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Design of a decision support system
for combined cycle schemes

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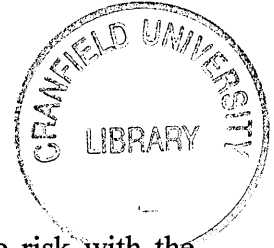
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ABSTRACT

The growing desire for sponsors of power generation projects to share risk with the lenders has promoted the use of computational tools, simulating and evaluating from a techno-economic viewpoint long-term, high-risk projects. Such models need to include reliable engine diagnostics, life cycle costing and risk analysis technique.

This work consisted in designing a Decision Support System (DSS) for the assessment of power generation projects using industrial gas turbines in combined cycle. The software, programmed in Visual Basic in Excel in a windows-frame, runs an external application named Pythia, which has been developed by the Department of Propulsion, Power, Energy and Automotive Engineering at Cranfield University. It can perform gas turbine performance simulations, including off-design conditions, with or without degradation effects providing thus reliable engine diagnostics.

Steam cycle models including different heat recovery steam generator configurations have been developed to simulate steam turbine design and off-design performance.

Plant performance simulation takes into account off-design conditions, part-load governing strategies and degradation effects.

Besides a robust economic mode and a life cycle costing model including maintenance-planning assessments offer a wide range of possible operating and economic scenarios.

The degree of uncertainty relating to technical and economic factors is assessed using normal distributions, and the level of risk is then evaluated using a risk analysis technique based upon the Monte Carlo method.

The DSS provides all sorts of charts and techno-economic figures in order to support the decision making through an effective user-friendly window-oriented interface.

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NOTATIONS

ARR	Accounting Rate of Return
C	Cash
CC	Combined Cycle
CF	Cash flow
CHP	Combined Heat and Power
COE	Cost Of Electricity
C _p	Heat Capacity (kJ/kg.K)
CW	Cooling Water
DCF	Discounted Cash Flow
DCFR	Discounted Cash Flow Rate
ε	Effectiveness
ECC	Equivalent Capitalised Cost
F	Shape Factor (-)
FBC	Fluid Bed Combustion
FCR	Fixed Charge Rate
FV	Future Value
GSA	Gas Supply Agreement
H	Enthalpy (kJ/kg)
HP	High Pressure
HRSG	Heat Recovery Steam Generator
IGCC	Integrated Gasification Combined Cycle
IGV	Inlet Guide Vane
IP	Intermediate Pressure
IPP	Independent Power Producer, Independent Power Project
IRR	Internal Rate of Return
LAC	Levelised Annual Cost
LCC	Life Cycle Cost
LevFCR	Levelised Fixed Charge Rate
LevCOE	Levelised Cost of Electricity
LP	Low Pressure
MTBF	Mean Time Between Failure
\dot{M}	Mass Flow (kg/s)
N	Number of passes
NGV	Nozzle Guide Vane
NPV	Net Present Value
NTU	Number of Transfer Unit
O&M	Operating and Maintenance, Overheads and Maintenance
P	Pressure (bar)
PFB	Pressurised Fluid Bed
PPA	Power Purchase Agreement
PV	Present Value

PWRR	Present Worth Revenue Requirement
\dot{Q}, q	Heat rate (kW)
ROI	Return On Investment
S	Surface (m ²)
SCC	Spencer, Cotton and Cannon
T	Temperature (°C or Kelvin)
TET	Turbine Entry Temperature
TFR	Equivalent Throttle Ratio
U	Heat Transfer Coefficient (W/m ²)
V	Velocity (m/s)
\dot{V}	Volume Flow rate (m ³ /s)
v	Specific Volume (m ³ /kg)
VIGV	Variable Inlet Guide Vane
η	Efficiency (-)

Subscript:

i	Year
k	Environment
n	Number of years
r	Discount Rate (%)
t	Tax Rate (%)
w	Water

INTRODUCTION

The electricity supply industry is being restructured all over the world. As a consequence, the manner in which investment decisions in power generation schemes are made is changing in some fundamental ways.

To meet the challenge of deregulation and customer demands for free competitive market, the electricity utility industry, which has to face a dramatic increase in the number of Independent Power Projects (IPPs) has to position itself as low-cost producer.

This least-cost policy increases the level of risk attached to any power generation scheme and force planners to assess any source of uncertainty in order to increase the level of confidence, necessary for ensuring the success of decision making.

Moreover decision making nowadays implies the utilisation of several computer programs or packages in order to assess and compare different projects or investment alternatives, under uncertainty if required.

Time is a continuous process and as each second passes, new factors enter into whatever it is that you are trying to do or decide on. Therefore gathering and processing information is essential to lower the risk of making wrong decisions in a field where making business decision is essential to survive.

A general approach for quantifying technical and financial risks of long-term capital investments relies on simulation techniques such as Monte Carlo method. Rather than just providing a safety factor to guard against unfavourable scenarios, this simulation approach allows the planners to define a consistent level of confidence in regard to achieve quantified project objectives.

The main objective of this work is to design a computational tool, known as Decision Support System (DSS) to appraise technically and economically combined cycle schemes.

This software called Hestia, Greek home goddess is aimed to manage different tasks in totally different fields.

In the technical field, Hestia must be versatile enough to tackle different plant configurations, types of application, off-design and deterioration conditions, and finally maintenance planning strategies. This should ensure robust and reliable performance results and accurate plant availability calculation. This is essential to correctly assess production costs and incomes from electricity sale.

In the economic field, the modelling must include basics from engineering finance and project accounting. Different economic techniques should provide all necessary key parameters that are commonly used by industry to support the decision making.

Finally as far as programming is concerned, usability and user-friendliness are the main goals to achieve. Therefore a window oriented programming language, i.e. Visual Basic in Excel 7.0, is regarded as a good choice to design such a tool, which must be used by novices without tedious and complex learning. Besides new programmers should be able to quickly up-date or even up-grade the code or the user-ineterface.

Chapter 1

WHY A DSS FOR COMBINED CYCLE SCHEMES

1.1. World energy demand and future trends

In the report *Energy 2020*, commissioned by PowerGen UK, researched by Oxford Economic Research Associates Limited which has been partly published in *Power Plant Technology* (1997), it has been predicted that global energy demand could double by 2020, with demand for electricity potentially trebling. This would lead to a \$3 trillion investment by the year 2020.

Figure 1 and Figure 2 show the upward trend of the energy consumption world-wide in time. The world energy consumption has been multiplied by 12 since 1900 up to 1993, with an increasing rate of 4 points (index 1900) per year between 1900 and 1950, and of 20 points between 1971 and 1993.

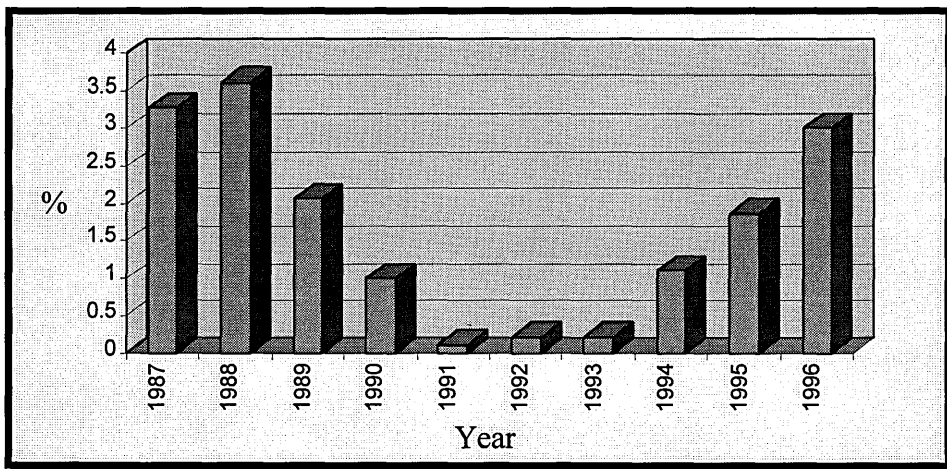


Figure 1: Annual Rate of Growth in Energy Consumption

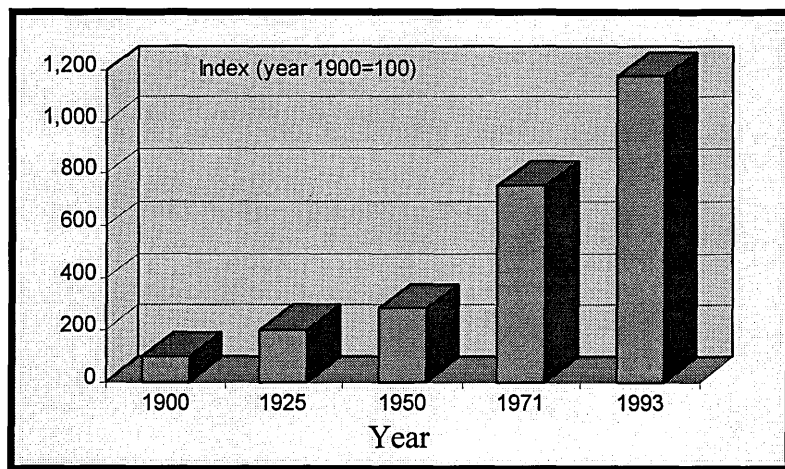


Figure 2: World Energy Consumption

Moreover if this upward trend is true world-wide, it is important to notice that the regional shares of the world's energy consumption (Appendix A) is not uniform, since

Indonesia has the fourth largest population in the world and does not even belong to the top 8 largest energy users. In Stein, (1996), it has been agreed that the highest growth in capacity will be in Southeast Asia and the Pacific Rim. Figure 3 shows that 53% of generating capacity addition by the year 2004 will take place in Asia against 12% in North America, which is the largest energy user.

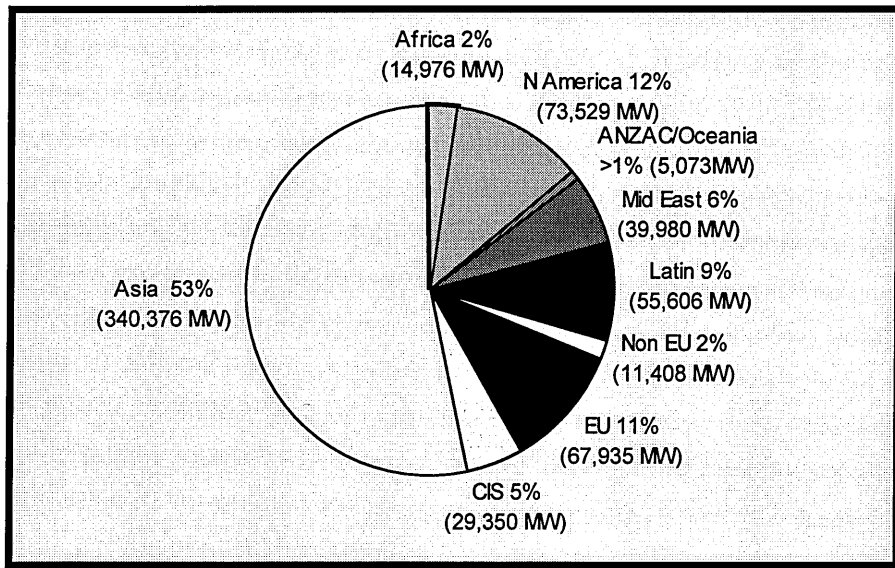


Figure 3: World-wide Generating Capacity Additions by Region: 1995-2004

Therefore it is clear that power generation schemes are becoming numerous and are dealing with very high capital costs, sometimes in locations where economic and political stability can not be achieved at the same time, increasing the risk attached to these large long-term investments.

1.2. Fuel Choice

Once the regional growing energy demand has been assessed, the second step is to recognise the benefit of using fossil fuels rather than nuclear energy. This is usually not an issue since for economic and technological reasons, nuclear power plant schemes are not competing against fossil fuel fired plant. On the other hand, the choice of burning coal, liquid fuels or natural gas is a real issue that planners need to assess.

Figure 4 shows that coal remains a major actor in fired power plant. Subcritical or supercritical Pulverised Fuel Combustion (PFB), atmospheric-pressure or pressurised Fluid Bed Combustion (FBC), and Integrated Gasification Combined Cycle (IGCC) provide different options for generating power, increasing the flexibility of burning coal with regard to specific constraints such as fuel flexibility, availability, capital cost, environmental performance and so on. This flexibility is well illustrated in OECD/IEA (1996).

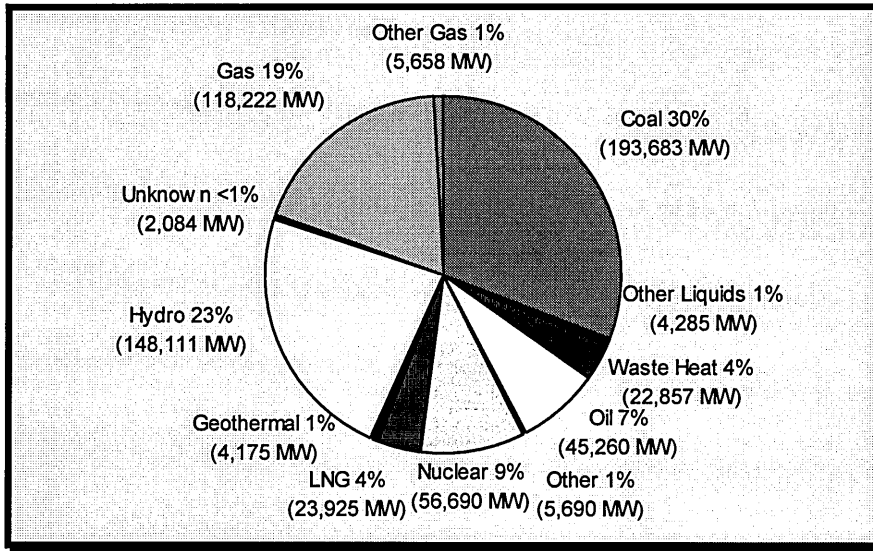


Figure 4: World-wide Generating Capacity Additions by Fuel: 1995-2004

Figure 4 shows also that the ratio covered by liquid fuels and gas is roughly the same as the ratio for coal, namely 30%.

This means that 50% of future power plants based upon fossil fuels will burn liquid fuels and natural gas, and then could be assessed by the DSS.

1.3. Combined Cycles and Combined Heat and Power plants

If 50% of fired power plant schemes are based on natural gas and liquid fuels, including crude oils, it is important to know the ratio of them which will use combustion turbines with or without steam turbines.

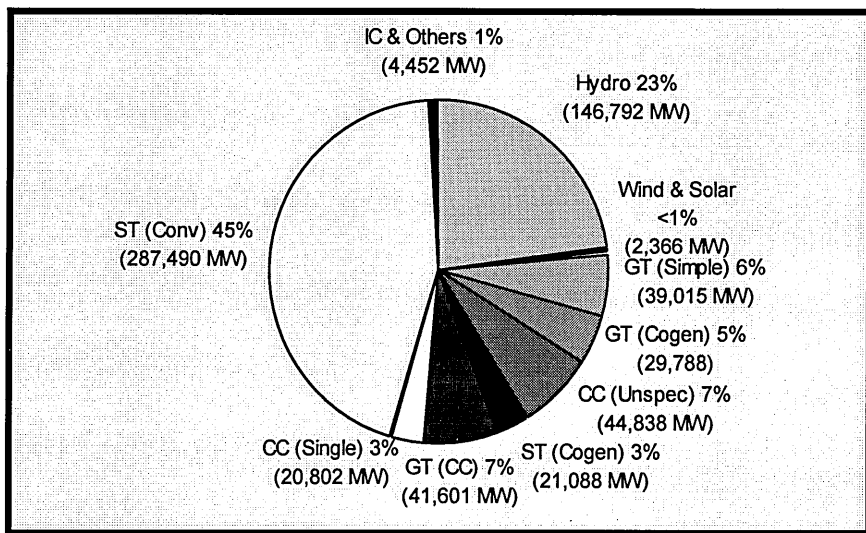


Figure 5: World-wide Generating Capacity Additions by Technology: 1995-2004

Figure 5 shows that about 30% of generating capacity addition will be based on combustion turbines technology in open or combined cycles. This means that stationary gas turbines are key actors in world power generation market, especially in developing countries, which are attracted by relatively low capital costs and fuel availability.

On top of that national acts like the United States Public Utilities Regulatory Policy Act (PURPA) of 1978 significantly changed the cogeneration market. The aim of PURPA is to promote conservation and effective utilisation of energy resources. In that perspective Combined Heat and Power (CHP) achieving sometimes more than 90% thermal efficiency is seen as one means to achieve this goal. Cogeneration also enables plants to raise extra revenue from electricity sold to the grid if not utilised by the plants themselves. Besides in some developed countries such as France, where power generation is still nationalised, the national grid owned by the public utility Electricité de France (EDF) had to buy the electricity from Independent Power Producers (IPPs) running CHP plants, leading to excessive misuses for profitability reasons. A recent law now obliges IPPs to clearly demonstrate that the main purpose of CHP is first the production of the heat and then power generation. Therefore the ratio between electricity and heat production must be greater than one third.

Combining gas and steam cycles is clearly the best option for energy savings but also for increasing the power output and the overall efficiency of the plant. Besides the choice of natural gas as fuel in combined cycle enables utilities to achieve high efficiency up to 60% with relatively low emissions. The dash for gas in UK and in USA is the proof that open and combined cycles are considered as a major actor in power generation market.

1.4. Privatisation and Liberalisation: IPPs

The electricity supply industry is world-wide being privatised. In some countries like United Kingdom, liberalisation is the next step of the deregulation policy. Privatisation means that anybody can be allowed to produce and sell power to the grid. Liberalisation will enable people to choose power generation company for their electricity supply. Similarly to telecommunication companies, some electricity suppliers must now introduce a new marketing dimension to their economic policy. Retail price advertisement and environment-friendly power generation could become new issues, which could lead to changes in industrial strategy. Capital cost could become the key issue if a very short payback was considered. But in all cases life-cycle cost remains the most critical issue for power plants, especially when the fuel price is rather uncertain in far future.

Privatisation has also promoted independent power projects (IPPs) increasing the competition, meaning that the potential growth for business in power generation industry is now limited. Smith (1995) proposed the following definition to characterise an IPP: *"an IPP is a private generation station that is owned by parties that are partially or fully independent of a local or national electric utility."*

As IPP can be single business, financing and risk sharing are major issues. It is certainly true when the contractual structure of the IPP is complex. Smith (1995) proposed a typical contractual structure, illustrated in Figure 6.

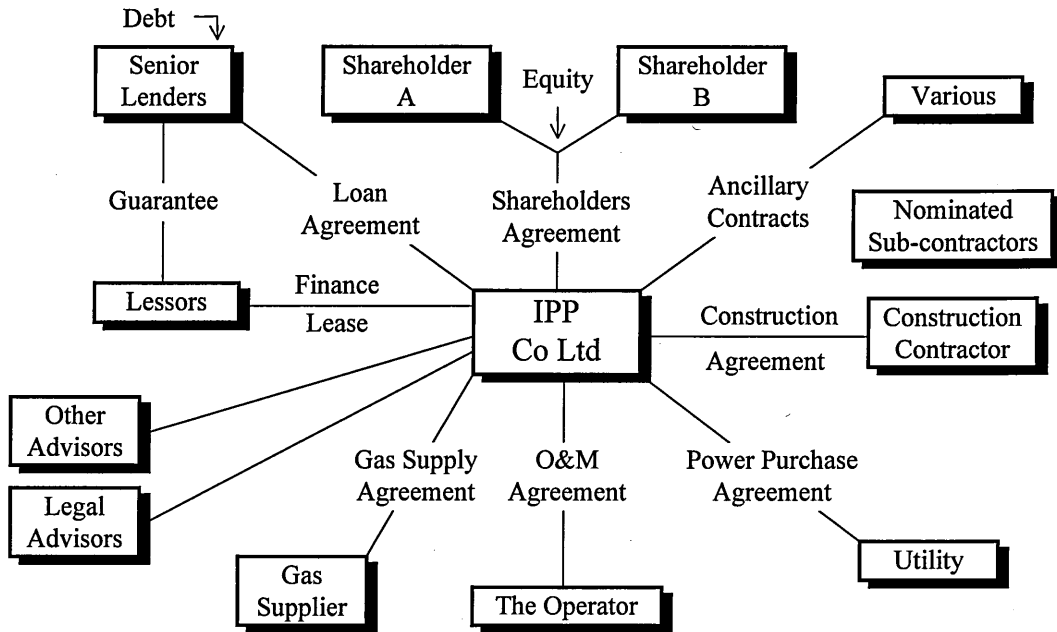


Figure 6: Contractual structure of a typical IPP

Different contracts have to be carefully negotiated.

Firstly the electricity purchaser must properly execute the Power Purchase Agreement (PPA). The PPA is the most important document in an IPP contract and requires long and difficult negotiations. This is mainly due to the fact that the PPA outlines the terms under which the majority of the project cash flows will be realised.

The second contract is the Gas Supply Agreement (GSA). Usually it covers at least the period required to repay the loans. The GSA will provide to the planners the pricing and indexation of the fuel, enabling them to price electricity production costs. It is of prime importance to get a good and long-term GSA in order to reduce the level of risk attached to uncertainty in production cost including fuel.

Construction contract and operation and maintenance contract are the last but not the least contracts to well assess and negotiate. Overrun in construction time and plant delivery can dramatically delay the payback of the project. Subcontracted O&M contracts have to be carefully assessed by the lenders for loan recovery concerns. Plant and performance degradation can reduce the revenue stream and then dramatically affect shareholders' returns and long-term debt repayments. In all cases, key players of IPP contracts in order to appraise feasibility and profitability patterns under given levels of uncertainty perform risk analyses.

A DSS designed to assess combined cycle schemes from a technical and economic viewpoint provides all graphic and numerical supports that decision making under

uncertainty requires. Such computational tool removes the tedious work of assessing plant performance and running economic and risk analyses in separate ways, requiring different software which have not been developed to answer the real set problems and questions that one company in particular could face up.

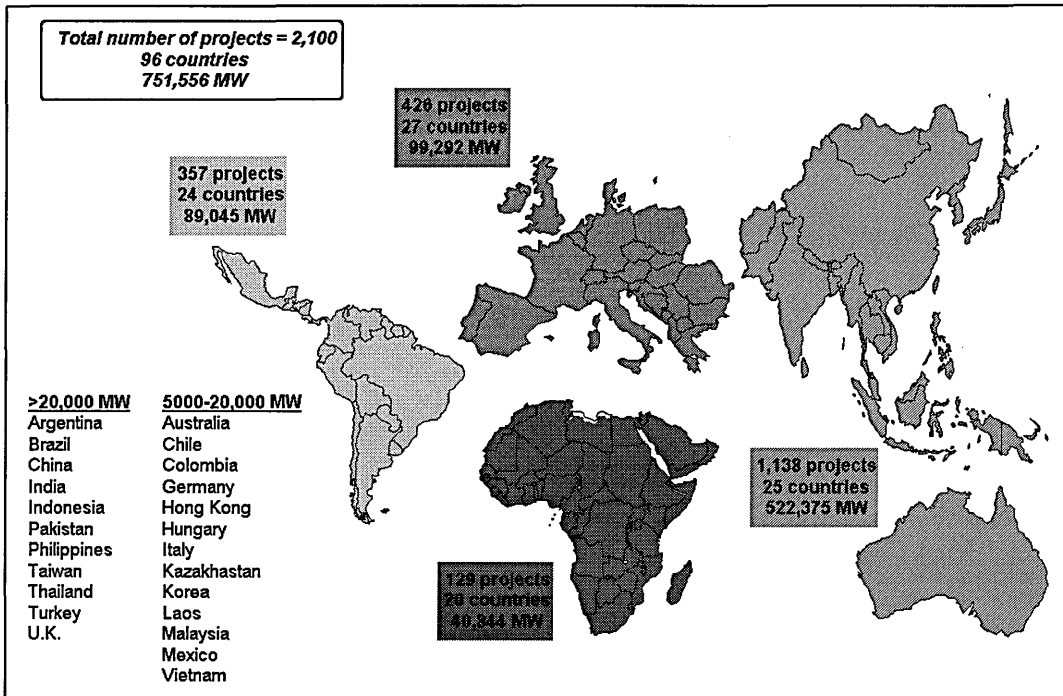


Figure 7: The Global Power Generation Market (exc. North America)

Figure 7 shows that the number of IPPs is very high and is world-wide distributed. It is clear that the 1,138 IPPs in Asia representing more than 50% of the total number of IPPs world-wide may have different requirements than the IPPs run in Europe for instance. Weather, fuel availability, fuel supply and currency stability could be the major concerns for East Asia where as West Europe is more likely to be concerned by pollutant emission and least-cost planning.

Considering the popularity of natural gas for power generation in open or combined cycles and considering the increasing number of IPPs world-wide which do not have the technology and resources for going to nuclear power, it is clear that such as a DSS represents a valuable tool for this growing industry, which requires more and more supports for decision making under technical and economic uncertainties in order to keep a good level of confidence necessary for successfully negotiating PPA and GSA contracts.

Chapter 2

ELEMENTARY THERMODYNAMICS

2.1. Introduction

Thermodynamic studies of thermal power plants are generally carried out in order to assess and maximise the plant efficiency, i.e. the ratio between the useful work and the chemical energy provided by the fuel. The aim of this chapter is to briefly recall basics of Carnot efficiency, Joule cycle and Rankine cycle in order to cover the main parameters, which are used to optimise the overall efficiency of the plant by modifying the different cycles.

2.2. Carnot Efficiency

The Carnot efficiency can be defined as the maximum efficiency of an ideal thermal process as illustrated in Figure 8:

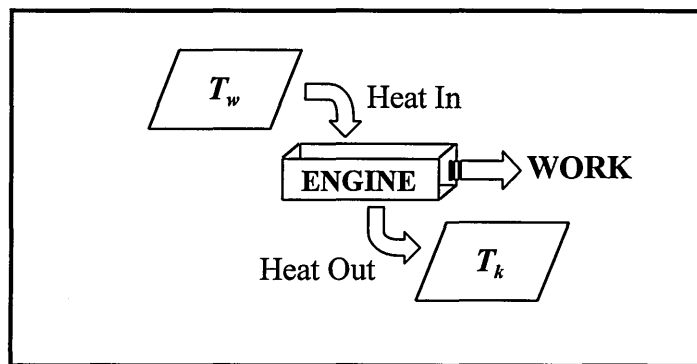


Figure 8: Carnot Efficiency Concept

$$\eta_c = \frac{T_w - T_k}{T_w} \quad (1)$$

Where:

η_c = Carnot efficiency

T_w = Temperature of the energy supplied

T_k = Temperature of the environment

In real processes, efficiencies are always lower because of energetic losses (radiation and convection) and exegeric losses (entropy), but it is clear that the higher T_w is, the better the Carnot efficiency. Besides if T_k is minimised, the efficiency is maximised.

Combined cycles provide a good trade-off since by nature no single cycle can make both improvements to an equal extent. Therefore it is relevant to combine both cycles i.e. a combustion turbine cycle with a high process temperature and a steam cycle with a good cold end as illustrated below:

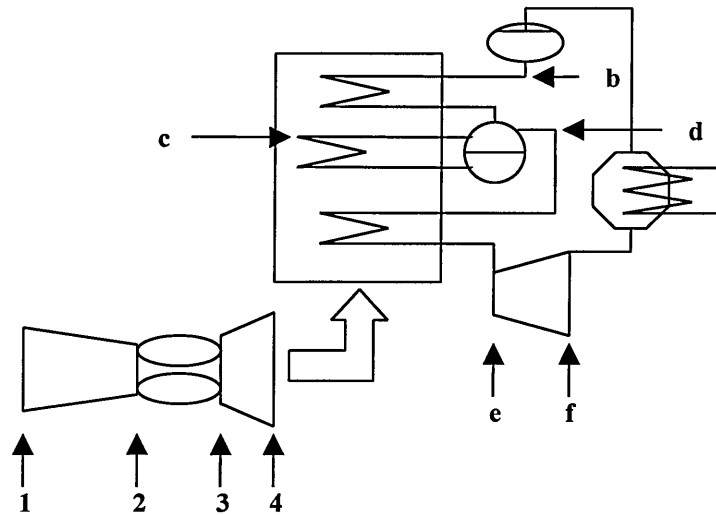


Figure 9: Cycles Notations

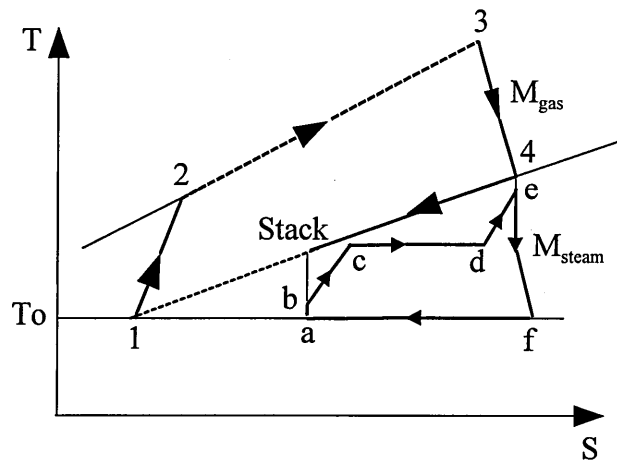


Figure 10: Open Circuit/Closed Cycle Combined Plant

This principle is clearly underlined in Kehlhofer (1991) and can be illustrated in Table 1.

In Appendix B are summarised the different pattern of these four cycles into T-S Diagram.

Table 1: Thermodynamic Comparison of GT, ST and CC Power Plants

	Gas Turbine	Steam Power Plant with reheat	Steam Power Plant without reheat	Combined-Cycle Power Plant
Average Temperature of the heat supplied (K)	950-1000	640-700	550-630	950-1000
Average Temperature of exhaust heat (K)	500-550	320-350	320-350	320-350
Carnot Efficiency (%)	42-47	45-54	37-50	63-68

2.3. Joule Cycle

To simplify the matter, a close cycle with a single perfect gas of constant specific heat capacity and with some irreversibility (compressor and turbine efficiencies, etc.) will be dealt with. Figure 11 gives the T-S pattern of the cycle:

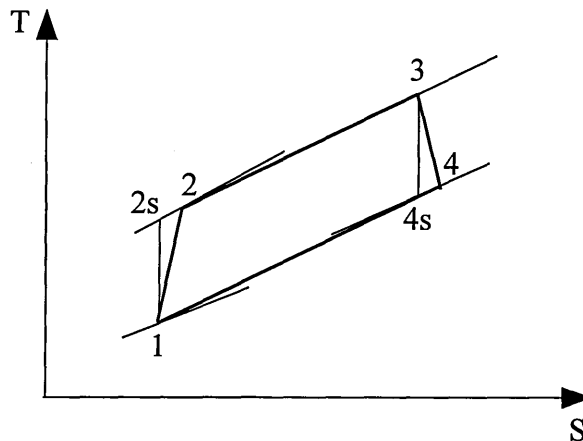


Figure 11: Simple Joule Cycle

T_3 is most commonly known as the Turbine Entry Temperature (TET).

According to the T-S diagram we can define isentropic efficiencies as follows:

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} \text{ For the compressor} \quad (2)$$

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}} \text{ For the turbine} \quad (3)$$

In Horlock (1992) it has been shown that there is a trade-off in the engine performance optimisation procedure. But it is clear that one way of increasing the efficiency could be

the increase in TET or the reduction in component irreversibility by reducing flow distortion and friction losses.

There are many ways of modifying this type of cycle in order to increase the thermal efficiency or/and the power output or both.

2.3.1. Regenerative cycle

The first concept is the regenerative cycle. This cycle is based on the assumption that the compressor delivery temperature T_2 on the T-S diagram is sufficiently lower than T_4 to introduce a heat exchanger in order to firstly increase the combustor inlet temperature and to secondly lower the temperature of heat rejection as illustrated in the following figure:

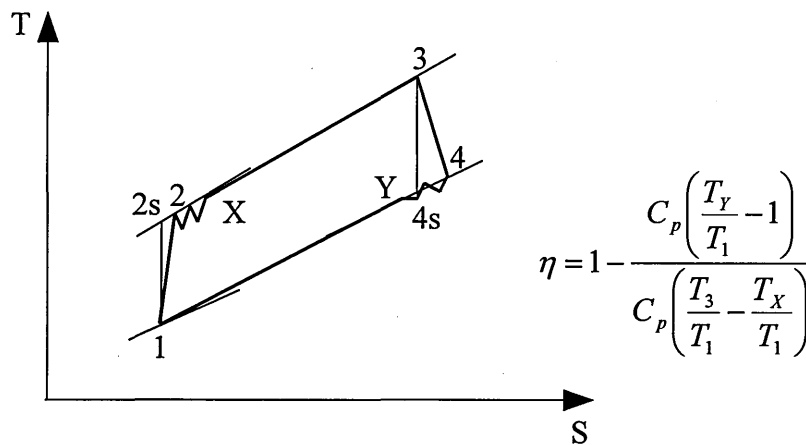


Figure 12: Regenerative cycle

In that case, the fuel consumption is reduced but the availability of exhaust gas is less if a bottoming cycle (steam cycle) is fitted to the Joule cycle. Moreover Horlock (1992) also showed that the regenerative cycle is very sensible to pressure ratio changes, since heat exchanger effectiveness is function of the turbine exhaust temperature, which mainly depends on the turbine pressure ratio. As quoted above there is even a limiting pressure ration at which heat exchange is not possible.

2.3.2. Reheating and Intercooling concepts

These two concepts are illustrated in Figure 13. They are based upon a very simple observation. Compression is always more effective with cold air where as expansion through the turbine is more effective with very hot air. This leads to consider cooling down the compressor mass flow using a heat exchanger, and heating up the turbine mass flow by an additional firing unit, both being located between LP stages and HP stages.

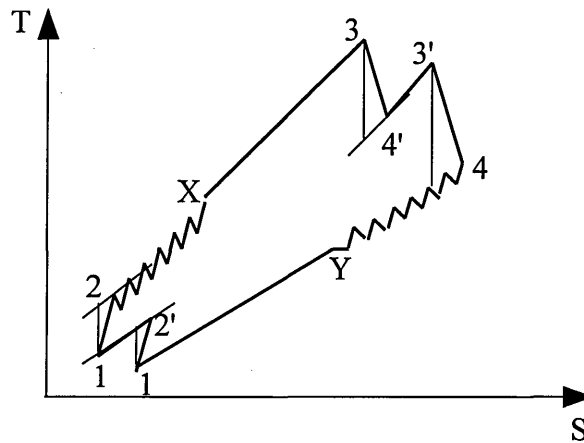


Figure 13: Reheated and intercooled cycle

Intercooling obviously will increase compressor efficiency and thus increase useful work of the turbine (power output) due to a reduction in compressor work. But cooling down the compressor mass flow will affect the compressor delivery temperature. A lower T_2 means that the amount of fuel to be burnt to raise the gases up to T_3 will be greater. This is a penalty that users have to carefully assess in the calculation of the overall production costs.

Reheating has the same effect as far as fuel consumption is concerned as intercooling. But the utilisation of reheat can dramatically increase the power output and provide a very high exhaust temperature, which can be beneficial in combined cycle. Moreover, reheating can be one means of increase the efficiency of a CC plant running at part-load conditions, since it prevents the Heat Recovery Steam Generator (HRSG) from suffering from low gas turbine exhaust temperature.

2.3.3. Fuel preheating

Nowadays it is very common to design CC and CHP plants implementing fuel-preheating process. Small amount of hot water extracted usually from economiser(s) end of the HRSG is used to preheat the fuel at quite high temperatures. Natural gas is expected to be heated up to 300°C in advanced, even if thermal stability is almost reached.

The gain in thermal efficiency is enough to implement it, despite the additional cost of a heat exchanger and small extraction of water from the steam cycle.

In conclusion, versatility of the Joule cycle makes the gas turbine a very flexible source of power and heat generation system. This versatility is provided by the DSS through a program called Pythia that will be presented in a following chapter.

2.4. Rankine Cycle

The Rankine cycle is often called bottoming cycle due to its location in the T-S diagram compared with the Joule Cycle characterising the gas open circuit (cf. Figure 10). The Rankine cycle is the closed water/steam cycle, which occurs into the HRSG, the steam turbine and the condenser as illustrated in Figure 14.

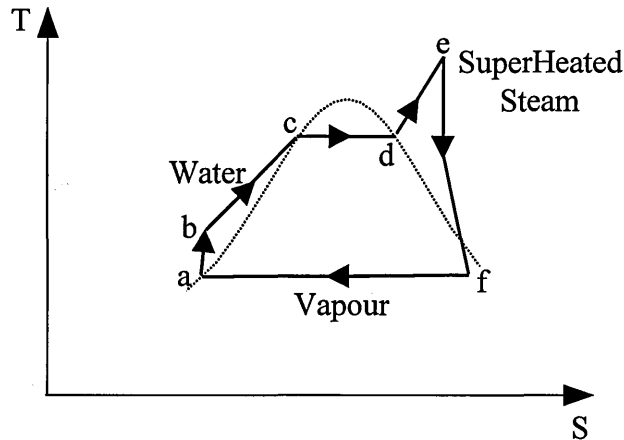


Figure 14: Basic Steam Cycle Temperature-Entropy Diagram

The water/steam path is the following one:

- a→b : deaeration and pumping of feeding water
- b→c : economiser
- c→d : evaporator
- d→e : superheater
- e→f : steam turbine
- f→a : condenser

In a practical steam cycle, some irreversibilities into components such as pumps, HRSG, turbine and condenser modify the Rankine cycle. Besides in pure combined cycles, due to deaeration, water makeup is necessary to maintain the level of water/steam into the circuit and usually that takes place after condensing process. In CHP, the makeup is obvious and can require a huge water source in order to keep the steam production and consumption constant.

Similarly to the Joule cycle, there are several possibilities for increasing power output and the efficiency in the steam cycle.

2.4.1. Reheating

Figure 15 shows the improvement of reheating the steam between the HP and the LP turbines. Besides to the efficiency and power output improvement of the steam turbine,

reheating extracts more heat from the exhaust gas through the HSRG reducing stack temperature and thus increasing Carnot efficiency.

Reheating is now becoming a basic addition of CC plant improvement when a three pressure CC is considered. Double reheat has not been proven as effective as single reheat in CC, but can be applied in CHP schemes for particular steam requirement reasons.

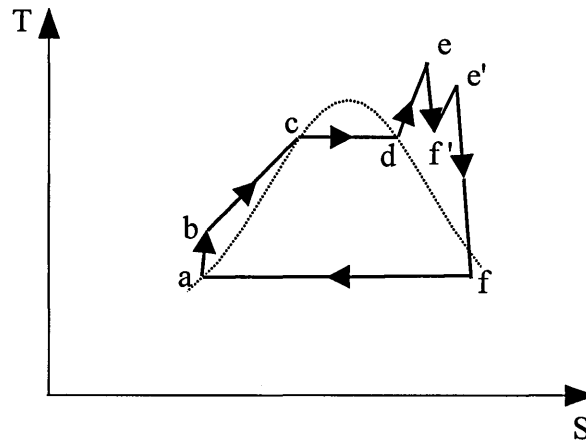


Figure 15: Cycle with reheat

2.4.2. Regeneration

As illustrated in Figure 16, it is possible to extract a small amount of the steam during its expansion in the steam turbine in order to heat up the feed water before it enters the HRSG. Regenerative feed heating in CC and in CHP usually is coupled with an additional economiser or evaporator in the deaeration process.

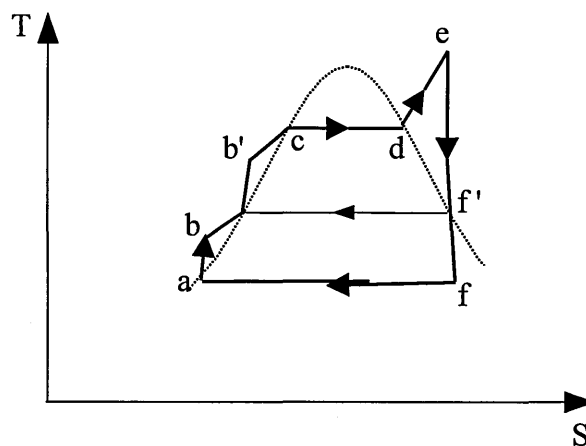


Figure 16: Regenerative cycle

2.4.3. Condenser pressure

The steam back pressure for a condensing steam turbine is of paramount importance. Firstly it sets the level of moisture of the steam when leaving the steam turbine, and for corrosion concerns, this level is usually kept less than 12 %. Secondly as Figure 14

shows, lowering the condensing line in the T-S diagram will increase the power extracted from the steam turbine. The pressure into the condenser is therefore very low, about 50 mbar. Unfortunately the condenser pressure greatly depends on cooling water conditions which can vary from a season to another one if extracted from a river for instance. On top of that a small condenser pressure variation dramatically affects the overall plant efficiency obliging CC owners to seriously consider cooling system efficiencies in off-design conditions.

