasification process (e.g.: gasifier supply, syngas purification, etc.).

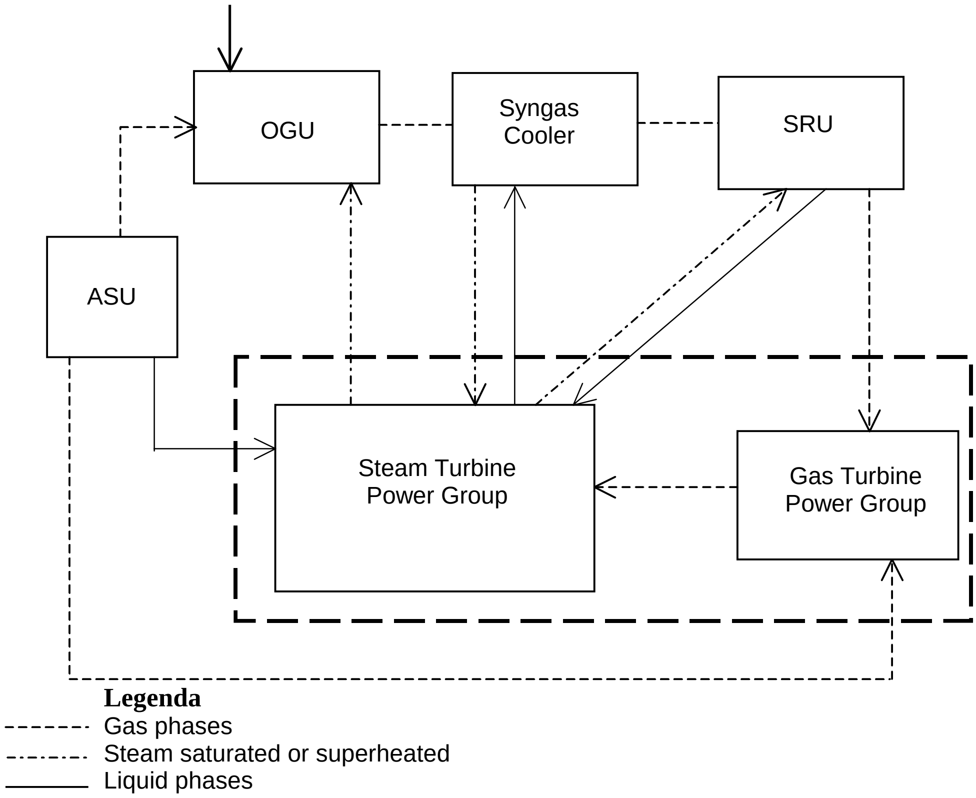


Fig. 2.1 - Integrated Gasification Combined Cycle, main fluidic interfaces

2.1 HRSG design criteria

In case of base load service the main *IGCC* performance parameter is the conversion efficiency, attainable only by the adoption of complex recovery path configuration (e.g.: multiple pressure levels, steam superheating, etc.). The two pressure level arrangement has been selected to achieve a good compromise between combined-cycle thermodynamics performance improvement and scheme simplicity (in terms of both lay-out and control system). Such an arrangement has nowadays a wide diffusion within the operating combined-cycle plant state of art.

The gained thermodynamic improvements result from an increased *quality* of heat recovery process typical of multi-level combined cycle, as a consequence of both the reduction of discharged heat and irreversibility of water vaporization phases. It has to be noted that:

- adopting a two pressure level configuration, the discharged thermal power share is reduced to 10% of the total available power from exhaust (a simple one level combined-cycle is affected by a 21% thermal discharge loss),
- the reachable efficiencies are greater than 50% with a two pressure level combined-cycle,
- the increased cost of *HRSG* (as a consequence of greater exchanging surfaces and structure complexity) is very moderate because of such component share only the 10% of total combined plant cost.

The preliminary design of the two pressure level *HRSG*, here presented, has been planned out on the basis of the ensuing criteria and requirements.

- The high pressure level (about *90 bar*) has been fixed in order to reach a good compromise between the thermodynamic cycle improvement (measured *kW/bar*) and component structural simplicity.
- The low pressure level (about *20 bar*) has been defined in order to optimized the HRSG exchanging efficiency, because its moderate influence on CCP power.
- The superheated steam temperature (about *520°C*) has been fixed in order to minimized the heat transfer surfaces, increasing the high pressure exchanged heat flux share.
- The discharge exhaust temperature at the stack hasn't inferior limit specified to avoid the acid condensate formation (exhaust originate from combustion of desulphurized gaseous fuel with excess air); it's advisable to design the *HRSG* in order to obtain discharge temperature in the range *(120°C ÷ 90°C)*.
- The steam limit conditions (9*0 bar, 520°C*) corresponds to small power steam plant, tipically adopted for combined-cycle scheme.
- The condensation condition (*0.1 bar*) refers to sea water operating as cooling fluid in open circuit.
- The de-aerating unit is provided directly from low temperature exhaust heat, thus allowing the elimination of regenerative steam bleeding from turbine because of its thermodynamic *non-sense* within combined cycle.

The *HRSG* has an orizontal configuration, with natural circulation obtained by several downcomers which also provide additional structural support for the heat transfer surface and steam drum. Such configuration enable the achievement of greatest component reliability and flexibiltiy in off-design operating condition, due to the lack of pumps, valves and control.

As far as the thermal top-bottom link is concerned, the chosen bottomer configuration results in the above specified steam generator characteristics. Fig. 2.1.1 shows the generator components arrangement.

The condensate pump lead the condensed water mass flow rates to the *HRSG* pre-heating section, and then to the de-areating. Starting from the de-areating unit two feed pumps complete water compression phase, leading to design HP and LP levels. Following the water paths within steam generators, the ensuing components could be defined.

Pre-heater, such section pre-heats the water mass flow rate using low temperature exhaust heat.

De-areating, such unit is designed to produce the required saturated steam mass flow rate for the removal of gaseous components from water and the completion of pre-heating phase.

Economizers, the LP economizer (eco-LP) is followed by the HP economizer (eco-HP) in the direction opposite to the exhaust path (such serial disposition is here preferred to the parallel because of its ability to avoid the effects of exhaust flow disuniformity in generator transverse sections).

Evaporators, the LP evaporator (ev-LP) follows the (eco-HP), while the HP evaporator (ev-HP) is arranged past the series of LP evaporator and super heater (such arrangement is suggested to fit the criteria of optimal heat exchanging).

Super heaters, the LP superheater (sh-LP), as already specified, is arranged before the high pressure evaporator (ev-HP) while the high pressure superheater follows it as last components of *HRSG*.