2.4.4. Multiple pressure levels

When implementing a single pressure CC or CHP, the stack temperature is usually between 180 and 150 °C meaning that heat utilisation from the gas turbine(s) is neither energetically nor exergetically optimum. According to Carnot efficiency definition one way of improving plant performance would be to achieve the lowest stack temperature possible. Therefore by multiplying the number of pressure levels at which steam is raised, the heat recovered by the HSRG is greater. The high pressure system recovers heat at high temperatures whilst the low pressure does so at low temperatures. This simple concept is applied for different types of CC and CHP configurations involving one, two or three pressure levels into the HRSG. Figure 17 shows that the power extracted from the steam turbine, now divided into different pressure sections is much greater and so is the overall CC efficiency.

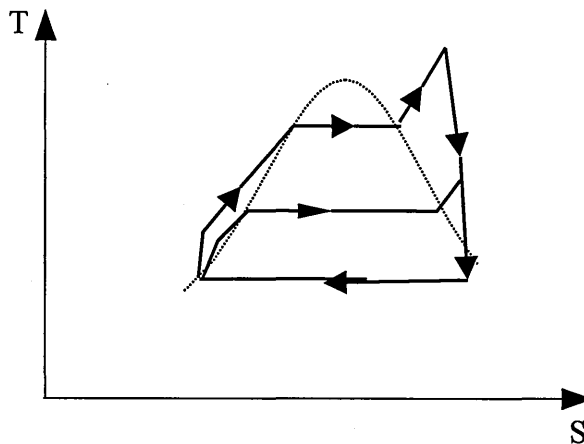


Figure 17: Dual-pressure parallel

2.4.5. Supercritical steam

The main idea of supercritical cycle is to reach pressures such as 220 bars and over in order to avoid condensation of the steam when expanding through the turbine, as illustrated below.

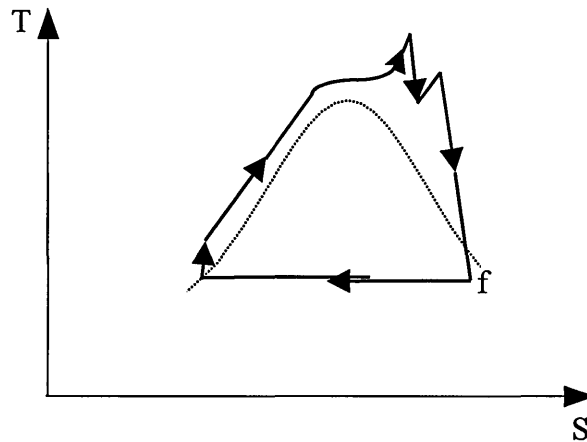


Figure 18: Supercritical cycle

Such a cycle will increase the overall performance of the plant by about 2% and will require a HRSG totally different than the conventional one, since there is no well-defined transition between water and steam and thus no need of drums any more. The type of HRSG used in supercritical conditions is called *once-through boiler*.

Very few plants based on supercritical steam technology are running and with the launch of F and H technologies in gas turbines reaching more than 60% efficiency in combined cycles, supercritical CC schemes are not planned in a near future.

2.5. Conclusion

Similarly to the Joule cycle, which characterises the gas path, the Rankine cycle can be implemented in very different ways involving different types of equipment and technologies. The versatility of the Joule and Rankine cycles makes combined cycles very attractive since it can always fit any specific needs and requirements in terms of power output, efficiency, capital cost, etc.

This configuration versatility in addition to high fuel flexibility increases the number of possible CC schemes that can be implemented. However the performance maximising process due to high competition between major power plants manufacturers has filtered the number of proposed CC plants by keeping the most advanced ones. Manufacturers such as GE Power Systems recognised that same configurations are always proposed with or without additional options in a similar range of power output.

Chapter 3

COMBINED CYCLE MODELLING

3.1. Configuration Selection

3.1.1. Introduction

As quoted in the previous chapter, manufacturers have ranged CC plants into power output ranges. Three pressure with reheat and fuel preheating is the usual layout of plant powered by >100 MW class engines such as the GE Frames 7 and 9.

Depending on the type of application CC or CHP, and on the power output, single or multiple pressure levels can be selected. CHP schemes are usually based on a single or dual pressure HRSG with supplementary firing. Heat requirement usually is perfectly achievable by a single pressured steam or by the combination of HP and LP steams. Hence middle range CC plants are run by two pressures HRSGs, whilst three pressures configuration are applied for heavy-duty CC plants.

Following meetings and discussions with some CC plant manufacturers and CC plants owners, it has been recognised that a DSS for economic appraisal does not require a generic design and simulation program such as Gate Cycle or GT-Pro, which can generate any type of plants, since most of the future plants to be sold will implement roughly same up-to-date technologies for performance maximising reasons, limiting thus the number of configurations proposed by manufacturers to electricity utilities and IPPs. In those conditions, the DSS proposes set configurations (Figure 19) which should satisfy more than 80% of plant design requirements.

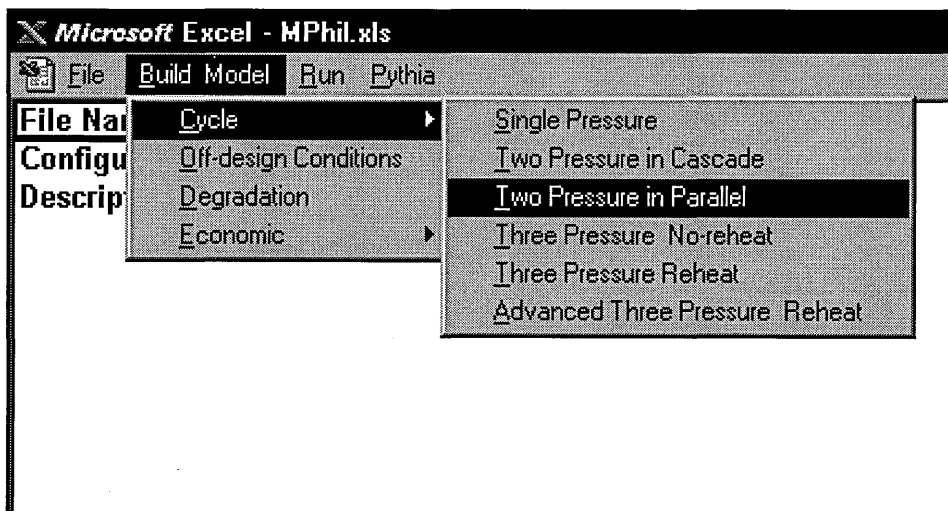


Figure 19: Build-In Cycles

3.1.2. Plant Layout options

When the type of CC has been selected through the menu illustrated in Figure 19, the user can set options that will personalise the plant layout. Figure 20 shows the type of

options provided in the DSS. The following choices will determine very important parameters, such as the configuration of plant units if more than one.

If the plant is not powered by single shaft STAG CC, i.e. if the gas turbine, the steam turbine and the generator are not mounted on the same shaft, several engines in parallel can be fitted to several HRSGs and a single steam turbine or be fitted to a single HRSG and a single steam turbine. As shown on the right hand-side box in Figure 20, these configurations are named: 1-1-1, 2-2-1, etc. to illustrate, firstly the number of gas turbines, secondly the number of HRSGs and finally the number of steam turbine, put together to make what the DSS calls a "plant unit".

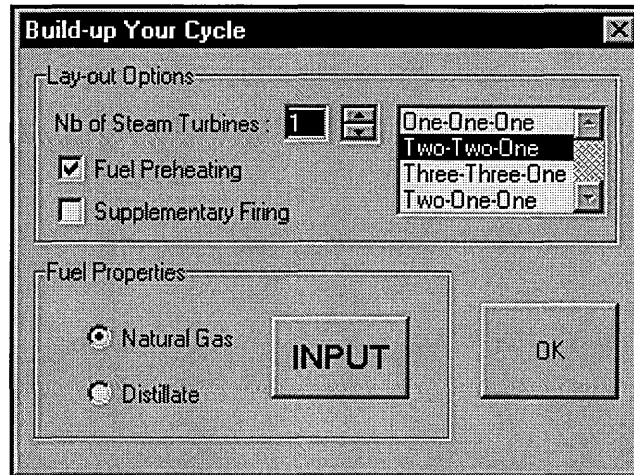


Figure 20: Plant Options Menu

Since each configuration proposed by the listing box always owns a single steam turbine, one way of setting up the number of plant units, is to input the number of steam turbines in parallel as shown in the left up hand-side input box.

One instance of power plant is shown in Figure 21, where eight multi-shaft units (1-1-1) have been put together. This power plant, the Korea Electric Power Corporation's (KEPCO) Seoinchon combined cycle plant is now producing nearly 4,000 MW power output, with the recent addition of eight new Frame 7F in single shaft CC.

In Appendix C, GE single-shaft and multi-shaft equipment arrangements are shown in details.

Besides the configuration option, the user can choose the type of fuel and input its chemical composition. Supplementary firing is also available as well as fuel preheating. All these options multiply the number of possible plant layouts that one user may have the need to.



Figure 21: Seoinchon Combined Cycle Plant

3.1.3. Single-Pressure Level Plant

The selected layout is illustrated in Figure 22.

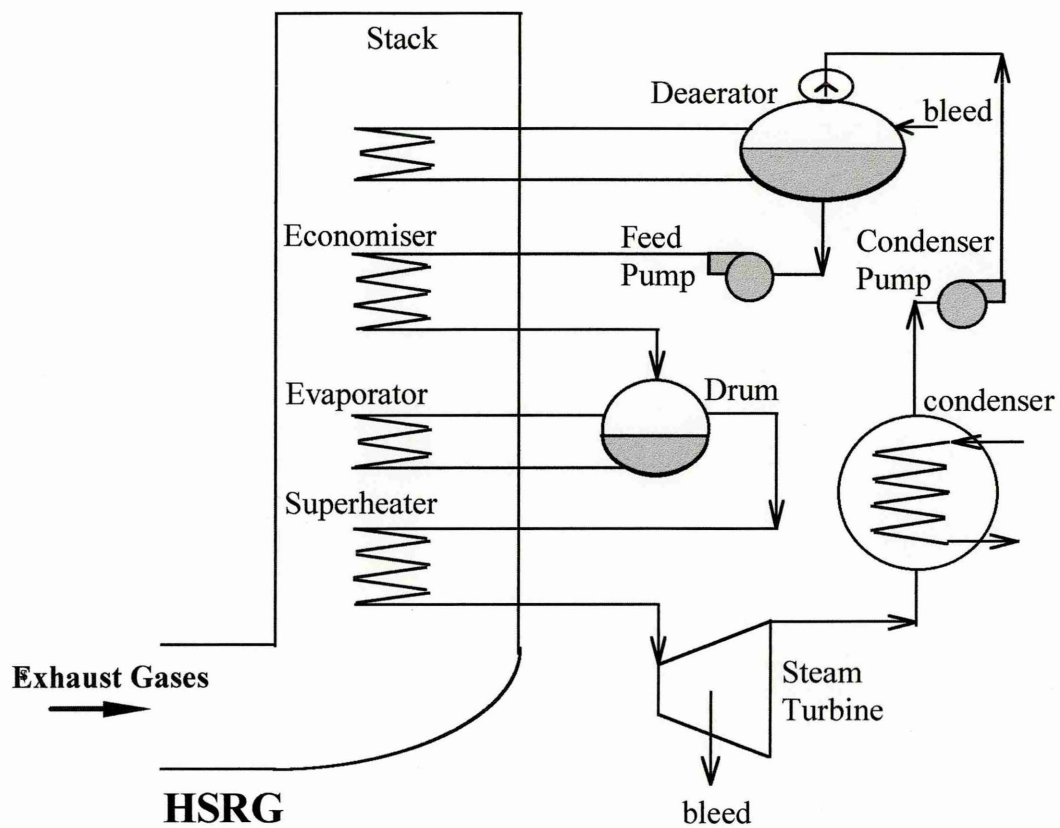


Figure 22: Single-Pressure Steam Cycle

Single-pressure plants are very often applied for CHP schemes. Steam generation is usually the main requirement and single pressure HSRGs can produce the maximum amount steam for a given gas turbine exhaust. Besides the quality of the steam generated can be effectively monitored by the use of supplementary firing at the back of the gas turbine(s) when running in off-design conditions.

According to Chase et al. (1996), single-pressure CC plants can be a sound economic selection when fuel is inexpensive and when applied in peaking duty.

The DSS proposed a single pressure plant but does not proposed a real CHP configuration. The main focus during the design of the DSS was combined cycle schemes but “doors” have been keep opened in order to easily modify the cycle to implement steam extractions, back-up drums and other features for steam process and handling. For instance a supplementary firing module was modelled so that the planner could set a constant HRSG inlet temperature by burning gas turbine exhaust gas with an additional fuel injection. The supplementary firing model is described in more details in a following chapter.

Figure 23 shows the final plant configuration programmed in Visual Basic. Deaeration takes place into an evapo-type deaerator, which is very common for such a plant whose stack temperature is expected to be very high (around 180°C). Such deaerator lowers the stack temperature by around 30°C, avoiding thus condensation of gas turbine exhaust gases.

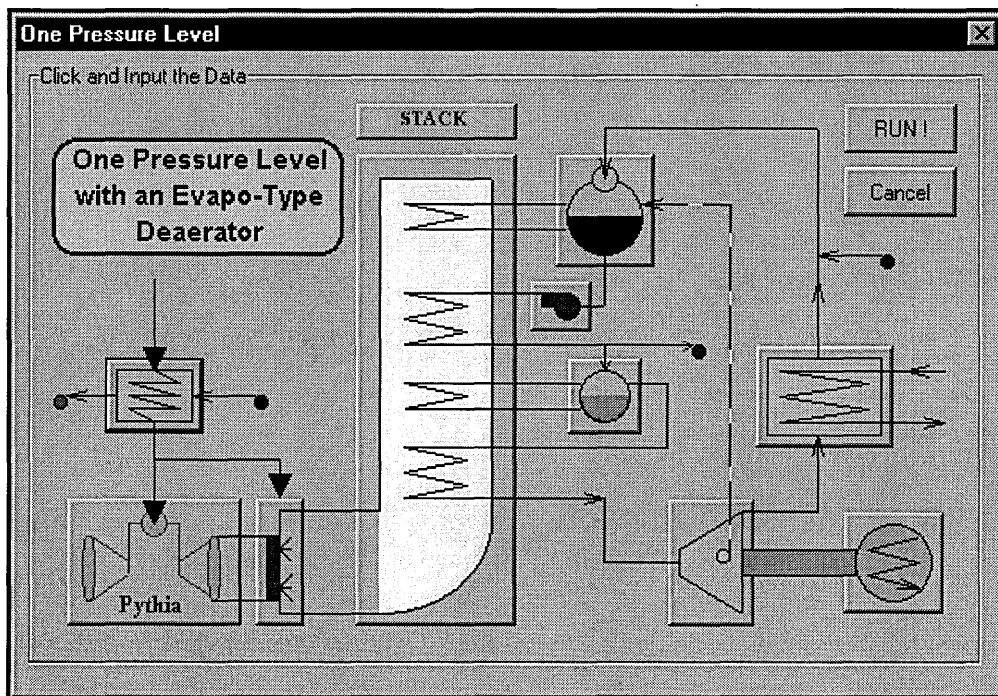


Figure 23: Single-Pressure Level Power Plant Model

3.1.4. Two-Pressure Level in Cascade

Two Pressures CC are widely proposed by manufacturers because of their relative low capital cost and their good trade-off between the great performance improvement compared to single pressure CC plants and their lighter equipment compared to three pressures CC plants.

There are two main configurations possible: the cascade configuration where the HP and LP circuits are totally disconnected to each other and the parallel configuration where the LP economiser is "shared" in parallel with a first HP economiser. Even so the cascade arrangement is simpler to implement, the parallel arrangement is now always preferred due to its higher effectiveness.

The plant proposed is not very different to the single pressure CC plant seen before, since its HRSG is not more than two single pressure level HRSGs put together. As shown in Figure 24, the steam cycle is almost identical except the deaerator.

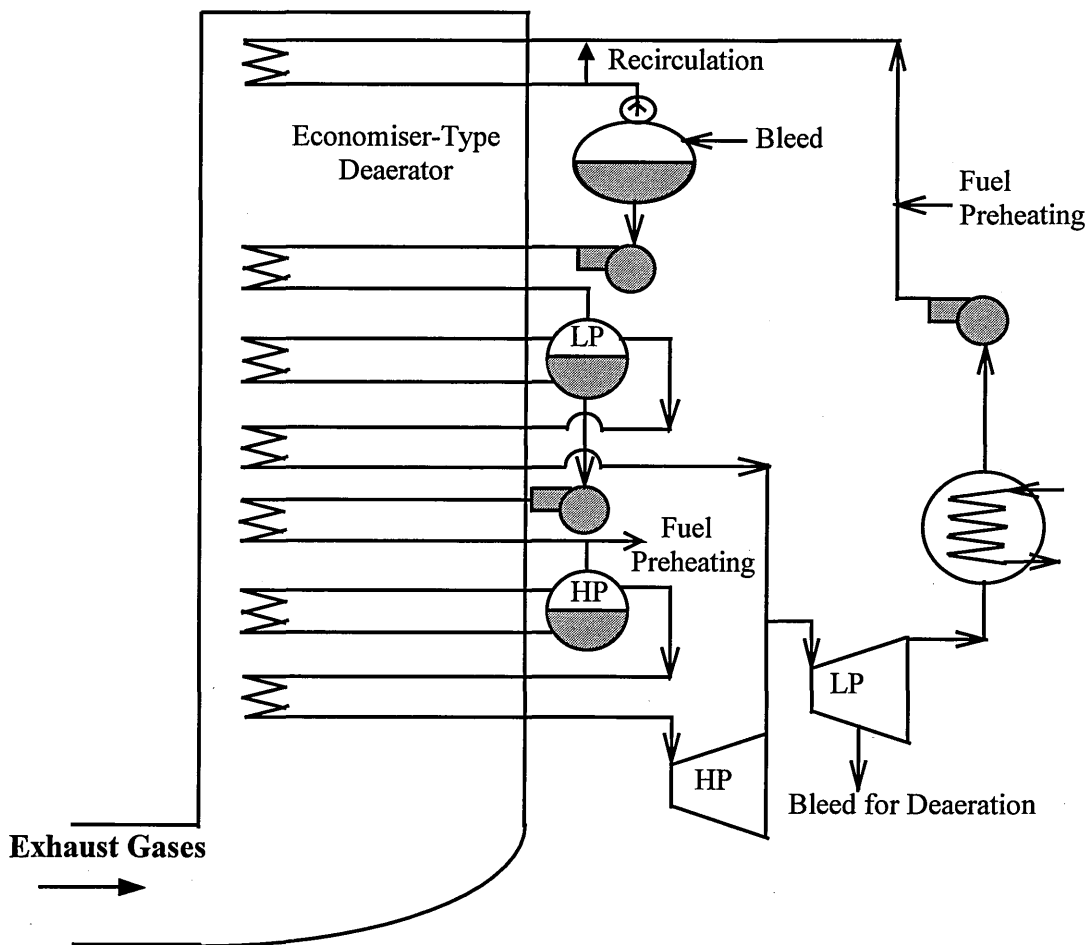


Figure 24: Two-Pressure Level in Cascade Steam Cycle

In multiple pressure HRSGs, stack temperature can be lowered under 100°C. In that case, an economiser heat exchanger is preferred to the evapo-type. Besides by means of

a recirculation circuit, exhaust gas condensation can be avoided when the plant is operating in off-design conditions.

The final selected layout is shown in Figure 25. Reheating is not provided because it is not very often implemented in such CC plants, which are attractive also for CHP schemes when HP steam and LP steam are required.

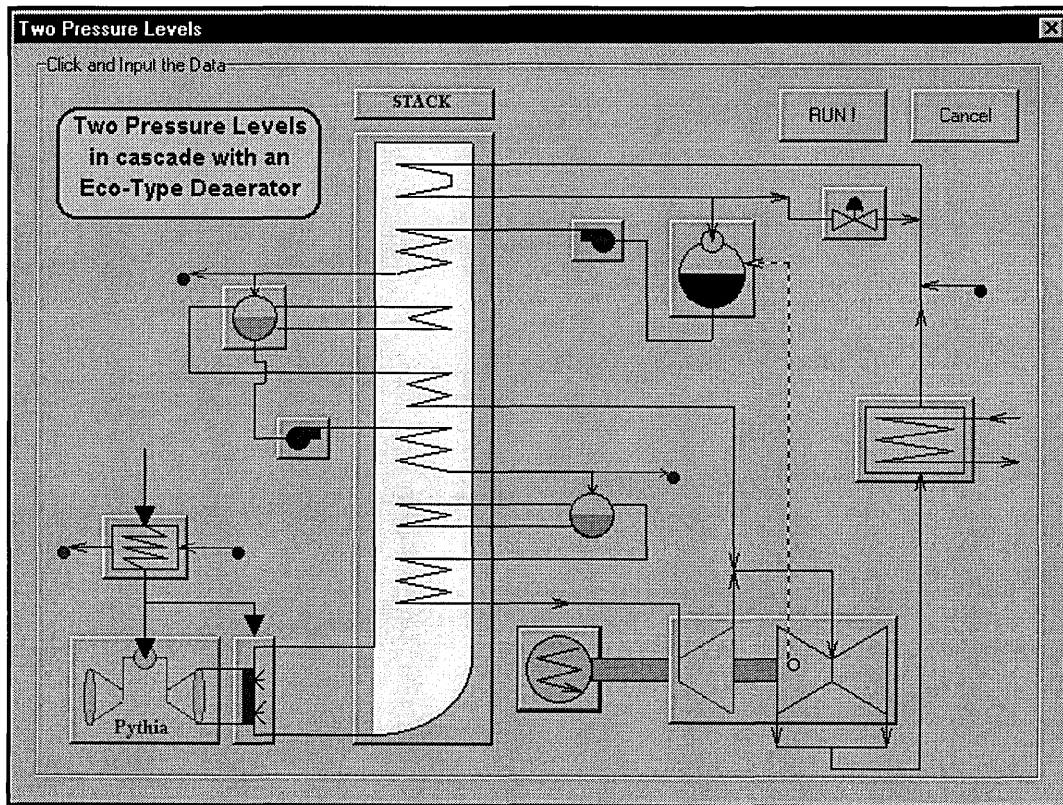


Figure 25: Two-Pressure Level in Cascade Power Plant Model

3.1.5. Two-Pressure Level in Parallel

This layout is quite similar to the previous one except that the heat exchangers in the HRSG are built-up differently as shown in Figure 26.

The “parallel” technology is nowadays always applied since it prevents the water circuit from handling HP, LP and sometimes even IP feed water in the same drum, i.e. the LP drum at the top of the HRSG. Such a large amount of water would certainly cause deposit problems into the LP drum, decreasing the LP steam generation by increasing the amount of water blown off. The VEGA 206-2P in Appendix C is an instance of a two pressure levels CC plant with a parallel heat exchangers configuration. The temperature of the HP feed water at the outlet of the HP economiser 1 (cf. Figure 26) is roughly the same as the LP drum inlet. Heat exchangers are basically built intrinsically, making thus a single economiser type heat exchanger.

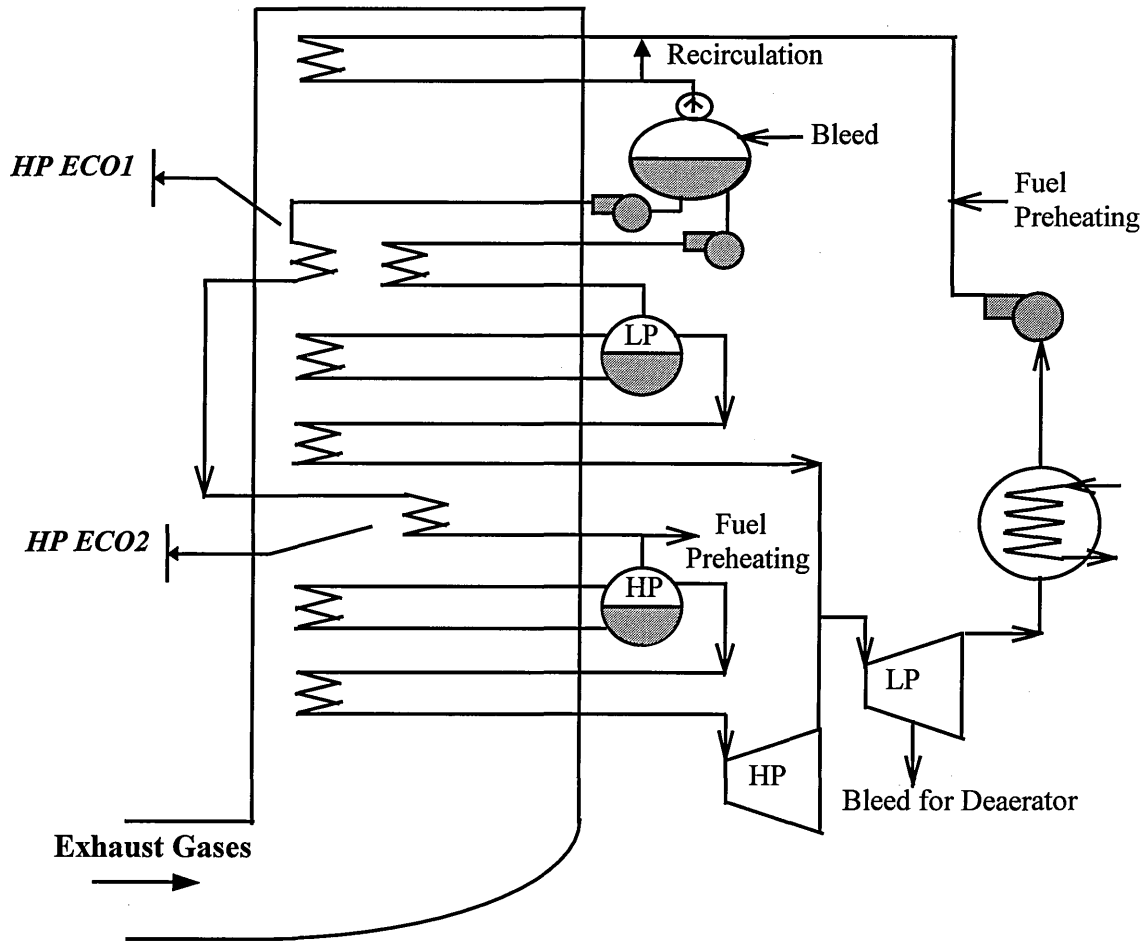


Figure 26: Two-Pressure Level in Parallel Steam Cycle

The modelling of the plant proposed by the DSS is illustrated in Figure 27:

3.1.6. Three-Pressure Level Models

As shown in Figure 28, the layout of three-pressure plant HRSGs are much more complicated. This is mainly due to the fact, that the HSRG must generate as much as possible steam by lowering gas turbine exhaust gases below 100°C with the smallest pinch temperature differences at any of the three pressure levels. For that purpose heat exchangers are configured in different ways compared to single and two pressures CC HRSGs. LP and IP superheater are located “before” IP and HP economisers in order to, firstly maximise the superheating temperatures by using the hottest gases possible, and secondly to minimise “pinch points” which correspond to the difference between the gas temperature and the saturation temperature at evaporator inlet (cf. §3.3.3).

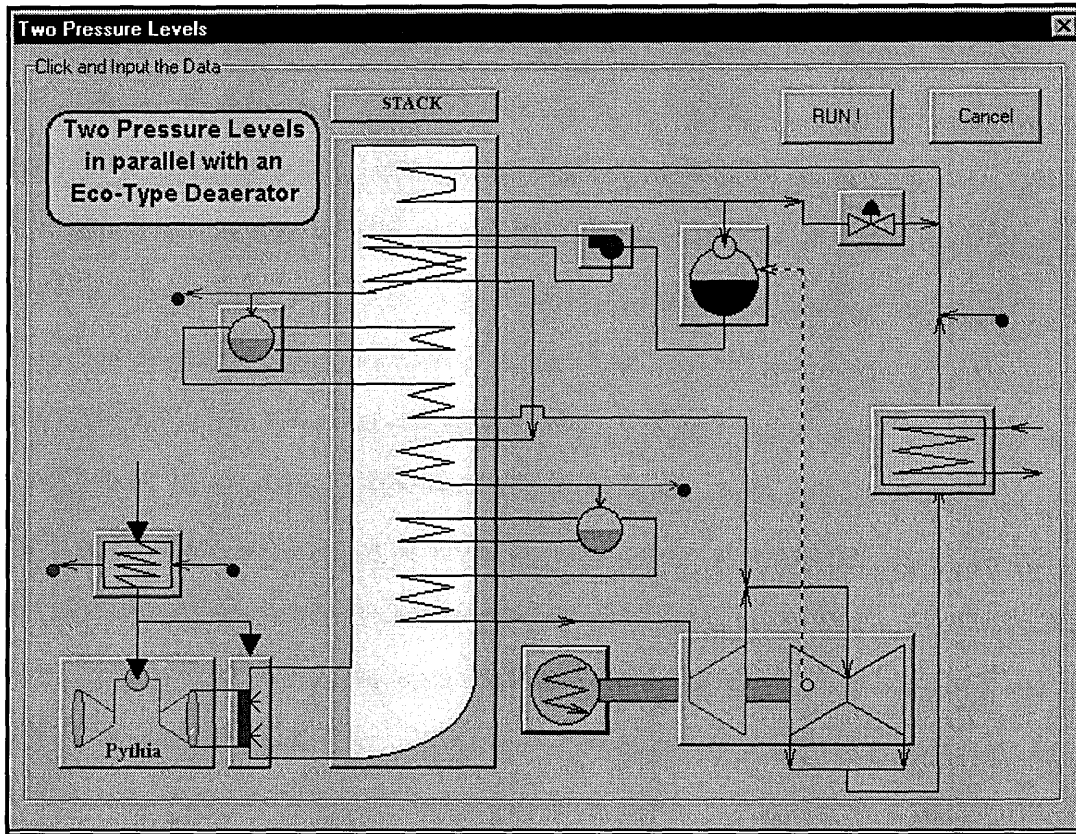


Figure 27: Two-Pressure Level in Parallel Power Plant Model

In the case of three pressures CC plants, reheating is commonly implemented. This will contribute to maximise the power output and the plant efficiency. Reheating is an option for the user therefore two different plant models are proposed. Figure 28 and Figure 29 only illustrate the reheated plant.

The layout is based on Briesch et al.(1995) and is presented as a 60% efficiency CC plant available by the year 2000. This Westinghouse layout is very similar to the General Electric one. In this particular modelling the deaerator is still a separate component of the plant and does not occur into the condenser as normally forecast by manufacturers by the year 2000. The purpose was to keep an advanced and proven HRSG configuration with a basic deaeration process since condenser-deaerator is a piece of advanced technology that has not been often presented in engineering publications. Due to the lack of information, only two different types of three-pressure CC plants have been modelled.

What is called the usual three pressures CC with or without reheat in the DSS is the plant illustrated in Figure 29.

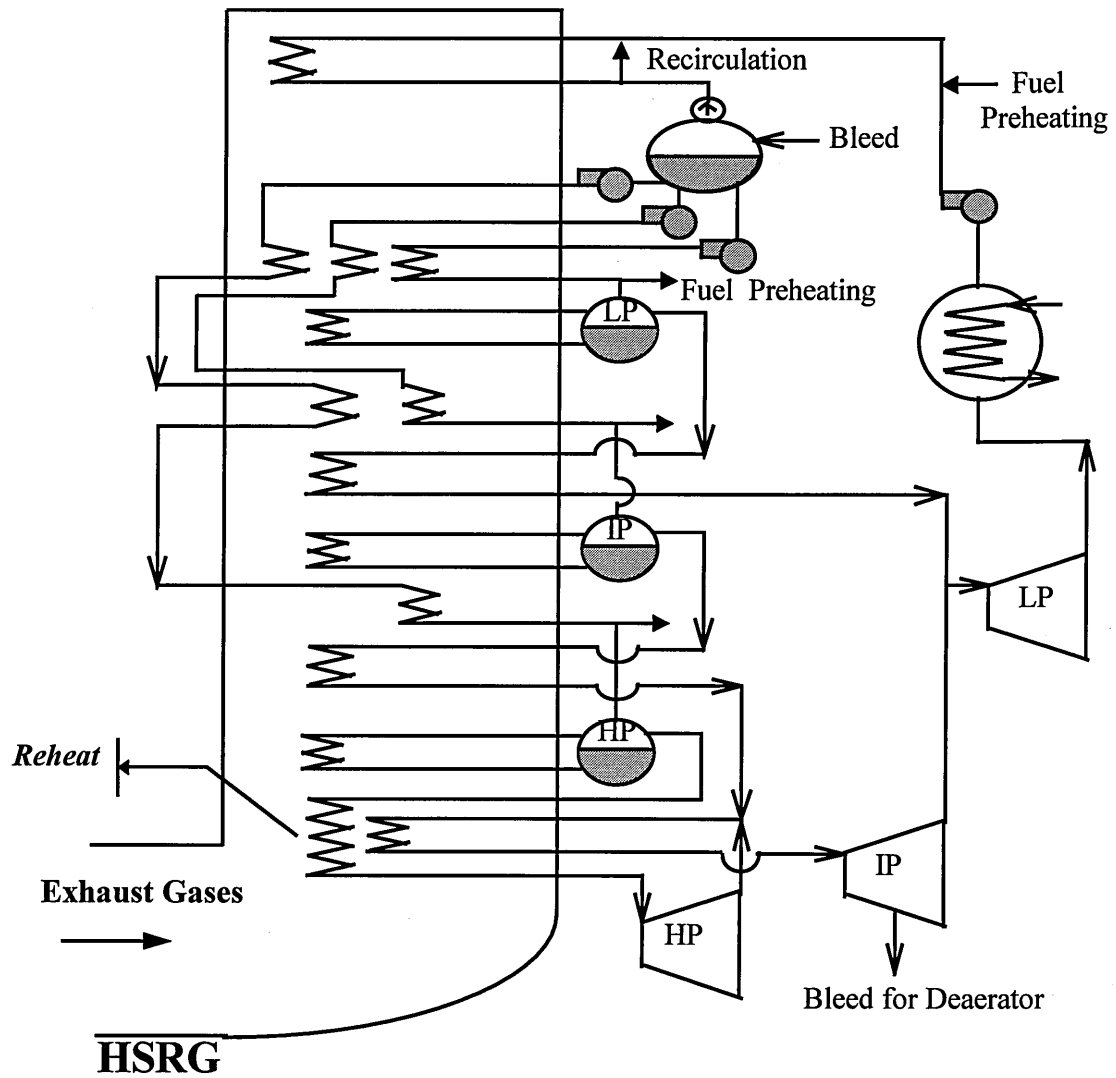


Figure 28: Three-Pressure Level Reheat Steam Cycle

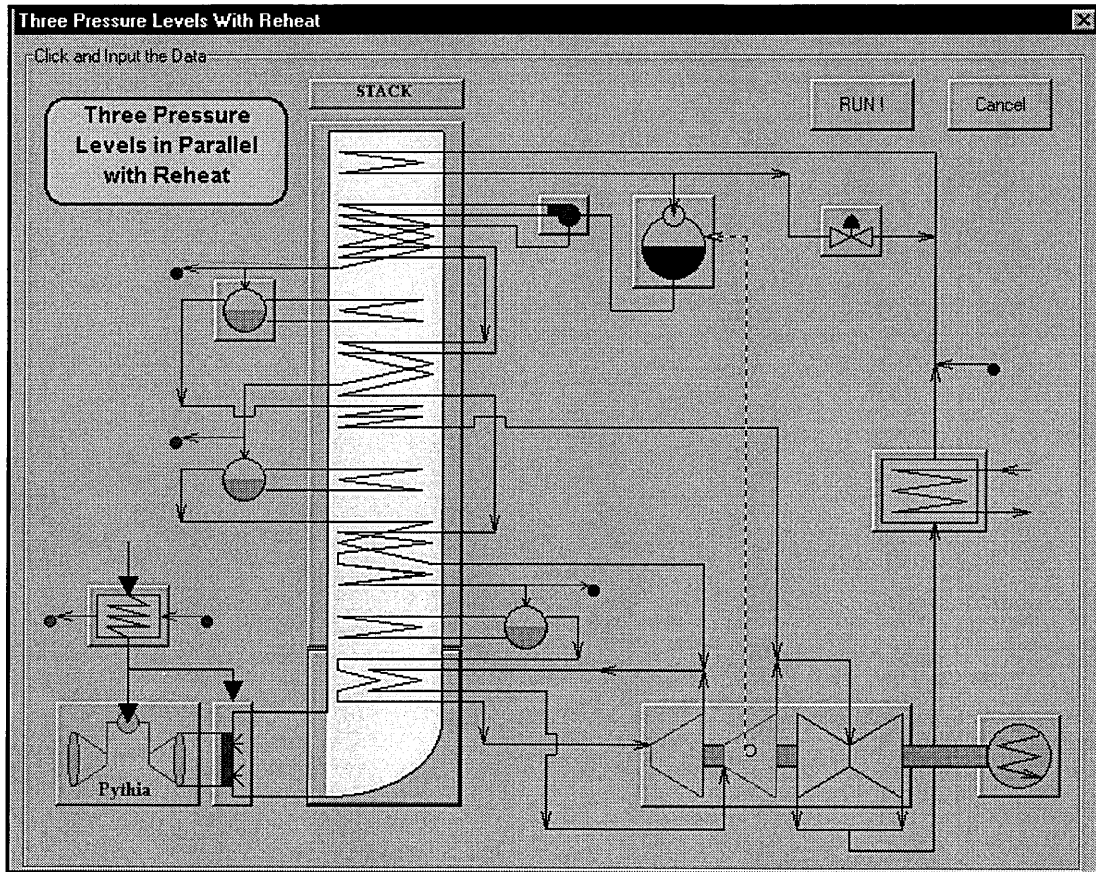


Figure 29: Three-Pressure Level Reheat Power Plant Model

3.1.7. Advanced Three-Pressure Reheat Model

The following three-pressure reheat plant is based on the GE STAG 209FA Multi-shaft Combined Cycle currently proposed by General Electric for large CC plants to be built by the year 2000-2005. In Tomlinson and McCullough (1996), plant performances are presented in details and this CC plant is said to exceed 60% in efficiency if powered by latest Frame 9FA engines.

The HSRG configuration (Figure 30) is quite complex and achieves a very good efficiency. Heat exchangers arrangement minimises pressure losses, radiation losses and heat exchanger surfaces. The LP drum feeds all economisers and water extraction at IP economiser end is done for fuel preheating purposes. As discussed previously, in this advanced plant, deaeration takes place into the condenser, hence called dea-condenser. An instance of such CC plant will be dealt with in a following chapter.

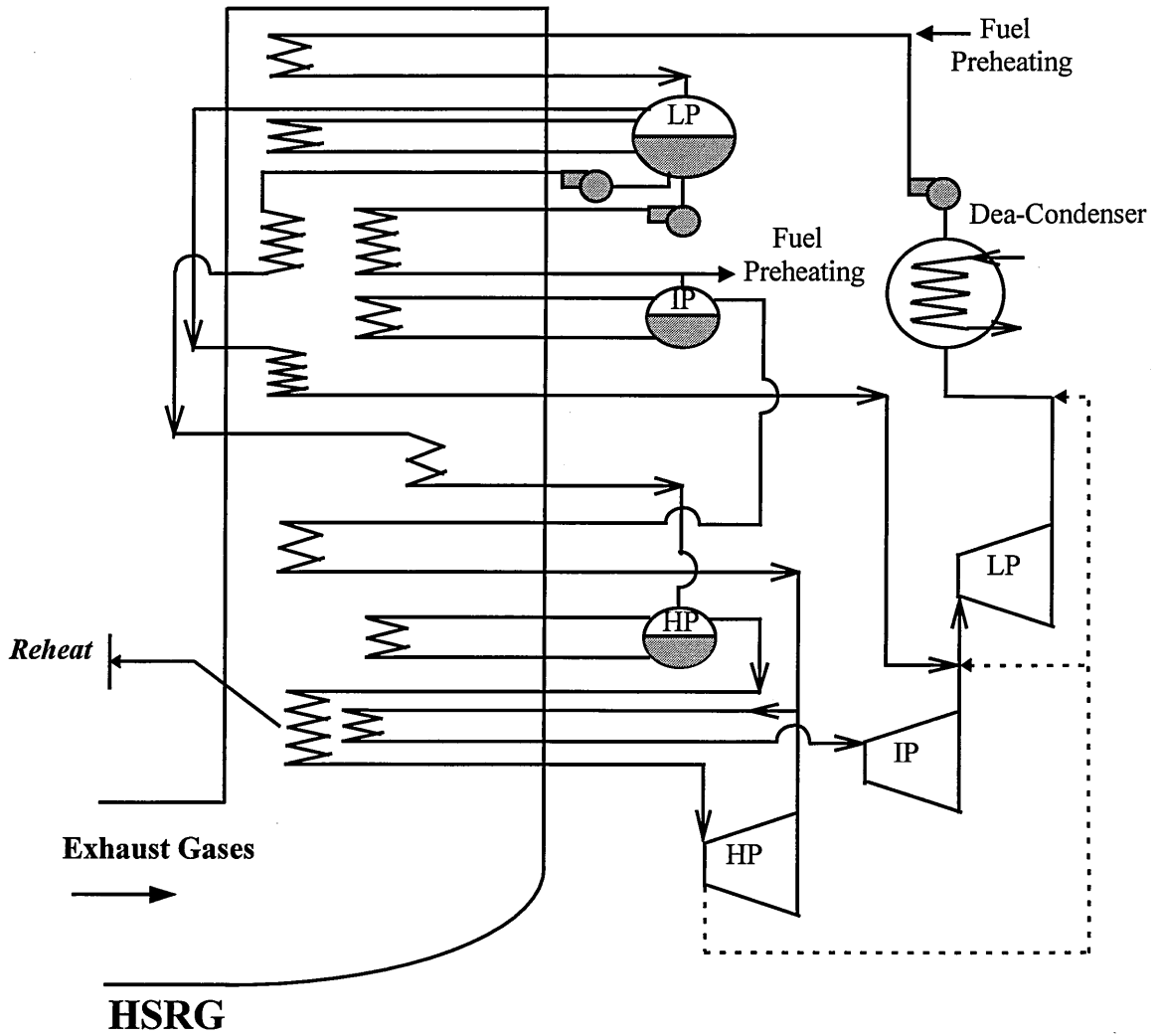


Figure 30: Advanced Three-Pressure Reheat Steam Cycle

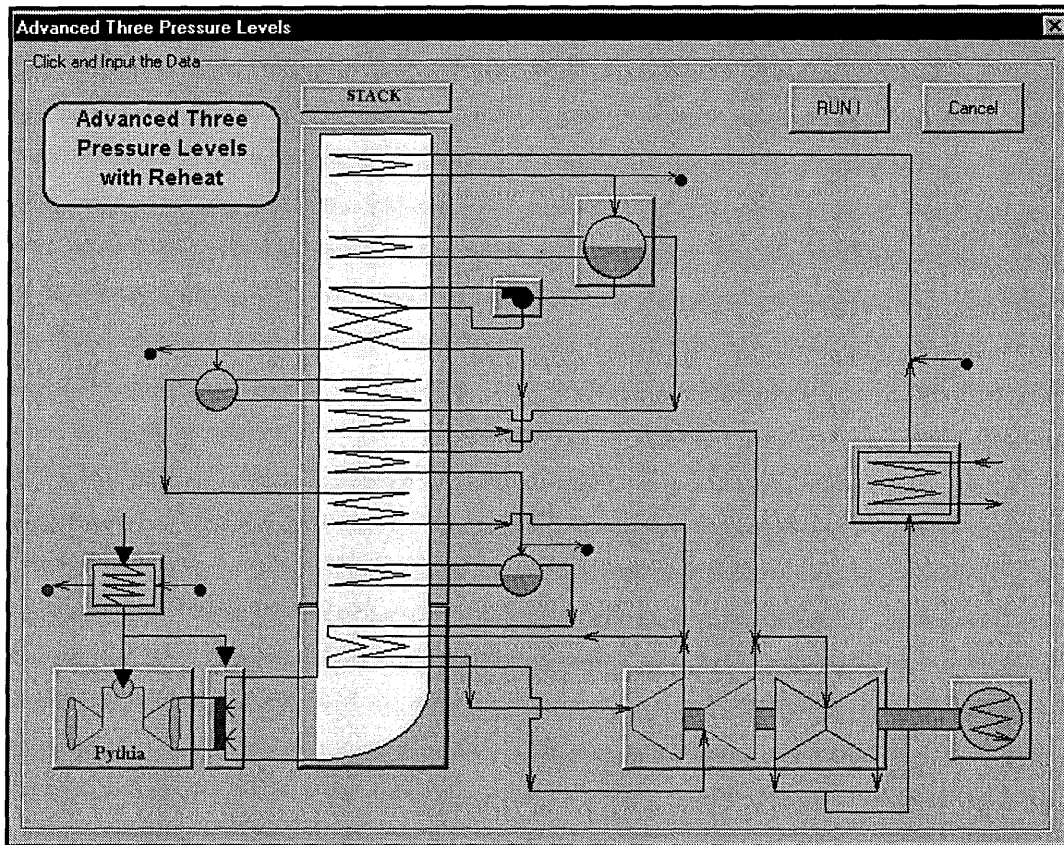


Figure 31: Advanced Three-Pressure Reheat Power Plant Model

3.1.8. Conclusion

The six CC models are hoped to provide a versatile design program that should fulfil most of user expectations. If not, the software enables the user to up-date or add his own plant models after learning how to do this by using the set of documentation provided with the software.

3.2. Gas Turbine Design and Simulation Programs

3.2.1. Introduction

As shown in Figure 31 in the previous chapter, steam cycle models are coupled to an engine design and simulation program called Pythia. As defined in Escher and Singh (1995), Pythia is an object-oriented diagnostics computer program that can simulate any kind of industrial gas turbine under any set conditions, off-design or/and deteriorated. Pythia is the Window version of an older program, named Turbomatch, satisfactorily used for engine performance calculation for more than 20 years. The DSS provides

several means of communication to these programs. Figure 32 is the screen printout of the window displayed when the gas turbine button is clicked (cf. Figure 31).

3.2.2. User interface

The user is offered two options. He can select an existing engine provided by a programmed library, or he can go into the design and simulation program in order to perform the gas turbine design performance calculation by his own. In that case, he will have to make sure that a well formatted output data file is created and successfully integrated into the DSS. Such a work can only be performed by someone who knows Pythia and Turbomatch quite well, that is why a library provides quick and an effective way of performing the whole calculation without using programs that require skills and knowledge that a user might not have. The library provides all sort of industrial machines sold by manufacturers such as General Electric, Westinghouse, ABB, Siemens etc. which have been modelled and simulated using Pythia. As Figure 32 shows, a GE Frame 9 FA has been selected and the user can display the results of the design-point simulation in order to compare it to manufacturer specifications if necessary. Would the model not be accurate enough, the user would then prefer doing his own design and performance calculation by calling Pythia and Turbomatch.

The screenshot shows a software window titled "Gas Turbine" with a close button in the top right corner. The window is divided into several sections:

- Design:** A large box containing the word "Pythia".
- Flue Gases Properties:** A box containing the label "Cp".
- Performance Results:** A box containing the label "Display".
- Pre-Designed GT:** A section with a checked checkbox "Library (Yes-No)", a dropdown menu set to "GE", and another dropdown menu set to "MS9001FA". Below this is an "OK" button.
- ISD Conditions:** A section with "Sea level", "Pa = 1 atm", and "60 % humidity".
- Exhaust Conditions:** A section with four input fields: "Inlet Mass Flow (kg/s): 601.9", "Exhaust Mass Flow (kg/s): 614.44", "Exhaust Temperature (°C): 589", and "Exhaust Pressure (atm): 1.018".
- Performance:** A section with three input fields: "Net Output (kW): 226500", "Net Efficiency (-): 0.357", and "Heat Rate (Btu/kWh): 9570".
- Auxiliaries:** A section with two input fields: "Auxiliary Power (kW): 325" and "Generator Efficiency (-): 0.98".
- Base Load Selection:** Two radio buttons: "Base Load Natural Gas" (selected) and "Base Load Distillate".

Figure 32: Engine Design or Selection Menu

In the software manual guide, the user can learn how to up-date the library or modify it. Engine simulation is the most important part of plant performance calculation. As a matter of fact in a combined cycle, the gas turbine acts as the "governing" component. Firstly the whole steam cycle depends on its exhaust conditions, temperature and mass flow, secondly the gas turbine is very often the CC component which manages part load conditions by the use of variable inlet guide vanes (IGVs), variable nozzle guide vanes (NGVs) or by simply decreasing the TET in order to decrease the power output. All these part-load governing strategies will be described in a later chapter.

3.2.3. Pythia

This program provides a very high quality and user-friendly windows-oriented interface. It makes gas turbine design straightforward and very quick. The process consists in dividing the engine into several basic components such as compressors, turbines etc. and modelling the engine using the modules provided by the software. In Figure 33 are summarised all available modules for building up engines.

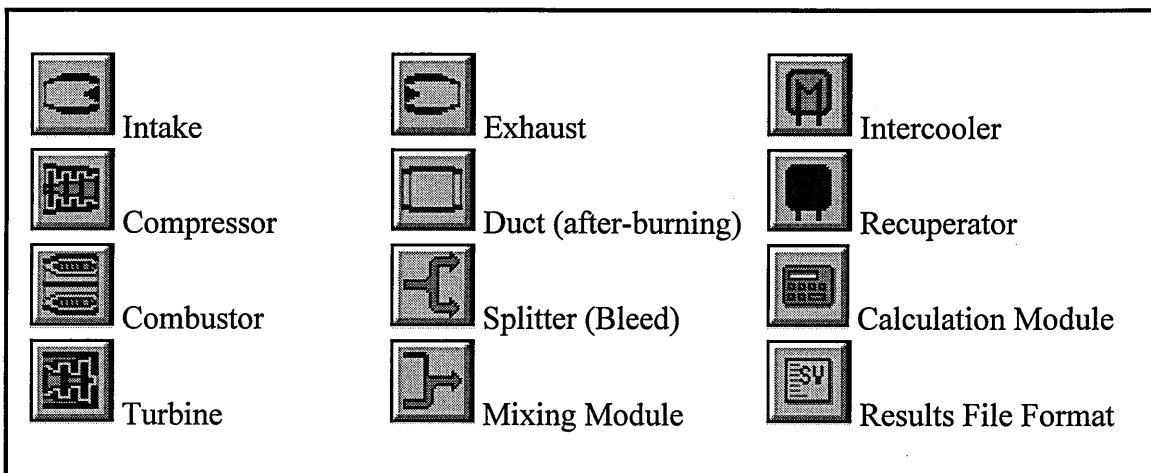


Figure 33: Pythia's Components Modelling

A complicated instance of modelling could be the latest Northrop-Grumman/Rolls-Royce mechanical drive gas turbine, the WR-21 implementing an intercooled recuperated cycle.

Based on Figure 34, it is possible to build an engine model using basic components that Pythia is proposing (cf. Figure 33).

It is important to notice that engine modelling is totally up to the user. Cooling system modelling for instance could be much more complicated and could reflect more effectively compressor bleeding systems. Even so several models can provide very accurate design-point performance results, only a very few can fit off-design performance curves, usually provided by engine manufacturers or published in engineering magazines.

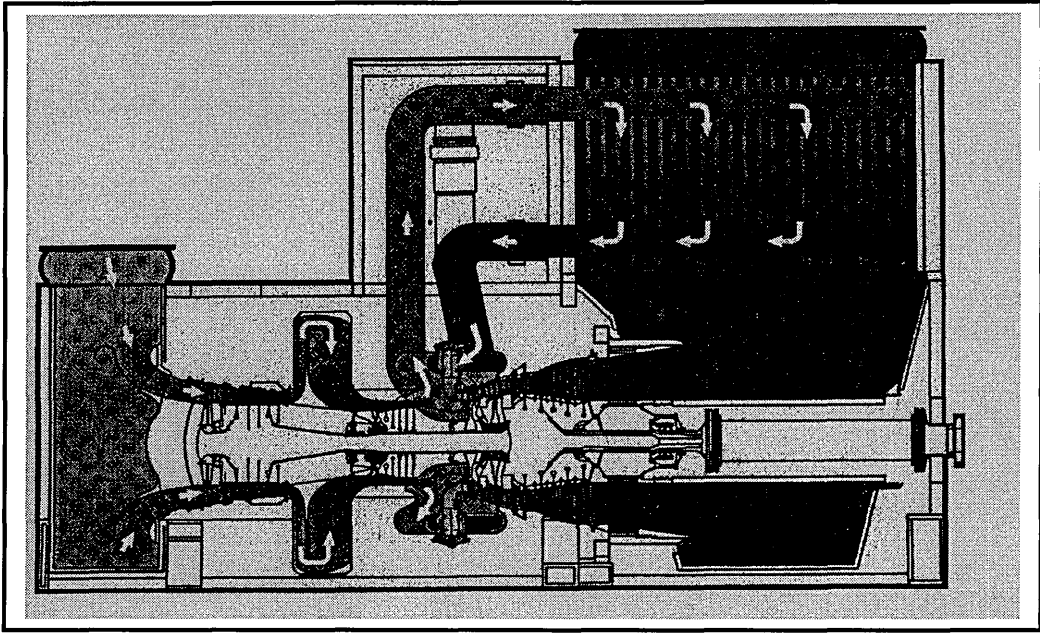


Figure 34: WR-21 Picture

One model could be the following one:

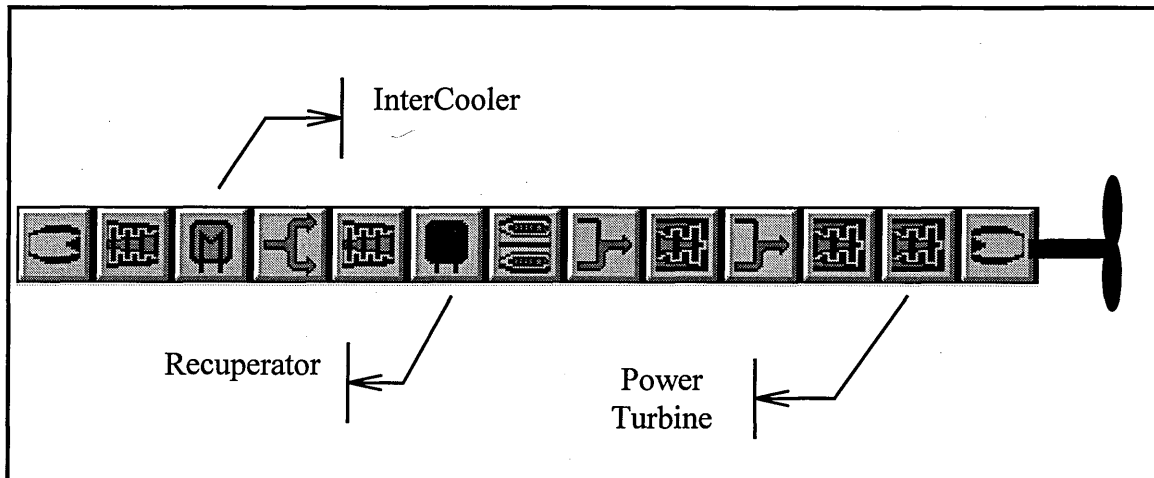


Figure 35: WR-21 Model Using Pythia

In Pablos (1996), the user can find all information necessary for designing and simulating engines. It is advised to use Pythia for engine design and for model validations, but for degradation calculation as well as for off-design performances Turbomatch is more suited. This is due to the fact that only Turbomatch enables the user to iterate performance calculation by changing degradation level or/and off-design parameters, such as compressor inlet temperature, IGVs angle etc.

3.2.4. The Turbomatch Scheme

It must be pointed out that Pythia runs Turbomatch in background for any performance calculation. The program can be then regarded as a window user-interface of Turbomatch.

As shown in Figure 36, the expected procedure for generating output files starts with Pythia for design and validation. Once the modelling is satisfying, a design-point is performed and an input file automatically named by Pythia: *FOR001.dat* is created. This file is basically the written input file that Turbomatch needs in order to run any performance calculation. It contains specific set of statements to characterise engine design and other statements for input data such as compressor efficiencies, pressure ratio, pressure losses, combustion efficiency etc.

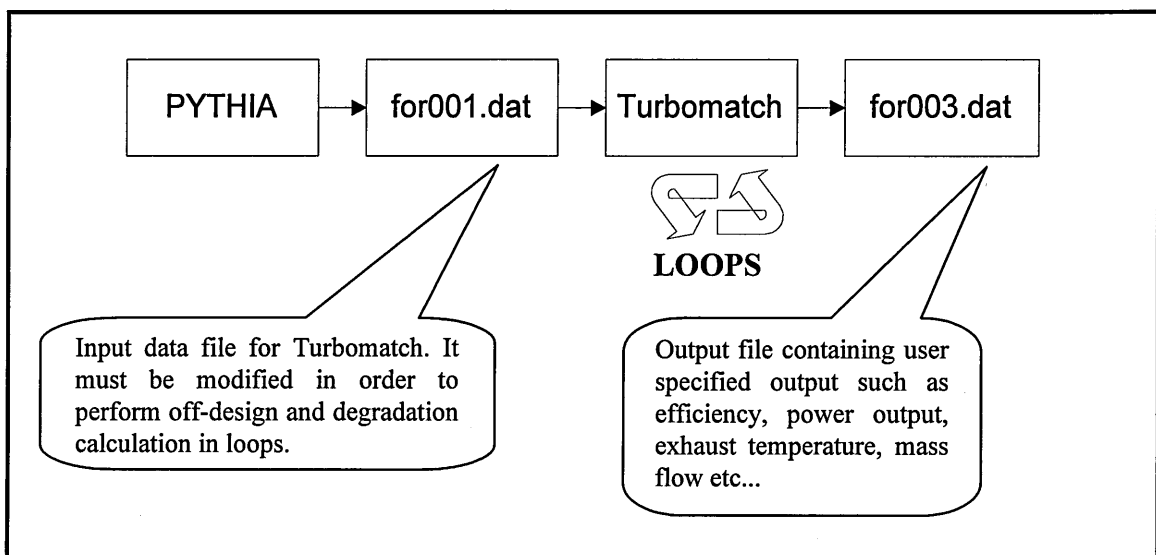


Figure 36: Performance Calculation Procedure

An instance of an input file (*for001.dat*) modified to perform off-design and deteriorated conditions in loops is proposed in Appendix D. This instance shows how tedious could be the work of a user, who would not want to use one of the gas turbines provided by the library. Moreover it demonstrates the value of Pythia in writing the file *for001.dat* in clean conditions since it prevents the user from debugging the file.

It must be highlighted that Turbomatch can perform several hundreds iterations in few seconds. This means that the user can implement a very comprehensive model, which will calculate the performance of the engine in off-design and deteriorated conditions in a daily basis for the whole life of the plant.

3.2.5. Engine Library

The purpose of the library is to firstly provide built-in engine models for the commonest industrial gas turbines available on the market, and to secondly fasten off-design and

deteriorated performance calculations by using mathematical expressions that are the results of calculations done on Turbomatch, worked out and correlated in Excel.

The integration of one gas turbine into the library requires lots of calculations and plots. This long and tedious procedure undertaken by software developers is well described in the manual guide, which is provided with the program. The results of such “pre-calculated” performance behaviours in off-design and deteriorated conditions can result to the following plot:

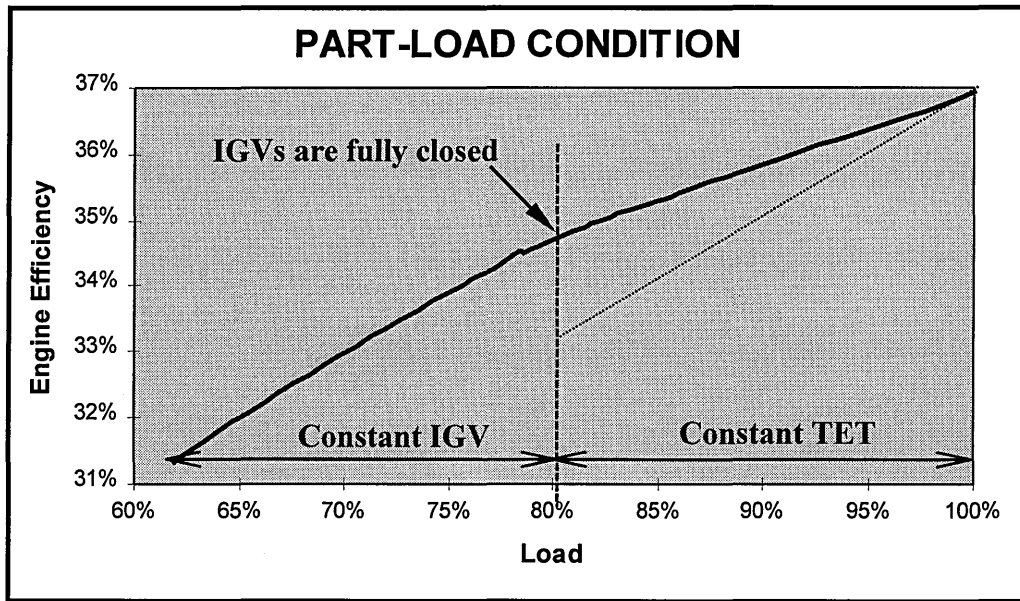


Figure 37: Part-load Behaviour of the MS9001FA

Figure 37 shows the efficiency of the MS9001FA as it has been designed and simulated by Pythia and Turbomatch, plotted against the load in clean conditions. This graph is the expected IGVs governed part-load behaviour. To decrease the load and to keep the efficiency as high as possible, gas turbine users firstly close compressor inlet guide vanes (IGVs) and when totally closed, around 80% load, the TET is then decreased. The dotted line on the plot shows the bad effect of decreasing the TET without using IGVs. Using inlet guide vanes can save 2% efficiency and sometimes more. The whole plant performance would follow the same trend as in Figure 37 at part-load if the steam cycle were running in sliding governing strategy.

As clearly illustrated in the above graph, curve fittings can be done and provides very accurate mathematical expressions which avoid heavy calculations that would have been necessary if the user had decided not to use the library but perform data processing directly in Turbomatch.

The main concept of using mathematical expressions for off-design performance calculation is based on the assumption that off-design main parameters such as compressor inlet temperatures, IGV and NGV angles, TET etc. have an independent influence on performance and exhaust conditions. Figure 38 is an instance of this basic

assumption. The exhaust temperature of the engine (GE MS9001FA) shows the same decreasing rate for different values of turbine entry temperature at different IGV angles.

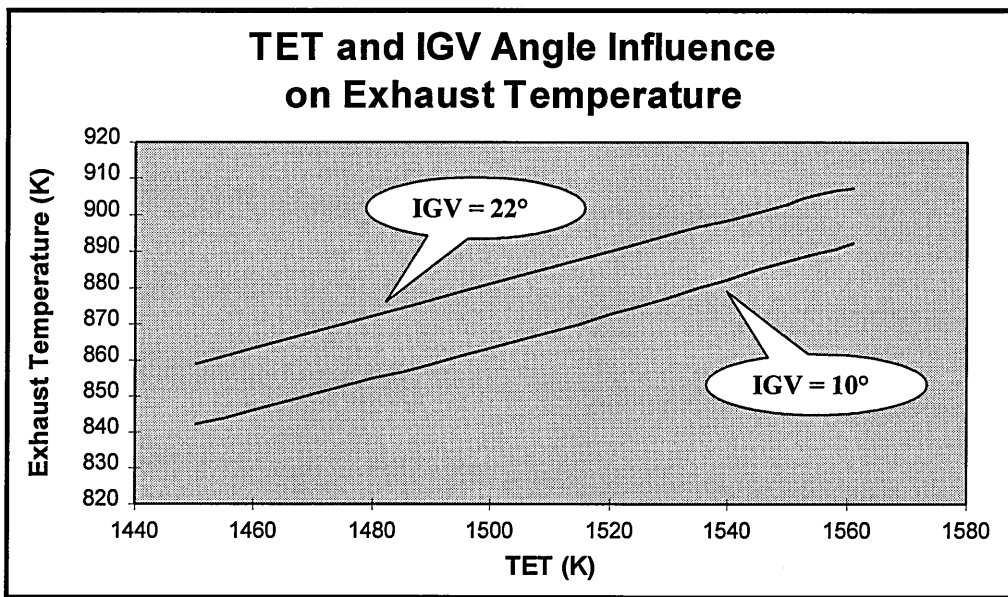


Figure 38: Instance of Independent Parameters on Output Data (MS9001FA)

The same sort of graphs can be plotted for exhaust mass flows, power output, etc. with regard to different parameters such as compressor isentropic efficiency degradation rate and IGV angle. The curve patterns will not be always linear, but for all known and processed calculations undertaken for building an engine model into the library, very accurate correlations were found.

3.2.6. Conclusion

Up-dating an engine model or adding a new one requires lots of data processing following a strict methodology described in the manual guide. In all cases this will demand very good knowledge and understanding of both Pythia and Turbomatch.

Besides it must be pointed out, that a large part of the reliability of steam cycle performance results depends upon gas turbine performance data accuracy. Exhaust gases entering the HRSG will govern steam generation and thus steam turbine performance, especially if at off-design conditions the plant is run in sliding governing strategy. This point will be developed in the off-design performance chapter.

In those conditions the user must have a good level of confidence concerning the reliability of both Pythia and Turbomatch, engine models and off-design philosophy followed by library designers.

3.3. Heat Recovery Steam Generator

3.3.1. Introduction

Abusively called boiler, the heat recovery steam generator is a major component of a combined cycle plant. It consists of a succession of heat exchangers through which gas turbine exhaust gases are exchanging heat with water and steam sometimes at different pressure levels.

HRSGs can be horizontal or vertical and therefore can be fitted with large fans in order to assist exhaust gases circulation. But natural circulation HRSGs are now becoming the commonest ones. The DSS only provides a modelling of this particular configuration.

As quoted previously, the major differences in the steam cycle modelling between the different CC layouts, proposed by the DSS are located in HRSG models since plant characteristics mainly consist of the number of pressure levels and heat exchanger configuration.

3.3.2. Heat Exchanger Types

3.3.2.1. Economisers

The economisers correspond to the heat exchangers that raise the temperature of pressurised water up to the saturation. Economiser outlet temperature is usually not exactly the saturation temperature but few degrees less. This temperature difference is called the "approach" (cf. Figure 39) and can be reduced to zero when the plant is running at part-load. In that case steam can be generated into the economisers but the rate must remain modest in order to keep pressure losses at a reasonable level and to avoid larger salt deposition leading to higher rate of tube corrosion.

Due to relatively low operating temperatures, economisers are often the last heat exchangers in the HRSG gas path.

3.3.2.2. Evaporators

The evaporators are connected to steam drums where steam is raised. These heat exchangers provide the heat necessary to completely transform water into steam. The medium passing through evaporator tubes is a mixture of steam and water.

Evaporators usually require a large amount of heat and thus the temperature drop for the gas side is large. This amount of energy mainly depends on the amount of steam to be generated and the saturation pressure at which the steam drum is set.

3.3.2.3. Superheaters

The superheaters are the heat exchangers, which heat up the steam that is leaving steam drums at the saturation temperature. In some cases, additional superheaters called reheaters are used in order to raise the temperature of the steam leaving the HP or IP steam turbine before returning to it for a further expansion in the IP or LP steam turbine. Superheating and reheating require very high gas temperature therefore they are located as close to HRSG inlet as possible. This explains why in most of three-pressure CC plants, LP and IP superheaters are located before the economisers.

For thermal resistance reasons, superheated and reheated temperatures can not exceed 560°C even so it could be possible to achieve higher temperatures by using supplementary firing for instance.

3.3.3. Heat Balance Diagrams and Equations

The heat balance diagram is very useful to assess the effectiveness of the HRSG. By plotting the bulk average temperature of gases and water and/or steam against the heat exchanged by both sides, it is possible to summarise the thermodynamics of the steam cycle. Figure 39 shows the simplest heat diagram possible.

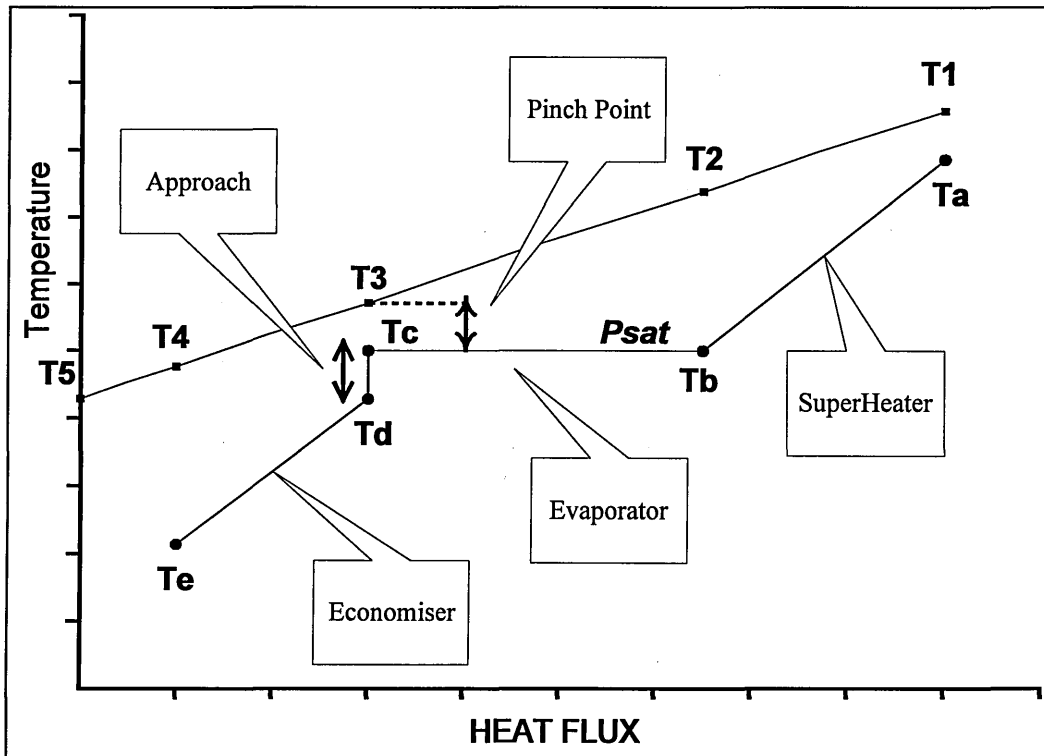


Figure 39: Single Pressure Level Heat Diagram

In that particular case the stack temperature would be T_5 and the superheated temperature T_a .

The effectiveness of the HRSG is very often assessed with regard to the stack temperature and the pinch point.

The notation used in Figure 39 will remain the same to characterise state position of both gas and water/steam sides for every type of cycle.

Heat transfer diagram can become much more complicated if the steam cycle is itself complicated. For instance the advanced three-pressure reheat heat diagram, as shown in Figure 40, highlights the fact that heat exchangers have different type of requirements and thus are in non-expected locations.

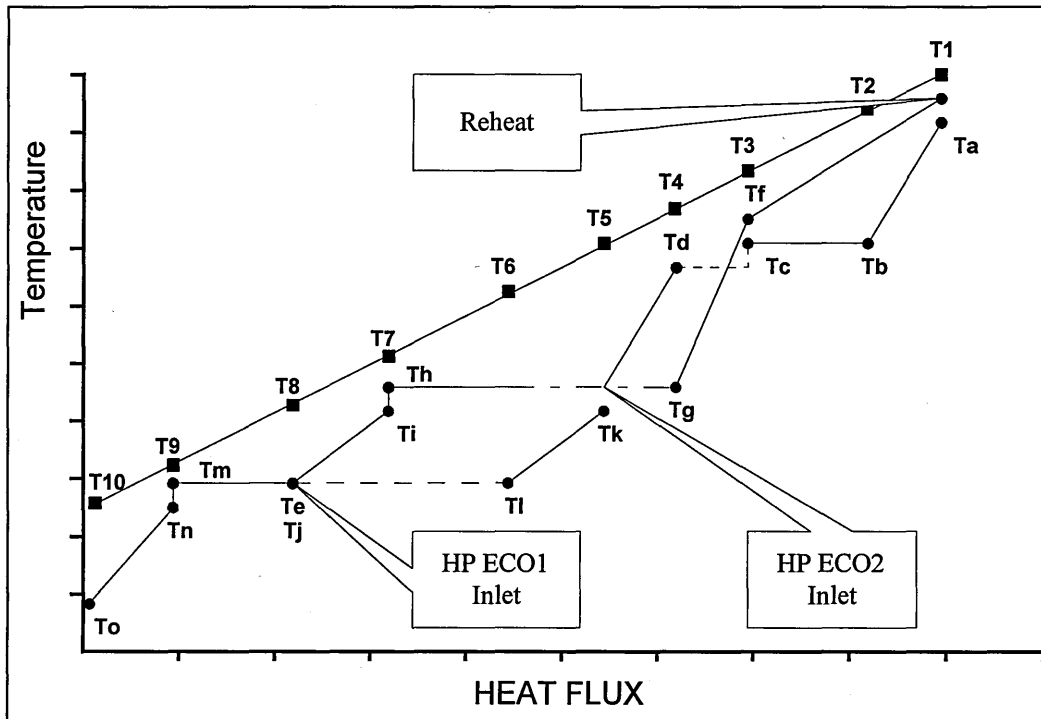


Figure 40: Advanced Three-Pressure Reheat Heat Diagram

Heat balance equations are straightforward. There are three types of expression for every type of heat exchanger, as follows:

$$1. \dot{Q} = \dot{M}_{gas} (H_1 - H_2) \quad (1)$$

$$2. \dot{Q} = \dot{M}_{steam} (H_a - H_b) \quad (2)$$

$$3. \dot{Q} = S_{super} U_{super} F_{super} \Delta T_{super}^{ln} \quad \text{where: } \Delta T_{super}^{ln} = \frac{(T_1 - T_a) - (T_2 - T_b)}{\ln \left(\frac{T_1 - T_a}{T_2 - T_b} \right)} \quad (3) \text{ \& (3bis)}$$

These three expressions provide the value of the heat exchanged between the gas side and the steam side of the superheater.

Similar expressions can be written for the economiser and the evaporator, except that for the evaporator, the shape factor F is equal to zero.

For the design-point calculation only the first two equations are used. Gas enthalpies are calculated as follows:

$$H_i = \dot{M}_{gas} C_p (T_i - T_{i+1}) (1 - \alpha_{rad}) \quad \text{where: } C_p = C_p \left(\frac{T_i + T_{i+1}}{2} \right) \quad (4)$$

Thermal losses by radiation are taken into account and are usually of the order of 1 or 2 percent of the overall HRSG duty.

The specific heat capacity is calculated at bulk temperatures. In Appendix E are summarised the mathematical expressions of thermodynamic parameters.

It is important to notice that the specific heat capacity factor C_p is highly dependent on the gas composition as well as the temperature. Figure 41 shows this sensitivity, especially with regard to water content, which for natural gas increases when fresh air ratio decreases.

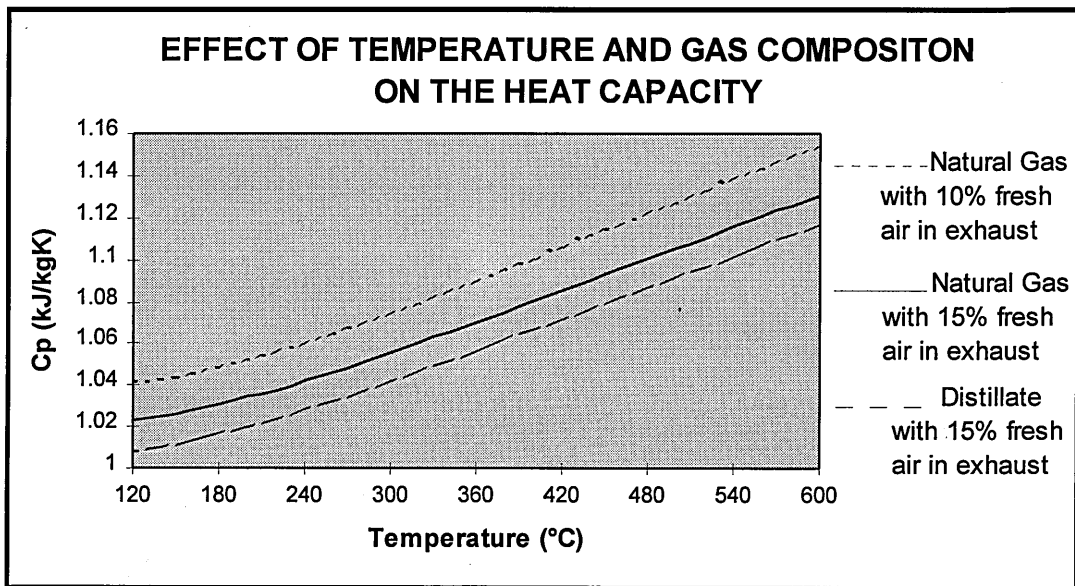


Figure 41: Cp as a Function of Gas Composition and Temperature

The exhaust gas composition is a very important parameter because it is one of the factors that determine the amount of heat exchanged at every level of the HRSG (cf. equation 4) and ultimately the stack temperature. It must be pointed out that convergence problems could be caused by wrong inputs in the exhaust gas compositions. Besides the user must bear in mind that gas turbine exhaust gases and HRSG inlet gases could have different chemical compositions if supplementary firing was used.

There are different ways of solving HRSG's equations. It depends on thermodynamic figures known by the user before running performance calculation.

A current way of calculating performance is to input pinch points and thus assess steam mass flows produced by the steam drums. But this is not consistent with the kind of figures that plant manufacturers are providing in their preliminary bids. Planners basically have access only to all figures necessary for calculating steam turbine

performance: pressure, temperature, mass flows and enthalpies of the steam entering and leaving the steam turbine. HRSG is almost kept as a black box as far as the thermodynamics is concerned. That is why pressure losses have to be guessed in order to assess saturation pressures at which steam is raised.

The DSS is a tool for assessing and comparing CC schemes during the decision making process. Therefore it must be designed with regard to the level of knowledge that planners get from plant manufacturers, i.e. the knowledge of thermodynamic conditions of superheated steam, expanded steam, bled steam and water circuit outside the HRSG.

Figure 42 represents one of the two input data windows that is displayed when the steam drum button is clicked. The second input data window (HRSG button) is illustrated in chapter 3.6.1. This lack of knowledge and the fact that assumptions must be made explains why plant performance calculation frequently requires iteration process. The stack temperature, provided by the manufacturers and not used in any performance calculation is usually a good convergence factor. It tells you if too much energy was extracted or if the steam cycle poorly performed, providing thus large pinch point values. It must be known that HRSG current technology gives pinch points around 10°C.

Pressure Level	SuperHeated Steam Temperature (°C)	SuperHeated Steam Pressure (Bar)	Steam Mass Flow (kg/s)	Approach Temp. Difference (°C)
High Pressure Level	520	50	70	5
Intermediate Pressure Level	330	34.7	10.7	5
Low Pressure Level	271.1	8.08	10.6	5

Figure 42: Steam Cycle Data Input Window

To summarise known data are assumed to be:

- gas turbine exhaust gas conditions (= pressure, temperature, mass flow),
- superheated and reheated steam conditions,
- inlet and outlet conditions of the steam turbine (HP, IP, LP if necessary),
- inlet and outlet conditions of the condenser,
- deaerator conditions,
- inlet conditions of the HRSG (feed water),
- stack temperature,
- inlet and outlet conditions of fuel preheater (if implemented).

And of course inlet and outlet of the gas turbine.

All these figures are useful to assess pressure drops, enthalpy drops, saturation pressures and temperatures, steam turbine efficiency, steam turbine outlet quality, steam turbine output, condenser efficiency, etc.

3.3.4. Algorithm

The algorithm of the design-point calculation is very easy to understand since it almost follows the gas path through the HRSG. The following instance is based on Figure 40. It is assumed that T_1 , M_{gas} , P_a , T_a , P_e , T_e , M_{steam} are known. Notice that T_e is almost the saturation temperature into the deaerator.