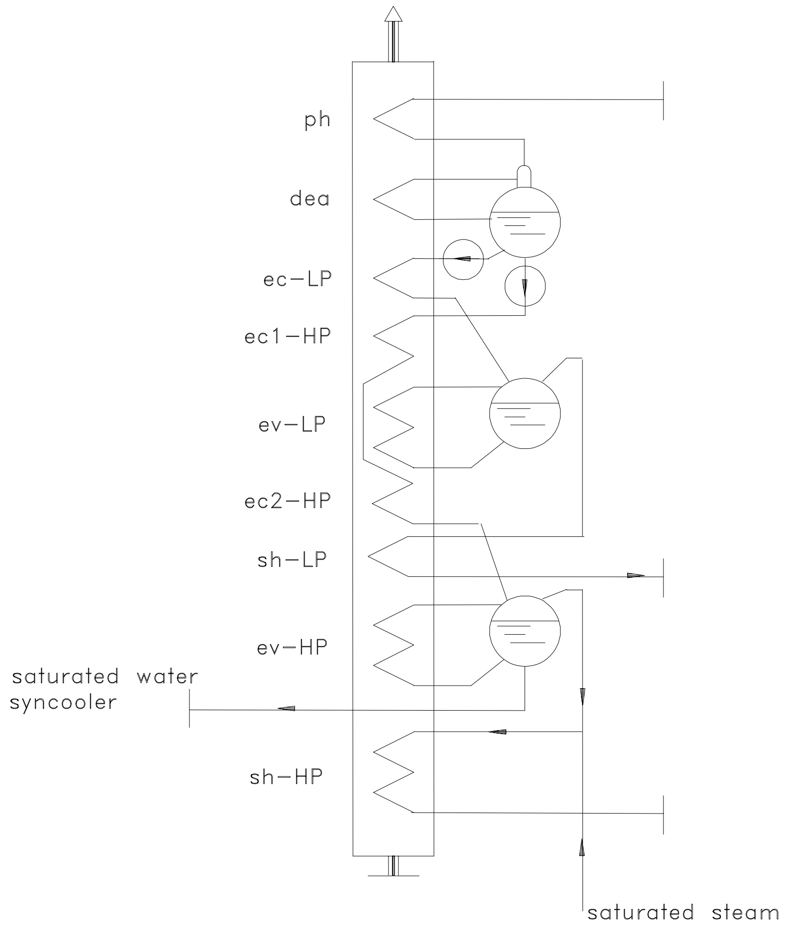
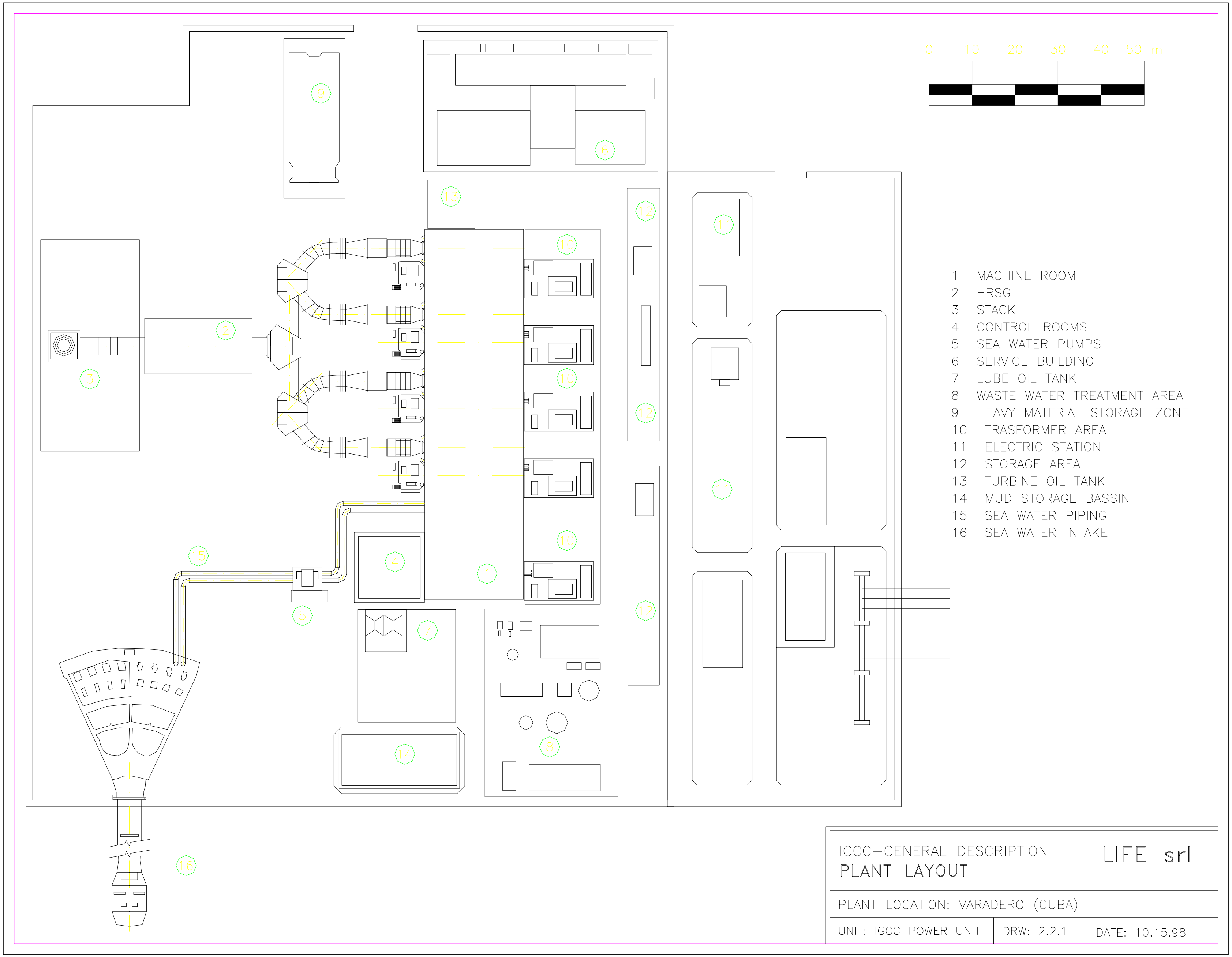


Fig. 2.1.1 - *HRSG* scheme

2.2 IGCC – Plant lay-out

The ensuing drawing (drw. 2.2.1) shows a preliminary arrangement of combined cycle power groups.


0 10 20 30 40 50 m
1 MACHINE ROOM
2 HRSG
3 STACK
4 CONTROL ROOMS
5 SEA WATER PUMPS
6 SERVICE BUILDING
7 LUBE OIL TANK
8 WASTE WATER TREATMENT AREA
9 HEAVY MATERIAL STORAGE ZONE
10 TRASFORMER AREA
11 ELECTRIC STATION
12 STORAGE AREA
13 TURBINE OIL TANK
14 MUD STORAGE BASSIN
15 SEA WATER PIPING
16 SEA WATER INTAKE
IGCC–GENERAL DESCRIPTION
PLANT LAYOUT
LIFE srl
PLANT LOCATION: VARADERO (CUBA)
UNIT: IGCC POWER UNIT
DRW: 2.2.1
DATE: 10.15.98

3. IGCC - Performance Data

3.1 IGCC - ISO rating

The *optimized* nominal design of IGCC has been carried out for ISO environmental conditions (ISO rating), and adopts the following set of data (Tab. 3.1.1) as problem boundary conditions which depends on the integration between power and gasification systems.

Tab. 3.1.1 - Design data basis

Measurement	**Value**
Elevation	sea level altitude
Ambient temperature (inlet compressor temperature)	15 (°C)
Ambient pressure	0.99 (bar)
Relative humidity	60%
Inlet system pressure drop	100 (mm H_2O), 0.0098 (bar)
Exhaust pressure drop	250 (mm H_2O), 0.0245 (bar)
Syngas fuel heating value (LHV)	16.8 (MJ/kg)
Combustion system type	Standard

The performance data in case of base load operating condition are below envisaged (Tab. 3.1.2).

Tab. 3.1.2 - IGCC Performance (ISO rating)

Measurement	**Value**
IGCC power output	285 (MW)
GT **net** power output	166.6 (MW)
ST **net** power output	115.9 (MW)
GT heat rate	11544.2 (kJ/kWh)
GT syngas mass flow rate	31.8 (kg/s)
GT heat consumption (10^{-6})	1923.3 (kJ/h)
Heat rate input to *HRSG*	335.6 (MW)
Heat rate input to *HRSG* (from **exhaust**)	283. (MW)
Heat rate input to *HRSG* (from **syngas cooler**)	52.6 (MW)
Exhaust mass flow rate	586.6 (kg/s)
Exhaust temperature	539.1 (°C)

The performance outputs have been evaluated at the electrical generator terminals and includes allowances for the auxiliary services normally supplied. The Tab. 3.1.3 that follows, shows data output obtained by the optimized thermodynamic design procedure obtained by using an in-house made code carried out at DMA (see appendix A.1).

Tab. 3.1.3 - Optimized thermodynamic design procedure output

IGCC - Optimized design procedure **DESIGN DATA**		
Plant location Varadero (CUBA)		
Unit IGCC power unit	Sheet 1 of 2	Date 15/10/98

INTEGRATED GASIFIED COMBINED CYCLE - MAIN CYCLE PARAMETERS

IGCC total power output (kW)	285000.00
Gas Turbine net power group output (kW)	166624.05
Steam Turbine net power group output (kW)	115915.95
IGCC auxiliary systems power demand (kW)	3291.
IGCC thermal efficiency	.446
GT thermal efficiency	.31
ST thermal efficiency	.34
GT power group exhaust mass flow rate(kg/s)	586.60
ST steam mass flow rate (kg/s)	102.67
ST steam mass flow rate - LP circuit (kg/s)	6.30
ST steam mass flow rate - HP circuit (kg/s)	96.36
ST de-areating mass flow rate (kg/s)	9.99

GAS TURBINE POWER GROUP

Compressor Inlet air temperature (°C)	15.00
Compressor Inlet air pressure (bar)	0.99
Compression ratio	11.8
Combustor outlet temperature (°C)	1109.97
Combustor outlet pressure (bar)	11.26
Turbine inlet temperature - TIT (°C)	1053.01
Expansion ratio	10.52
Turbine outlet temperature - TOT (°C)	539.12
Turbine cooling - bleeding air mass flow rate (kg/s)	47.26
Air fuel ratio	15.96
Fuel mass flow rate (kg/s)	31.79
Compressor inlet mass flow rate (kg/s)	587.04

Tab. 3.1.3 - Optimized thermodynamic design procedure output (cont.)