- **Step 1:** based on P_a and P_e , pressure losses through the HRSG are guessed and input into the program. Therefore the saturation pressure (P_b and P_c) is calculated and using water/steam thermodynamic properties at saturation (cf. Appendix E), T_b and T_c can be found.
- **Step 2:** using other pressure loss input data, P_d and P_e are calculated as well as T_d based on the approach.
- **Step 3:** once all pressure and temperature values of the water/steam cycle are known, enthalpies can be calculated using properties functions in Appendix E.
- **Step 4:** by combining equations 1 and 2 for the superheater, the evaporator and the economiser, T_2 , T_3 and T_4 can be calculated.
- **Step 5:** stack temperature T_5 is based on heat balance calculation into the deaerator. If the deaerator receives steam bled from the steam turbine, step 5 follows the steam turbine performance calculation in order to assess the enthalpy of the steam bled and send into the deaerator.

The procedure is almost the same for all cycle models, but as Figure 41 shows, some complicated steam cycles require more attention. For instance in a parallel configuration, LP and/or IP economisers heat up LP feeding water as well as HP and/or IP feeding water, adding some terms into equation 2.

The convergence of the whole calculation can be affected by two parameters.

First water/steam properties equations require water/steam pressure and temperature except for saturation conditions where only one of both is necessary, in order to calculate enthalpies and entropies. The user must know the Mollier diagram from which these mathematical expressions have been extracted. This T-S diagram shows that enthalpy and pressure curves can provide two or even three different results for a given set of temperature and entropy values. On top of that these curves become very sharp and thus very difficult to mathematically be modelled when approaching the triple point.

The second potential source of non-convergence could come from Cp calculation. For the same reasons as for water/steam properties, the specific heat is calculated with gas temperature and gas chemical composition. The later remains unchanged once the gases enter the HRSG, whilst the gas temperature varies. The overall temperature decreases through the HRSG but in the mathematical solving procedure this temperature can reach extreme values due to convergence problems. Equation 1 showed that the heat exchanged was function of gas temperature difference and function of the specific heat capacity, which was also calculated with gas temperature difference. This implies that the solution can not immediately be found, but requires an iterative process as shown as follows:

$$T_2^{n+1} = T_1 - \left(\frac{\dot{M}_{steam}}{\dot{M}_{gas}} \right) \left[\frac{1}{C_p \left(\frac{T_1 + T_2^n}{2} \right)} \right] \left(\frac{1}{1 - \alpha_{rad}} \right) (H_a - H_b) \quad (5)$$

Therefore if for some reasons the temperature does not converge and reach extreme values, the specific heat capacity calculation will cause specific error messages, stopping the whole calculation and providing the parameter that did not converge.

3.3.5. Conclusion

HRSG modelling and design-point calculation is quite straightforward. It is easy to modify an existing model or to add a new one. But convergence can not always be reached, especially if badly assessed input data are processed. The user must therefore pay attention on input data in order to understand the reasons why the design-point calculation did no converge.

3.4. Steam Turbine Model

3.4.1. Introduction

The steam turbine is the second source of power output in a combined cycle plant. It represents about one third of total plant output and thus increase plant efficiency by

almost the same ratio. Steam turbines nowadays range from 2 up to 1,500 MW when equipping large nuclear power plants. Steam turbine efficiency is affected by friction losses and exhaust losses due to water condensation that is why its not surprising that HP or IP non-condensing steam turbine sections reach very high efficiencies (>80%) compared with condensing turbines.

Steam turbines are based on very mature technology since they were introduced in power generation market roughly one century ago. Contrarily to the gas turbine, the steam turbine is not the subject of leading edge engineering implementation occurring every couple of years. First row blade materials are usually the main focus of steam turbine related research and development in order to increase the permissible inlet temperature, levelled off at 540-560°C for many years and reaching now 600°C (and 300 bar) for the largest ones.

3.4.2. Steam Turbine Principles

The aim of this chapter is not to deal with the machinery in details but just to give an overview of steam turbine principles in order to ease the understanding of part-load behaviour and part-load governing strategies involving the steam turbine and presented in a following chapter.

3.4.2.1. Steam Turbine Layouts

The following layouts are the commonest ones:

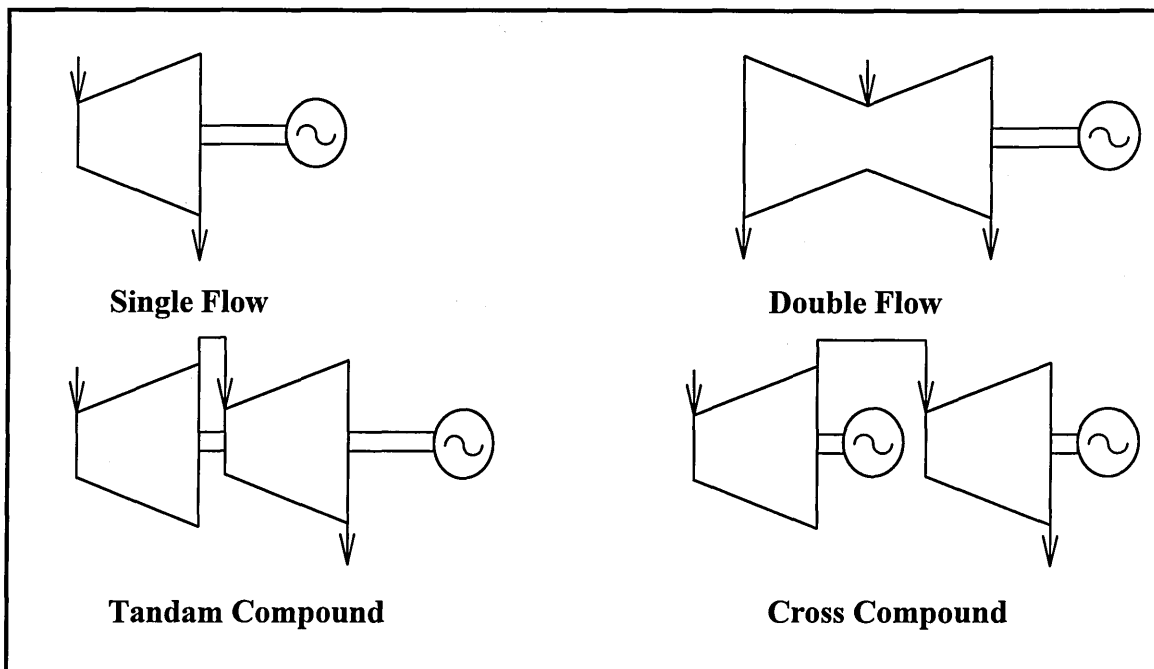


Figure 43: Steam Turbine layouts

3.4.2.2. Steam Turbine Types

The steam turbine basically utilises the steam pressure to transform thermal energy into kinetic energy and a change in momentum to provide forces on the blades. There are two major types of steam turbine: impulse and reaction turbines.

Impulse turbines can be divided into several types:

- simple impulse
 - velocity-compounded
 - pressure-compounded
 - pressure-velocity compounded
1. The simple impulse turbine consists of one row of nozzles followed by one row moving blades, as shown in Figure 44. After expansion from inlet to outlet pressure across the set of nozzles, the steam reaches a very high velocity. This high velocity steam is going through the moving blades and the remaining kinetic energy when the steam leaves the blades is lost. This type of steam turbine usually runs at extremely high rotational speeds, often over 30,000 rpm.

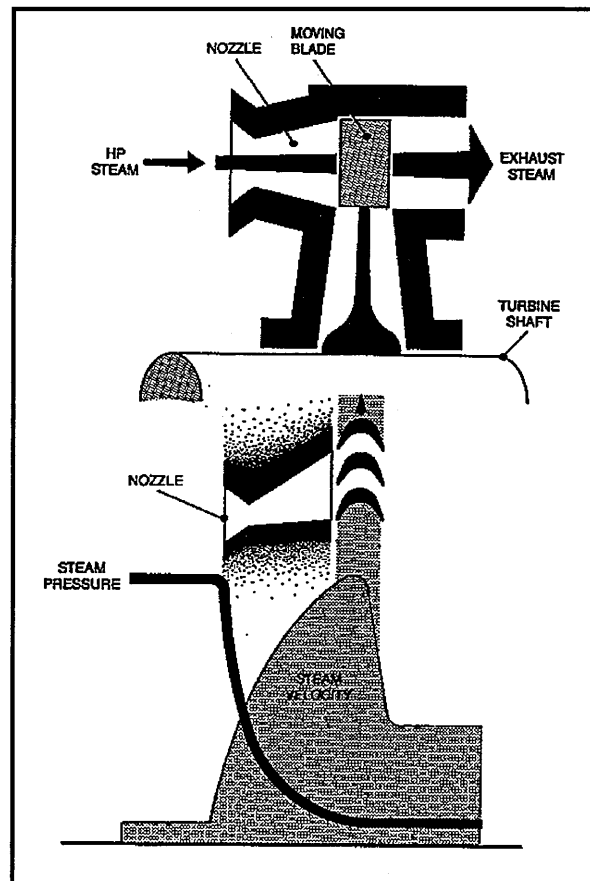


Figure 44: Simple Impulse Turbine

- The velocity-compounded turbine has one set of nozzles at the inlet followed by series of moving and fixed blades as shown in Figure 45. The fixed rows of blades are modifying the direction of the steam in order to make it suitable for following moving blades admissions. It must be noticed that the steam pressure remains constant after crossing the nozzle.

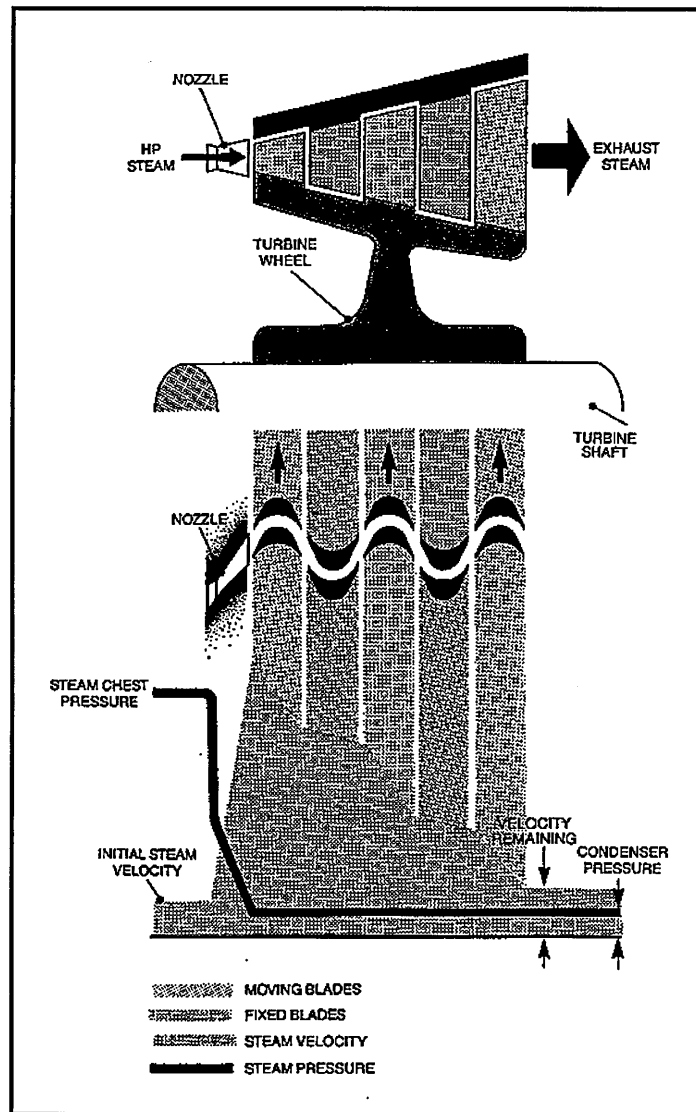


Figure 45: Velocity-Compounded Turbine

- The pressure-compounded turbine, referred sometimes to as the Rateau turbine consists roughly of a series of simple-impulse turbines i.e. a row of nozzles and a row of moving blades, commonly called turbine stage. The steam pressure drops when passing through each set of nozzles, and the resulting velocity increase is utilised in the following row of moving blades, as shown in Figure 46. As at each stage the steam velocity is rather low and so the blade velocities that prevent excessive steam friction losses.

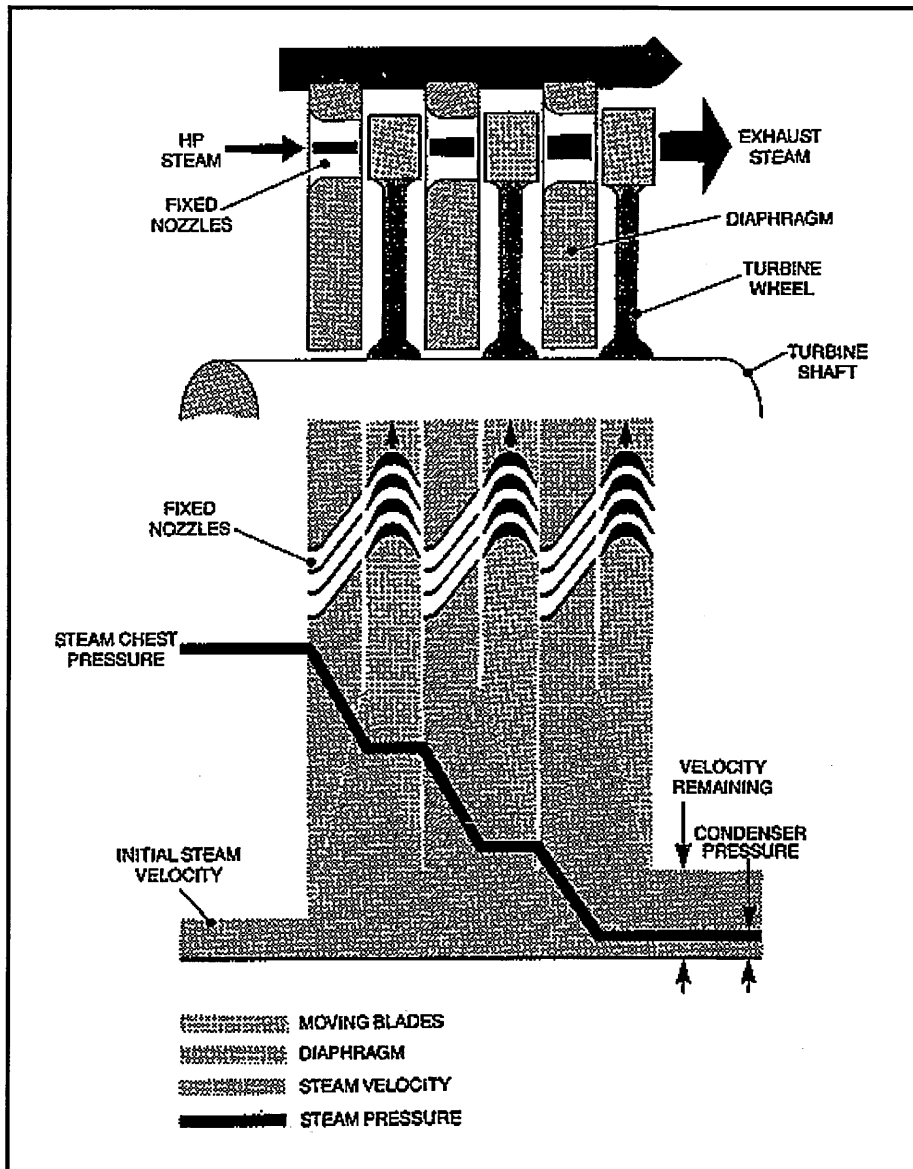


Figure 46: Pressure-Compounded Turbine

4. The pressure-velocity compounded turbine consists of two velocity-compounded turbines in series, as illustrated in Figure 47. The first set of nozzle reduces the steam pressure to an intermediary value whilst the second set reduces it to the back pressure.

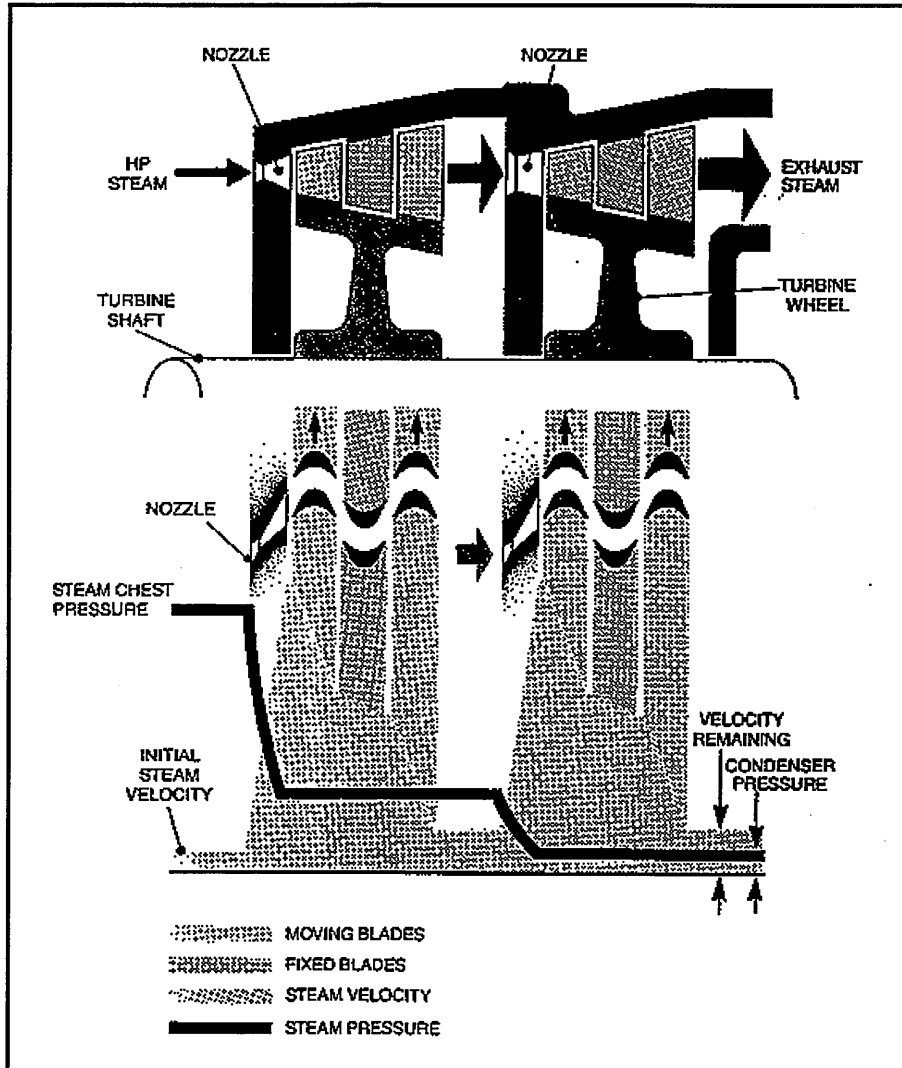


Figure 47: Pressure-Velocity Compounded Turbine

To finish with, there is also the impulse-reaction turbine type first built by Sir Charles A. Parsons. This type of turbine is shown in Figure 48. In that case the pressure drops while passing across both the fixed and the moving blades. The pressure drop across every row means that the clearance between the tips of the blades and the rotor or cylinder must be kept very small to prevent undue steam leakage from stage to stage.

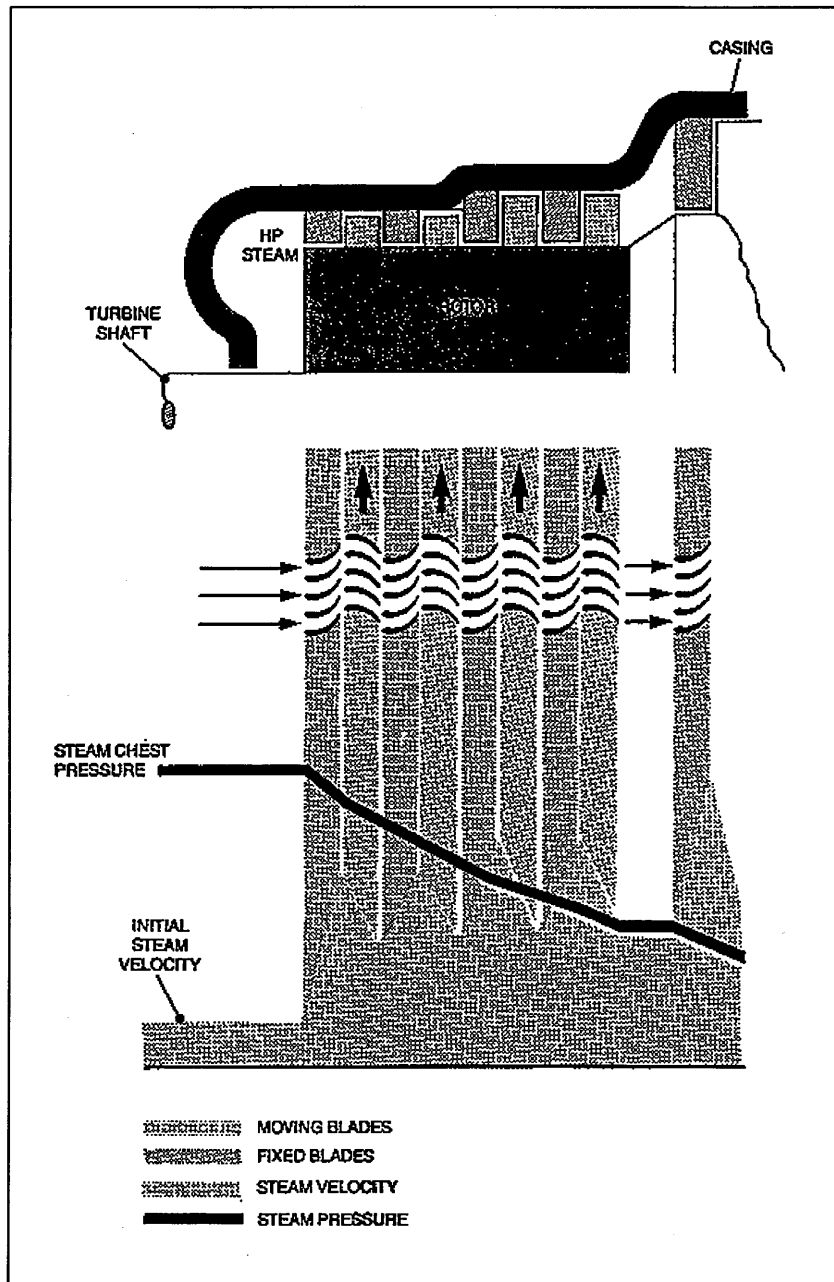


Figure 48: Reaction Turbine

3.4.3. Steam Turbine Modelling

3.4.3.1. Steam Turbine Efficiency Definition

Due to steam friction on the blades and to water condensation at back stages of the steam turbine the outlet entropy of the water/steam mixture is increased, reducing thus

the outlet temperature and the outlet enthalpy. But depending on the type of steam turbine the entropy increase can be significantly different, especially at part-load. In all cases steam turbine efficiency is expressed as follows:

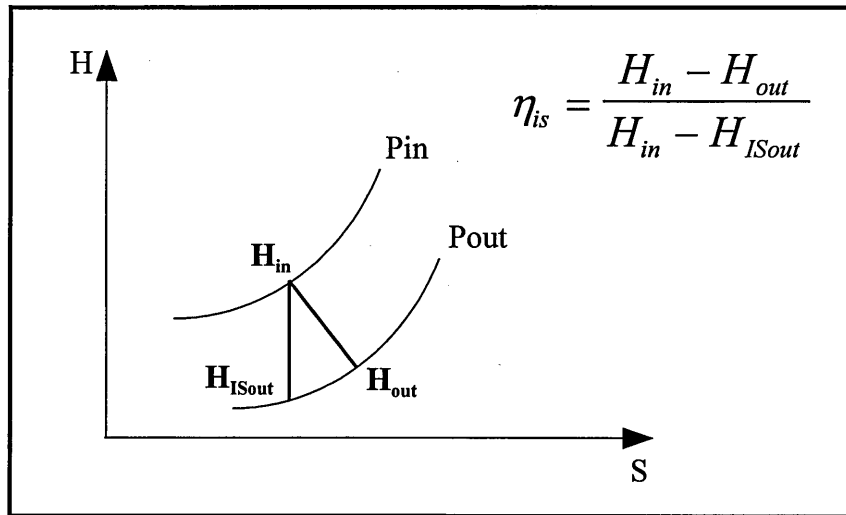


Figure 49: Steam Turbine Efficiency Diagram

3.4.3.2. Steam Turbine Models

When the user clicks on the steam turbine button, a certain number of input data is requested. Figure 50 shows the input data window displayed for a single-pressure plant. Even so most of manufacturers provide inlet and/or outlet steam turbine conditions, some other calculation modes are provided through a series of option buttons.

There are three options, based on:

1. the knowledge of the outlet enthalpy (option 2),
2. the knowledge of the steam turbine efficiency (option 3),
3. or the lack of thermodynamic and performance data and the knowledge of the steam turbine as a piece of turbomachinery (option 1).

As illustrated in Figure 50, option 1 is called “GE Spencer, Cotton & Cannon” (SCC) calculation mode. The steam turbine efficiency calculation is based on Spencer et al. (1974).

Steam Turbine (s)

High Pressure

Inlet Conditions

Inlet Temperature (°C): 520

Inlet Pressure (Bar): 50

Outlet Conditions

Outlet Pressure (Bar): 0.05

Steam Bleed

Bleed Pressure (Bar): 6

Bleed Mass Flow (kg/s): 0.4

Calculation Mode

GE Spencer, Cotton & Cannon

Outlet Enthalpy (kJ/Kg) 2372

Isentropic Efficiency (-) 0.8371

OK Cancel

Figure 50: Steam Turbine Input Data Window

The SCC methodology has been developed to predict the design and normal part-load efficiency of large steam turbines for conventional fossil-plant operation. The turbine performance analysis is carried out section by section (HP, IP and LP), that is why depending on the type of plant, the input data window that is displayed when the SCC option button is clicked can be different. In Figure 50, the HP condensing model would be called and the following input data window would be displayed.

Once all required data have been corrected input, when pushing the “OK” button, the SCC calculation is performed and a message window displays thermodynamic and performance results, so that the user can have the opportunity to assess the quality of the methodology for its own steam turbine based case.

Due to copyright restrictions, charts, figures and mathematical models, upon which the SCC calculation is based, can not be reproduced directly. A good summary of these key mathematical expressions can be found in GateCycle User's Guide.

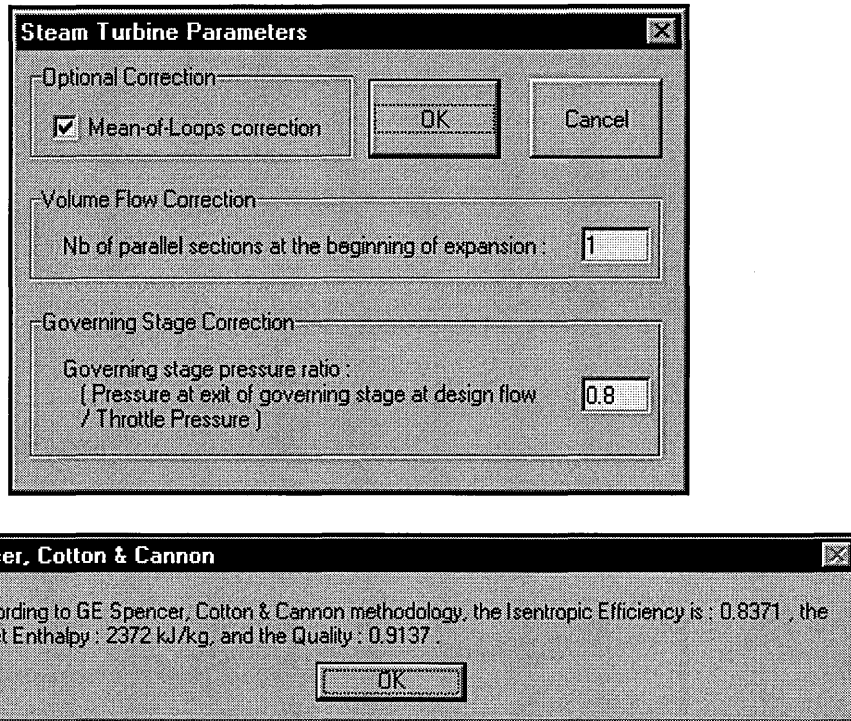


Figure 51: SCC Model for an HP Condensing Turbine

It must be pointed out that, if selected for the design-point calculation, the SCC calculation mode is also applied for off-design conditions, whatever the part-load control is.

3.4.4. Conclusion

Steam turbine performance calculation is always done in clean conditions. No degradation model has been developed. It must be known that steam turbine major overhaul does occur after 40,000 hours operation, sometimes later. The steam turbine contrarily to the gas turbine does not require off-line frequent inspections, that is why it is assumed that major overhauls are carried out concurrently on gas turbines and steam turbines. Calculation modes proposed by the DSS are not less numerous than what is proposed by existing plant simulation packages such as GateCycle. This means that a good level of accuracy can be expected as far plant performance results are concerned, at design and off-design conditions.

3.5. Other Components

3.5.1. Condenser

After the steam has surrendered its useful heat to the turbine, it passes to the condenser. The lower the back pressure, the higher the steam turbine output. This means that condenser operating pressure is very low, circa 50 mbar leading to a relatively low condensate temperature, circa 35°C.

Theoretically, an infinite condensing surface would give a condensing temperature equal to the temperature of the inlet cooling water (CW). However, as Figure 52 shows for a water cooled surface type condenser, that a practical margin, around 15°C is kept between the operating condensing temperature and the inlet CW temperature, but even so the size of condensing plant is considerable.

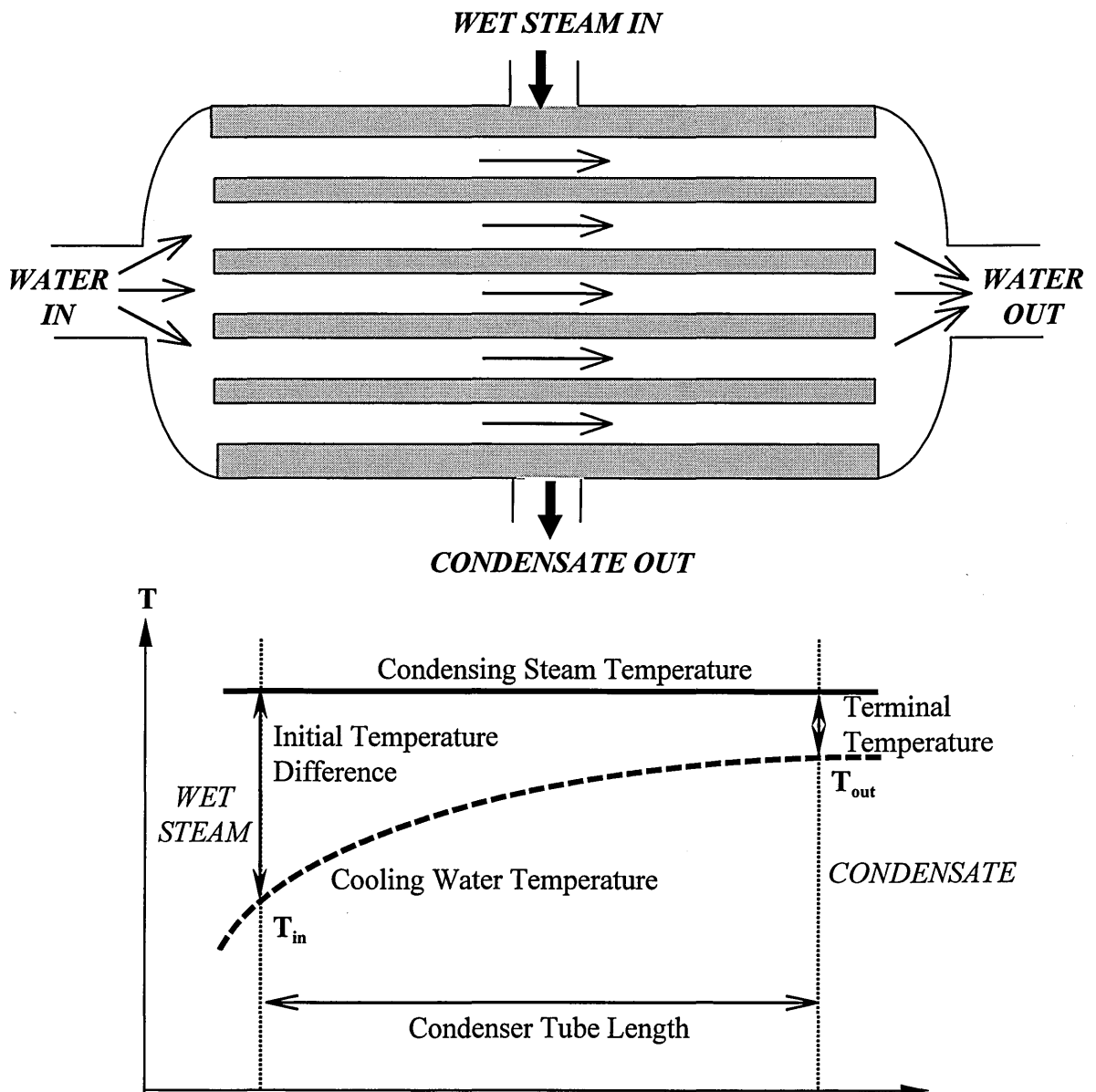


Figure 52: Condenser Temperature Diagram

In Modern Plant Practice, an instance of a traditional 660 MW unit shows that the condenser may have $20,000 \times 25$ mm dia. tubes, each 20 meters long giving thus a total condensing surface of about $31,500 \text{ m}^2$.

The major reason of this massive sizing is well illustrated in

Figure 53, since for a generator output of 660 MW, about 780 MW of energy is surrendered to the cooling water.

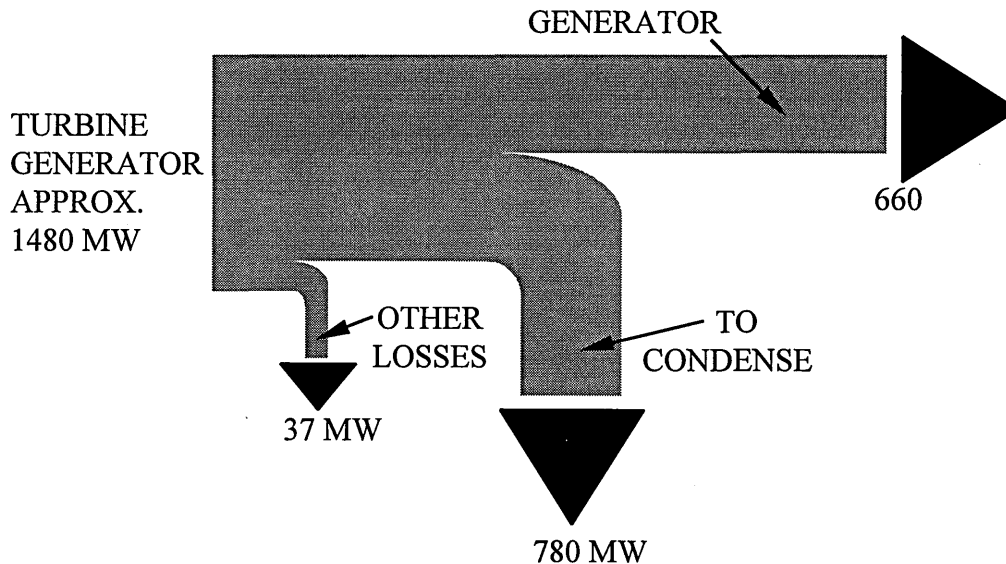


Figure 53: Heat-energy Distribution Diagram

Turbine back pressure or condensing pressure small variations can have dramatic economic impact. For a 2,000 MW station, if the back pressure is worsened by just 2 mbar, the cost of the additional heat required for keeping the power output constant would be about £250,000 per year (in extra fuel).

Undoubtedly condenser performance is a major operating parameter for a given station, so it is of paramount importance to clearly assess the factors which could worsen the back pressure.

Condensers must be designed to withstand steam generator overloads and their performance must be variable, in order to account for variations in the operating conditions, which exhibit frequent changes in temperature, pressure and steam quality.

The model implemented in the DSS requires inlet and outlet temperatures of the coolant as shown in Figure 54.

Figure 54: Condenser Data Input Window

As for plant simulation commercial packages, heat transfer coefficients are input data. However there are design codes, standards and specifications about condensers delivered by some official documents such as:

- the Heat Exchanger Institute (HEI) Standards for Steam Surface Condensers,
- the British Electrical and Allied Manufacturers Association (BEAMA) publication on the Recommended Practice for the Design of Surface Type Steam Condensing Plant.

Recognising the proven use of BEAMA and HEI design codes to obtain well designed condensers, further development work has resulted in a single individual formulae being established for each design code. These equations are:

$$1. U = 2.15(V^{0.5}) \left(0.7586 + 0.0135.T - 0.0001.T^2 \right) \text{ kW/m}^2\text{K (BEAMA)} \quad (6)$$

$$2. U = 2.70(V^{0.5}) \left(0.5707 + 0.0274.T - 0.00036.T^2 \right) \text{ kW/m}^2\text{K (HEI)} \quad (7)$$

Where V is the velocity of the cooling fluid through the tubes and T the mean cooling water temperature for BEAMA and the inlet cooling water temperature for HEI. HEI mathematical expression is the result of data sampling and correlation, but there is a method that can provide an overall heat transfer coefficient in a more accurate way based on the following equation:

$$U = C_1 C_2 C_3 C_4 \sqrt{V} \quad (8)$$

Where:

- V is the velocity of the cooling water through the tubes
- C_1 is a constant based on tube outside diameter
- C_2 is a correction factor for the cooling water temperature
- C_3 is a constant depending on the tube material and thickness

- C4 is the “cleanliness” factor

The way of calculating these correction factors is detailed in Appendix F.

Design and of-design performance of the condenser are highly dependent on coolant inlet condition. If the coolant is water and if the water is extracted from a river, the CW inlet temperature can vary by few degrees with seasons. Besides, environment regulations limit the CW outlet temperature by limiting the temperature difference under a permissible value, typically between 10 and 15°C, in order to prevent river's ecosystem from being damaged by hot water rejection.

This means that condensing operating pressure and temperature is set up with regard to CW inlet temperature and maximum permissible temperature rise.

The DSS does not provide a model for cooling towers or other types of cooling unit. The user only input CW inlet and outlet temperatures as shown in Figure 54. Off-design performance is calculated from an optimising process view-point, always using design-point CW inlet temperature.

It is assumed that the plant owners and users are tempted in keeping condenser duty as its best, whatever part-load conditions. The second assumption is that CW temperature difference is also kept constant in order to avoid non-permissible CW outlet temperatures.

In Modern Plant Practice, curves of optimum back-pressure versus load, for various CW inlet temperature are provided. From these curves, mathematical expressions have been made in order to assess back-pressure variations with regard to load variations at a given design-point back-pressure and at a given CW inlet temperature. Figure 55 shows the resulting curves.

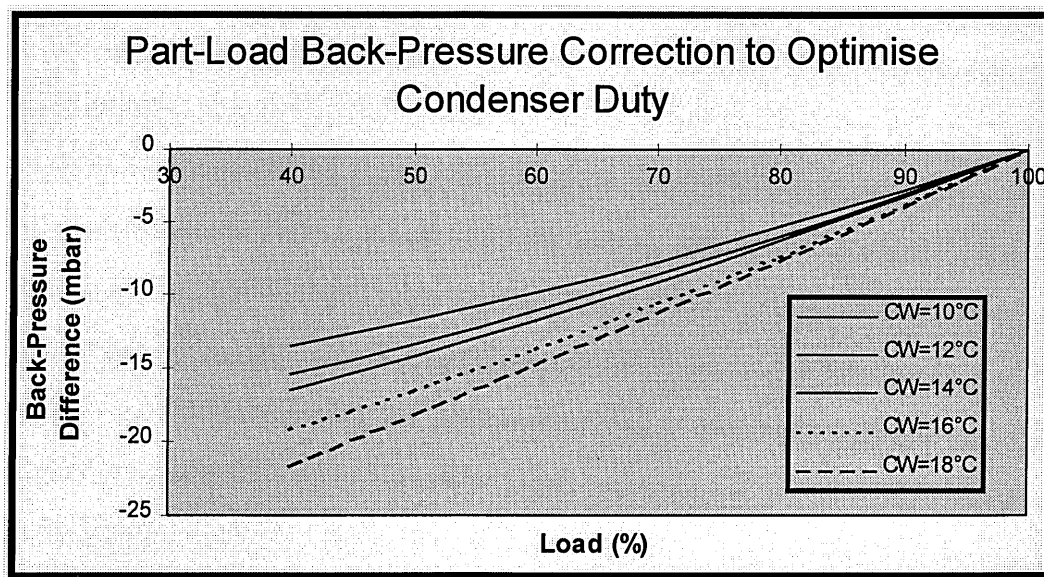


Figure 55: Curves of Optimised Back-Pressure versus load and CW Inlet Temperature

This part-load modelling obviously means that very low back-pressures are reached at low load. This model becomes weaker when the load becomes less than 50% since, as Figure 55 shows, the optimised back-pressure should be far less than the design-point pressure which is very difficult to achieve in reality. Therefore it must be kept in mind that at some part-load conditions, because of this model, condensing pressure and steam exhaust quality can be inaccurately estimated.

3.5.2. Deaerator

Deaerators are open feedwater heaters. They are operating near saturation in order to make the removal of non-condensables possible. For example high concentration of dissolved O_2 can dramatically increase corrosion rates, therefore its removal is extremely important.

Deaeration pressure is usually quite low, around two bars, but sufficiently high to play the role of feedwater heater.

In plant models proposed by the DSS, two types of deaerator can be found: an economiser-type and an evapo-type. In both cases, steam extracted from the steam turbine at a higher pressure than the deaerator operating pressure can be injected. It is assumed that in off-design conditions, deaeration pressure that sets up operating conditions is kept constant. This assumption is based on real off-design operating conditions.

Figure 56 shows the input data required by the modelling:

Deaerator	
Pressure Setting	
Deaerator Pressure (Bar):	2
Feeding	
Number of HRSG to be fed:	1
Heat Exchanger Sizing	
Heat Transfer Coefficient (W/m ² .K):	80
Number of Passes:	4

Figure 56: Deaerator Data Input Window

The second entry concerns cases where one deaerator is fed by several steam turbines and thus has to heat up and deaerate feedwater for several HRSGs. That is the case of the plant that is illustrated in Appendix C.

More information about deaerators can be found in Dechamps (1997) and in Modern Plant Practice Vol. D & G.

3.5.3. Recirculation

When a economiser-type deaerator is used for a multiple pressure CC plant, stack temperature in off-design conditions can be so low that flue gases can condense and thus become highly corrosive for HRSG materials. Using a recirculation loop enables the user to set up a fixed economiser inlet temperature, between deaerator and condenser temperatures, typically 60-80°C for natural gas, in order to control the heat extracted by the economiser.

It is assumed that a water back up is provided and that steam cycle performance is not affected. A dynamic modelling obviously would have been totally different.

Figure 57 is the displayed window when the recirculation button is pushed.

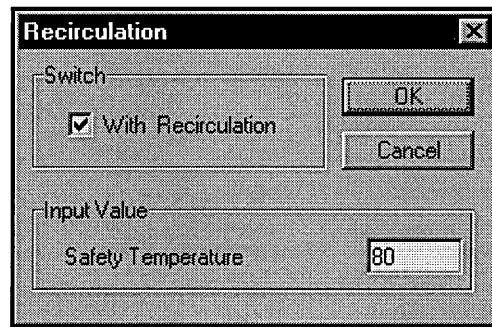


Figure 57: Recirculation Data Input Window

3.5.4. Pumps

In CC plants, there are several pumps with different pressure ratios. A typical condensate pump for a typical CC plant using deaeration compress condensates from 0.05 bar up to 2 bars. In case of advanced plant fitted with a dea-condenser, outlet pressure of the condensate pump can be as high as 18 bar, leading to a compressor ratio of about 360 if the condensing operating pressure is 50 mbar. But there are pumps which only deals with pressure ratios of 4 or 5, or even less.

Because isentropic efficiency of the pumps is not 100 %, there is an increase of water temperature, which is calculated in the DSS model as follows:

$$T_{out} = T_{in} + \frac{\Delta p \cdot v}{C_p \cdot \eta_{is}} \quad (9)$$

where:

- Δp is the pressure ratio
- v is water specific volume
- C_p is water specific heat capacity set as a constant
- η_{is} is the isentropic efficiency

It must be noticed that pump temperature ratios are quite small, sometimes even impossible to be read in the display of results.

Figure 58 shows pump model window:

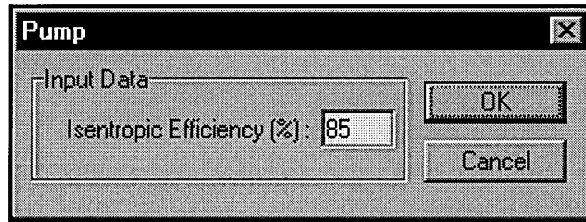


Figure 58: Pump Data Input Window

3.5.5. Fuel Preheater

Fuel preheater modelling is regarded as a normal heat exchanger. One fluid is hot water extracted from LP, LP or HP economiser of the HRSG and the other medium is the fuel, liquid or gaseous.

Plant manufacturers usually provide inlet and outlet conditions and thus it is assumed that these data can be known and input in the model as illustrated in Figure 59. Fuel heat capacity is calculated by taking into account both fuel chemical composition and temperature. Thermodynamic property relations are summarised in Appendix E.

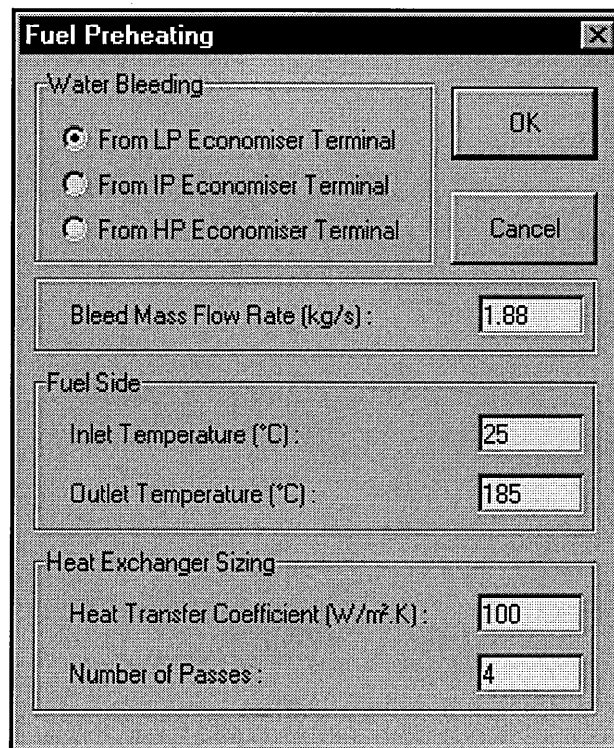


Figure 59: Fuel Preheater Data Input Window

3.5.6. Supplementary Firing

The model implemented in the DSS considers that after-burning, if switched on, will always occur whatever the ratio of fresh air available into gas turbine exhaust gases is. An outlet temperature is input as shown in Figure 60, and according to the fuel heat value and to the temperature ratio, a supplementary fuel mass flow is calculated.

Based on it, a stoichiometric O₂ mass flow is then assessed and compared with the amount of fresh air available for the combustion. If there is a lack of oxygen, a warning message box is displayed, and the level of oxygen after the post-combustion is set to zero, meaning that only the amount of air required for the stoichiometric combustion has been injected.

Figure 60: Supplementary Firing Data Input Window

3.6. Heat Exchanger Sizing

3.6.1. Introduction

CC plants involve a large number of different heat exchangers. Superheaters, economisers, evaporators, condenser and fuel preheater are among them.

Plant manufacturers usually do not provide any detail about heat exchanger surface, efficiencies and internal specific design. This information can be available on request, only if HRSG manufacturers are willing to providing them.

All plant simulation packages available on the market assess heat exchange surfaces with heat transfer coefficients as input data. This is obviously a weakness since heat transfer coefficients are very difficult to assess. Flows through tubes can have very different turbulence levels leading to temperature profiles that can be only predicted by a heavy numerical analysis based on turbulence and fluid flow dynamic physics.

The DSS will provide the same model as provided by commercial packages. Heat transfer coefficients are to be evaluated and input, as well as the number of passes, required if the ϵ -NTU approach is used. This is shown in Figure 61.

There are two major methods of calculating heat exchangers. The first is called the ϵ -NTU approach, where NTU stands for Number of Transfer Units. The second is called Log-Mean Δt approach or ΔT_{lm} . Both approaches have been programmed and it is up to

the user to decide which one is suitable for the design-point calculation. As far as the off-design calculation is concerned, the ΔT_{lm} approach is always used. This will be described in more details in the off-design modelling chapter.

Figure 61: HRSG Data Input Window

3.6.2. The ϵ -NTU Approach

The following method is extracted from Kays and London (1984) and GateCycle User's Guide.

The maximum rate of transfer occurs when one fluid leaves the heat exchanger at the same temperature as the temperature of the other fluid when it enters. The maximum heat transfer is therefore:

$$q_{\max} = (\dot{m} \cdot C_p)_h (T_{h,in} - T_{c,in}) = \dot{C}_h (T_{h,in} - T_{c,in}) \quad (10)$$

or

$$q_{\max} = (\dot{m} \cdot C_p)_c (T_{h,in} - T_{c,in}) = \dot{C}_c (T_{h,in} - T_{c,in}) \quad (11)$$

whichever is smaller.

Where:

- C_p is assumed constant
- h and c represent the hot and cold streams.

Therefore if $\dot{C}_{\min} = \text{MIN}(\dot{C}_h, \dot{C}_c)$, the rate of maximum heat transfer can be written:

$$q_{\max} = \dot{C}_{\min} (T_{h,in} - T_{c,in}) \quad (12)$$

Heat exchangers effectiveness can thus be defined as follows:

$$\varepsilon = \frac{q_{\text{actual}}}{q_{\max}} = \frac{\dot{C}_c (T_{c,out} - T_{c,in})}{\dot{C}_{\min} (T_{h,in} - T_{c,in})} = \frac{\dot{C}_h (T_{h,in} - T_{h,out})}{\dot{C}_{\min} (T_{h,in} - T_{c,in})} \quad (13)$$

Kays and London (1984) showed that for a constant overall heat transfer coefficient U, the effectiveness can also be written:

$$\varepsilon = f\left(NTU, \frac{\dot{C}_{\min}}{\dot{C}_{\max}}, \text{geometry}\right) \quad (14)$$

where:

$$NTU = \frac{U \cdot \text{Surface}}{\dot{C}_{\min}} \quad (15)$$

Kays and London (1984) proposed mathematical expression for equation 14 for different type of arrangements: counterflow, parallel flow, crossflow etc.

But most of heat exchangers involved in CC plants are crossflow types with one fluid mixed and the other unmixed. Following the same strategy as commercial packages, the following arrangement is the only one taken into account in heat exchanger sizing calculation.

Users can easily add further models if necessary.

The following set of equations is based on the arrangement illustrated in Figure 62.

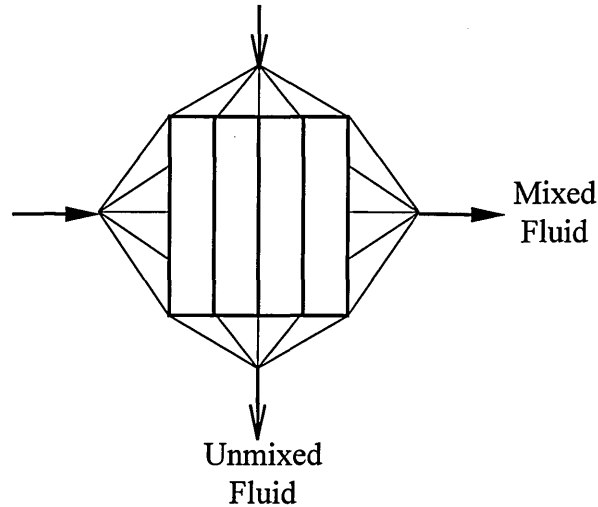


Figure 62: Crossflow Arrangement, One Fluid Mixed, the Other Unmixed

The ε -NTU equations, for a single pass are as follows:

- **Case 1:** $C_{\max} = C_{\text{unmixed}}, C_{\min} = C_{\text{mixed}}$

$$NTU = -\frac{C_{\max}}{C_{\min}} \ln(1 - \Gamma) \quad (16)$$

Where:

$$\Gamma = -\frac{C_{\min}}{C_{\max}} \ln(1 - \varepsilon) \quad (17)$$

- **Case 2:** $C_{\max} = C_{\text{mixed}}, C_{\min} = C_{\text{unmixed}}$

$$NTU = -\ln(1 - \Gamma) \quad (18)$$

Where:

$$\Gamma = -\frac{C_{\max}}{C_{\min}} \ln\left(1 - \frac{C_{\min}}{C_{\max}} \varepsilon\right) \quad (19)$$

- **Case 3:** $C_{\max} \gg C_{\min}$ (evaporator, condenser)

$$NTU = -\ln(1 - \varepsilon) \quad (20)$$

These equations combined with equation 15 makes the calculation of surface possible for any cases, but if the arrangement is a crossflow with multiple passes (N) others equations must be derived. Based on GateCycle User's Guide and on Kays and London (1984), the following methodology is proposed:

$$\lambda = \frac{\left(C_{ratio} - \frac{1}{\varepsilon}\right)}{\left(1 - \frac{1}{\varepsilon}\right)} \quad (21)$$

Where:

$$C_{ratio} = \frac{C_{min}}{C_{max}} \quad (22)$$

$$\varepsilon_p = 1 - \left(\frac{1 - \varepsilon_p C_{ratio}}{\lambda}\right)^{1/N} \quad (23)$$

$$\gamma = -\frac{C_{mix}}{C_{unmix}} \ln(1 - \varepsilon_p) \quad (24)$$

$$NTU = -N \frac{C_{unmix}}{C_{mix}} \ln(1 - \gamma) \quad (25)$$

ε_p is defined as the effectiveness of the single pass heat exchanger, and equation 23 is solved through an iterative process.

Equation 20 remains unchanged for boiling/condensing heat exchangers.

3.6.3. The ΔT_{lm} Approach

The ΔT_{lm} approach is fully explained in Kern (1950). The following expressions are derived from Figure 63.

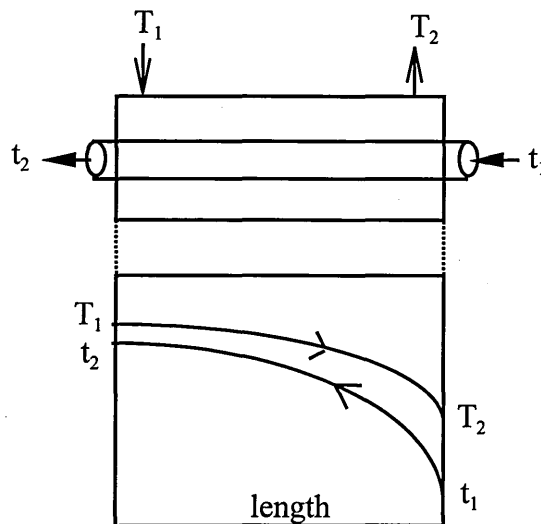


Figure 63: Counter-Current Type

Log-Mean Δt approach is derived as follows:

$$\Delta T_h = (T_1 - t_2) \quad (26)$$

$$\Delta T_c = (T_2 - t_1) \quad (27)$$

$$\Delta T_{lm} = \frac{GTTD - LTTD}{\ln(GTTD/LTTD)} \quad (28)$$

Where:

- GTTD = greater of ΔT_h , ΔT_c
- LTTD = lesser of ΔT_h , ΔT_c

The heat transfer can be written:

$$\dot{q} = U \cdot \text{Surface} \cdot F \cdot \Delta T_{lm} \quad (29)$$

To be accurate, the ΔT_{lm} approach requires a shape factor F that should be assessed with regard to the type of heat exchanger and its geometry. It is very difficult to find published mathematical models to calculate these shape factors. Saunders (1988) proposes one model for $\frac{1}{2}^+$ shells in series with any number of passes (N) (cf. Figure 64).

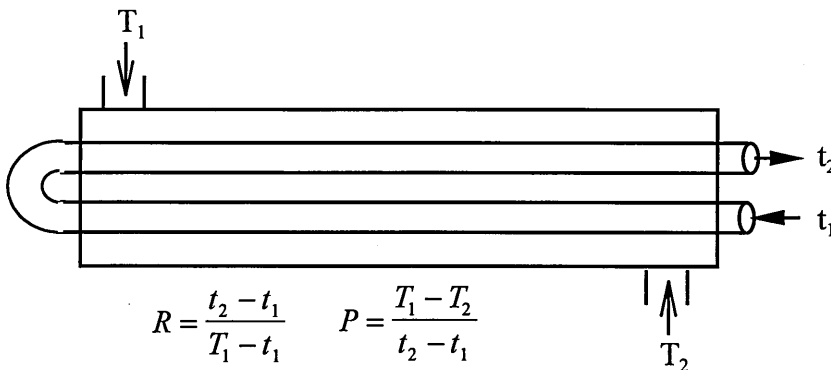


Figure 64: Shape Factor Model Definition

If $N=1$, equations 31a or 31b are used with $P = P^*$.

If $N>1$, equations 30a or 30b, and equations 31a or 31b are used as appropriate.

$$P^* = \frac{\left(\frac{1-PR}{1-P}\right)^{1/N} - 1}{\left(\frac{1-PR}{1-P}\right)^{1/N} - R} \quad \text{for } R \neq 1 \quad (30a)$$

$$P^* = \frac{P}{P - NP + N} \quad \text{for } R=1 \quad (30b)$$

$$F = \frac{(R^2 + 1)^{1/2} \ln\left(\frac{1 - P^* R}{1 - P^*}\right)}{(1 - R) \ln\left[\frac{2 - P^* \left[R + 1 - (R^2 + 1)^{1/2}\right]}{2 - P^* \left[R + 1 + (R^2 + 1)^{1/2}\right]}\right]} \quad \text{for } R \neq 1 \quad (31a)$$

$$F = \frac{2^{1/2} P^*}{(1 - P^*) \ln\left[\frac{2 - P^* (2 - 2^{1/2})}{2 - P^* (2 + 2^{1/2})}\right]} \quad \text{for } R = 1 \quad (31b)$$

F equals the unity when a phase change (evaporators and condenser) occurs in either one of the two heat exchanger fluids.

Therefore once the shape factor is calculated, the heat exchanger surface can be found using equation 29.

3.6.4. Results Display

Once design-point calculation is performed, the user can display sorts of results and among them is the heat exchanger sizing. Figure 65 shows all the results derived from models described previously.

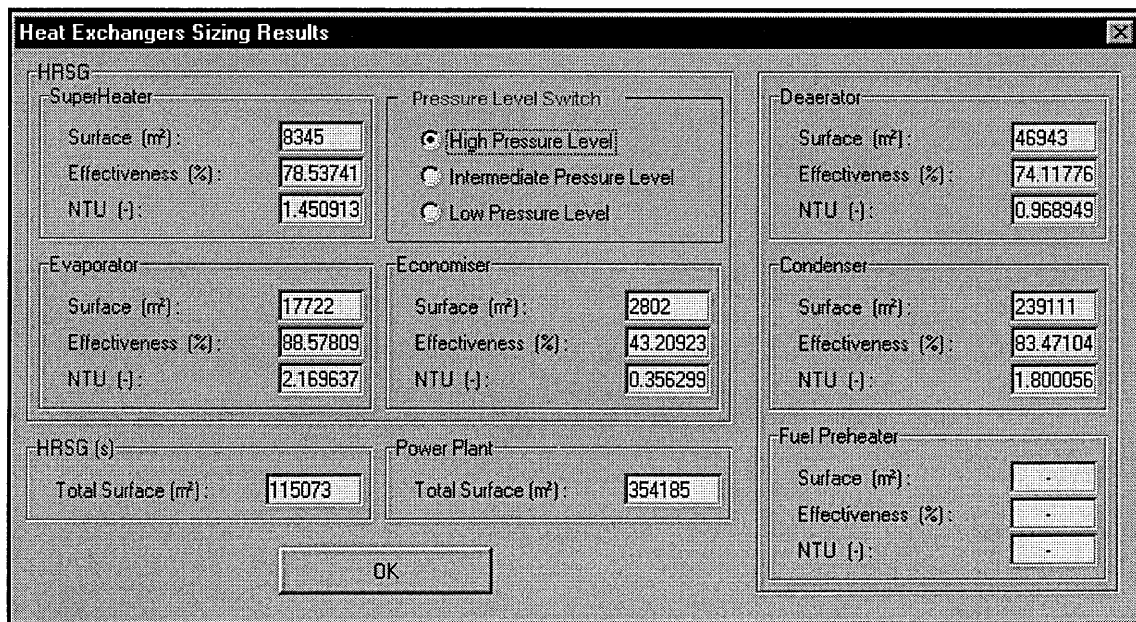


Figure 65: Heat Exchanger Sizing Results

The HRSG is displayed by pressure level and HRSG's total heat exchanger surface is provided as well plant total surface. It must be underlined that a techno-economic optimisation based on these calculated surfaces and pinch points is proposed in Dechamps (1995). But in that case, a rigorous heat transfer coefficients must be preliminary carried out in order to insure the consistency of economic results.

3.6.5. Conclusion

Even so two different approaches have been programmed, the choice is not directly given to the user. As matter of fact, off-design calculation only derives from the ΔT_{lm} approach. That is why only heat exchanger surfaces calculated by the ΔT_{lm} approach are taken into account and displayed. ϵ -NTU is also run during the design-point calculation in order to provide NTU values that can be compared with published data or plant simulation output data for validation purposes.

Further developments of the present software will certainly include the modelling of an off-design calculation based upon the ϵ -NTU approach, which obviously presents many advantages in terms of numerical analysis, since no guesses have to be done, but also in terms of accuracy since the ΔT_{lm} approach suffers from the lack of mathematical models for assessing all sorts of shape factors.

Chapter 4

OFF DESIGN PERFORMANCE SIMULATION

4.1. Introduction

Until now, only design conditions have been dealt with. They represent conditions under which the plant reaches its best overall performance. Even if these design-point performances are chosen for the selection of the CC plant, they are almost “theoretical” values, firstly because external conditions are rarely 15°C, sea level and 60% humidity, and secondly because technologies involved in combine cycle plants, such as the gas turbine are very sensitive to all other environmental factor such as air quality, fuel contaminants etc. In those conditions, we can expect the plant running at off-design condition most of the time, affecting thus overall plant efficiency and power output.

This chapter will not present in great details all the parameters causing off-design conditions and their relative impact on plant performance. All these details can easily be found in literature, especially in Dechamps (1997). The aim of this chapter is to present the choices made for modelling gas turbine, HRSG and steam turbine off-design control systems, according to what was found the most appropriate for a DSS, which obviously can not provide the same versatility as pure plant simulation packages.

4.2. Off-design Performance Theory

4.2.1. Ambient Temperature

The ambient temperature is certainly the most frequent off-design cause for the gas turbine. An ambient temperature of 15°C can not obviously be kept as compressor inlet temperature at every hour of the day for each month of the year. Equatorial regions that are reputed to have almost constant temperatures are much warmer and humid, 25-30°C and 70-80 % moisture.

The ambient temperature has a strong influence on gas turbine performance as shown in Figure 66.

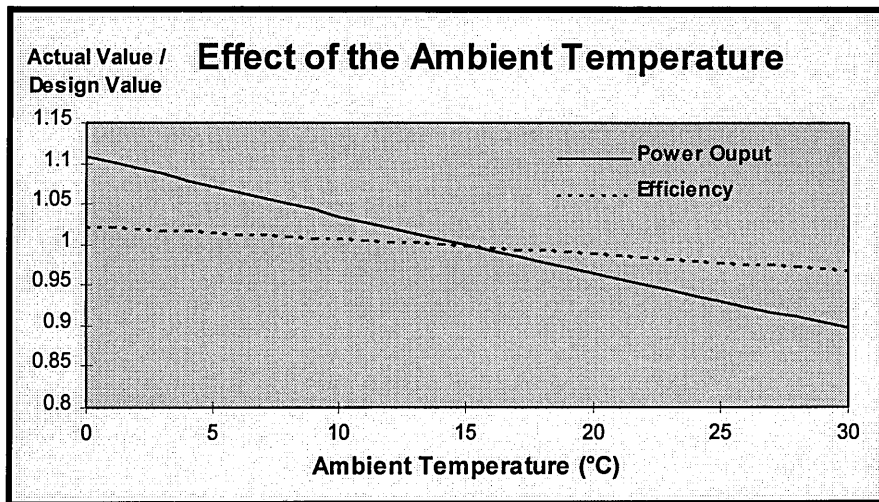


Figure 66: Effect of the Ambient Temperature on the MS9001FA

It is obvious that hot days can dramatically affect gas turbine performance by decreasing the inlet mass flow and the pressure ratio. That explains why “chilling” plant based on a cooled inlet mass flow provide better efficiency and power output than normal plants.

Besides, the ambient temperature affects gas turbine exhaust conditions which set up the performance of the steam cycle. According to Figure 67, even a small change in ambient temperature modifies both exhaust mass flow and temperature and thus makes the steam cycle running in off-design. However due to the fact that both parameters behave in an opposite way (cf. Figure 67), overall CC plant efficiency is not as sharply affected as plant power output is for instance.

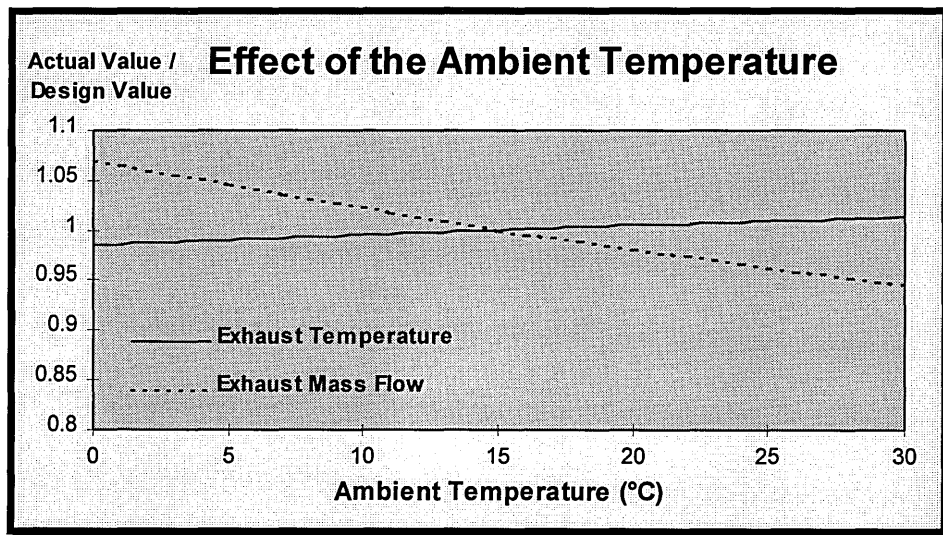


Figure 67: Effect of the Ambient Temperature on the MS9001FA

In conclusion, hot days mean for the plant less power output and maybe a poorer efficiency.

4.2.2. Gas Turbine Inlet Pressure Drop

Gas turbine inlet pressure drop or inversely inlet pressure recovery factor is mainly due to inlet air filters or/and ducts. A common value for this pressure loss is between 1 and 2 %. But monitored plant performance often shows that dirty filters can add non-expected performance losses. As illustrated in Figure 68, inlet pressure drop will reduce plant output but for the same reasons as the ambient temperature effect on exhaust conditions, the overall plant efficiency will remain almost unchanged.

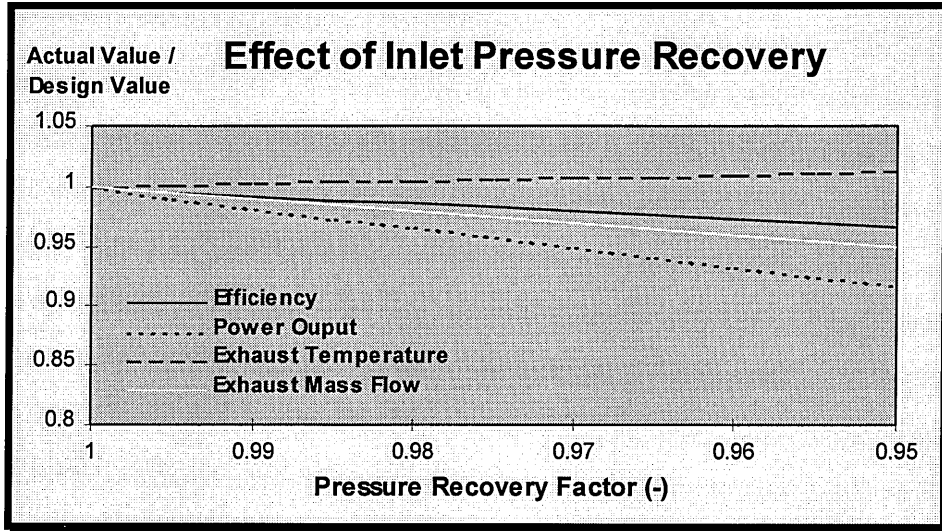


Figure 68: Effect of Inlet Pressure Drop on the MS9001FA

4.2.3. Altitude

Using a correction factor curve usually provided by plant manufacturers assesses the effect of altitude. It is clear that, once the plant site is known, a correction factor can be calculated and then be applied for design and off-design performances.

Figure 69 shows the effect on performance and gas turbine exhaust conditions.

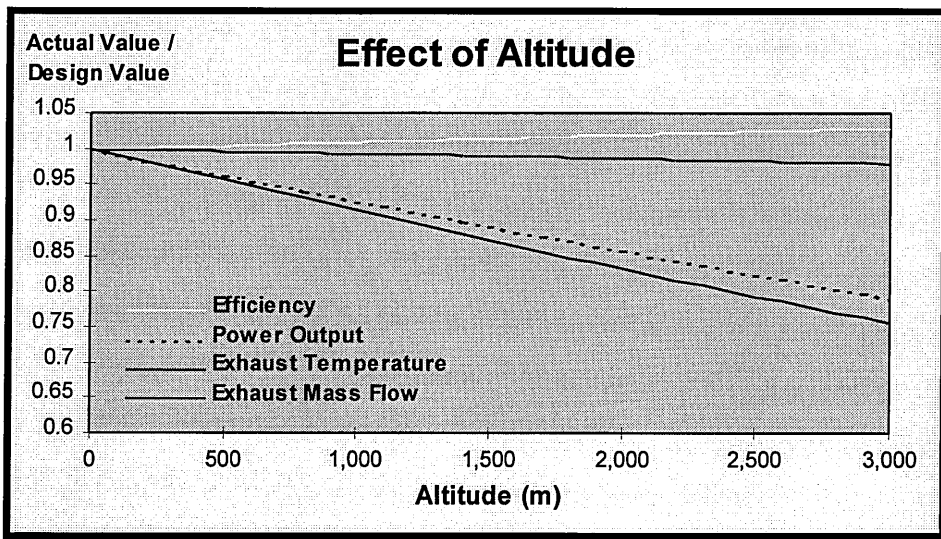


Figure 69: Effect of Altitude on the MS9001FA

The most dramatic impacts of altitude on plant performance is clearly on power output due to the drop of air density, that is why for power plants in high altitude locations like Mexico, additional units are usually considered only to recover this huge power loss.

4.2.4. Other Parameters

There are several other parameters such as condensing pressure that has already been discussed in a previous chapter, ambient pressure, ambient humidity, fuel quality, etc. Studies about these other parameters have been carried out in Dechamps (1997) for instance.

But in all cases, plant performance is affected, in a good or in a bad way and gas turbine exhaust conditions are always different than the design-point conditions.

Pythia and Turbomatch can tackle gas turbine off-design simulation, cumulating ambient temperature, altitude, inlet pressure, inlet pressure loss etc. effects at the same time. The only limitation of gas turbine off-design performance simulation is the knowledge of input data. It has been assumed that average monthly temperature can be easily found using published weather statistics, therefore a specific input window was designed and can be displayed as shown in Figure 70.

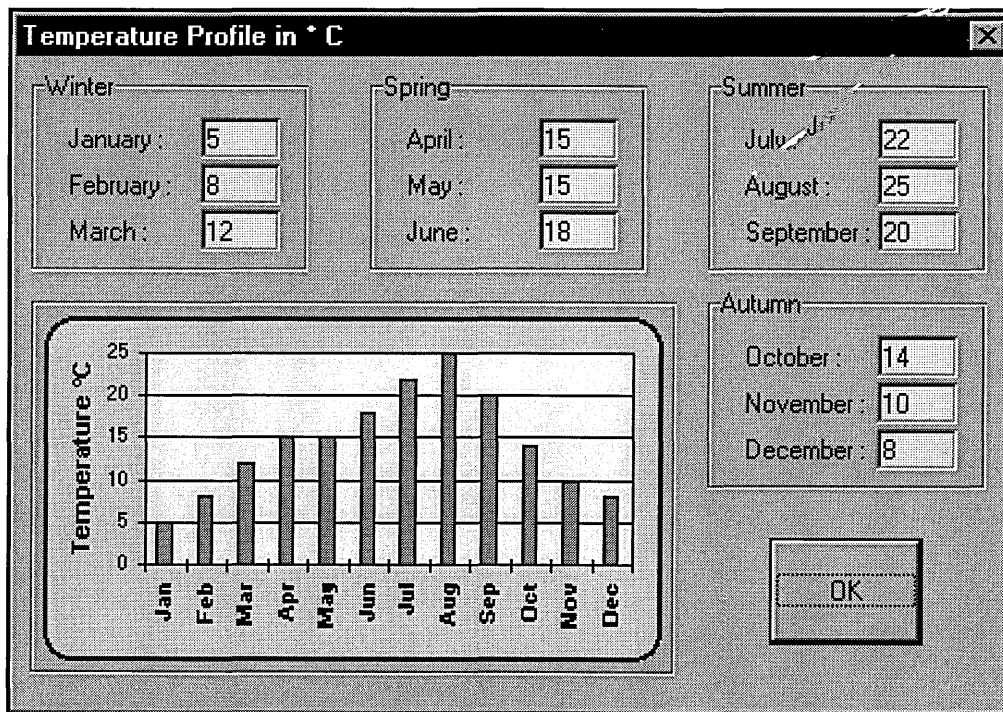


Figure 70: Temperature Profile Data Input Window

The chart is immediately updated when a new figure is input. If a user wants a daily or a weekly temperature profile, he must write the appropriate input file For001.dat, run Turbomatch and make sure that data processing is carried out, consistently with DSS models requirements.

Plant degradation and part-load duties are other sources of off-design conditions for the plant. Engine degradation is inevitable, therefore clean performance is as theoretical as design-point conditions. Part-load is a planned operating condition, but because engines are usually not running full load in order to follow load demand curves, flue gases going

through HRSG(s) are at different conditions than design conditions, making the steam cycle operating in off-design mode.

Both off-design condition sources required specific studies and specific modelling, since they represent very complicated and challenging issues that software developers and CC plant operators have both to face up.

4.3. Degradation Model

4.3.1. Introduction

Degradation is an inevitable phenomenon. All plant components are “living” entities suffering from wearing and damages. But rates and effects of these component deteriorations have totally different amplitudes. The gas turbine is recognised as the most sensible piece of mechanics in any CC plant. Because gas turbine operating conditions are extreme in terms of velocity and temperature, components such as compressors, combustion chambers and turbines are very sensitive to air impurities, creep and thermal fatigues. Engine performance degradation can easily reach 8% before operators decide to carry out maintenance work. Plant location, type of duties (base load, middle merit or peaking), maintenance inspection intervals, fuel type, engine model are very important factors which can dramatically modify component degradation rates. Degradation sources come from operating conditions, which can not be anticipated all the time, and from obvious facts, such as the type of duty, geographical location etc.

The DSS provides a very versatile and accurate engine degradation model through Pythia and Turbomatch but does not include a steam cycle degradation model.

4.3.2. Engine Degradation Methodology

Engine degradation calculation is based on the degradation model build in Pythia and Turbomatch. This model takes into account compressor, combustion chamber, turbine and exhaust deterioration. As shown in Figure 71, degradation ratios can be input for different type of degradation. In that case the compressor non-dimensional mass flow is reduced by 5%, resulting from compressor fouling for instance, reducing thus the isentropic efficiency by 1%.

By the same token, turbine non-dimensional mass flow and efficiency can be deteriorated. For instance turbine corrosion could increase by 1 or 2 % the non-dimensional mass flow, where as turbine fouling could decrease it by 3 or 4%, giving to an overall decrease of about 2 %.

Compressor, turbine, combustor and exhaust duct degradation rates must be assessed and both engine simulation programs can then calculate degraded performance.

In reality, CC scheme planners are more likely to know non-recoverable plant performance degradation rates because plant manufacturers provide them in order to support economic appraisals carried out during preliminary studies. This means that

these performance rates must be converted into component degradation rates, supposing then that individual component degradation are well known in terms of their effect on the plant performance, but also in terms of their likelihood to occur at a particular point in time. To carry out such an assessment, planners must have some experience with the type of machines run in combined cycle in a roughly similar type of duty and environment.

Compressor

Brick Data:

5	Z	0.85
6	PCN	1.
7	Pressure Ratio	3.659
8	Isentropic Eff.	0.85
9	Error Selection	0.
10	Compressor Map	1.

Station Numbering:

2	Inlet
3	Outlet

Engine Vector Results:

301	COMWK
-----	-------

Possible Variables:

- Z
- PEN

Deterioration [+/- %]:

- Non-Dimensional Mass Flow
- Efficiency

Buttons: Save, Cancel, Help

Figure 71: Pythia's Compressor Degradation Model

Besides monitored gas path data could be essential to exactly assess degradation origins such as fouling, corrosion or erosion. The DSS for the time being is not fitted with a model that would translate input plant degradation curves into gas turbine degradation rates occurring in specific components. However a methodology has been developed for designing degradation data banks for the engines available in the DSS library.

Deterioration is affecting clean engine design-point and off-design exhaust conditions. The steam cycle is totally depending upon gas turbine exhaust mass flow and temperature, therefore the most accurate the calculation of these two figures is, the most accurate steam cycle performance assessment is likely to be.

The main concept is that every type of engine component degradation independently affect performance and exhaust conditions. This can be illustrated by the following example.

Figure 72 and Figure 73 show deteriorated performance of the GE MS9001FA for different types of component degradation. In that particular case, and according to Escher (1995) a drop by 5% of compressor non-dimensional mass flow (fouling effect for instance) implies that compressor isentropic efficiency is reduced by 1%. Similarly

an increase by 2 % of the turbine non-dimensional mass flow (erosion effect for instance) implies that turbine isentropic efficiency is decreased by 1%. It is assumed that both non-dimensional mass flows are deteriorated at the same rate, and that isentropic efficiencies are linearly reduced.

Figure 74 illustrates the case where both compressor and turbine non-dimensional mass flows are similarly and simultaneously deteriorated. This chart shows that overall performance degradation levels are almost the addition of the first two performance degradation levels where components were separately deteriorated. This is confirmed in Figure 75 for the case of the exhaust mass flow.