IGCC - Optimized design procedure **DESIGN DATA**		
Plant location Varadero (CUBA)		
Unit IGCC power unit	Sheet 2 of 2	Date 15/10/98

STEAM TURBINE POWER GROUP

Condensing temperature (°C)	45.46
Condensing pressure (bar)	.10
Pre-heater pressure (bar)	1.40
Pre-heater outlet temperature (°C)	56.85
De-areating pressure (bar)	1.40
De-areating temperature (°C)	108.75
ev-LP vaporization temperature (°C)	144.29
sh-LP steam outlet temperature (°C)	289.88
LP steam pressure (bar)	4.07
ev-HP vaporization temperature (°C)	296.82
sh-HP steam outlet temperature (°C)	519.12
HP steam pressure (bar)	82.12
LP pinch-point (°C)	15.00
HP pinch-point (°C)	10.00
LP approach-point (°C)	25.00
HP approach-point (°C)	20.00

HEAT RECOVERY STEAM GENERATOR

Exhaust inlet temperature (°C)	539.12
Exhaust outlet temperature (°C)	113.85
Exhaust I eco. outlet temperature (°C)	155.85
Exhaust thermal power (kW)	283011.33
HRSG Heat rate (kW)	263200.56
Pre-heater thermal power (kW)	4886.94
De-areating thermal power (kW)	22284.20
eco-LP thermal power (kW)	953.14
ev-LP thermal power (kW)	13425.85
sh-LP thermal power (kW)	1919.59
eco-HP thermal power (kW)	78920.27
ev-HP thermal power (kW)	73881.36
sh-HP thermal power (kW)	59837.95

3.2 IGCC - Environmental de-rating

The performance variation of gas turbine with ambient air conditions (thus compressor inlet air conditions) causes an associated variation in combined cycle performance which should be considered when matching the combined cycle system to site atmospheric conditions.

The site atmospheric conditions adopted for the evaluation of performance de-rating are listed below.

Tab. 3.2.1 - Site atmospheric conditions

Measurement	Value
Elevation	sea level altitude
Ambient temperature (inlet compressor temperature)	35 (°C)
Ambient pressure	0.99 (bar)
Relative humidity	80%

The evaluation of de-rating has been carried out considering the compressor inlet temperature as the leading parameter of performance degradation. As a matter of fact, the influence of non-ISO specific humidity on GT operating parameters (output and heat rate) is two order of magnitude lower compared to the effect of ambient temperature variation.

The behavior of combined cycle system in off-design conditions due to the variation of compressor inlet temperature, depends on the *GT* controlling strategy. In what follow, the *GT* control system is assumed to operate on firing temperature in order to preserve constant TIT. As a consequence of such control strategy the following main effects could be highlighted:

- gas turbine air mass flow rate decreases with increasing of temperature,
- *GT* compression ratio decreases as a consequence of air mass flow rate reduction with increasing of air temperature (and constant TIT).

The performance data in case of environmental de-rating with base load operating condition are below envisaged.

Tab. 3.2.2 - Environmetal de-rating - effect of compressor inlet temperature

Measurement	Percent design	Value
GT power group output	87	144.9 (MW)
GT heat rate	104	12005.97 (kJ/kWh)
GT syngas mass flow rate	89.9	28.6 (kg/s)
GT heat consumption (10^{-6})	90	1730.97 (kJ/h)
Exhaust flow	92	539.7 (kg/s)
Exhaust temperature		552.8 (°C)

As far as the IGCC performance are concerned, typically there is very little heat rate or efficiency variation with ambient temperature. This occurs because the maximum cycle temperature remains approximately constant and the performance variations simply reflect the variation in efficiencies of components with varying operating conditions (e.g.: the *GT* power group).

3.3 IGCC - Performance degradation

The *IGCC* are afflicted by a performance loss during extended period operational periods mainly due to *GT* power group compressor fouling. For such reason during normal operation the performance loss is minimized by periodic on-line and off-line compressor water washes.

The scheduling usually adopted for maintanance operations on *IGCC* - *GT* power group is below resumed.

Tab. 3.3.1

Maintainance operation	GT Fired hours
On-line compressor wash	1.000
Off-line compressor wash	5.000
Inspection and compressor scour	24.000

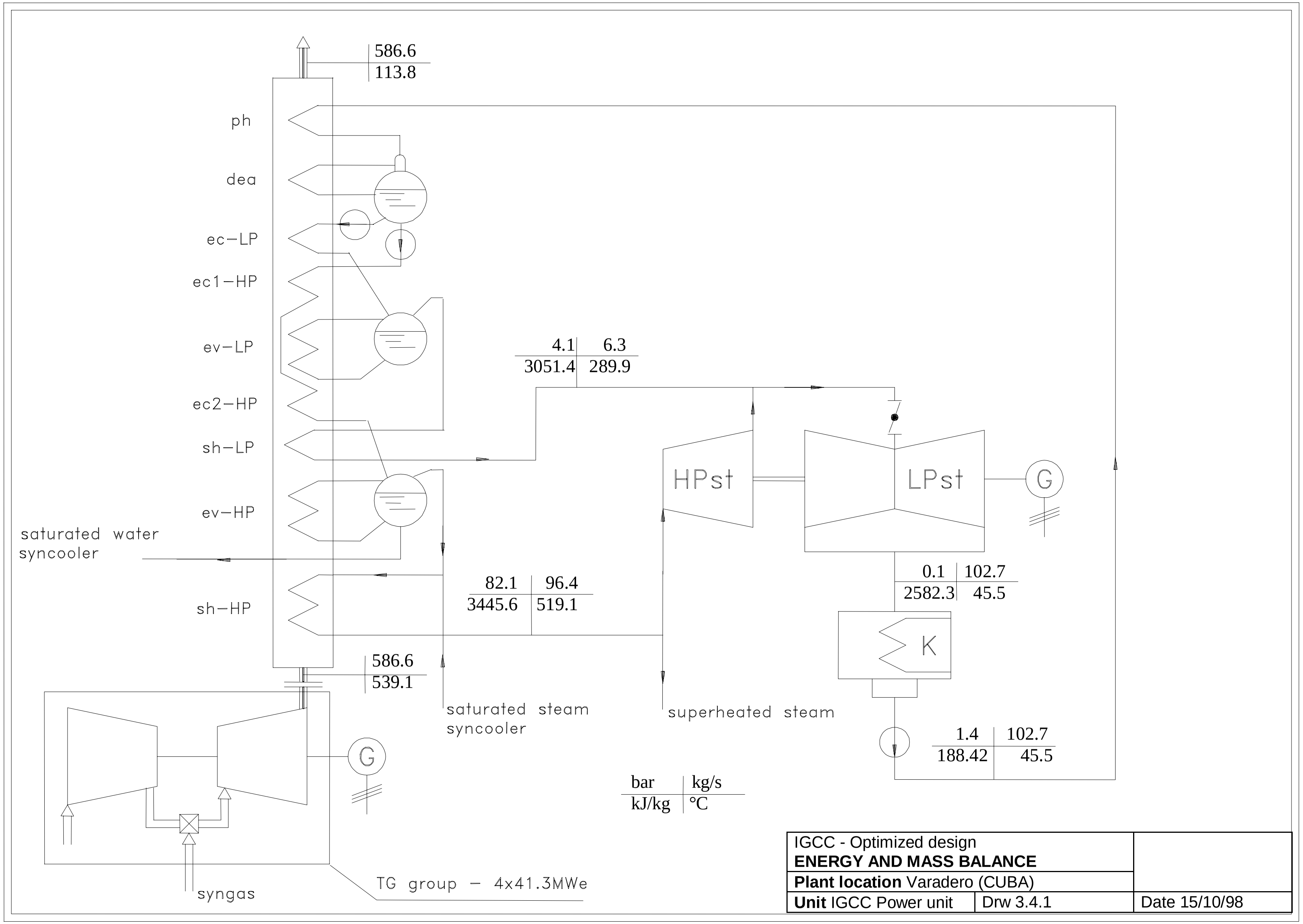
The expected aged performance have been derived on the basis of *IGCC* location, which account for environmental conditions with high humidity and corrosive atmosphere. The *IGCC* expected performance losses have been evaluated during an operational period of 6 years.