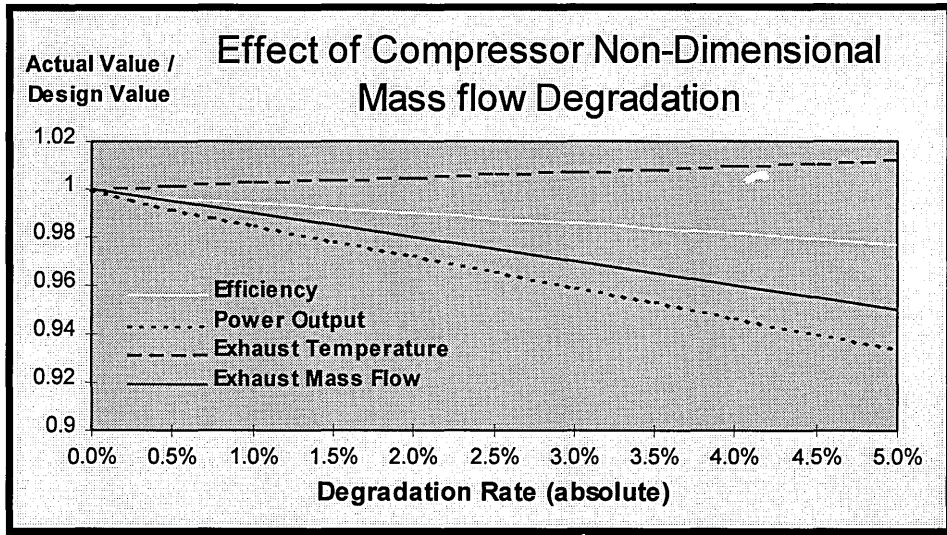


Figure 72: MS9001FA Deteriorated Performance

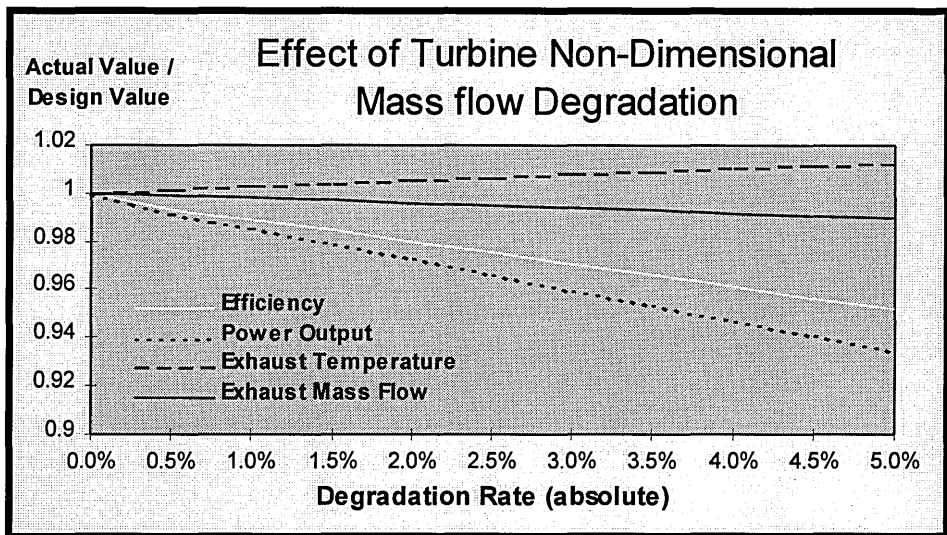


Figure 73: MS9001FA Deteriorated Performance

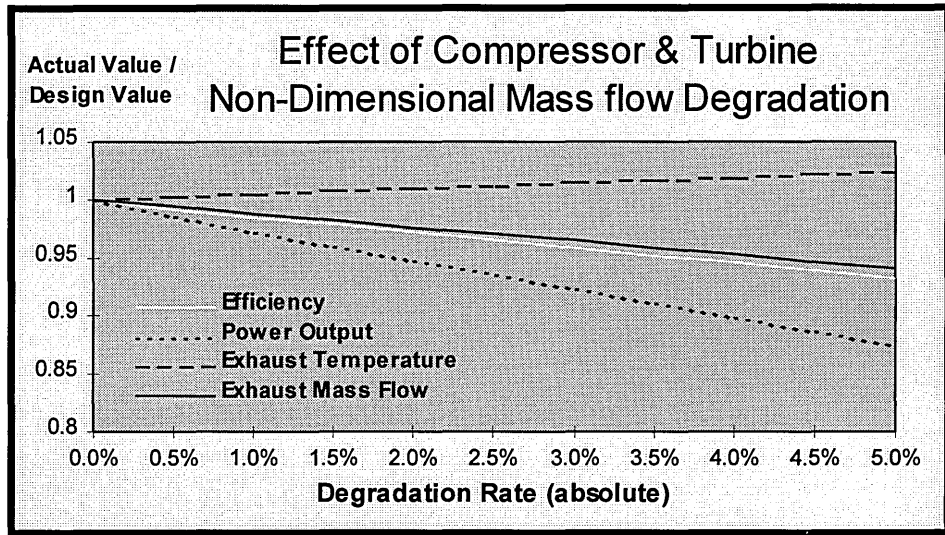


Figure 74: MS9001FA Deteriorated Performance

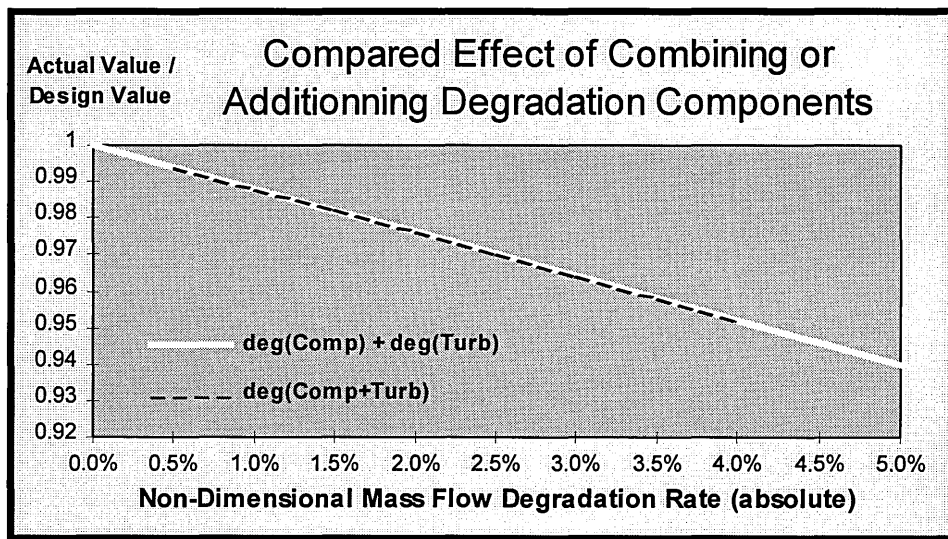


Figure 75: MS9001FA Deteriorated Performance

The main idea is firstly to translate plant performance losses into equivalent engine performance losses. Plant performance degradation curves must be only considered as base lines. Afterwards with regard to user's selections, as shown in Figure 76, specific rates of degradation are attributed to every engine component that can be deteriorated, namely compressor(s), turbines(s), combustor(s) and exhaust duct(s).

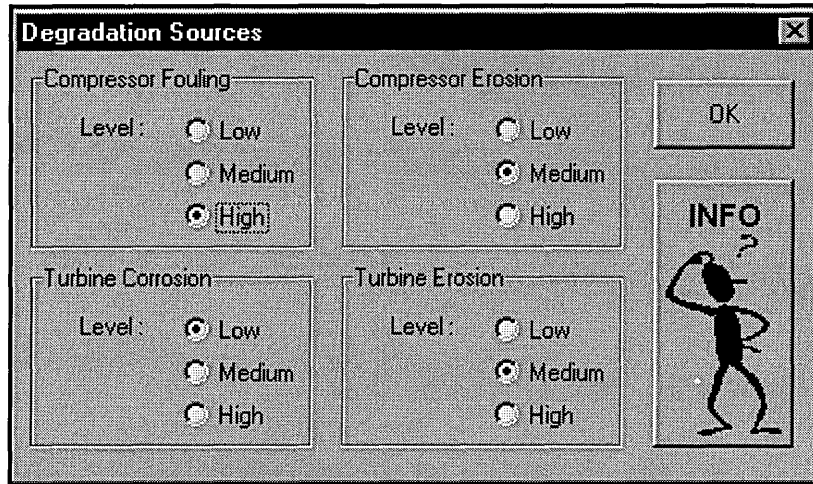


Figure 76: Degradation Level Data Input Window

Finally based on plots like in Figure 72 and Figure 73, exhaust mass flow and exhaust temperature differences are calculated and then added in order to provide final HRSG inlet conditions as illustrated in the following mathematical expressions:

$$\Delta T_{deg}^{GTout} = \Delta T_{deg}^{Comp} + \Delta T_{deg}^{Turb} + \Delta T_{deg}^{Comb} + \Delta T_{deg}^{Exh} \quad (32)$$

provided by curve fitting
mathematical expression,
read in the library for a

$$T_{in}^{HRSG} = T_{clean}^{GTout} + \Delta T_{deg}^{GTout} \quad (33)$$

The same type of expressions can be written for the exhaust mass flow.

A good degradation model should include a proper assessment of different component degradation rates as a function of engine fired hours. The DSS does not provide that for the time being, even if a specific data input window has been designed. A user can however follow the previous methodology with his specific component degradation rates by inputting them into Turbomatch and by running the simulation for several points in time. Then by processing the results file, For003.dat into Excel, charts similar to those in Figure 72 to Figure 75 can be plotted and curve fitting can be executed in order to find simple and straightforward mathematical expressions that will provide for any level of engine degradation, HRSG inlet conditions necessary to calculate steam cycle performance.

Even so the DSS does not provide for the time being, a specific model that would execute the methodology described previously, it does however include a user interface capable of reading plant performance degradation curves.

4.3.3. Plant Degradation Model

Plant manufacturers provide expected CC non-recoverable performance loss plotted against the number of fired hours. This kind of plot is illustrated in Figure 77.

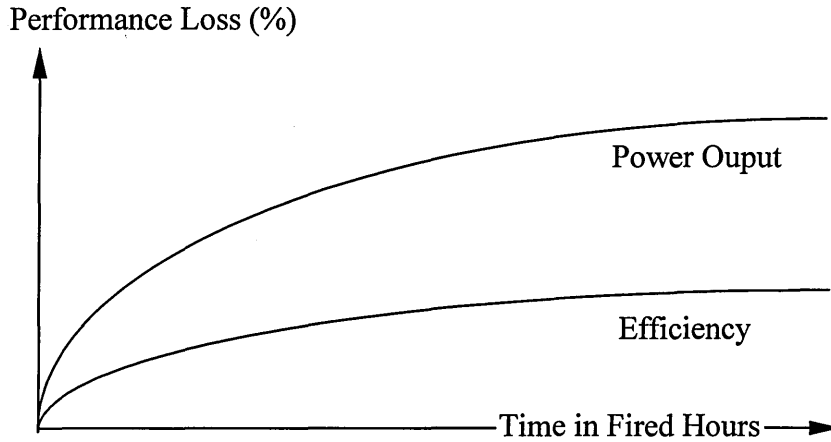


Figure 77: Expected Non-Recoverable Performance Loss Pattern

In compressors, non-recoverable performance losses are typically caused by non-deposit-related blade surface roughness, erosion and blade tip rubs. In turbines, nozzle throat area changes, bucket tip clearance increases and leakage are potential causes. But erosion, especially in desert regions due to sand, corrosion, very high in off-shore applications and transient conditions, extensive in peaking duties, are other factors that can dramatically deteriorate the engine in an irreversible way.

The DSS model is based on the following chart (Figure 78):

It shows that in average, the loss during operation has the same pattern as the non-recoverable loss curve. This kind of exponential curves followed by a levelling-off can be expressed as follows:

$$y = A_0 \left[1 - \exp\left(-\frac{t}{\tau}\right) \right] \quad (34)$$

Where:

- y is the performance loss,
- t is time in fired hours,
- A_0 is the levelling-off value,
- τ is a characteristic value that can be assessed using the value of the derived function y' at $t = 0$, as follows:

$$y'(t = 0) = \frac{A_0}{\tau} \quad (35)$$

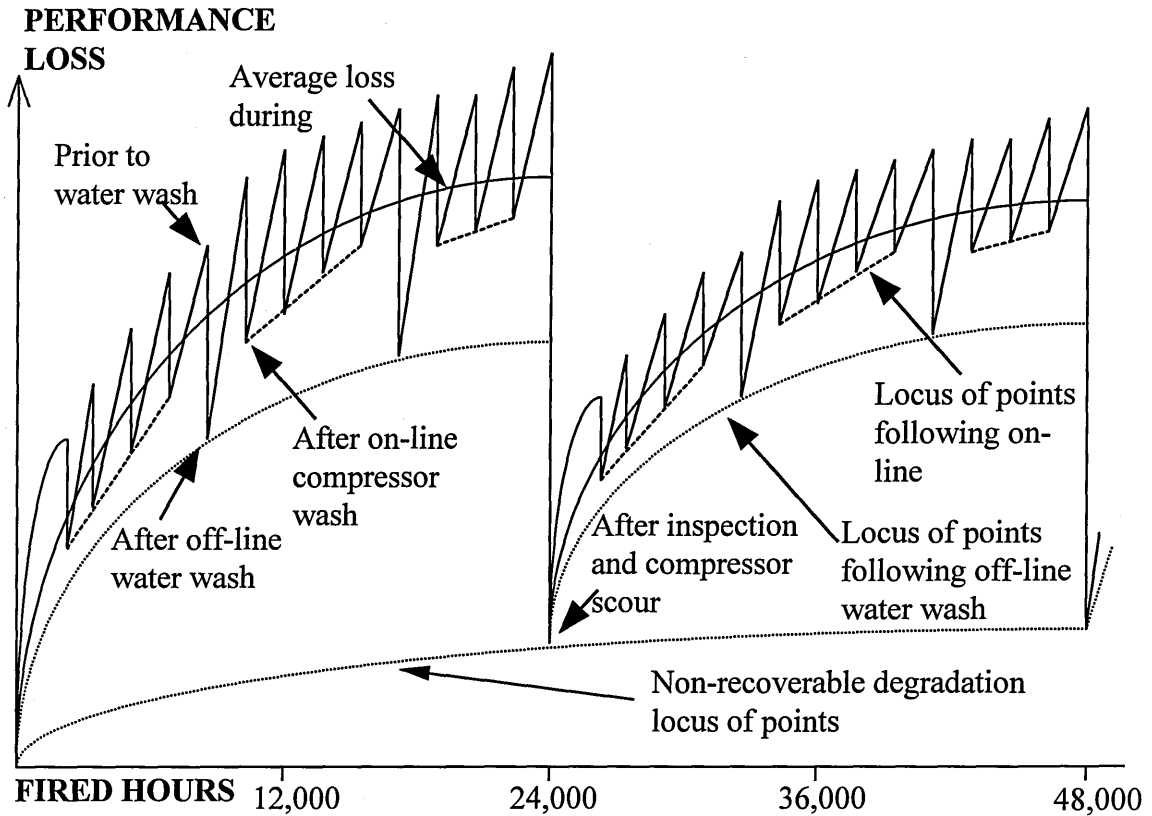


Figure 78: Recoverable and Non-recoverable Performance Loss Pattern

Figure 79 shows that this mathematical expression can reproduce quite accurately degradation curves provided by manufacturers.

NB: Values for confidentiality can not be displayed.

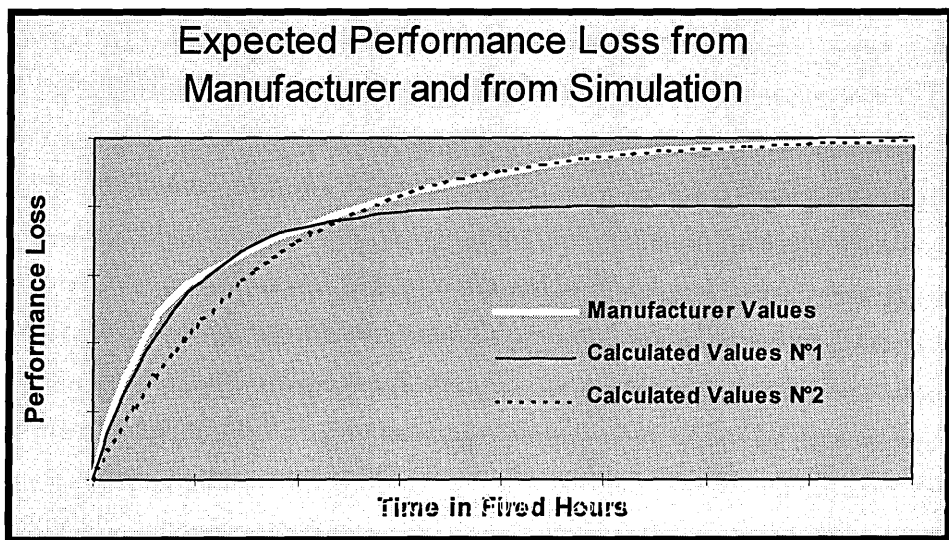


Figure 79: Comparison between Manufacturer and Simulated Curves

It is important to notice that non-recoverable degradation curves range from the start to the end of the plant life. For base load applications, plant life can be about 200,000 fired hours for a 30 years project. The levelling-off of non-recoverable performance loss curve is usually reached after 40,000 fired hours, therefore it is important that simulated curve fit well manufacturer data when operating time exceeds 20,000 or 30,000 fired hours. On the contrary, average loss during operation is partially or totally recovered after maintenance works. Hot gas path components are usually checked and maintain every 24,000 fired hours for base load, and much less than that, for cycling or peak duties. That is why in that particular case, the accuracy of the simulated curve must be in the first 30,000 hours since the levelling-off of average loss during operation curve is never reached, as illustrated in Figure 78.

Two sets of simulated curves have been plotted in Figure 79, in order to highlight the fact that only part of manufacturer degradation curves must be well fitted by simulations. Set N°1 could represent the average loss during operation curve, where time scale is 1,000 fired hours, whilst set N°2 could illustrate non-recoverable performance loss simulated curve with a time scale of 10,000 fired hours.

The user interface provides tools to enable this change of time scale in order to enhance values reading in locations of the curve that only interest the user. Figure 80 shows the DSS window of the degradation modelling.

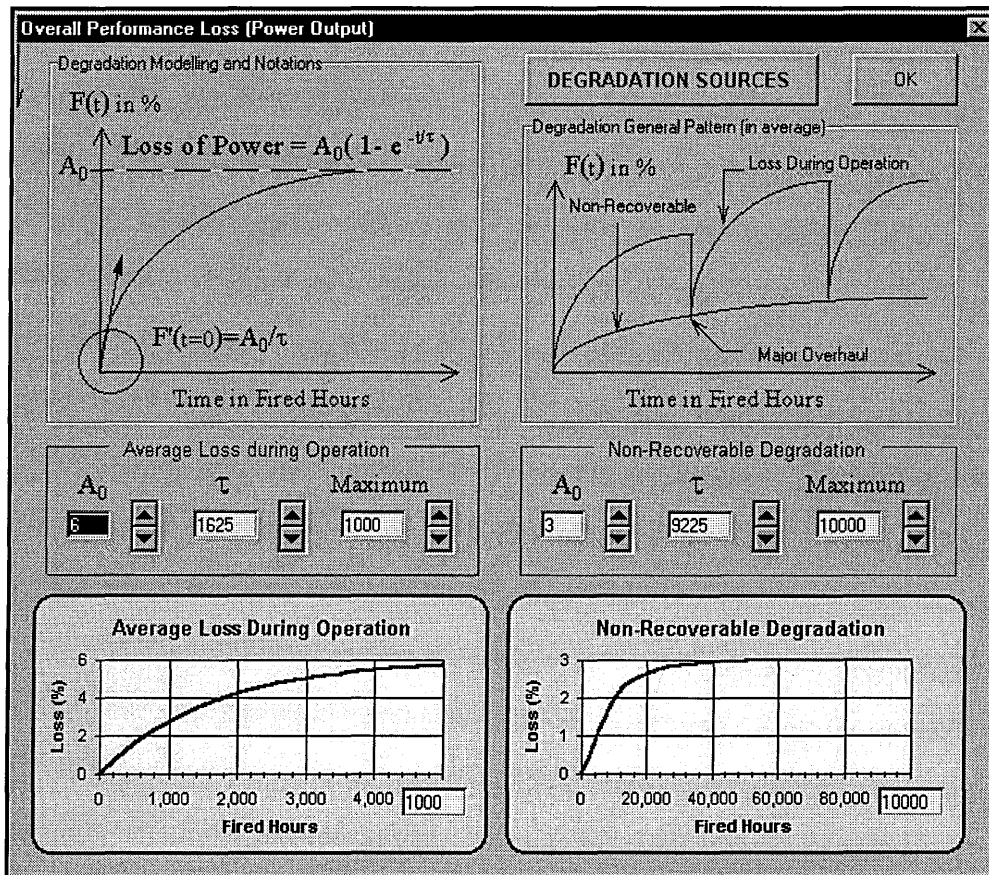


Figure 80: Degradation Model Data Input Window

4.3.4. Conclusion

Plant component deterioration is of major concern for planners but it is an irreversible phenomenon, which is still not well identified and anticipated. However based on previous power generation experience and based also on plant manufacturer experience and their data bank, it is possible to assess degradation rates.

The DSS dealt with engine deterioration in a very powerful and accurate way, but technical knowledge are required, as well as few performance loss data to initiate the assessment. The degradation model has been programmed in such a way that future developers should be able to apply any type of degradation modelling.

Because of engine degradation, steam cycle runs in off-design conditions, thus the overall plant output is not likely to achieve clean off-design targets. That is why planners tend to oversized power plants, especially for base load applications, since the viability of the scheme relies on plant availability and capacity factor, which is the ratio of the actual power output on the clean full power output at same off-design conditions and at the same point in time.

4.4. Part-Load Operating Conditions

4.4.1. Introduction

Part-load is the operating condition where full power is not required. Most of the time, plants operating at part-load have a "master-slave" behaviour which means that the actual control of the load is made through gas turbine(s) settings whilst HRSG(s) and steam turbine(s) are just following exhaust gases condition changes.

The exception of this type of behaviour is when a throttling control is used. In that case, operators can modify inlet steam turbine enthalpies using a throttle valve, in order to reduce its power output, without modifying the gas cycle. Throttling reduces plant efficiency quite dramatically compared to other part-load governing strategies like sliding pressure or supplementary firing. But for low load demands, throttling is proven to be better than any other sort of part-load controls.

The DSS does not provide throttling control because it has been recognised that even cycling duties usually operate over 40% load most of the time. If a user feels that efficiencies have been over or underestimated due to the lack of throttling control model, the latter can be modified before running the calculation.

The steam cycle is therefore always running in sliding control mode.

4.4.2. Part-load Control Systems

4.4.2.1. VIVG and TET Control

Industrial gas turbines are nowadays fitted with one or several rows of variable inlet guide vanes (VIGVs), which can be closed in order to reduce compressor mass flow and thus power output. They are a very efficient ways of controlling thermal efficiency as well as compressor surge margin. When the engine is equipped with several rows of VIGVs, the load can be reduced down to 70% before they are fully closed.

Obviously when the load demand is less than 70%, operators must use another control system. They usually then reduce the TET in order to derate the turbine power output. The consequence in this case is not the reduction of the mass flow but a sharp drop in exhaust temperature. That is why operators tend not to use TET reduction for part-loads ranging from 99 to 70-80% because CC efficiency is very sensitive to engine exhaust gas temperature.

Sometimes the TET is slightly increased when VIGVs are being closed in order to keep exhaust temperature constant, assuring good steam cycle efficiency. Another way would be the use of nozzle guide vanes (NGVs), at the front of the turbine, which would limit exhaust temperature drop, when being closed.

As illustrated in Figure 81 and Figure 82, both part-load control systems have a dramatic opposite effect on exhaust temperature. That can explain why despite the decrease of mass flow, VIGV control due to higher exhaust temperature provide an overall better efficiency than TET control. This is clearly illustrated in Figure 83.

As these plots show, all curves can be accurately modelled by linear mathematical functions. Correlated coefficients are stored into the engine library and are used by the DSS when capacity factors are calculated after having taken into account off-design and degradation conditions.

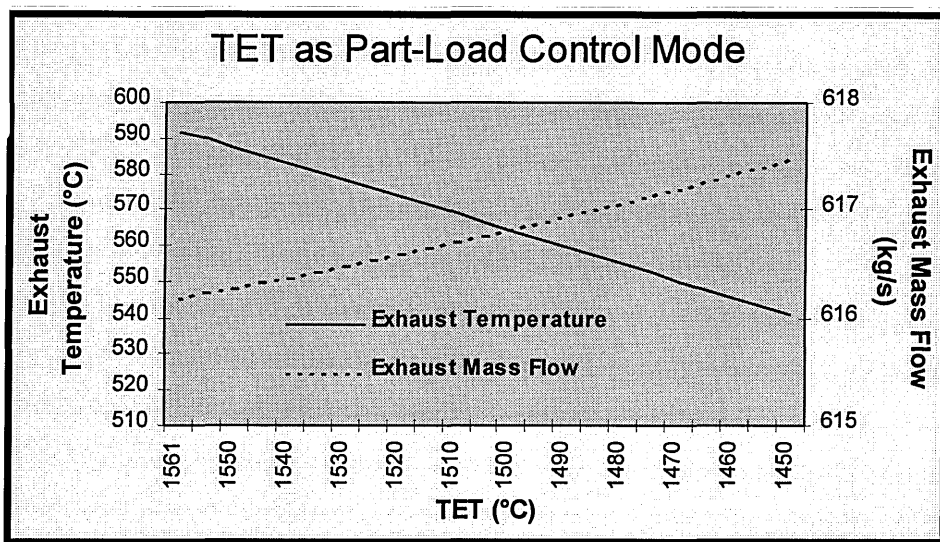


Figure 81: TET Control for MS9001FA Part-Load

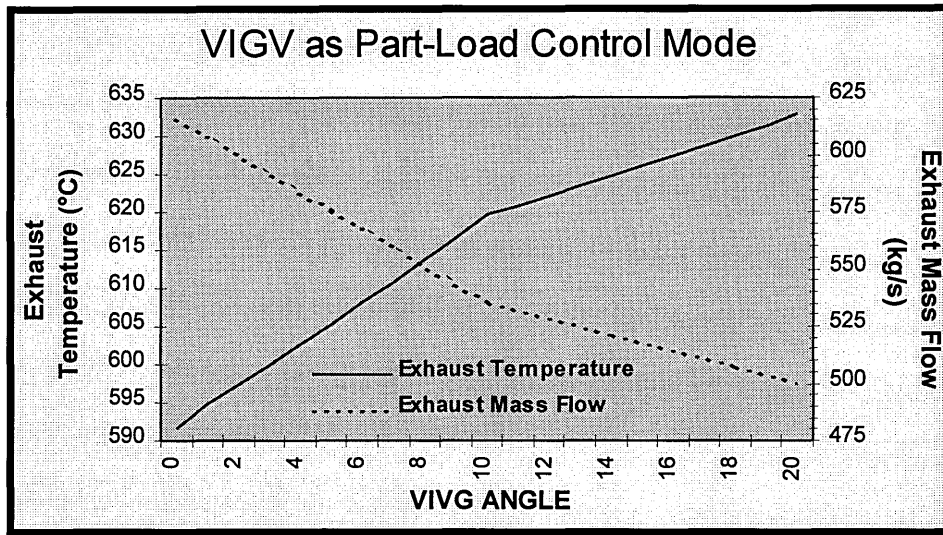


Figure 82: VIGV Control for MS9001FA Part-Load

For a given capacity factor, which is actually the load demanded at one point in time, an engine efficiency is allocated and gas turbine exhaust conditions are modified according to the same methodology as for degradation and off-design conditions. This means that exhaust temperature and mass flow differences are assessed for a specific engine part-load setting, and added to full load deteriorated exhaust temperature and mass flow.

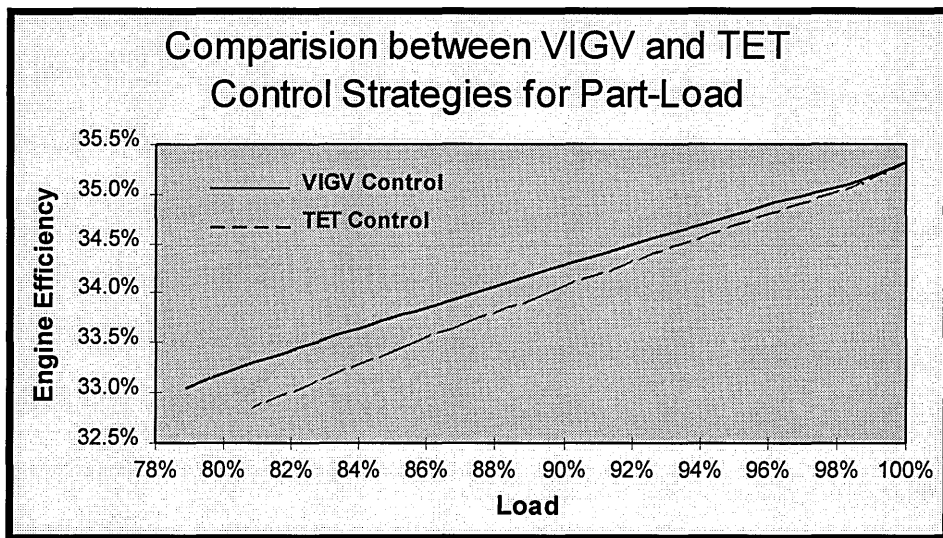


Figure 83: Compared Strategies for MS9001FA Part-Load Applications

A combined utilisation of VIGV and TET part-load control systems is illustrated in Figure 37 and the improvement in efficiency is obviously cost-effective.

It must be highlighted that mid-merit CC plant schemes that could become very popular, especially in load following duties, require high quality part-load performance

simulation in order to accurately assessed internal rate of return, usually very dependent on fuel cost and electricity retail price. This can only be reached if plant availability is optimised with regard to plant efficiency at part-load. One way of managing both is the use of a sequential loading operating strategy.

4.4.2.2. Sequential Loading

The main idea of sequential loading resides on the well-known fact that gas turbine performance decrease when the load is reduced. In those conditions, if the plant owns several units running in parallel like for the Korean Seoinchon CC plant which counts eight multi-shaft units, each powered by one MS7001F gas turbine, it is worth keeping some of the units at full load or closed to full load, whilst others are simply shut down. When 50 % load is required, four gas turbines running at full load will always provide a much higher plant efficiency than eight units operating at mid-load. This concept, well applied in large CC plants, is illustrated in Figure 84.

This concept however hides quite complicated issues. It becomes very difficult to manage several units in parallel when powered by several gas turbines power a single steam turbine as for 2-2-1 or 3-3-1 configurations presented in a previous chapter. In that case, sequential loading could be extended to the scale of a unit making performance assessments very complicated.

Besides sequential loading give the opportunity for plant operators to optimise maintenance inspection and overhauls, increasing thus plant availability and profits.

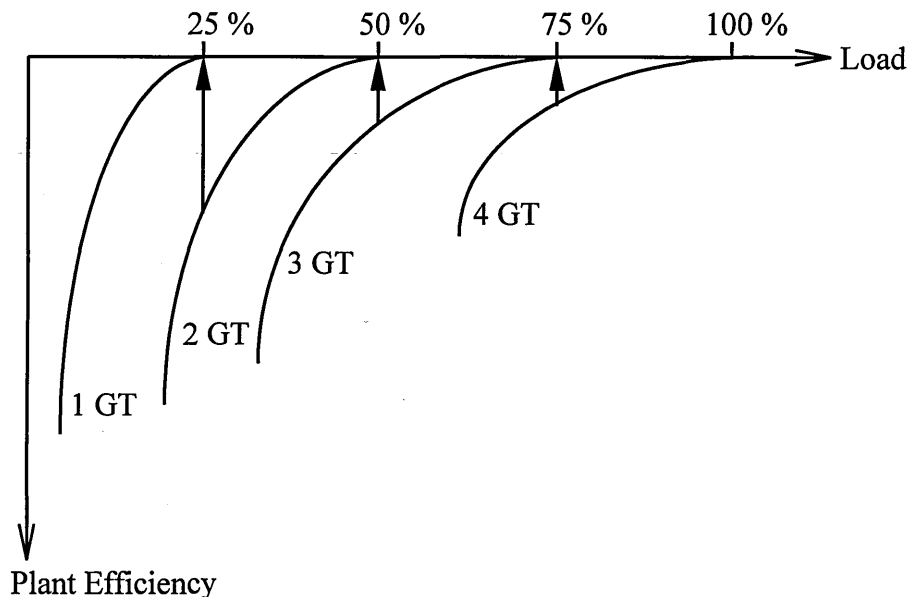


Figure 84: Sequential Loading Concept

4.4.2.3. Other Controls

There are plenty of other part-load control systems such as compressor inlet air heating, compressor bleeding, supplementary firing etc. But none of them except supplementary firing in specific applications, studied in Dechamps (1997) provide better efficiency than VIGV and TET control, with sequential loading if the plant makes it possible.

4.4.3. Part-load Model

The DSS as illustrated in Figure 85 provides base load and cycling load models. Cycling load requires hourly load for the twelve months of the year. Power plants, which follow load demand from big cities for instance, can be simulated. Even peaking duties can be assessed since load up to 110 % can be input as shown in Figure 85. When base load application is selected, a overall capacity factor is required. It is expected to be very close to 100%.

When cycling duty is selected, the user can specify if he wants a VIGV/TET control and then input the value of the load after which the TET starts to be reduced. He can also select the utilisation of sequential loading as shown in Figure 85.

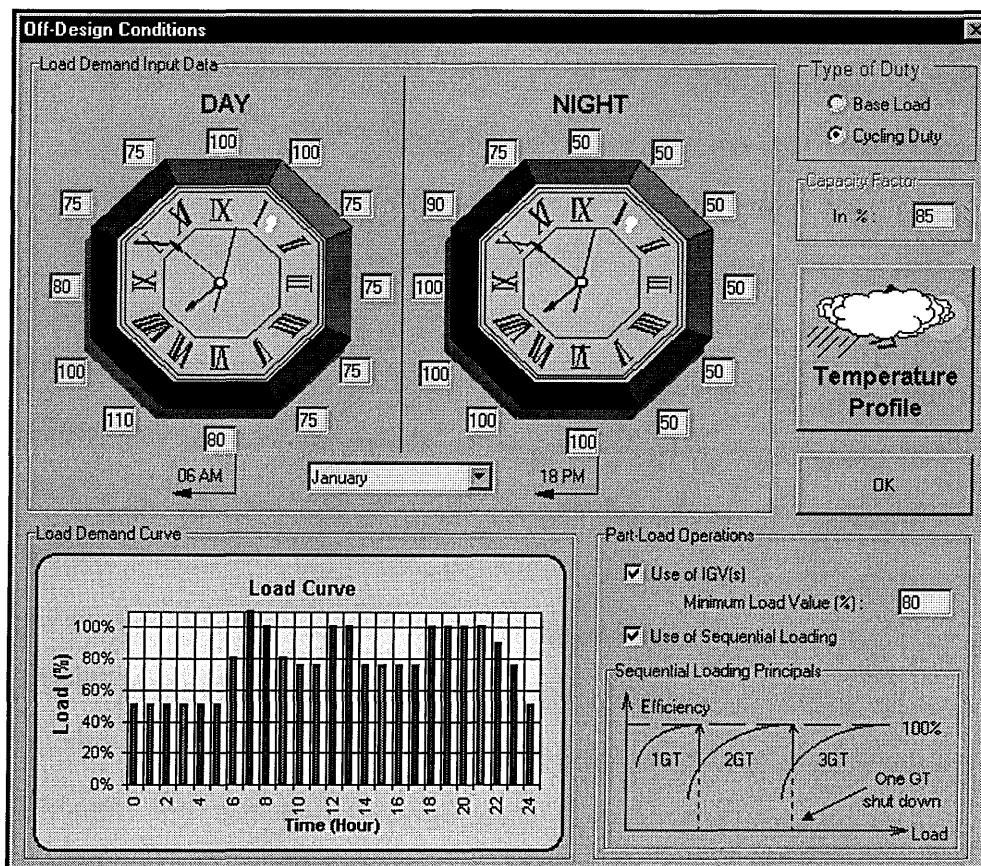


Figure 85: Off-design Data Input Window

4.5. Off-Design Steam Cycle Modelling

4.5.1. Introduction

Off-design conditions are caused by gas turbine inlet air conditions (temperature, pressure, altitude, humidity), by engine deterioration and finally by part-load control systems when applications are not base load. All these effects can be tackled by the DSS, either directly as illustrated in Figure 72 to Figure 75, Figure 80 and Figure 85, either by following proposed additional or alternative methodologies. The aim of those off-design models is to assess as accurately as possible HRSG inlet conditions namely engine exhaust temperature and mass flow.

Once these parameters are calculated, it is possible to simulate HRSG operating conditions in off-design by using numerical analysis algorithm. This should solve several heat exchange equations for each pressure level as illustrated in Equations 1, 2 and 3, and thus provide new inlet conditions for the steam turbine that would also operate in off-design conditions.

The purpose of this chapter is to present the mathematical modelling of the HRSG and steam turbine operating in off-design conditions.

4.5.2. Heat Recovery Steam Generator Off-design Model

4.5.2.1. Principles

Because the HRSG is following off-design engine behaviour by modifying steam cycle thermodynamic conditions, superheated steam temperature(s) and pressure(s), as well as its mass flow(s), are different to input data at design-point conditions. HRSG “slave” behaviour is commonly called “sliding pressure”, because drums where steam is raised, accommodate themselves to flue gases new inlet conditions, by reducing saturation temperatures and pressures.

It must be highlighted that in off-design conditions, all thermodynamic parameters of water and steam are unknown. Besides gas temperatures are also unknown, with the exception of inlet gas temperature noted T_1 in all heat transfer diagrams and charts plotted previously.

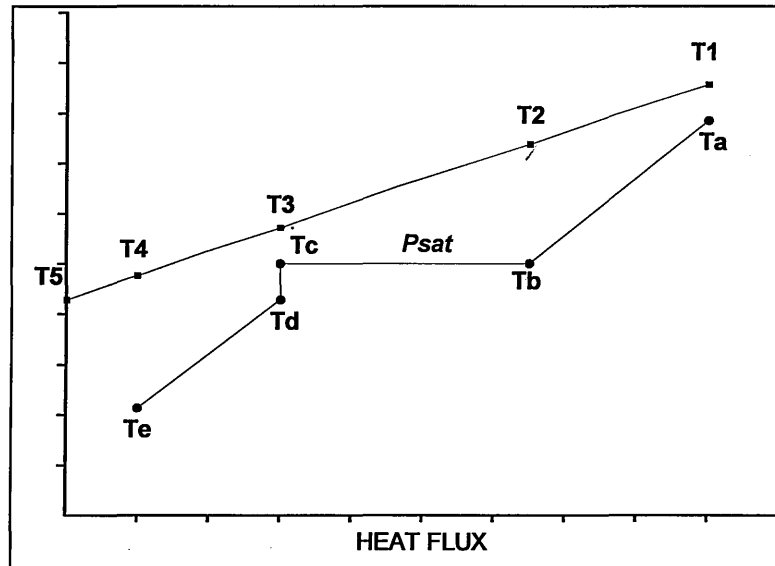


Figure 86: Single-Pressure Heat Transfer Diagram

According to Figure 86 and equations 1, 2 and 3, the unknowns for a single-pressure HRSG would be:

- Superheater: T_a and P_a
- Evaporator: $T_b = T_c$, P_{sat} and steam mass flow
- Economiser: T_d , P_d and P_e
- Gas side: T_2 , T_3 and T_4
- All heat quantities: Q_{super} , Q_{evap} and Q_{eco}
- All new heat transfer coefficients: U_{super} , U_{evap} and U_{eco} .

NB: Deaeration conditions are kept constant in off-design conditions, therefore T_e is known and T_5 can be easily calculated with T_4 .

This means that for a single-pressure HRSG, we need 17 equations in order to solve these 17 unknowns.

It can be noticed that multiple-pressure HRSGs present a highly challenging numerical analysis since the number of unknowns and thus the number of equations required to solve the system, increases by 17 every time a pressure level is added to the steam cycle.

4.5.2.2. Algorithm for single and dual-pressure HRSGs

The following algorithm is based on the ΔT_{lm} approach described in the chapter about heat exchanger sizing. It is illustrated in equation 21 where a Log-Mean Temperature difference and a shape factor are used. ΔT_{lm} and ϵ -NTU methods provide an additional equation to the system for each type of heat exchanger. Only one can be used since both approaches, if different in shape utilises exactly the same type of data, i.e. inlet and outlet temperature differences. Off-design calculations using the ΔT_{lm} concept is applied

in Dechamps (1994), whilst commercial packages such as Gate Cycle applies the ϵ -NTU method.

A choice had to be done, and ΔT_{lm} appeared to be more straightforward than using intermediary parameters like heat exchanger effectiveness. As it will be highlighted later, this choice is not the best one for solving three-pressure HRSGs or advanced HRSGs.

There are three ways of expressing heat equation for each heat exchanger, as shown in equations 1, 2 and 3.

On top of that, saturation properties provide a direct means of calculating drum temperature as a function of drum pressure. This adds one equation for every pressure level of the HRSG.

In Erbes and Gay (1987), off-design correlations for assessing heat exchanger pressure drops and overall heat transfer coefficients are proposed. They are calculated using design pressure drops, temperatures and mass flows, as follows:

$$\frac{\Delta p}{\Delta p_{des}} = \left(\frac{\dot{M}}{\dot{M}_{des}} \right)^{1.8} \left(\frac{T}{T_{des}} \right) \left(\frac{P}{P_{des}} \right)^{-1} \quad (36)$$

$$\frac{U}{U_{des}} = \left(\frac{\dot{M}}{\dot{M}_{des}} \right)^{0.805} \quad (37)$$

Each correlation provides three additional equations for every pressure level.

Finally assuming that the flow at the steam turbine inlet is choked and that the steam is a perfect gas, the Stodola relation can be utilised to complete the equations system. This relation can be written:

$$\dot{M}_{steam} \left(\frac{\sqrt{T}}{P} \right) = Const \quad (38)$$

This relation can be applied at every steam turbine inlet for multiple-pressure plants.