Tab. 3.3.2 - Expected IGCC aged performance

GT Fired period (1 year = 8000 hrs)	IGCC power output loss (%)	IGCC thermal efficiency loss (%)
1	3.4	1.7
2	4.2	2.1
3	4.6	2.3
4	5	2.5
5	5.2	2.55
6 (48.000hrs)	5.3	2.6

3.4 IGCC - Mass and energy balances

The preliminary combined cycle plant solution is here described in terms of combined cycle energy and mass balances (drw. 3.4.1). In such diagrams each component is identified on the basis of coding further employed in par. **6**.



586.6
113.8
ph
dea
ec–LP
ec1–HP
ev–LP
ec2–HP
sh–LP
ev–HP
sh–HP
saturated water
syncooler
4.1 | 6.3
3051.4 | 289.9
82.1 | 96.4
3445.6 | 519.1
586.6
539.1
saturated steam
syncooler
superheated steam
HPst
LPst
G
K
0.1 | 102.7
2582.3 | 45.5
1.4 | 102.7
188.42 | 45.5
bar | kg/s
kJ/kg | °C
syngas
TG group – 4x41.3MWe
IGCC - Optimized design
ENERGY AND MASS BALANCE
Plant location Varadero (CUBA)
Unit IGCC Power unit
Drw 3.4.1
Date 15/10/98

4. IGCC - Plant operating strategy

The combined cycle system control requirements involve the following main control loops (Tab. 4.0.1).

Tab. 4.0.1 – IGCC main control loops

Control loops – Components	Control actions
GT control	Fuel Compressor inlet guide vane
HRSG	Feedwater Economizer recirculation Damper modulation Superheated steam maximum temperature De-areating hotwell level
ST	Steam pressure (HP and LP levels) Steam by-pass valve Speed (starting and synchronizing)
Auxiliaries	Condensate pump recirculation Boiler feed-water pump recirculation Condenser hotwell level

In the *IGCC* control strategy the *GT* power group plays the leading role in the modulation of power outputs, and as a consequence the *HRSG* and *ST* power group simply follow the gas turbines and generate power depending on the amount of heat received from the *topper* itself.

4.1 Gas Turbine control

Such a control strategy is therefore based on *GT* modulating capabilities:

- the *GT* compressor inlet guide vane (*VIGV*) is modulated to control air mass flow rate over a limited range, from approximately 80 to 100% of the load range,

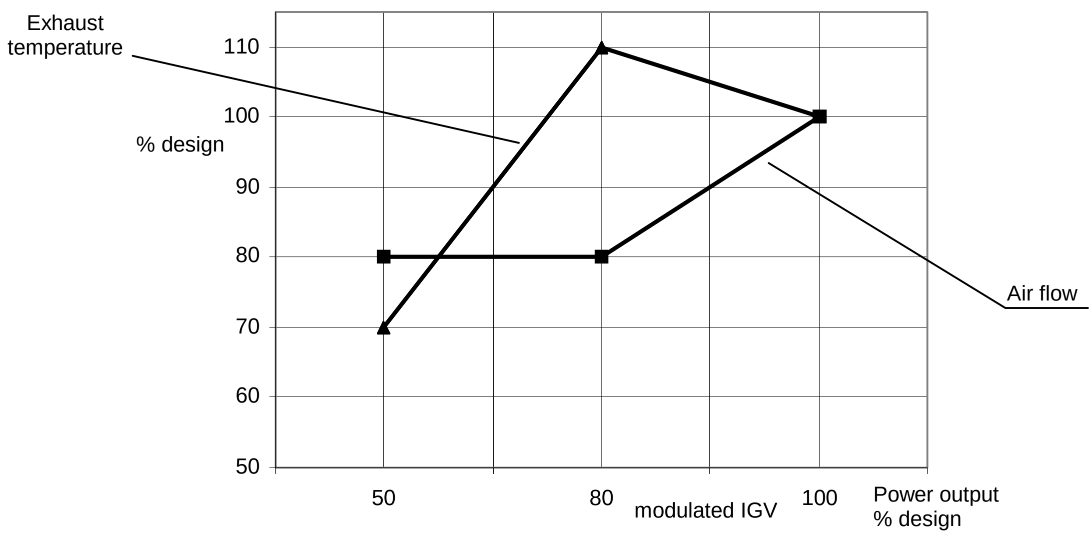


Fig. 4.1.1 – *GT* load modulating procedure

- the *GT* fuel control modulates the energy input to the combustor and to the unfired *HRSG*, from approximately 50 to 80% of the load range.

An example of *GT* modulating procedure is reported in Fig. 4.1.1, with reference to heavy-duty, open cycle one-shaft configuration. It should be noted that the reduction of air mass flow rate during *IGV* modulation, results in increased exhaust temperature. As a consequence of such behavior the thermal recovery is improved and the stack thermal losses decreases, due to the higher exhaust temperature at the inlet of *HRSG* that results in exhaust heat exchanging line with increased slope.

4.2 HRSG and Steam Turbine control

The control strategy for *HRSG* and *steam power group* load modulation includes the ensuing capabilities.

- The control of super-heated steam temperature with water injection, in order to carried out the steam temperation.
- The stabilization of *HP steam circuit* pressure to nominal value, in order to produce high pressure superheated steam at constant pressure and temperature durig load modulation to supply the gasification process (*Oil Gasification Unit*). The pressure level at the inlet section of *HP steam turbine* is controlled acting via a lamination valve.
- The stabilization of *LP steam circuit* pressure to nominal value, in order to preserve during the modulation the *LP* steam drum stability condition. In particular, the control of the pressure of *LP* steam circuit allows to operate at nominal pressure conditon the condensate pump, the pre-heater and the de-areating portion of water circuit thus gaining constant inlet pressure for the feed-water pumps. The pressure level at the inlet section of *LP steam turbine* is controlled acting via a lamination valve.

4.3 Plant operative control loops

The IGCC plant is considered shared into a number of *process group* (*PG*), each controlling a zone of the plant. The general performance carried out by each *PG* are below listed:

- start and stop of motors, modulation of pumps,
- control of valves,
- check motors and valves manoeuvring on the basis of pre-imposed control logic,
- check and control of alarm signals,
- limit the set values changes within the permissible range,
- check the sequence logic for *PG*s starting,
- preventive check for the operating conditions that allow *PG* start.

A short description of the main *IGCC* plant *Process Groups* follows.

Gas turbine GT

Starts and stops the turbine.

Controls loading, unloading and synchronizing.

Controls *VIGV* position.

Controls fuel mass flow rate.

Oil system

Starts and stops the lube oil and hydraulic oil pumps.

Controls the emergency oil pump.

Condensate

Controls the condensate pumps and valves.

Level control in condenser hotwell.

Feedwater

Controls the feedwater pumps.

Level control in feedwater tank (de-areating hotwell).

HRSG

Drums level control.

Temperature control of superheated steam.

Steam turbine ST

By-pass valve control during startup with respect to boiler pressure and steam pressure.

Control of steam temperation.

Blow-off valve control.

Control of lamination valves on *HP* and *LP* steam lines.

Overall logic

Starts the *Process Groups* in the correct order.

Controls the sequence of *Process Groups* stopping to achieve the plant stop.

Controls the correctness of starting conditions.

4.4 Starts and stop sequences

The automatic starting sequence comprehends the following actions:

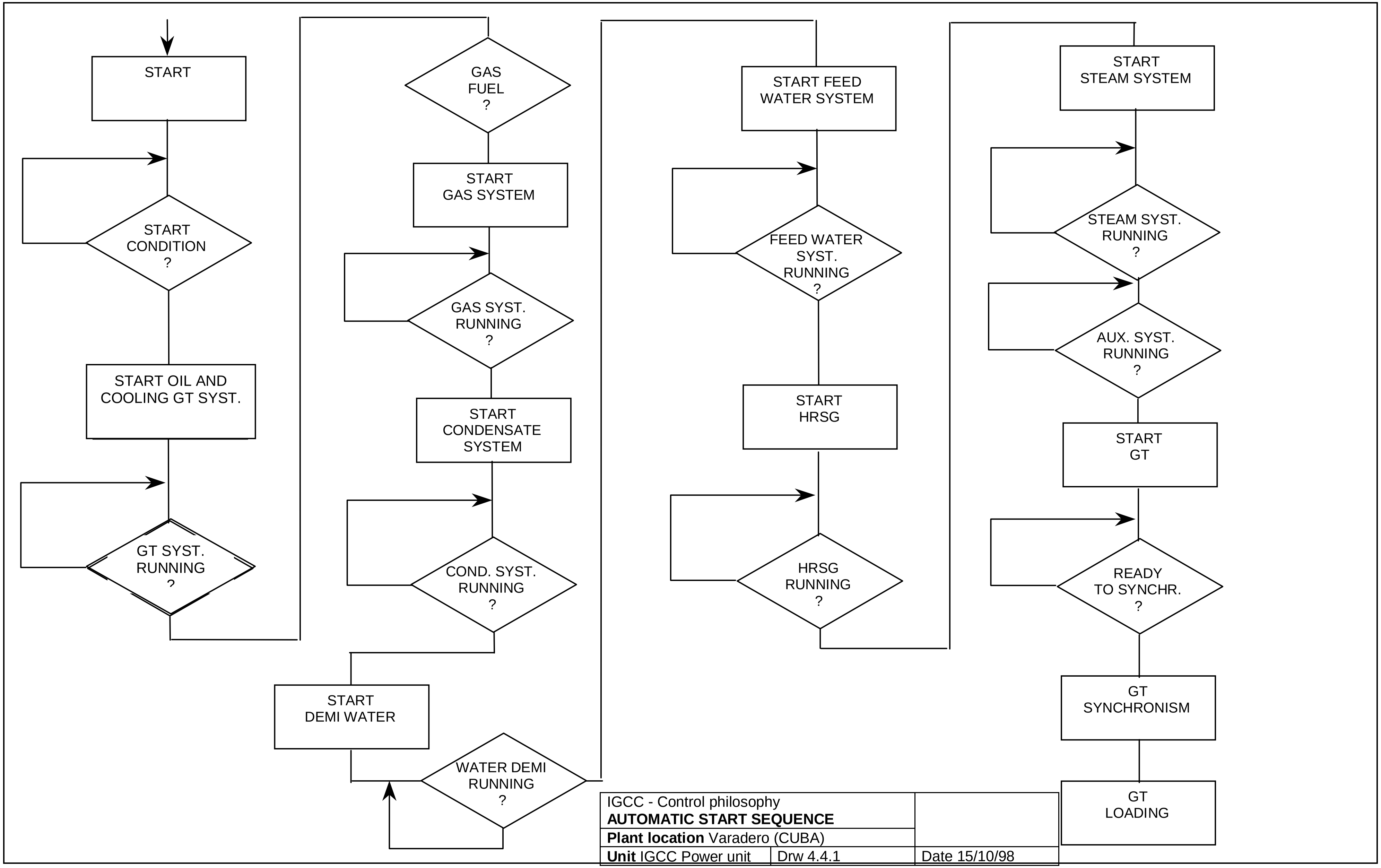
- *GT* lube oil system start,
- *GT* cooling air system start,
- *GT* fuel system start,
- *ST* start condensate system,
- *ST* Water Demineralization unit start,
- *ST* - *HRSG* system start,
- *ST* start system (lube oil, running),
- *GT* start and runs up to synchronized speed.

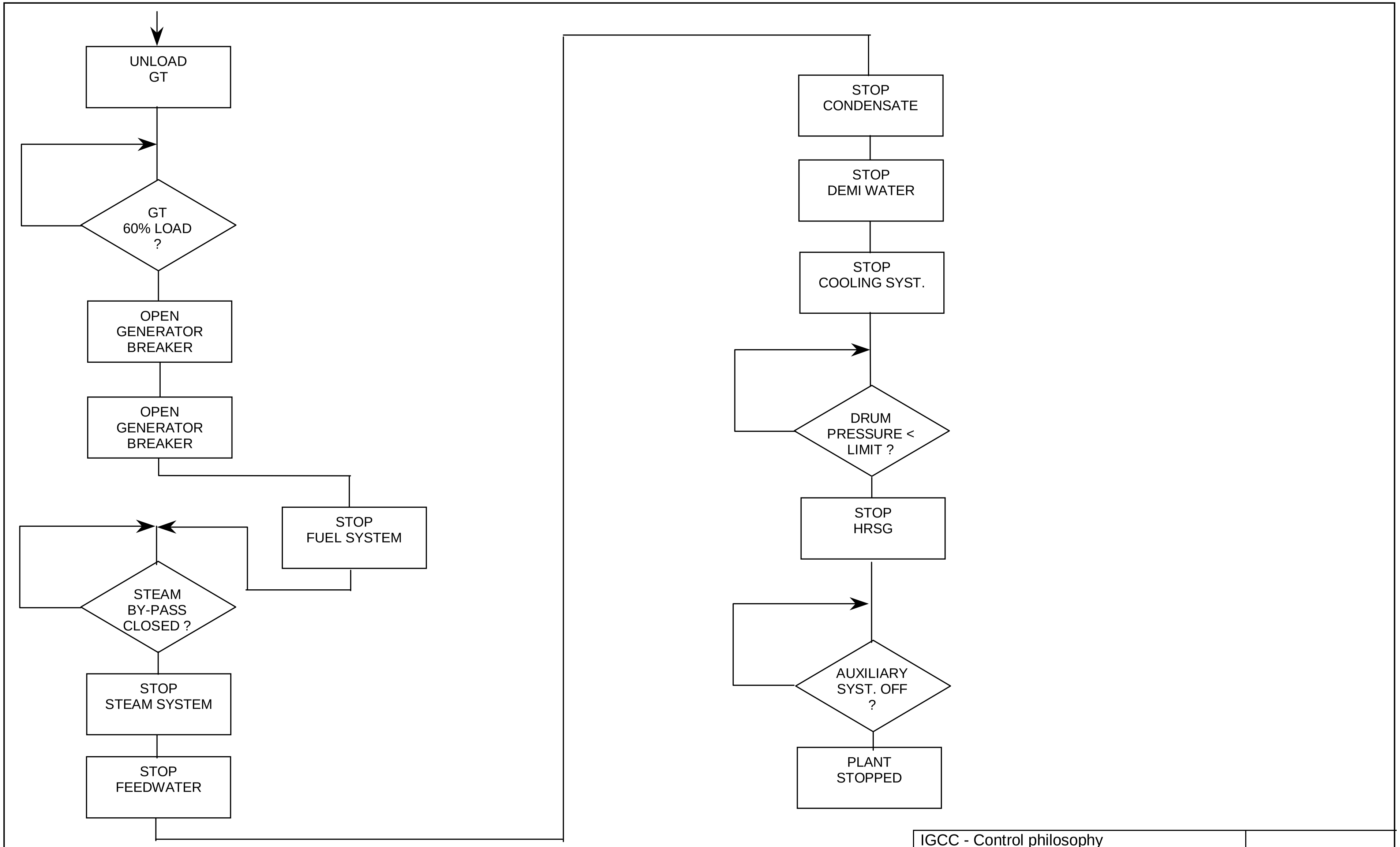
The scheme 4.4.1 represents the block diagram for the automatic starting sequence.

The automatic stop sequence comprehends the following actions:

- *GT* unload,
- open electric generator circuit breaker,
- *GT* stop,
- *ST* stop system,
- *ST* stop feedwater,
- *ST* stop condensate,
- *ST* stop Water Demineralization unit,
- *ST* system stop (lube oil),
- *GT* lube oil system stop,
- *GT* cooling system stop,
- *ST* - *HRSG* stop.