In summarise, for a single-pressure HRSG the following system provides enough equations to solve the off-design calculation:

1. Heat Transfer Coefficients : $U = f \left(U_{des}, \frac{\dot{M}}{\dot{M}_{des}} \right)$, 3 equations
2. Pressure Drops: $\Delta p = f \left(\Delta p_{des}, \frac{\dot{M}}{\dot{M}_{des}}, \frac{T}{T_{des}}, \frac{P}{P_{des}} \right)$, 3 equations
3. Stodola Relation: $\dot{M} \left(\frac{\sqrt{T_a}}{P_a} \right) = \dot{M}_{des} \left(\frac{\sqrt{T_{a,des}}}{P_{a,des}} \right)$, 1 equation

4. Saturation: $T_{sat} = f(P_{sat})$, 1 equation

5. Heat Equations: 9 equations

- Superheater:

$$\dot{Q}_{sup} = \dot{M}_{gas} C_p (T_1 - T_2) (1 - \alpha_{sup}) \quad (\text{Sup1})$$

$$\dot{Q}_{sup} = \dot{M} (H_a - H_b) \quad (\text{Sup2})$$

$$\dot{Q}_{sup} = U_{sup} S_{sup} \Delta T_{lm} F_{sup} \quad (\text{Sup3})$$
- Evaporator:

$$\dot{Q}_{evap} = \dot{M}_{gas} C_p (T_2 - T_3) (1 - \alpha_{evap}) \quad (\text{Evap1})$$

$$\dot{Q}_{evap} = \dot{M} (H_b - H_d) \quad (\text{Evap2})$$

$$\dot{Q}_{evap} = U_{evap} S_{evap} \Delta T_{lm} \quad (\text{Evap3})$$
- Economiser:

$$\dot{Q}_{eco} = \dot{M}_{gas} C_p (T_3 - T_4) (1 - \alpha_{eco}) \quad (\text{Eco1})$$

$$\dot{Q}_{eco} = \dot{M} (H_d - H_e) \quad (\text{Eco2})$$

$$\dot{Q}_{eco} = U_{eco} S_{eco} \Delta T_{lm} F_{eco} \quad (\text{Eco3})$$

Off-design heat transfer coefficients can be directly calculated using design-point output data stored in DSS database. However, other equations require a numerical analysis method since the unknowns are intrinsically linked.

To simplify the system, it is assumed that pressure drop ratios are not significantly different between superheater, economiser and evaporator. Therefore only one of those is calculated, for the superheater for example and is applied for the calculation of the others.

This means that an algorithm must be defined to solve the 9 heat equations, the Stodola relation and the saturation relation, giving a system with 12 equations with 12 unknowns.

One way of solving this system is the use of a matrix-based approach that would simultaneously provide all results. Another way is to solve the equations sequentially by guessing and checking unknown values.

Matrix approach requiring rigorous knowledge in numerical analysis could not be implemented in the time available for the DSS design. However that remains the best and the safest option to solve such large systems when applied for multiple-pressure plants.

Sequential approach has been chosen and was implemented with success for single and dual-pressure HRSGs. For convergence reasons, no successful algorithm for three-pressure plants could be implemented. That would require a large amount of time that the MPhil. could not provide. This means that the user must have the knowledge of at least one off-design operating condition.

4.5.2.3. Methodology for three-pressure HRSGs

In a sliding pressure mode, pressure, temperature and mass flow profiles are almost linear with engine load. Therefore with only two points, the design and one off-design

steam conditions in the HRSG, a linear coefficient can be found and applied for any other engine load.

This methodology is actually even utilised for single and dual-pressure HRSG off-design calculation to reduce computational time. It must be pointed out that solving twelve equations in a sequential approach that has been optimised, require few seconds for only one off-design point. According to the part-load model, a user can input 24 different values of load in 12 different months, leading to 8640 different off-design points to be calculated (including degradation) in a 30 years project. Computational time would become too important and totally useless since the correlated curve approach is very accurate as Figure 87 shows:

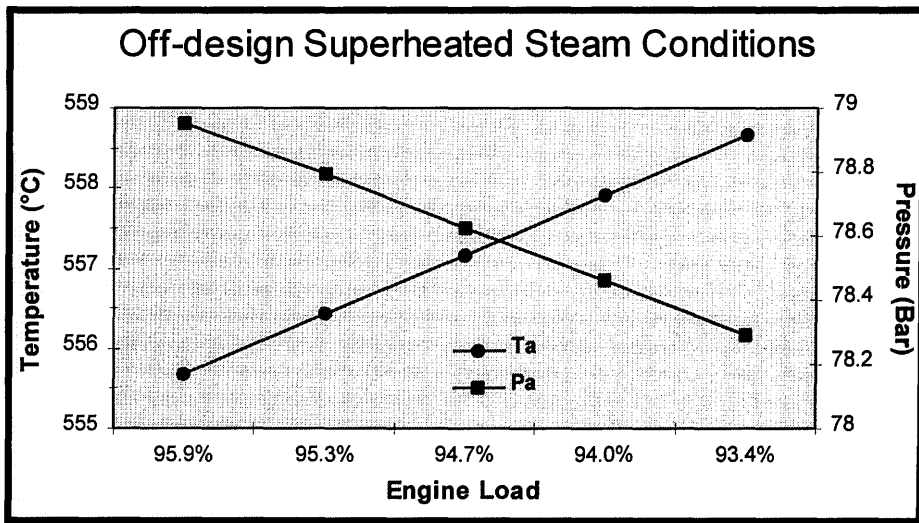


Figure 87: Example of an Off-design Calculation using a MS9001FA

Superheated mass flow follows also a linear trend with engine load.

Therefore a single HRSG off-design point is required to assess quite accurately steam conditions for off-design performance. As the actual trend is to propose almost the same equipment based on similar technologies, it is not difficult to find in technical papers or magazines, instances of three-pressure CC plants, powered with a similar type of gas turbine that could provide a rough idea of what the steam turbine inlet conditions could be with regard to a specific engine power rating.

The exact algorithms used to calculate single-pressure and dual-pressure HRSGs are detailed in Appendix G.

4.5.3. Off-design Steam Turbine Efficiency Calculation

The DSS provides two different ways of calculating the steam turbine efficiency depending on the calculation mode selected for the design-point calculation (cf. Figure 50).

If the GE Spencer, Cotton & Cannon calculation mode is selected, this model is still used for off-design steam turbine efficiency calculation. For copyright restrictions, it is not possible to explain how the calculation is carried out in details, but this empirical method basically utilises an Equivalent Throttle Ratio (TFR) (equation 39) as main input.

$$TFR = \left(\frac{\text{CurrentFlow}}{\text{ActualFlow}} \right) \sqrt{\frac{\left(\frac{P}{v} \right)_{\text{rated}}}{\left(\frac{P}{v} \right)_{\text{non-rated}}}} \quad (39)$$

A series of efficiency correction factors are then calculated using empirical based mathematical expressions, to finally provide the steam expansion line through the steam turbine. As shown in Figure 51, the user can also select a mean-of-loops correction. This additional correction is intended for use with steam turbines that have 4, 6 or 8 control valves. Normally, the GE methods predict the part-load efficiency at valve points (where the valves are fully open or fully closed). To accurately determine the efficiency at intermediate points, it would be necessary to model the performance of each valve individually. Instead, the mean-of-loops correction is used as an efficiency penalty factor, which corrects the calculated valve-point efficiency to a mean along the valve-loop efficiency curve. This corrected efficiency would be too low if the turbine were actually operating at a valve-point, and it would be too high if the turbine were operating at its worst efficiency point between two valve points, as illustrated in the following figure:

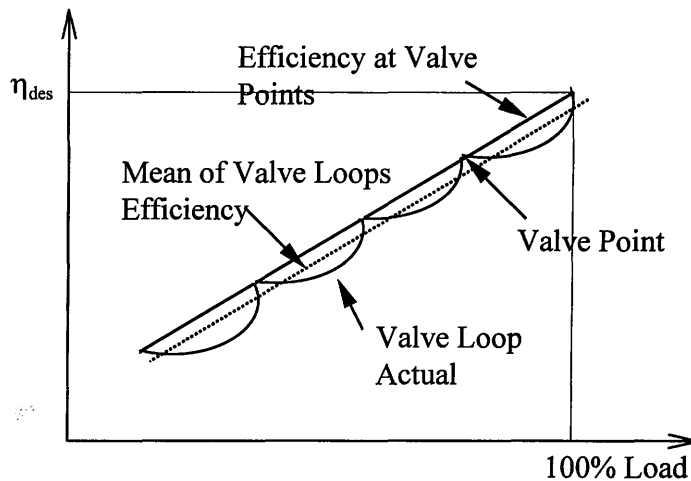


Figure 88: Mean-of-loops Concept

If another calculation were selected, a correlation extracted from Erbes and Gay (1987) would be utilised as follows:

$$\frac{\eta}{\eta_{des}} = K_1 \left(\frac{\dot{V}}{\dot{V}_{des}} \right)^x + K_2 \left(\frac{\dot{M}}{\dot{M}_{des}} \right)^y + K_3 \quad (40)$$

Where \dot{V} is the volumic flow rate.

The default values for the constants in equation 40 are the same as those used in Erbes and Gay (1987) and in Dechamps (1995) namely:

- $K_1 = -0.9$
- $K_2 = 1.9$
- $K_3 = 0$
- $x = -2$
- $y = -1$

This correlation begins to break down when the flows are very low, therefore user-input are possible to correct it, if noticed after a preliminary calculation.

4.5.4. Conclusion

Steam cycle off-design modelling requires specific models for the heat recovery steam generator and for the steam turbine.

The HRSG off-design calculation can become very complicated and thus could demand a powerful numerical analysis method. That method could not be programmed in the time available for the MPhil. but it is certainly the first task that further developers will tackle.

As far as the steam turbine is concerned, a degradation model could be developed to enhance the accuracy of off-design performance calculations. But in all cases where the turbine is clean, the GE methodology provides very good results for a large number of different steam turbine types and applications.

Chapter 5

LIFE CYCLE COSTING

5.1. Definition

The Life Cycle Cost (LCC) could be defined as the cost of plant ownership, from capital costs to spare parts costs during the lifespan of the project. This concept, developed in Spector (1989), is now becoming widely utilised in all industrial areas. The advantage of this method compared to the traditional “dollars per kilowatt” approach, is that it separates all independent types of cost, highlighting thus the most critical ones for a particular project. LCC for combined cycle schemes are greatly affected by fuel costs, whilst LCC for an integrated coal gasification combined cycle plant would show that the economics would be mainly influenced by the very high capital cost.

The DSS defined its life cycle costing as the costs of fuel, overheads, maintenance actions, financing and additional optional features such as fuel treatment cost, cost of emissions etc.

5.2. Maintenance Planning and Plant Availability

5.2.1. Model

Capital cost and cost of financing are often expressed per installed kilowatt. It only depends on the plant size, plant technologies and borrowing conditions (duration and interest rate). These costs are said fixed because of their independence from plant availability and capacity factor that are function of time.

Sometimes O&M costs are divided into fixed and variable costs, since labour and overheads can be regarded as fixed charges for the utility. The DSS does not make the difference between those types of cost, and only provides variable cost expressed thus per kilowatt-hour. However, due to its structure, the DSS enables the user to modify the production cost final expression by adding an additional operating cost into a specific spreadsheet (cf. §8.2). But in all other cases, operating parameters, plant availability and capacity factor must be accurately assessed, since they define the number of fired hours and kilowatts produced per year during the plant lifespan.

Plant annual capacity factor is normally input in the part-load user-interface. Base load or cycling load can be selected and in both cases, a fixed value or monthly load demand curves must be input. The difference that the DSS makes between the load and the plant capacity factor is due to performance losses that depend on degradation rates and number of annual and cumulated fired hours. For example if a base load application is selected with an expected 100% clean capacity factor and if at one point time the degradation level is such that plant performance loss reaches 10%, in order to provide the load demanded by the grid, the plant must operate in peaking duty at 110% load. That is why, the overall clean capacity factor for base load applications must be assessed with regard to expected performance degradation rates and levels.

Maintenance strategies are of paramount importance for achieving good plant availability and reliability. Both parameters can sometimes have antagonist effects. Reliability is greatly affected by inspection intervals and maintenance action effectiveness, therefore in order to achieve very good plant reliability by reducing forced outages, forecast down-time ratio can be sometimes high. Of course that depends on the monitoring system which normally should provide appropriate warnings in order to anticipate and thus optimise off-line maintenance actions. In Hoeft (1994), availability, CC plant availability and CC plant reliability definitions are proposed:

1. Probability of being available, independent of whether the unit is needed:

$$Availability = 1 - \frac{TotalUnavailableHours}{PeriodHours} \quad (41)$$

2. Probability of a multi-shaft combined cycle power plant not being totally forced out of service when the unit is required:

$$CCreliability = 1 - \left(\frac{(FOH)_{GT}}{(PH)_{GT}} \right) + 0.3 \times \left(\frac{(FOH)_{HRSG}}{(PH)_{HRSG}} + \frac{(FOH)_{ST}}{(PH)_{ST}} \right) \quad (42)$$

Where: • FOH = forced outage hours.
 • PH = period hours.
 • It is assumed that steam cycle MW output contribution is about 30%.

3. Probability of a multi-shaft combined cycle power plant being available for power generation (independent of whether unit is needed):

$$CCavailability = 1 - \left(\frac{(UH)_{GT}}{(PH)_{GT}} \right) + 0.3 \times \left(\frac{(UH)_{HRSG}}{(PH)_{HRSG}} + \frac{(UH)_{ST}}{(PH)_{ST}} \right) \quad (43)$$

Where: • UH = total unavailable hours.

These definitions show that to increase plant availability, scheduled maintenance hours or/and forced outage hours must be reduced. That is why, in order to achieve good availability and good reliability, maintenance action duration must be as short as possible and as effective as possible to reduce the Mean Time Between Failure (MTBF). The type of application will dictate how this trade-off between reliability and availability must be balanced. Base load duties require the highest availability possible, therefore it is not striking to see very long periods between maintenance actions and inspections. But because reliability is also very important, monitoring systems are becoming real issues for planners in order to reduce component unexpected failures. That is why it is well known that base load plants have to face up both maintainability and availability issues much more than peaking or emergency duties for instance, that typically rely on 100 % reliability.

Another issue is the type of strategy adopted in carrying out planned maintenance actions namely, in a concurrent way or in series.

This type of choice is very often based on spare parts availability. Base load CC plant powered by a single unit whatever the configuration (1-1-1, 2-2-1, etc.) would probably carry out maintenance works concurrently using three-shift labour forces. Larger plants with several units in parallel may prefer to schedule maintenance actions with regard firstly to the load demand, and secondly to the level of deterioration of each unit. It is obvious that both strategies can have totally different outputs in terms of cost and availability. Moreover these strategies depend also on the technical expertise and labour forces owned by the utility. If maintenance is subcontracted, service providers might decide what is the best for the utility with regard to its PPA contract. In other cases, past experience usually provides the answer.

Five types of maintenance action are proposed through the user interface as shown in Figure 89:

- On-line compressor wash
- Off-line compressor wash
- Combustion inspection
- Hot-gas path inspection
- Major overhaul

Operating and Maintenance Input Data

Fuel Operating Parameters

Fixed Price Contract Duration (yrs): []

Select Units System:

\$ / million Btu Price: [2.5]

\$ / GJ

Fuel Price Trend after Contract

Fuel Inflation Annual Value (%): [1.5]

Escalation Factor Value (+/- constant/year): []

Fixed Price Value (in Units System): []

Commercial Operation Start

Contract Award Date: [May] Number of Months from the Contract Award: [33]

Installed Power Output (design) [kW]: [61000]

Maintenance Planning

GE Methodology As an Input Strategy Recovery Factors

Compressor Fouling

On-Line Compressor Washing: Time Interval (fired hrs): [120]

Off-Line Compressor Washing: Time Interval (fired hrs): [504]

Off-Line Inspections

	Combustion:	Hot-Gas Path:	Major:
Time Interval (fired hrs):	[8000]	[24000]	[48000]
Duration (in 8-Hour shifts):	[8]	[42]	[168]
Extra Cost (cents/kWh):	[0.1]	[0.25]	[0.5]

Maintenance Work Strategy

Concurrently for all Units 8-Hour Shifts/day: [3]

Overheads and Maintenance Costing

As a Fixed Cost D&M Cost (cents/kWh): [2]

Electricity Price

As a Fixed Value Price (cents/kWh): [5.5]

Costs Trend Following ...:

an Inflation Rate Annual Value (%): []

an Escalation Factor Value (+/- constant/year): [0.1]

OK

Figure 89: Operating Data User-Interface

As the only degradation model available for the time being is based on an average loss during operation curve, on and off-line compressor washes are not taking into account. But input data are read and stored therefore further developers can easily utilise them for additional maintenance planning models.

The user can input the number of shifts during maintenance actions. It is important that plant operators reduce the time required for maintenance work by a factor of two or even three, by fulfilling the 24 hours available every day.

As far as action strategy is concerned, there is no choice. The DSS providing only one maintenance-planning model based on a concurrent strategy, it is always assumed that all units are maintained at one time. But once again because all input data are read and written into a specific spreadsheet, further developers should be able to utilise selections for the strategy (cf. Figure 89) and thus applied a different planning model base on a series approach for instance.

5.2.2. GE Methodology

As for the steam turbine off-design efficiency calculation, GE Power Systems published a methodology based on empirical expressions in order to assess the hot-gas path inspection interval. For copyright restrictions, these expressions can not be reproduced but they can be found in Hoeft (1994).

According to Hoeft (1994), hot-gas path inspection calculation must be based on both number of fired hours and starts. Due to different impacts on hot-gas path component, both criteria must be considered separately and the most constraining one must provide the minimum interval that operators should stick with for safety and life cycle costing optimisation reasons.

Number of fired hours is the criterion for continuous duty machines and thus is the main cause of creep, corrosion and oxidation of hot-gas path components. Peaking machines contrarily are characterised by their high number of starts that obviously affect thermal mechanical fatigue life on engine components.

As shown in Figure 90, a number of input data are requested in order to provide a fired-hours-based inspection interval, and a starts-based inspection interval. The lowest of both is then the interval advised by GE.

GE Maintenance Planning Method for Hot-Gas Path Inspection Interval Calculation

Starts-Based Criterion		Hours-Based Criterion	
Annual Number of Normal Starts (per year) :	<input type="text" value="100"/>	Annual Hours on Natural Gas (Base Load) (per year) :	<input type="text" value="3200"/>
Annual Number of Peak Load Starts (per year) :	<input type="text" value="0"/>	Annual Hours on Light Distillate (per year) :	<input type="text" value="350"/>
Annual Number of Part Load Starts (per year) :	<input type="text" value="40"/>	Annual Hours on Heavy Fuel (per year) :	<input type="text" value="0"/>
Annual Number of Emergency Starts (per year) :	<input type="text" value="2"/>	Annual Hours on Peak Load (per year) :	<input type="text" value="120"/>
Annual Number of Fast Load Starts (per year) :	<input type="text" value="5"/>		
Trips		Steam Injection Rate (%) :	<input type="text" value="2.4"/>
Annual Number of Trips from Load (per year) :	<input type="text" value="20"/>	Gas Turbine Type :	<input type="text" value="MS7EA"/>
Average Load (%) :	<input type="text" value="80"/>		
Material of 2nd & 3rd Nozzels	Unit Operation Control	RESULTS	
<input type="radio"/> GTD-222	<input checked="" type="radio"/> Dry Control Curve	The Hours-Based adjusted Inspection Interval is :	<input type="text" value="5.231E"/> yr
<input checked="" type="radio"/> FSX-414	<input type="radio"/> Wet Control Curve	The Starts-Based adjusted Inspection Interval is :	<input type="text" value="3.597E"/> yr
		The Hot-Gas Path Advised Inspection Interval is :	<input type="text" value="3.597E"/> yr
<p>! The Major Inspection Interval commonly is twice as much as the Hot-Gas Path Inspection Interval.</p> <input type="button" value="OK"/>			

Figure 90: GE Inspection Calculation Methodology Interface

This methodology very often applied in industry provides a very good idea about the hot-gas path inspection with regard to the type of machines and application. It is usual to assess major overhaul interval as twice as much as hot-gas path interval and combustion interval as third times less. If the user uses these common applied rules, the DSS provides a very accurate maintenance planning and then a very accurate plant availability. In Appendix H are summed up indicative maintenance requirements for gas turbines operating on natural gas (Griffith, 1995).

Figure 91 shows an instance of maintenance planning calculation and the resulted plant availability (Figure 92).

The plotted plant availability is on an annual scale but the calculation is made and stored in a monthly basis. The user therefore can have access and modify those figures before any performance calculation if necessary.

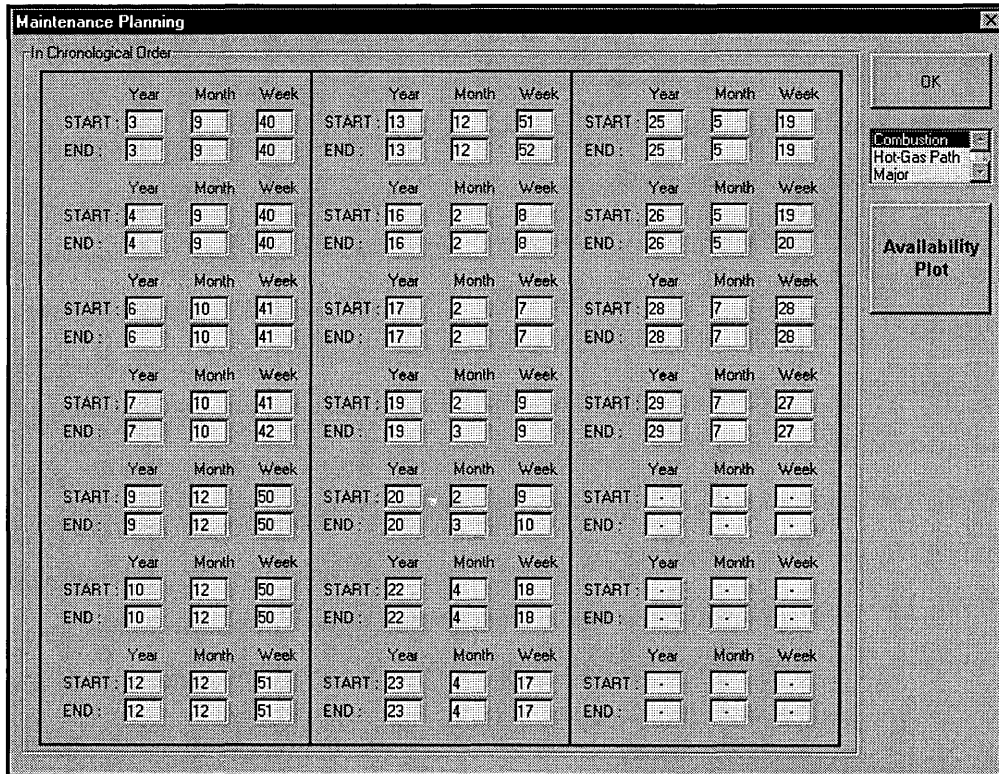


Figure 91: Instance of Maintenance Planning

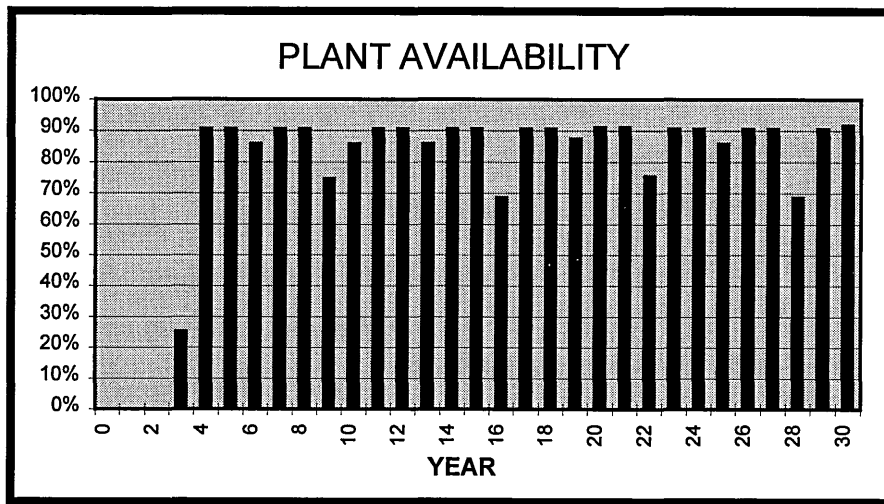


Figure 92: DSS Plant Availability Plot

5.3. Operating Cost

5.3.1. Overheads and Maintenance Cost

These cost are input in the same frame as maintenance planning as illustrated in Figure 91.

There are two types of O&M costs: those related to annual expected overheads and maintenance and those that are considered as “extra” in periods when one of the three major maintenance actions is undertaken. All these parameters are expressed in cents per kilowatt-hour.

Because O&M costs can vary during the plant lifespan, two different costing trends are proposed:

- following an inflation factor,
- following an escalation factor.

Figure 93 shows the possible O&M cost curves over the plant life.

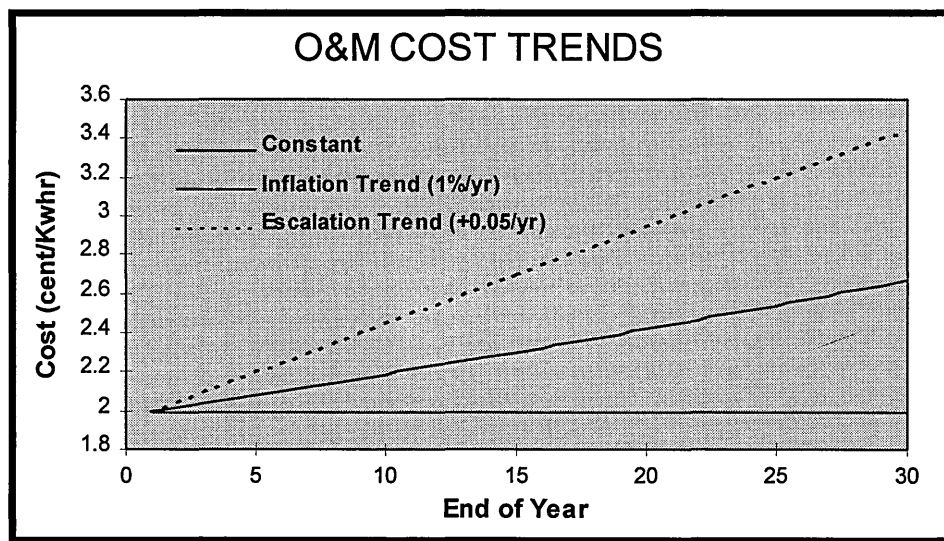


Figure 93: O&M Costing Proposed Trends

5.3.2. Fuel Cost

Accurate fuel costing is of paramount importance for CC schemes because they can be accounted for more than 60% of all operating and maintenance costs. Very often, fuel costing can be accurately assessed in preliminary economic appraisals based on GSA contract. Fixed fuel price contract is nowadays the most critical contract for such large capital cost investments. They provide to the planners a very accurate evaluation of fuel costs, providing thus a good level of confidence as far as economic appraisals are concerned. A usual fixed price contract sets up an annual fuel inflation rate indexed to a safe and non-risky economic criterion. For instance a typical GSA for an IPP in South East Asia where currencies and economy are fluctuating very much, will index the fuel inflation rate to the annual price inflation rate of United States, and great proportions of payments will be made in American dollars.

Similarly to O&M costs, the DSS proposed costing trends. Moreover it is possible to select a fixed fuel price contract for a given duration less than the life of the project and select then a following trend for the after-contract time.

The proposed trends are as follows:

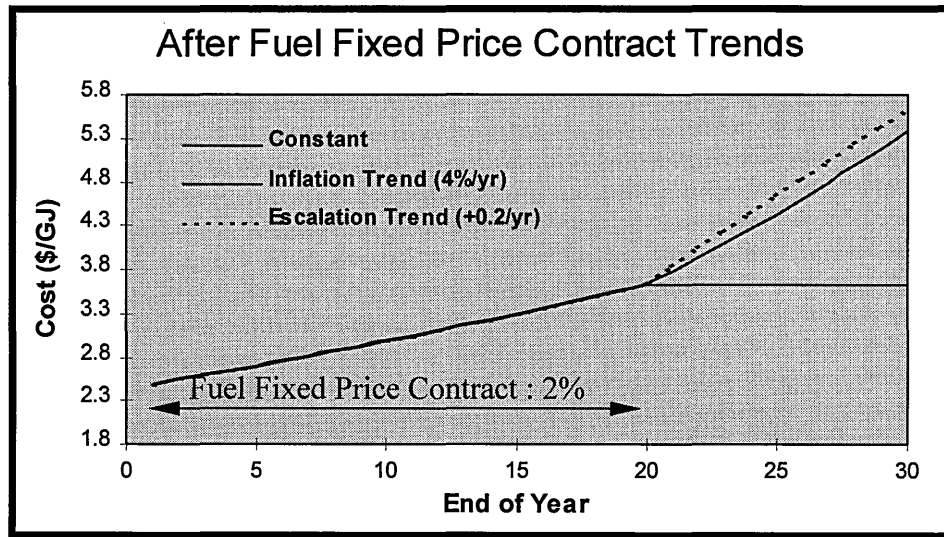


Figure 94: Proposed Trends for After Fuel Fixed Price Contract

The user can input the fuel price either in SI (\$/GJ) units or in Imperial units (\$/mBtu). The escalation factor remains in the selected units system.

5.3.3. Additional Production Costs : Environmental Costs

Considering efforts of engine manufacturers in research and development to design cleaner engines, environmental regulations are becoming one of the major issue for future power generation projects.

It is clear that utility owners must now endure that power is generated within the framework of current and future legislation. According to Holmes and Wittington (1994), there are two proposed methods in order to reduce the emission levels: trading permits and European Carbon Tax.

The first method is a flexible approach based on government emission limits setting. The second one is to be a combination of a tax on the carbon content of fossil fuels, and a tax on all renewable forms of energy. But in both cases, this operating "extra" cost must be considered in any economic appraisal, especially in countries where emission regulations are very demanding like in Scandinavia.

Emission costs could therefore become an economic issue for life cycle costing assessment. According to Bernow et al. (1991), it is time for utility owners to incorporate "environmental externalities" in power generation operation. Power plant scheme planners should cost all types of emissions that could be internalised by means of new environmental laws.

In Table 2 are summarised emission cost values for major air pollutant that the Massachusetts Department of Public Utilities adopted in 1990 (Bernow et al., 1991). These figures give a good idea about the penalty in electricity pricing and then the importance of those environmental considerations for the feasibility of CC schemes. In ASME B133.9 (1994), a method for measuring exhaust emissions from stationary gas turbine engines is proposed.

Table 2: Costs of Emissions (Massachusetts DPU, 1990)

Pollutant Type	Cent/kg of pollutant
NO _x	393.7
CO ₂	1.4
SO ₂	92.6
CH ₄	13.3
CO	52.7
Particulates	243.2

5.4. Electricity Retail Pricing

The electricity retail price is not a life cycle cost parameter but it is assessed using the overall cost of production per kilowatt-hour, that is why it is located in the operating input data interface as illustrated in Figure 89.

The cost of electricity is a very important economic parameter because it provides the break-even baseline for electricity retail pricing negotiations with the grid for GSA contract. This economic parameter will be detailed in the following chapter.

The DSS provides the same types of future trends as for O&M cost, i.e. a constant price, a price following an inflation factor or an escalation factor.

Chapter 6

ECONOMIC MODEL

6.1. Introduction

Economic appraisal of CC schemes greatly relies on the quality of performance simulations. The DSS provides all the tools necessary to carry out consistent and reliable plant simulations in any set conditions: deteriorated, off-design, part-load, load following, etc. But decision making is usually supported by economic performance. The closest links between economics and technical performance can be found during PPA and GSA contracts negotiations, since both expertise are required to make these contracts a success.

There are two issues in economic modelling. The first one is the financial model. This type of model is greatly dictated by national fiscal systems that are responsible for issuing accounting procedures and book-keeping rules. Therefore the quality of such a model can be only judged on its capability of embedding different fiscal systems owning specific accounting features. To provide this versatility, the DSS accepts British and American fiscal systems.

The second issue concerns economic techniques. There are several different economic techniques based on different economic appreciation and they are all utilised by planners according to their specific internal decision making strategy. However very few of them are always considered, since they provide a very good flavour of schemes feasibility and profitability. The DSS therefore provides about ten different economic appraisal techniques but only keeps four of them for the risk analysis.

6.2. Economic Evaluation

The aim of this chapter is to recall basic expressions utilised for assessing cash available at every end of year for the lifespan of the project. All these expressions are based on Stoll (1989) and on Marsh (1980).

6.2.1. Project financing

The way of financing any project is multiple. Every company generally with regard to its assets and its short or long term financing policy utilises different composite financing types. Most of utilities find their funds in three major types of financing : long-term bonds, preferred stocks and common stocks.

6.2.1.1. Long-term Liabilities

According to Whitehead (1992), the commonest types of long-term liability are:

- bank loans
- private loans
- finance house loans

- mortgages

In our particular case, long-term bonds represent the major source of funds. Banks are the major financing partners. Finance houses generally are second. Borrowing life is commonly the same as GSA or PPA contract life for economic safety reasons.

6.2.1.2.Common Stock

Common stock is part of common equity of the utility. The utilities issue stock on the stock exchanges, which permits public ownership of electric utility. The proceeds from issuing common stock to the public are recorded under the equity section entitled common stock and paid in capital. Under “common stock” is listed the par value times the number of shares. Under “paid in capital” is the excess between market value and par value. Par value is a nominal value per share, usually equal to the original stock price at first issue, while the market value is the price on the stock exchange. The utility issues common stock at the market value from the sale of all the stock over the entire history of the corporation is recorded under these categories.

6.2.1.3.Preferred Stock

The difference between common stock and preferred stock is that the second has a fixed yield associated with each offering and usually on vote in company business. To highlight the contrast, just consider that common stock has no fixed yield, and dividends and profits on common stock are entirely dependant on the financial success of the utility. In the case of preferred stock, a fixed yield is guaranteed. If the utility does very well in a particular year, the preferred stockholder still receives only the stipulated dividend.

6.2.1.4.Composite Cost of Capital

Once the financing of the project is defined, the utility can evaluate the composite cost of capital, which is commonly used to evaluate a discount rate and represents the mix of financing a project.

The composite cost of capital is then defined as follows:

$$\text{Composite cost of capital} = bB + pP + cC \quad (44)$$

Where :

Table 3: Types of project financing

Financing type	% of each type	Interest rate
Long-term Bonds	b	B
Preferred Stock	p	P
Common Stock	c	C

6.2.2. Depreciation

Depreciation is one key consideration in electric utility accounting. As deducted from the taxable profit, it can substantially reduce income taxes.

Book depreciation is assumed to be linear as follows:

$$DB_i = DB = \frac{\text{PlantCapitalCost}}{\text{BookLife}} \quad (45)$$

It must be noticed that this depreciation method, called “straight-line depreciation” is always used, by accounting convention, to calculate book depreciation expense in the income statement.

But this method is not the commonest utilised for tax depreciation calculations. There other depreciation techniques, and these proposed by the DSS are the following ones:

- Sinking-Funds (SF) method,
- Sum-of-Years Digits (SYD) method,
- Double-Declining-Balance (DDB) method.

6.2.2.1. Sinking-Fund Method

This method is based on the sinking-fund factor. It is used without regard to whether the corporation has or has not bonded indebtedness requiring real sinking funds. It is merely a systematic, rational method of determining the annual charge for depreciation. It is based on a fictitious sinking fund, presumed to be set up to recover the initial investment at interest rate, r over the life of the asset, n .

The depreciation charge each year is the level sinking-fund annuity plus the interest on the accumulated depreciation. The level sinking-fund annuity is the initial investment times the Sinking-Fund Factor (SFF), which may be calculated as follows:

$$SFF = \frac{r}{(1+r)^n - 1} \quad (46)$$

The depreciation factor D and the accumulated depreciation rate A are written below:

$$D(i) = SFF(1+r)^{i-1} \quad (47)$$

$$A(i) = SFF \frac{(1+r)^i - 1}{i} \quad (48)$$

6.2.2.2. Sum-of-Years Digits Method

Sinking-fund depreciation is "slow" relative to straight-line; i.e. the depreciation accumulates less rapidly in the early years of life. In contrast are two methods of "fast" depreciation. The first is called sum-of-years digits (SYD). In this method, the annual charge is a fraction of initial investment whose numerator is the remaining life and whose denominator is the Sum-of-Years Digits:

$$D(i) = \frac{n-i+1}{(1+2+3+\dots+n)} = \frac{n-i+1}{\frac{n(n+1)}{2}} \quad (49)$$

$$A(i) = 1 - \frac{(n-i)(n-i+1)}{n(n+1)} \quad (50)$$

6.2.2.3. Declining-Balance Method

The second fast depreciation method is Declining-Balance depreciation (DB). Here, a fixed rate is applied not to the initial investment but to the balance of the investment after the depreciation charges of prior years have been subtracted. If the annual rate is f , depreciation factors are calculated as follows:

$$D(i) = f(1-f)^{i-1} \quad (51)$$

$$A(i) = 1 - (1-f)^i \quad (52)$$

The usual practice is to switch to straight-line depreciation in the year in which the straight-line depreciation on the remaining balance is just equal to the declining-balance depreciation. Equation 53 gives the expression to assess this specific year:

$$I_0 = n - \frac{1-f}{f} \quad (53)$$

Income tax regulations permit a maximum rate equal to twice the straight-line rate; hence it is there commonly called Double-Declining-Balance (DDB) and this is the method always applied, therefore:

$$f = 2 \times \frac{1}{n} \quad (54)$$

All these depreciation methods are graphically compared in Figure 95.

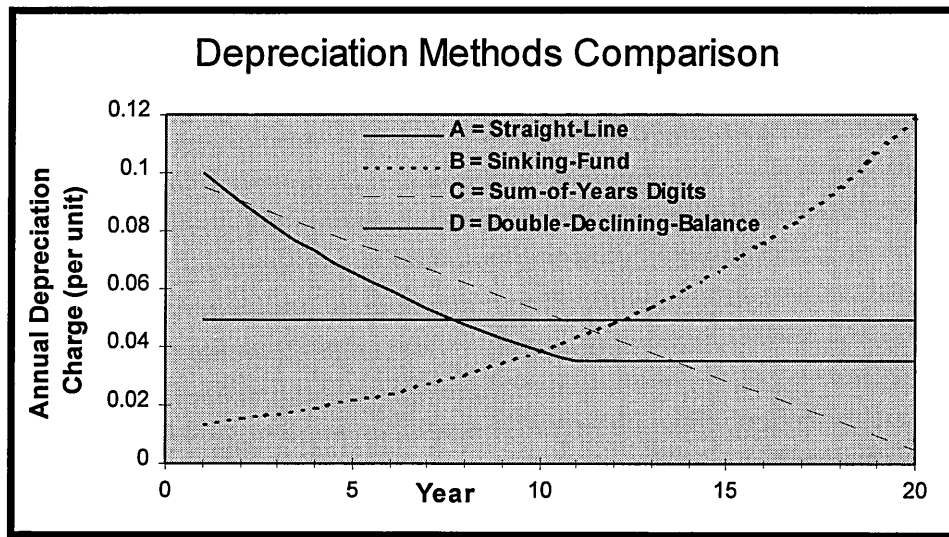


Figure 95: Comparison between Tax Depreciation Methods

6.2.3. Tax Accounting Procedure

6.2.3.1. Normalised Accounting

The choice of tax depreciation methods can affect the magnitude of current versus future earnings levels by influencing current versus future income tax expenses. Here the concern is that fast depreciation understates current taxes, hence overstates current earnings, thus permitting lower current revenues at the expense of users who come later. This refreshing advocacy of the welfare of future generations finds expression in a “have-your-cake-and-eat-it” accounting procedure known as normalising.

In this procedure, fast depreciation is used in calculating income taxes actually to be paid. But the income statement for shareholders is prepared as though the straight-line had been used for calculating taxes. Thus in early years, the tax expense shown on the income statement is larger than the tax actually paid to the taxing authority. In later years, the reverse is true, and the desired avoidance of swings in earnings and electricity rates is achieved, while also obtaining the cash-flow benefit of fast tax depreciation.