The scheme 4.4.2 represents the block diagram for the automatic stop sequence.

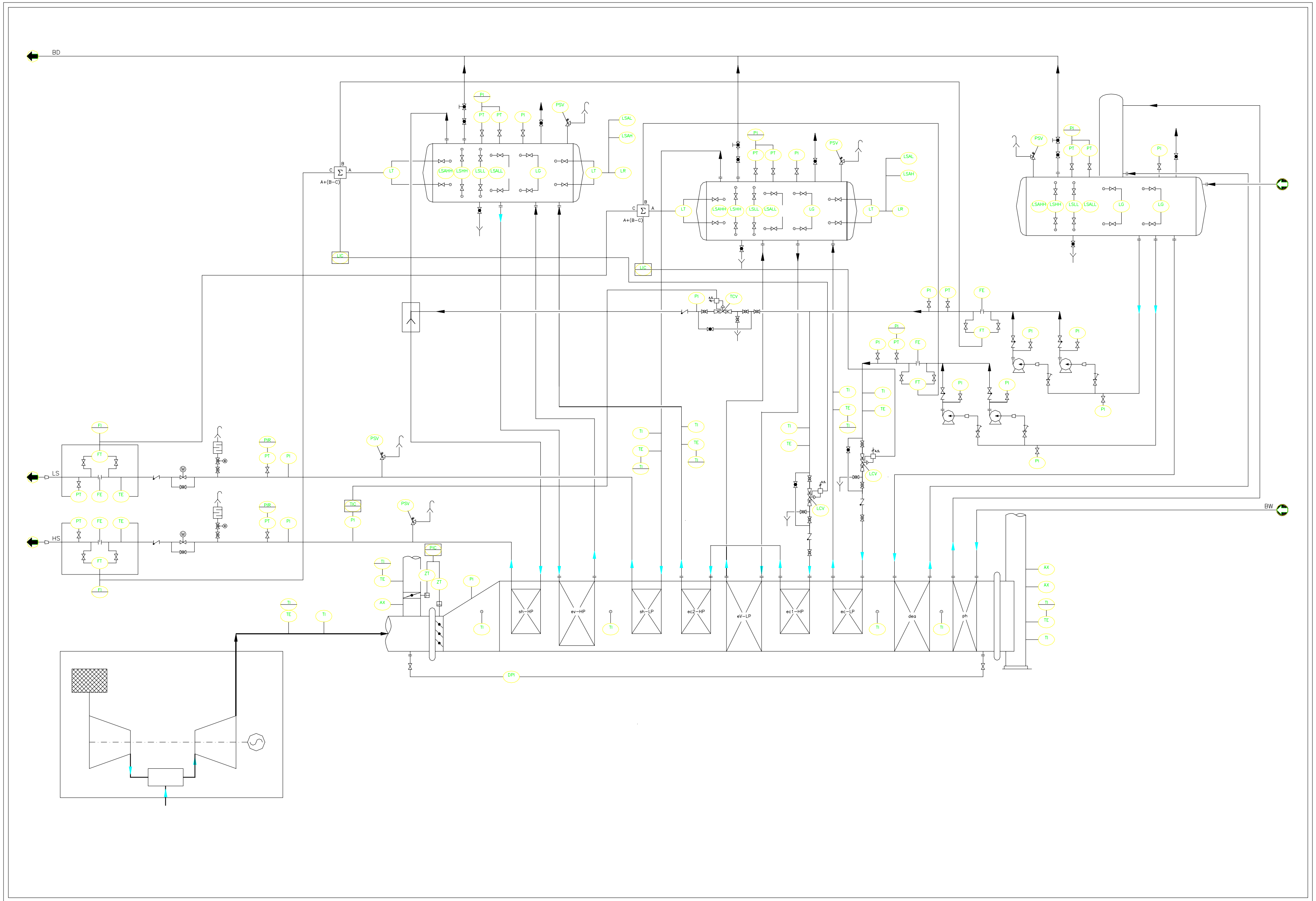

START
START CONDITION ?
START OIL AND COOLING GT SYST.
GT SYST. RUNNING ?
GAS FUEL ?
START GAS SYSTEM
GAS SYST. RUNNING ?
START CONDENSATE SYSTEM
COND. SYST. RUNNING ?
START DEMI WATER
WATER DEMI RUNNING ?
START FEED WATER SYSTEM
FEED WATER SYST. RUNNING ?
START HRSG
HRSG RUNNING ?
START STEAM SYSTEM
STEAM SYST. RUNNING ?
AUX. SYST. RUNNING ?
START GT
READY TO SYNCHR. ?
GT SYNCHRONISM
GT LOADING
IGCC - Control philosophy
AUTOMATIC START SEQUENCE
Plant location Varadero (CUBA)
Unit IGCC Power unit
Drw 4.4.1
Date 15/10/98



IGCC - Control philosophy **AUTOMATIC STOP SEQUENCE**		
Plant location Varadero (CUBA)		
Unit IGCC Power unit	Drw 4.4.2	Date 15/10/98

5. IGCC - P&I diagrams

The process description is showed in drw. 5.0.1.


BD
LS
HS
BW

5.1 IGCC - Measuring equipment

The IGCC measuring equipment have to include the ensuing groups of instruments.

IGCC - Process and Instruments **MEASURING EQUIPMENTS**		
Plant location Varadero (CUBA)		
Unit IGCC power unit	Sheet 1 of 1	Date 15/10/98

Measured variable	**Instruments**	**System and location**
Temperature	Thermo-couples (TC)	Exhaust temperature
Temperature	RTDs	GT bearings Compressor delivery Compressor inlet Seal air supply Starter motor Gas fuel ST bearings Lube oil system Alternator
Speed	Magnetic pick-ups	GT speed ST speed
Control valve position	Position transducers	Gaseous fuel control valve Inlet guide vane GT bleed valves
Pressure	Manometers	Compressor delivery Compressor inlet Compressor air flow Exhaust gas flow Gaseous fuel system losses Δp Lube oil system losses Δp GT air intake system losses Δp Cooling air system losses Δp
Level	Potentiometer sensors	Condenser hotwell De-areating drum LP and HP steam drums
Flame intensity	Flame detectors	Combustor Ignitor torch

6. IGCC - Requirements

The paragraph aims to report the description of preliminary plant set-up, through the requirements of its components.

6.1 Gas Turbine (GT)

The topper power group is carried out sharing *topper* power group output within 4 heavy duty single-shaft gas turbine, for base load service (Fig. 6.1.1). The *GT* power group operate in outdoor installation, equipped with lube oil cooling system, and air cooling system.

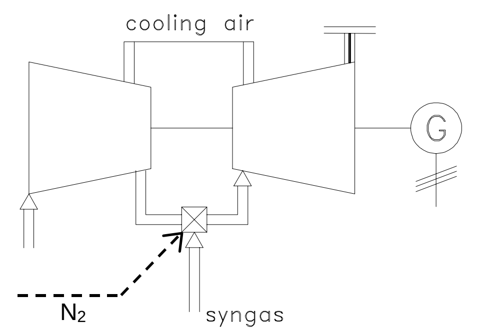


Fig. 6.1.1 - Gas Turbine

The main requirements are listed in Tab. 6.1.1, with reference to group nominal operating condition.

Tab. 6.1.1 - GT main requirements

	Gas turbine	*GT group*
Shaft power **(MW)**	41.65	166.6
Inlet mass flow rate **(kg/s)**	146.8	545.9
Compression ratio β	11.8	–
Air-fuel ratio	15.96	
Fuel mass flow rate **(kg/s)**	7.95	31.8
Bleeded air mass flow rate **(kg/s)**	11.8	47.3
Efficiency η_{GT} %	31.	–
Outlet gas temperature - TOT **(°C)**	539.1	–
RPM	5100	–

The suggested overall dimensions and weights for each gas turbines of topper power group are resumed in tab. 6.1.2.

Tab. 6.1.2 - GT dimension and weights

Dimensions and weight	Data GT	Data auxiliary
Lenght (m)	7.6	8.3
Width (m)	3.2	3.2
Height (m)	3.8	3.8
Weight (kg)	85 000	38 000

6.2 Heat Recovery Steam Generator (HRSG)

The recovery burnerless generator is used for the production of super-heated high purity steam for turbine and process supply. The generator has to guarantee high level of reliability and automation in the complete range of operating conditions.

The *HRSG* nominal operating condition are reported in Tab. 6.2.0:

Tab. 6.2.0 - HRSG nominal operating conditions

	HRSG
Inlet exhaust mass flow rate (kg/s)	586.6
Inlet exhaust temperature (°C)	539.1
Outlet exhaust temperature (°C)	113.8
Exhaust thermal power (MWth)	283.
Heat rate (MWth)	263.2

Is important to note that, along the high pressure line, a saturated water mass flow rate is subtracted from high pressure drum to supply the syngas cooler.

Saturated water mass flow rate to syngas cooler (kg/s)	37.6
Heat rate from syncooler (MWth)	52.6

The cooling mass flow rate evaporates through the syngas cooler and, then, is feeded back for the super-heating.

Additional requirements for the *HRSG* are.

- The steam drums should have sufficient size for start up, shut down and transient conditions.
- The drums must be available to handle sudden pressure drop (2 bar) without unacceptable level fluctuation.
- By-pass stack and the necessary damper system.

- Each by-pass stack have to be equipped with a silencer.
- The sound pressure level under free field conditions 1m from the boiler, must not exceed 80 dB(A).
- Sufficient bolted inspection doors and manholes of the proper size to reach each tube bank and the drum internals, thus permitting the maximum access and visibility to *HRSG* and its equipment.

☯

The main generator components are (Fig.s 6.2.1, 6.2.2, 6.2.3):

- **Pre-heater**,
- **De-areating**,
- **Economizers** (LP and HP),
- **Evaporators** (LP and HP),
- **Super-heaters** (LP and HP),
- **Stack**.

6.2.1 Pre-heater (ph)

Such heat exchanger pre-heats the water before its inlet in de-areating unit (Fig. 6.2.1). The heat exchanger design requirements are listed in Tab. 6.2.1.

Tab. 6.2.1 - (ph) heat exchanger requirements

	Pre-heater
Water mass flow rate (kg/s)	102.7
Water pressure (bar)	1.4
Water inlet temperature (°C)	45.5
Water outlet temperature (°C)	56.8
Heat rate (MWth)	4.9

6.2.2 De-aerating (dea)

The de-aerating unit is designed to carry out two different tasks: the regenerative pre-heating of water; the removal of gaseous components from water. The de-areating unit in the chosen *HRSG* configuration, interact directly with low temperature exhaust flow to produce the necessary saturated steam mass flow rate (see Fig. 6.2.1). The water quality required in de-areating design inlvolve the ensuing limit concentrations:

- **O_2 < 0.02 ppm**
- **CO_2 < 1.00 ppm**

Tab. 6.2.2 reports the de-areating design requirements.

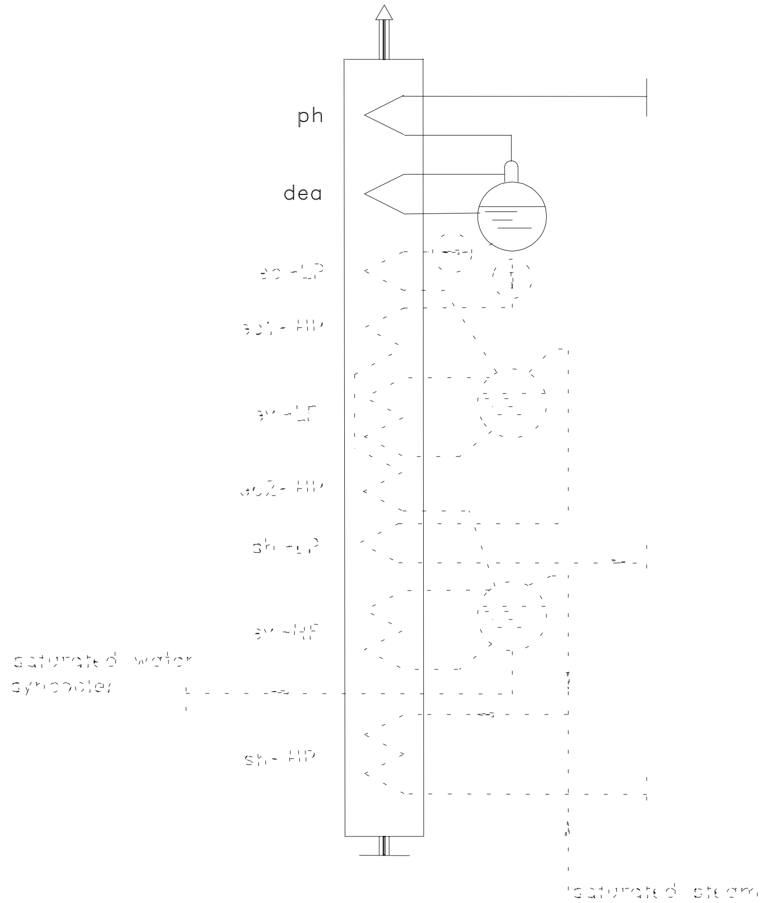


Fig. 6.2.1 - HRSG, pre-heating circuit

Tab. 6.2.2 - (dea) design requirements

	De-aerating
Pressure **(bar)**	1.4
Temperature **(°C)**	108.8
Water mass flow rate **(kg/s)**	102.7
Steam mass flow rate **(kg/s)**	9.99
Heat rate **(MWth)**	22.3

The adoption of de-areating guarantees greater reliability compared to cycle with de-areating condenser, because of:

- the de-areating greater effciency, in off-design conditions (load modulation),
- the de-areating greater effciency in carbon dioxide elimination.

The *HP* and *LP* feed water pumps are supplied by de-areating hotwell, that is another water storage volume in the *HRSG*.

6.2.3 Economizers (ec)

The eco-LP is installed at the end of exhaust path in *HRSG* (low temperature zone), while the high pressure ones (eco-HP) are installed upstream. For such reason two limit phenomena have to be considered in economizers design:

- inlet water temperatures to eco-LP and eco-HP, such temperatures have to be greater enough to prevent cold exhaust gas condensation,
- outlet temperatures from eco-LP and eco-HP, in order to satisfy the approach interval to vaporization temperatures thus avoiding an early vaporization.

In the proposed scheme the eco-HP is interfaced with the eco-LP by the low pressure steam separator.

Tab. 6.2.3 - (ec) design requirements

	Low Pressure - Steam	*High Pressure - Steam*
Water mass flow (kg/s)	6.3	96.4
Inlet Temperature (°C)	108.8	108.8
Outlet Temperature (°C)	144.3	296.8
Pressure (bar)	4.1	82.1
Heat rate (MWth)	0.953	80.9

6.2.4 Evaporators (ev)

The evaporators, ev-LP and ev-HP have the characteristics resumed in Tab. 6.2.4.

Tab. 6.2.4 - (ev) design requirements

	Low Pressure - Steam	*High Pressure - Steam*
Water mass flow (kg/s)	6.3	96.4
Inlet Temperature (°C)	144.3	296.8
Outlet Temperature (°C)	144.3	296.8
Pressure (bar)	4.1	82.1
Pinch-point (°C)	15	15
Heat rate (MWth)	13.4	78.9

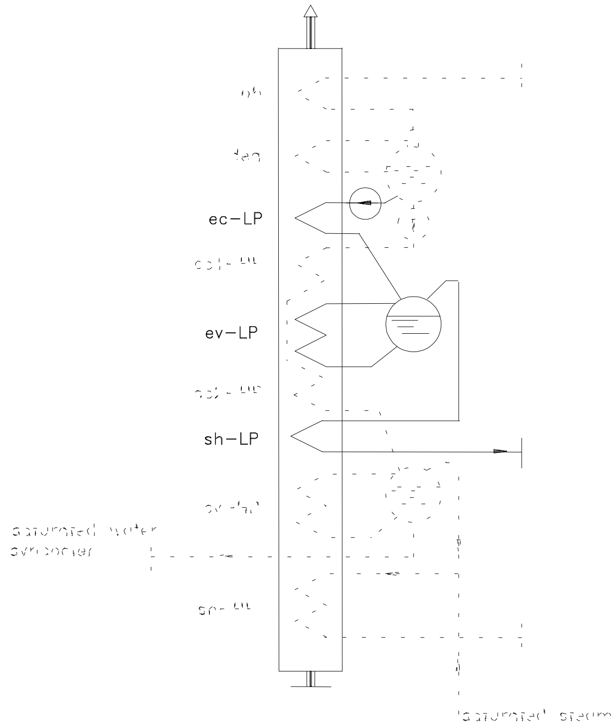


Fig. 6.2.2 - HRSG, low pressure steam circuit

6.2.5 Super-heaters (sh)

The super-heaters sh-LP and sh-HP supply respectively LP and HP steam turbines with constant temperature steam flow rate. Both the super-heating sections have a water injection system to control maximum allowable steam temperature.

Tab. 6.2.5 - (sh) design requirements

	Low Pressure - Steam	*High Pressure - Steam*
Steam mass flow **(kg/s)**	6.3	96.4
Inlet Temperature **(°C)**	144.3	296.8
Outlet Temperature **(°C)**	289.9	519.1
Pressure **(bar)**	4.1	82.1
Approach point **(°C)**	25	20
Heat rate **(MWth)**	1.9	59.8

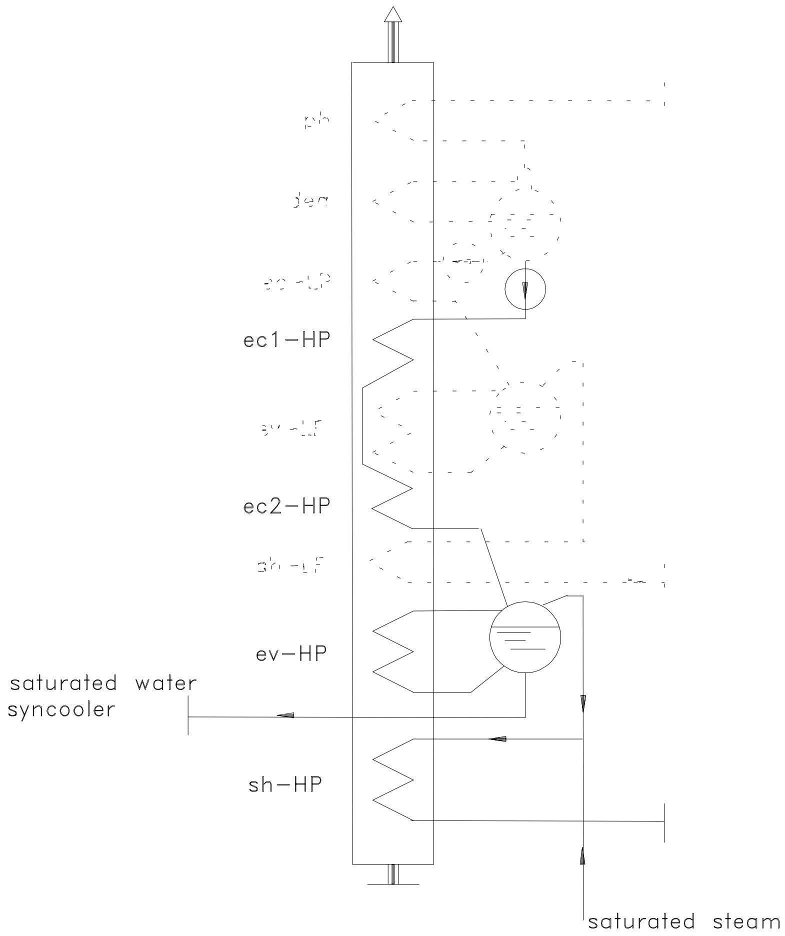


Fig. 6.2.3 - HRSG, high pressure steam circuit

6.2.6 HRSG design data sheet

The *HRSG* thermal design has been carried out by the LMTD method, and the heat transfer surfaces have been considered with finned tubes.

IGCC - HRSG design data sheet **HEAT TRANSFER SURFACES**		
Plant location Varadero (CUBA)		
Unit IGCC power unit	Sheet 1 of 1	Date 15/10/98

HRSG component	**Exchanging surface (10^3 m^2)**	**Exhaust temperature difference (°C)**
Pre-heater (ph)	3.96	out, 113.97 in, 136.89
De-areating (dea)	15.84	out, 136.89 in, 158.22
LP economizer (ec-LP)	1.075	out, 158.22 in, 159.75
LP evaporator (ev-LP)	13.755	out, 159.75 in, 181.06
HP economizer (ec1-HP, ec2-HP)	52.498	out, 181.06 in, 315.75
LP superheater (sh-LP)	1.167	out, 315.75 in, 318.87
HP evaporator (ev-HP)	28.346	out, 318.87 in, 444.85
HP superheater (sh-HP)	33.123	out, 444.85 in, 539.1

Tab. 6.2.6 resumes the dimension and weight data sheet for the *HRSG*.

Tab. 6.2.6 - HRSG dimensions and weights

Dimensions and weight	**Data**
Lenght (m)	30
Width (m)	15
Height (m)	30
Weight (kg)	1 500 000

6.3 Turbine-generator

The steam turbine, with the two bodies (LP and HP) in one shaft arrangement, is directly linked to the electric generator and is used to supply a national electric network. The super-heated high pressure steam mass flow rate is partially used in order to supply all the process high pressure steam requirements. For such reason, the available high pressure steam for back-pressure turbine feeding is reduced to the *87%* of total high pressure steam production. As far as low pressure turbine is concerned, the steam mass flow rate that supplies LP turbine results from the mixing of steam discharged from HP turbine with those coming from the super heating phase of low pressure circuit.

The steam condition at the inlet of each turbine bodies are specified in Tab. 6.3.1.

Tab. 6.3.1 - Steam conditions

	Low Pressure – Steam	*High Pressure - Steam*
Inlet steam pressure (bar)	4.1	82.1
Inlet steam temperature (°C)	289.9	519.1
Enthalpy (kJ/kg)	3051.4	3445.6

The condensation condition corresponds to pressure fixed at *0.1 bar* and temperature *45.5 °C*. It has to be noted, also, that no steam bleeding has been used to supply the de-aerating.

The suggested steam turbine architecture is here described:

- **HP turbine**, with reacting front section,
- **LP turbine**, shared in two condensing bodies,
- **Electric-generator**.

6.3.1 Turbines (HPst, LPst)

The steam turbines must have the characteristics shown in Tab. 6.3.2.

Tab. 6.3.2 - (HPst, LPst) design requirements

	Low Pressure – Turbine	*High Pressure – Turbine*
Inlet superheated steam Pressure (bar)	4.1	82.1
Inlet superheated steam Temperature (°C)	289.9	519.1
Steam mass flow rate (kg/s)	90.2	83.9
Bleeded steam mass flow rate (kg/s)	–	–

Gros shaft power (MW)	119.2

The preliminary overall dimensions and weights for the steam turbine bodies are resumed in Tab.

6.3.3.

Tab. 6.3.3 - (HPst, LPst) dimensions and weights

Dimensions and weight	Data
Lenght (m)	4 ÷ 5
Width (m)	4
Height (m)	4 ÷ 5
Weight (kg)	120 000 ÷ 130 000

6.4 Condenser and water circuit

The set of transformation on operating fluid (steam-water) downwind from the turbines involve the following components:

- **Condenser**,
- **Condensate pump** and **Boiler feed pump**.

6.4.1 Condenser (K)

The refrigerant condenser is a sea water-cooled shell and tube exchanger. Its nominal operating conditions are here specified in Tab. 6.4.1.

Tab. 6.4.1 - (K) design requirements

	Condenser
Pressure (bar)	0.1
Temperature (°C)	45.5
Steam mass flow rate (kg/s)	102.7
Inlet saturated mixture ratio	0.84
Sea water cooling flow rate (kg/s) **(m^3/s)**	7034. 7.
Heat rate (MWth)	245.8

6.4.2 Condenser design data sheet

IGCC – Condenser design data sheet **HEAT TRANSFER SURFACES**		
Plant location Varadero (CUBA)		
Unit IGCC power unit	Sheet 1 of 1	Date 15/10/98
Condenser	**Exchanging surface (10^3 m^2)**	**Cooling water Temperature difference (°C)**
	16.32	in, 20 out, 28

Notes. The condenser design has been carried out by the LMTD method. The heat transfer surfaces have been carried out with finned tubes.