Statements for early years in the life of the plant, when tax depreciation is greater than book depreciation (based on straight-line method), the deferred tax is positive. Deferred tax is the difference between the tax that would have been calculated, if the book depreciation had been used in the income tax statement, and the actual tax.

In later years, the deferred tax becomes negative because tax depreciation is less than book depreciation.

6.2.3.2. Flow-Through Accounting

The normalisation procedure is not always used with fast tax depreciation. The alternative is simply to show on the income statement the taxes as actually incurred and paid. This non-normalised treatment is referred to as “flow-through”.

6.2.3.3. Flow-Through vs Normalised Tax Accounting

This question has been extensively studied in Brigham and Nantell (1974) and this study showed that normalisation is a sounder policy than flow-through.

Both methods enable a company to take advantage of accelerated depreciation for tax purposes, and both pass according to Brigham and Nantell's model, the benefits of accelerated depreciation on to the utility's customers. However, flow-through gives current customers benefits, which are really, too large, and future customers are forced to make up for these excessively large current rate reductions.

To sum up, the rate payer would prefer the flow-through accelerated accounting method because it leads to the lowest revenue requirements, or lower electric rates, consistent with providing the utility a fair return on its investment, or fair net income. On the other hand, the utility investor would prefer the normalised treatment because it provides a fair rate of return on net income, as well as a larger cash flow from operations. Larger operations cash flow means that the utility will need less external financing to complete its construction projects and expansion plan. Less external financing benefits current investors.

6.2.4. Investment Tax Credit

Part of the philosophy of fast tax depreciation is the idea that, by increasing cash flow, it encourages investment in new facilities, a frequent objective of national tax policy. Even more encouraging is another device, the investment tax credit, which allows an actual, direct reduction in income taxes when there is investment in new facilities. The procedure is to calculate income taxes in the usual way, then subtract the investment tax credit, which is some percentage of the investment made that year.

The whole investment tax credit is deferred in the year in which it occurs; then it is amortised, or taken as a credit, gradually, over a future period of years which is generally equal to the estimated life of the asset in which the investment is made.

For the time being, investment tax credit are not allowed therefore even so the economic model includes it, in the user interface, input boxes for that particular feature have been

deactivated. But it is up to the user to modify that, if state authorities again allow this tax credit.

6.2.5. Income Statement

Error! Reference source not found. shows the income statement:

Table 4: Income Statement

Income Statement (year i)	
Revenue	REV _i
Less Expenses	
Production costs	PC _i
Book depreciation	DB _i
Income taxes paid	IT _i - ITC _i
Deferred investment tax credit	ITC _i • NORMITC
Amortisation of investment tax credit	AITC _i • NORMITC
Deferred income taxes	DFIT _i
Ad valorem taxes and insurance	AV _i
= Operating Income	OPIN _i
Less interest expense	INT _i
= Net Income	NI _i

Therefore the Net Income (equity return) statement is :

$$NI_i = REV_i - PC_i - DB_i - (IT_i - ITC_i) - DFIT_i - AV_i - ITC_i \cdot NORMITC + AITC_i \cdot NORMITC + INT_i \quad (45)$$

Where:

- NORMITC = 1 if the investment tax credit is normalised
 , 0 if the investment tax credit is flow-through
- DFIT_i = t • (DT_i - DBT_i) • NORMDEPR
- NORMDEPR = 1 if tax depreciation savings are normalised
 , 0 if tax depreciation savings are flow-through
- AITC_i = (ITC/YRITC) if i ≤ YRITC = amortisation of investment tax credit
 = 0 if i > YRITC
- YRITC = years to amortise the investment tax credit (YRITC =1 if flow-through)

Net income contains the dividends of the preferred stockholders and the earnings available to common stockholders.

6.2.6. Income Tax Statement

Income tax statement is necessary to calculate taxable income and furthermore income tax.

Table 5: Income Tax Statement

Income Tax Statement (year i)	
Revenue	REV _i
Less deductible expenses	
Production costs	PC _i
Tax Depreciation	
Ad valorem taxes & insurance	AV _i
Interest expenses	INT _i
= Taxable income	TI _i

This implies :

$$TI_i = REV_i - PC_i - DT_i - AV_i - INT_i \quad (46)$$

Income Tax is (tax rate = t) :

$$IT_i = t \cdot TI_i \quad (47)$$

Less investment tax credit: ITC_i

Income Tax paid: FIT_i = IT_i - ITC_i (48)

The income tax statement presented in Table 5 is based on the formula of revenue less deductible expenses equals taxable income. Income tax is computed as the tax rate times the taxable income. Normally the taxes paid are reduced by any applicable investment tax credit (normalised accounting procedure is thus compulsory), but in our case it is equals to zero, unless new tax reforms do allow it again in order to stimulate new plant investments by industry, including electric utilities.

In the economic model user interface, it is asked if the project is part of an existing business, owning other different projects. It is clear that if the project, which is part of an existing business, generates negative taxable incomes, usually in the early years of the scheme, income taxes will be negative, as if money was earned. This accounting procedure is consistent with the reality of book keeping since all running projects constitute a whole business, generating positive and negative taxable incomes that all together provide the company taxable income. Therefore negative income taxes are fictitious at the scale of the project, but they must be accounted like that for the calculation of overall income taxes paid by utilities.

If the project is a single business, negative taxable incomes will therefore provide income taxes equal to zero.

6.2.7. Cash Report

The cash report (Table 6) also called a “source and application of funds” statement, shows how the transfer of funds impact on cash.

Table 6: Cash Report

Cash report (year I+1)	
Cash (start of year)	CASH _I
Cash from operations	
Net income	NI _I
Depreciation (book)	DB _I
Deferred income taxes	DFIT _I
Deferred investment tax credit	ITC _I • NORMITC
Less amortisation of investment tax credit	AITC _I • NORMITC
Cash from external sources	
New financing	FINANCE _I
Less application of funds	
Capital expenditures	CAEXP _I
Refunding of financing	REFUND _I
Dividends	DIVIDEND _I
Decrease in working capital	ΔWKCAP _I
Cash at year - end	CASH _{I+1}

In order to simplify the model, some assumptions have to be made :

- FINANCE_I = 0
- CAEXP_I = 0
- ΔWKCAP_I = 0

It has been assumed that the utility will not need to change its working capital as a result of its new \$ 1 addition. There are no new capital expenditures or new financing necessary after the plant is placed into service.

It is assumed that each year, the business will refund part of the plant financing.

In those conditions, the net cash is :

$$CASH_{I+1} - CASH_I = NI_I - DIVIDEND_I - REFUND_I + DB_I + DFIT_I + ITC_I \cdot NORMITC - AITC_I \cdot NORMITC \quad (48)$$

This represents the “net profits” generated by the project that can be distributed as wished by utility owners to common stockholders with regard to their re-investment policy.

It is assumed that dividends are paid to preferred stockholders whatever the annual net income, according to preferred stock fixed yield.

Figure 96 shows the user interface for the economic model.

Economic and Finance Input Data

Financing Data

	% of Investment	Yield (%)
Preferred Stock :	20	15
Common Stock :	30	12
Long-term Bonds :	50	10

Capital Cost (\$/kW) : 300
 Tax Rate (%) : 40
 Inflation Rate (%) : 2.5
 Ad Valorem (%) : 5

Other Accounting Considerations

Investment Tax Credit
 Tax Credit Rate (%) : .
 Amortization Time (yrs) : .

Tax Depreciation
 Depreciation Life (years) : 20

Tax Depreciation Method

Sinking Fund Depreciation
 Straight Line
 Sum-of-Years Digits
 Double Declining Balance

Is this Project part of an existing Utility running other Businesses ?
 YES NO

Book Life (years) : 30
 Loans Life (years) : 20

Tax Accounting Method

Flow-Through Accounting
 Normalized Accounting

Discount Rate Options

Based on the CCC (after Tax)
 As an Input Data
 Including the Inflation Rate

Composite Cost of Capital or CCC (%) : 11.6
 Discount Rate (%) : 12.1
 OK

Figure 96: Economic Model User Interface

6.3. Economic Appraisal Methods

6.3.1. Investment appraisal

Investment appraisal identifies both the resources needed and the expected benefits, and makes this comparison. Economic evaluation is that vital part of investment appraisal that is concerned with factors, which can be quantified, measured and compared in money terms.

The purpose of investment appraisal is to provide information for making good investment decisions. The objectives of appraisal are first to ensure that a project is “worthwhile”, that is, the expected future profits justify the prior expenditure of resources; second, that it is the “best” project for the purpose among alternative ways of achieving the desired future benefits.

6.3.2. Introduction to Project Cash Flow Techniques

6.3.2.1. Definition of cash flow

Allen (1991) defines cash flow as a phrase meaning the actual movement of cash in and out of an enterprise. Cash flow in (or positive cash flow) is cash received, and cash out (negative cash flow) is cash paid out. The difference between these two flows is termed the Net Cash Flow. In our concern, cash flow includes accounting parameters such as depreciation. But the principle remains absolutely the same.

6.3.2.2. Concept of Cumulative Cash Flow

Cumulative Cash is the addition of the Net Cash, or “profits” of every year of the project lifetime. A simplified cumulative cash flow diagram for a major project is shown in Figure 97.

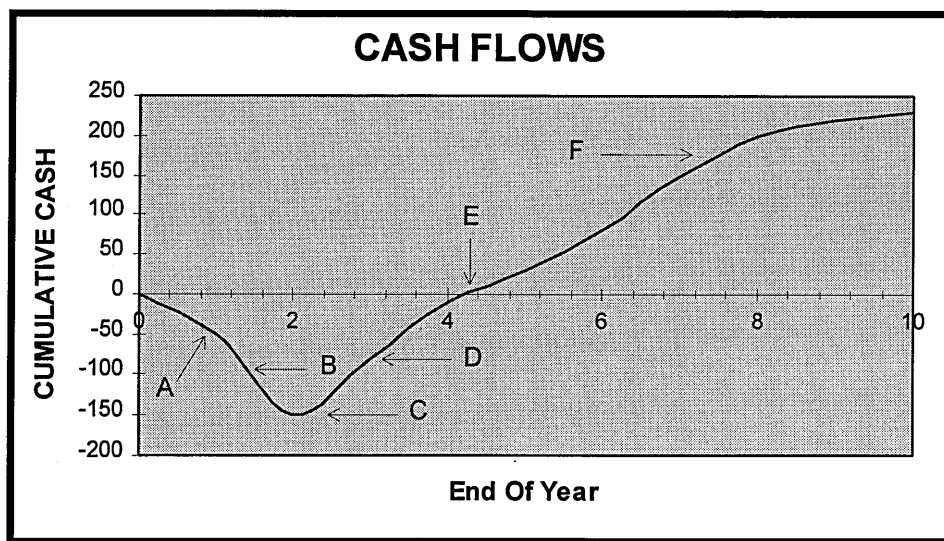


Figure 97: Cumulative Cash Flow Diagram

The early stages of this project consist of development, design and other preliminary activities which all involve the expenditure of money, and so the cumulative cash flow curve is negative and dips to point A. This is followed by the main phase of capital investment in buildings, plant and equipment. The rate of expenditure here is greater and the curve drops more steeply to point B. The lowest point C on the curve represents the maximum cumulative net expenditure level or the maximum debt incurred by the project. Beyond this point in time, income resulting from sales of resulting product exceeds the combined production costs and other operating costs. The net cash flow curve turns upwards. Beyond point D the curve rises with a constant gradient, indicating that the net rate of cash flow (income minus costs) is positive and constant. At point E the cumulative cash flow changes from being negative to positive. This is the breakeven point or payback time for the project, at which time, the total income it has so far generated, just balances all the previous project expenditure. Beyond this point in time, there is an increasingly positive net cash position as the project continues to accumulate net income. Eventually towards the end of the project's useful life, at point F, the net rate of cash inflow may decrease due, for example to increasing maintenance costs as the plant begins to wear out.

The complete cumulative cash flow diagram of Figure 97 could be obtained accurately from the records of the project at the end of the project's life. However such an historical evaluation would be useless for assisting project decision-making as all these decisions would be in the past. Economic evaluation to assist investment decision-making is concerned with analysing the economic consequences of investment decisions before they are made, that is with future cash flows and not with past ones. For those reasons, the dimension of uncertainty has to be introduced in the economic model, and economic evaluation techniques have to be adapted to take into account of them.

6.3.3. Long-term Decision Making: Discounting Cash Flows

For large long-term projects, the timing of cash receipts and payments often becomes critical and this chapter deals with a technique that specifically allows for this factor in decision-making - the Discounted Cash Flow (DCF) technique.

6.3.3.1. Time value of money

The basic concept underlying discounted cash flow analyses is that £1 received in the future date is not as valuable as £1 received now. Everyone has one's own personal time value of money. But in all cases, for large long-term projects or capital projects, it is a business concept (Allen, 1991). If a company invests its own funds in a project there will be no external interest charges to pay, but the company nevertheless loses the opportunity to earn a return on these funds through making an alternative investment, for example lending them to a bank and receiving interest, or indeed investing them in another profitable project. This loss of alternative earning power for funds is an opportunity cost, which like paying interest is also a legitimate charge on a project.

6.3.3.2. Discount Rate

In utility terminology, the time value of money is also referred to as the “discount or present worth rate”. Very often, time value of money is related with the cost of investment or the composite cost of capital defined previously. The commonest way of calculating the discount rate is to include tax rate in it. The after-tax rate accounts for the fact that bank loans and bonds (debt) interests are tax deductible items. Therefore, $(1 - t)$ is the net cost of debt to the utility.

The after-tax discount rate can be defined as follows:

$$\text{Discount Rate} = (1 - t) \cdot bB + pP + cC \quad (49)$$

It is important to underline that the time value of money has nothing to do with inflation (= decrease of purchasing power), which has to be taken into account as separate factor. Inflation can be added to the discount rate providing the Nominal Discount Rate.

6.3.3.3. Net Present Value Method

Before defining the NPV, it is necessary to clearly define what a Present Value is.

If £1 in the future has not the same value as £1 now, there can be no valid direct comparison between future amounts and present amounts. In order to make such a comparison it is necessary to revalue all future amounts in terms of the present £ (£1 now). Such revalued amounts are termed Present Value (PV). A first simple relation can be defined :

$$PV = \frac{FV}{(1+r)^t} \quad (50)$$

Where :

- FV = future value in t years' time
- PV = present value now
- r = the annual discount rate expressed as a decimal fraction
- $\frac{1}{(1+r)^t}$ = the discount factor

If applied to the cash flow, FV becomes C_t , cash at time t .

The Net Present Value of a complete project is the sum of the present values of the project's individual annual cash flows. Therefore:

$$NPV = \sum PV = \sum_{t=0}^{t=n} \frac{C_t}{(1+r)^t} \quad (51)$$

Where n = the complete project life in years.

6.3.3.4. Discounted Cash Flow Rate of Return Method

DCFR is also known by other names, such as “interest rate of return” and “internal rate of return” (IRR).

DCFR is defined from the basic relationship for the discounting of project cash flows (Equation 51). DCFR is the per cent value of r , the yearly discount rate, which equates the project NPV to zero.

$$NPV = \sum_{t=0}^{t=n} \frac{C_t}{(1+R)^t} = 0 \quad (52)$$

Where R is the project DCFR.

The DCFR for a project can be compared with its cost of capital. If the DCFR is greater than the cost of capital, this means that the project is expected to generate more in economic returns than it will cost to service the investment, and it is thus expected to be economically viable. It is of prime importance to highlight that the DCFR is a measure of rate of return and not of overall profit, i.e. it is relative and not absolute. This relative nature of the DCFR is also evident if the initial outlays differ. It is clearly better to have an 80 per cent return on £1,000 than 100 per cent on £1. If the DCFR merely measures the relative rate, the NPV measures the absolute profit.

6.3.3.5. NPV or DCFR Method ?

One argument in favour of the DCFR method is that the ordinary businessman is more familiar with a rate of return than he is with a present value. Nevertheless, as it has been seen, the interpretation of the DCFR is so hedged about with qualifications that there is probably a greater danger that a given DCFR figure will be misinterpreted than an NPV figure will be misunderstood (Harper, 1989). Indeed, some authorities assert that the DCFR method is based on totally invalid theory. Consequently the DSS will propose a long-term decision-making based on both methodologies leaving the final choice of whether or not the IRR is relevant to support the decision making to the user.

6.3.4. Payback Time

It is now generally accepted that a DCF technique is needed to enable correct capital project decisions to be made. However, an other technique is also sometimes used: the payback time.

In this technique, the time that must elapse before the net cash flows from a project result in the entire initial outlay being repaid is found. This time figure is called the payback time or breakeven, and planners look for short payback periods when making long-term project decisions.

The main objections to this technique are as follows:

- It takes no consideration of cash flows after the end of the pay-back time.
- It ignores the timing of receipts, thus ignores importance of time value of money.

The main advantage claimed however for this technique, is that if there is considerable uncertainty about the future, then an early replacement of the initial outlay by the end of a period in which events are relatively predictable means that the chance of at least regaining one's money is reasonably high.

In actual fact one should not really compare the payback and the DCF technique since they measure different things. DCF is a profitability measure while payback is a liquidity measure. The intelligent manager, then, will normally assess a project in terms of both techniques. This will be then the case in the DSS design since both discounted cash flow (NPV + IRR) and payback time techniques are highlighted as shown in Figure 98.

Economic Parameters	
From the Cash Flows	
Net Present Value (NPV) in Present \$:	\$72,200,312.00
With a Discount Rate of (%) :	12.1
Internal Rate of Return (IRR) in % :	18.63
Pay-Back Time (PBT) in Years :	9.91
Net Present Value Index in (-) :	0.3945
Accounting Rate of Return in % :	41.95
Return On Investment (ROI) incl. Deprec. in % :	20.97
Revenue Requirement Method	
Levelized Fixed Charge Rate (FCR) in % :	17.2
Present-Worth Revenue Requirement in Present \$:	\$1,341,317,727
Levelized Annual Cost Method in Present \$:	\$167,750,693.0
Equivalent Capitalized Cost Method in Present \$:	\$975,444,922.6
Electricity Pricing	
Levelized Cost of Electricity (COE) in Present \$ /MWh :	4.3
<input type="button" value="FCR Chart"/> <input type="button" value="COE Chart"/> <input type="button" value="OK"/>	

Figure 98: Economic Results For Decision-Making

6.3.5. Fixed-Charge Rate and Levelised Fixed-Charge Rate

The fixed-charge rate (FCR) is an economic parameter which depends only on capital investment financing, accounting procedure and some other economic features, but it does not take into account operating and production parameters. It can be regarded as a pure financing charge rate for the scheme. It is useful in order to select the best depreciation methods, tax accounting procedure, ways of financing and the interest rates attached to them, etc.

The FCR is expected to decrease with time because of loan refund and interest charges decrease. Expressions of the FCR and levelised FCR are extracted from Stoll (1989).

The annual fixed-charge rate is a powerful tool but it is difficult to apply because it changes every year. On way of avoiding this problem, is to "levelise" the FCR. This concept of levelisation is based on the concept of time value of money and on the use of

a present worth discount rate. The general expression for levelising any variable V_i is given by Stoll (1989) as follows:

$$\bar{V} = CRF_N \cdot \sum_{j=1}^N \frac{V_j}{(1+r)^j} \quad (53)$$

Where:

- V_j = dollar amount of variable V in year j
- r = present worth discount rate
- CRF_N = capital recovery factor for N years at rate r

$$CFR_N = \frac{r(1+r)^N}{(1+r)^N - 1} \quad (54)$$

- N = number of years to be levelised
- \bar{V} = levelised value of variable V

6.3.5.1. Return on Investment

The FCR is derived from the cash report (Table 6) with the following assumptions:

- $NI_i = \text{DIVIDEND}_i$
- $\text{CASH}_{i+1} = \text{CASH}_i$
- $\Delta \text{WKCAP}_i = 0$
- $\text{CAPEXP}_i = 0$
- $\text{FINANCE}_i = 0$
- $\text{REFUND}_i = \text{DB}_i + \text{DFIT}_i + (\text{ITC}_i - \text{AITC}_i) \cdot \text{NORMITC}$

In the most general case, financing outstanding can be written:

$$\text{Financing Outstanding}_j = 1 - \sum_{i=1}^{j-1} \left[\text{DB}_i + \text{DFIT}_i + (\text{ITC}_i - \text{AITC}_i) \cdot \text{NORMITC} \right] \quad (55)$$

At year j, the bond interest, preferred dividend and net income requirements are then:

$$\text{Interest Expense}_j = bB \left\{ 1 - \sum_{i=1}^{j-1} \left[\text{DB}_i + \text{DFIT}_i + (\text{ITC}_i - \text{AITC}_i) \cdot \text{NORMITC} \right] \right\} \quad (56)$$

And,

$$\text{Net Income}_j = \text{Preferred Dividends}_j + \text{Common Stock Earnings}_j$$

$$\text{Net Income}_j = (pP + cC) \left\{ 1 - \sum_{i=1}^{j-1} \left[\text{DB}_i + \text{DFIT}_i + (\text{ITC}_i - \text{AITC}_i) \cdot \text{NORMITC} \right] \right\} \quad (57)$$

6.3.5.2. Depreciation

As quoted before, the depreciation of book purposes follows a straight-line depreciation method.

$$\text{Book Depreciation} = DB_j = \frac{1}{\text{BookLife}} \quad (58)$$

6.3.5.3. Taxes

The following expression of the income tax is based on Equation 45 and Equation 46:

$$IT_j = \frac{t}{1-t} \left[DB_j - DT_j + DFIT_j - ITC_j(1 - NORMITC) - AITC_j \cdot NORMITC + NI_j \right] \quad (58)$$

Where NI_j is expressed in Equation 57.

Therefore:

$$\text{Income Taxes Paid}_j = IT_j - ITC_j \quad (60)$$

And the deferred income tax is:

$$DFIT_j = t \times (DT_j - DBT_j) \cdot NORMDEPR \quad (61)$$

The deferred investment tax credit is:

$$\text{Deferred Investment Tax Credit}_j = ITC_j \cdot NORMITC \quad (62)$$

And finally, the amortisation of investment tax credit is:

$$\text{Amount of } ITC_j = AITC_j \cdot NORMITC \quad (63)$$

6.3.5.4. Ad Valorem Taxes and Insurance

These taxes are typically based on plant's initial capital cost. Common values range from 2 up to 4% per year.

$$\text{Ad Valorem} = AV_j \quad (64)$$

6.3.5.5.Total Annual Fixed-Charge Rate

The total annual fixed-charge rate is the sum of the following components:

$$\begin{aligned}
 \text{Annual Fixed-Charge Rate}_j &= \text{Interest Expense (Equation 56)} \\
 &+ \text{Net Income (Equation 57)} \\
 &+ \text{Book Depreciation (Equation 58)} \\
 &+ \text{Income Taxes Paid (Equation 60)} \\
 &+ \text{Deferred Income Tax (Equation 61)} \\
 &+ \text{Deferred Investment Tax Credit (Equation 62)} \\
 &+ \text{Amortisation of Investment Tax Credit (Equation 63)} \\
 &+ \text{Ad Valorem (Equation 64)}
 \end{aligned}$$

6.3.5.6.Levelised annual Fixed-Charge Rate

Derived from previous expression, with a discount rate r and a number of years N to be levelised, final expression components can be written as follows:

Levelised Interest and Net Income =

$$(bB + cC + pP) \left\{ 1 - \left[\frac{\overline{DB}}{r} - \frac{SFF_N}{r} + \frac{t}{r} (\overline{DT} - \overline{DBT}) \cdot NORMDEPR \right. \right. \\
 \left. \left. + ITC \cdot \left(1 - \frac{CRF_N}{1-r} - \frac{\overline{DITC} - SFF_N}{r} \right) \cdot NORMITC \right] \right\} \quad (65)$$

Where: SFF_N = Sinking Fund Factor

$$SFF_N = \frac{r}{(1+r)^N - 1} \quad (66)$$

Levelised Income taxes =

$$\overline{IT} = \frac{t}{1-t} \left[\frac{\overline{DB} - \overline{DT} + t \times (\overline{DT} - \overline{DBT}) \cdot NORMDEPR - ITC \cdot (1 - NORMITC) \cdot \frac{CRF_N}{1+r}}{- ITC \cdot \overline{DITC} \cdot NORMITC + \left(\frac{pP + cC}{pP + cC + bB} \right) \times (Eq.65)} \right] \quad (67)$$

$$\text{Levelised Income Taxes Paid} = \overline{IT} - ITC \cdot \frac{CRF_N}{1+r} \quad (68)$$

$$\text{Deferred Income Taxes} = t \times (\overline{DT} - \overline{DBT}) \quad (69)$$

$$\text{Deferred Investment Tax Credit} = ITC \cdot \frac{CRF_N}{1+r} \cdot NORMITC \quad (70)$$

$$\text{Amortisation of Investment Tax Credit} = ITC \cdot \overline{DITC} \cdot NORMITC \quad (71)$$

$$\text{Ad Valorem Taxes and Insurance} = \overline{AV} \quad (72)$$

The levelised fixed-charge rate is sum up of :

$$\begin{aligned} \text{Levelised FCR} = & \text{Levelised Interest and Net Income (Equation 65)} \\ & + \text{Levelised Book Depreciation (} \overline{DB} = 1/N \text{)} \\ & + \text{Levelised Income Taxes Paid (Equation 68)} \\ & + \text{Levelised Deferred Income Taxes (Equation 69)} \\ & + \text{Levelised Deferred Investment Tax Credit (Equation 70)} \\ & + \text{Levelised Amortisation of Investment Tax Credit (Equation 71)} \\ & + \text{Levelised Ad Valorem and Insurance (Equation 72)} \end{aligned}$$

Figure 99 is the sort of plot that the DSS provides as far as the fixed-charge rate economic technique is concerned. In that particular case, the life of the project is thirty years and tax accounting is normalised over 20 years, explaining the sharp increase at year 21. This chart can be very useful in order to select tax depreciation methods and type of tax accounting procedure. The levelised value therefore provides a key parameter for this type of very important preliminary accounting decision making. The Figure 98 shows that the levelised FCR is provided under “the revenue requirements method”, because the latter is a derived economic technique, which include production cost in addition.

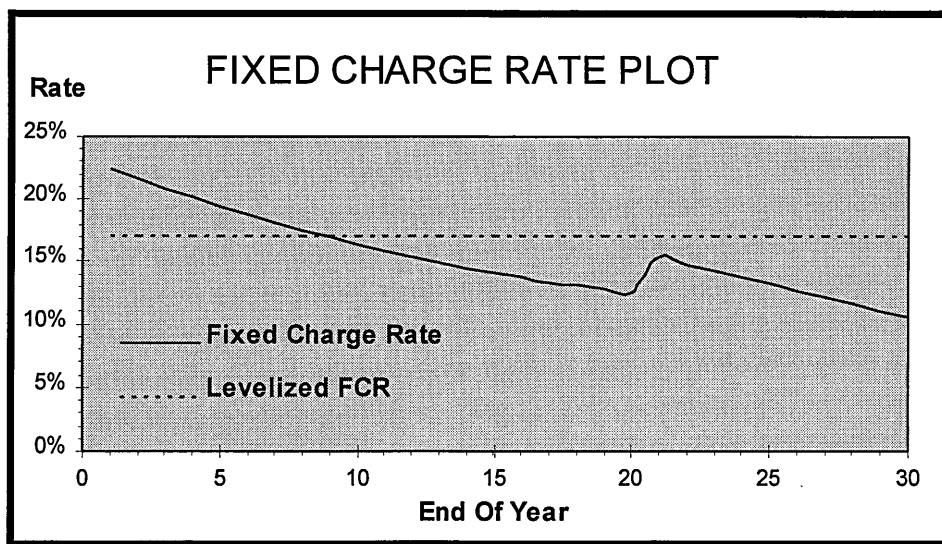


Figure 99: Instance of FCR Chart displayed by the DSS

6.3.6. Revenue Requirements Method

This method is the economic evaluation method used predominately by the utility industry. Revenue requirements consist of two items: the annual fixed-charge rate on a new investment and the annual expenses for production, operating and maintenance. The decision criterion among alternatives is to implement projects that have the least present-worth revenue requirements.

Present-worth revenue requirements can be calculated and presented in three formats:

1. the cumulative present-worth method,
2. the levelised annual cost method,
3. the equivalent capitalised cost method.

6.3.6.1. The Cumulative Present-Worth Method

PWRR = N-year Present-Worth of Fixed-Charge + Cum. Present-Worth Operating Cost

$$PWRR = FCR \cdot CapCost \cdot PWF_N + \sum_{j=1}^N \frac{PC_j}{(1+r)^j} \quad (73)$$

Where: PWF = Present-Worth Factor

$$PWF_N = \frac{(1+r)^N - 1}{r \cdot (1+r)^N} = \frac{1}{CRF_N} \quad (74)$$

6.3.6.2. The levelised Annual Cost Method

LAC = Levelised Fixed-Charge Rate + Levelised Production Cost

$$LAC = LevFCR + CRF_N \cdot \sum_{j=1}^N \frac{PC_j}{(1+r)^j} \quad (75)$$

6.3.6.3. The Equivalent Capitalised Cost Method

ECC = (Cumulative Present-Worth Cost) / (Levelised Annual FCR · Present-Worth Factor)

$$ECC = \frac{\sum_{j=1}^N \frac{PC_j}{(1+r)^j}}{LevFCR \cdot PWF_N} \quad (76)$$

In Stoll (1989) is explained the advantages and drawbacks of these parameters. In all cases they must be utilised to compare different alternatives, thus they do not provide a direct answer as NPV, Pay-Back, IRR economic evaluation methods, but only means of assessing one particular scheme against many others.

6.3.7. Direct Units Comparison Economic Evaluation Methods

6.3.7.1. Cost of Electricity

The cost of electricity (COE) or electricity pricing method is a very important during PPA and GSA contracts negotiations, since it provides the base-line over the lifespan of the project, of what the cost of producing electricity under off-design, degradation, etc. conditions, would be. Before fixing electricity retail prices, it is advised to run a first simulation with fictitious prices, in order to only assess production costs and plant availability that will provide the annual production and O&M cost per kilowatt-hour, known as cost of electricity and plotted as follows:

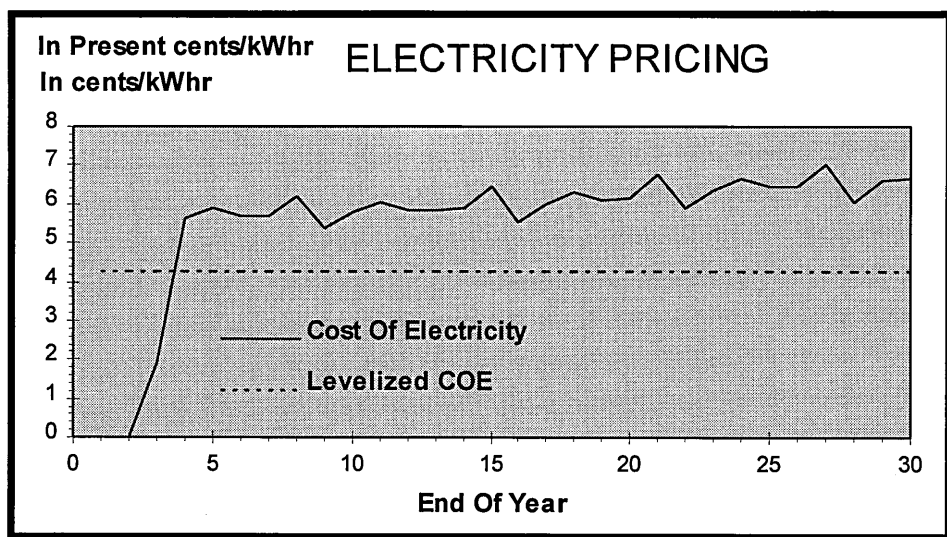


Figure 100: Cost of Electricity Trend provided by the DSS

Using these curves, it then possible to apply specific rate of return in order to ensure a profitability margin. Running therefore further simulations could provide all sorts of economic parameters that would reveal real effects of this margin on the overall economics of the project.

The levelised COE is there, once again a key figure for comparing different scenarios or alternatives.

6.3.7.2. Net Present Value Index

The NPV index is simply the ratio of the NPV and capital cost. That provides a ratio of profit or loss per capitalised present dollar. Another formula is sometimes proposed, where this NPV ratio is added to unity. One must understand that a positive NPV means that in present money, all capital costs have been recovered and some profits have been generated. That is why, unless the ratio is negative, profits in present money can be expected at the end of the project.

$$Index = \frac{NPV}{CapCost} \quad (77)$$

6.3.7.3. Return On Investment

The return on investment calculated in the DSS includes depreciation. It is a measure of the annual rate of return on capital employed by a project. Allen (1991) defines it as “per cent ratio of average yearly profit (net cash inflow) over the productive life of the project, divided by the total initial investment.

$$ROI = \frac{CumCash_N / BookLife}{CapInvestment} \times 100 \quad (78)$$

6.3.7.4. Accounting Rate of Return

This concept is extracted from Gazely (1993). It seems that this type of rate of return is used in some industries but no detail whatsoever could be found to explain the following expression:

$$ARR = \frac{CumCash_N / BookLife}{\frac{CapInvestment}{2}} \times 100 \quad (79)$$

6.4. Conclusion

The economic model proposed by the DSS is very versatile and provide tens of economic parameters based on different economic evaluation methods and give a large number charts, clearly represented. This represents a very powerful tool for such a DSS, since CC schemes involve large capital investments with long-term contracts, and thus they require flexible and accurate economic appraisals to support decision making. It is clear that reliable plant simulations and robust economic appraisals already provide an

invaluable tool for planners to manage negotiations of GSA and PPA contracts with a higher level of confidence.

However, decision making under uncertainty must also be taken into account in order to assess all possible scenarios that could occur during the project lifespan. The DSS therefore includes a simulation risk analysis technique based on Monte-Carlo method that utilises both technical and economic models, previously detailed, under different types and levels of uncertainty.

Chapter 7

RISK ANALYSIS

7.1. Introduction to Risk and Uncertainty

As the process of project appraisal necessarily involves dealing with the future there will be a degree of risk attached to many projections. Most of the information needed for an evaluation and concerned with the future are in the form forecasts and estimates. Besides some of those forecasts and estimates will be more uncertain than others. *The "risk" situation is one where the complete range of possible outcomes for a project can be foreseen and their relative chances estimated with reasonable confidence. The "uncertain" situation is again where the complete range of possible outcomes can be identified, but their relative chances are not known, or can only be guessed at.*

In preliminary studies, using statistical data published, or using experience, it is possible to plot "hazard" maps that could reveal important factors that occur frequently and that affect performance quite enough to be regarded as high risk factor. To illustrate the matter, Figure 101 is a plot based on annual north American unit performance statistics, more especially on the top 25 outage and de-rating cause statistics.

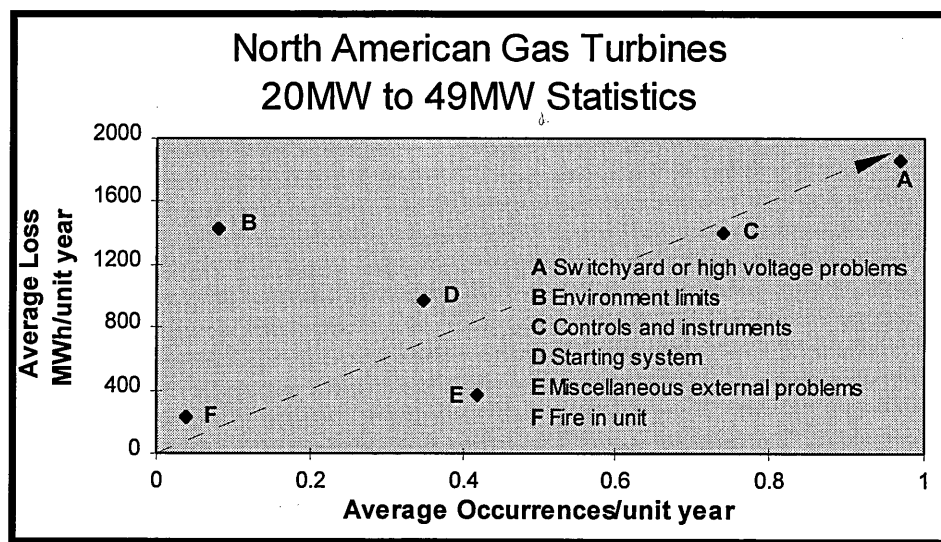


Figure 101: Instance of Risk Map

The dotted arrow provides the location of high risk factors that do not present any kind of trade-off between performance loss impact and frequency of occurrence. For instance switchyard or high voltage factor would require a precise assessment in order to compute the level and the pattern of uncertainty attached to it into a proper risk analysis model.

A number of options are available in dealing with risk. Two of them will be discussed, but only one is totally integrated into the DSS:

- **Sensitivity analysis:** Carry out sensitivity analysis by a systematic analysis of the effects of varying the values assumed for certain variables on the overall result of the proposal. Clearly, by this process it would be possible to identify those factors to which the project is most at risk. If the project does, subsequently, go ahead, these factors would be the key statistics to monitor. This technique very useful as a preliminary risk analysis can be easily implemented by just modifying one particular input parameter and seeing output data.
- **Simulation techniques:** Undertake simulation using a technique such as Monte Carlo analysis where each factor affecting the investment decision is examined and random numbers are used to generate an estimated NPV, probability distribution and standard deviation for the project. This kind of risk analysis requires calculation simulation model, contrarily to the sensitivity risk analysis.

7.2. Sensitivity Analysis

This method is used to explore the effects of uncertainty in different areas on project economic viability, but it does no attempt to quantify the uncertainty in those areas.

Because of the way many factors interact in cash flow and economic evaluation calculations, the conclusions from a sensitivity analysis may not always be obvious beforehand, and it is advisable to do analysis systematically, preferably using spreadsheet. The main drawback of sensitivity analysis is that it gives no indication of the likelihood or probability that any variation of a particular magnitude will actually occur. This is left entirely to the unstructured opinions or views of the user or decision-maker.

In order to deal systematically and rationally with uncertainty and economic risk in economic evaluation, it appears that it is of prime importance to quantify uncertainty in terms of the (subjective) probabilities of occurrence of possible alternative situations that could arise.

Simulation techniques such as the Monte-Carlo method respond to these requirements.

7.3. Simulation Technique - The Monte-Carlo Method

7.3.1. Subjective Probability Distributions

Uncertainty in an estimate or forecast can be represented by a continuous or semi-continuous subjective probability distribution which expresses the relative chances that the variable being estimated will turn out to have any particular value. Although it may be convenient and appropriate to express a subjective probability distribution as a particular mathematical function, for example as a normal distribution, Figure 102 shows different types of subjective probability distributions.

In theory according to the perception of the situation, a subjective probability distribution may be of any appropriate shape. If all the input subjective probability distributions to an economic evaluation can be expressed analytically in mathematical form, then it should be possible to calculate the mathematical form of the output NPV distribution. In practice, however, mathematics become very complex. Approaching numerically those continuous distributions by representing them using histograms can solve this problem. But to be computed, these histograms would have required so many input data that the DSS would have lost usability. That is why, only normal distribution is proposed in the DSS, given that, this is the commonest distribution utilised in statistics, since it describes most of statistical phenomena. Moreover Excel 7.0 proposes very efficient random generation function following normal law.

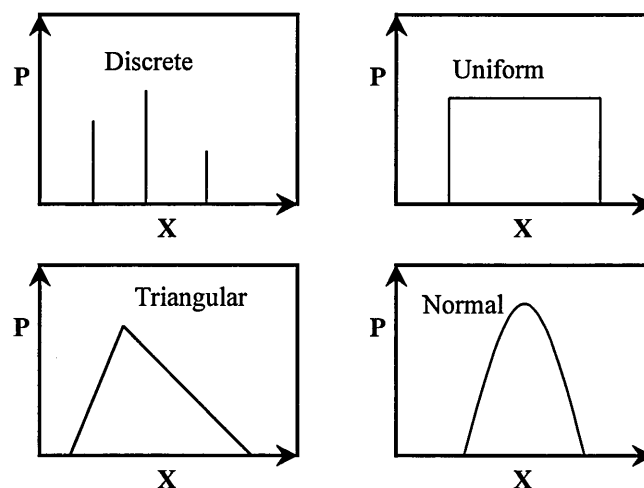


Figure 102: Some types of subjective probability distributions

7.3.2. The Monte-Carlo Method

The basic idea of using the Monte-Carlo simulation technique is to carry out a large number of individual project evaluations, each with different input values, selected from their subjective probability distribution in random combinations.

To complete Monte-Carlo simulation procedure applied to the economic evaluation and risk analysis of a project is shown in Figure 103.

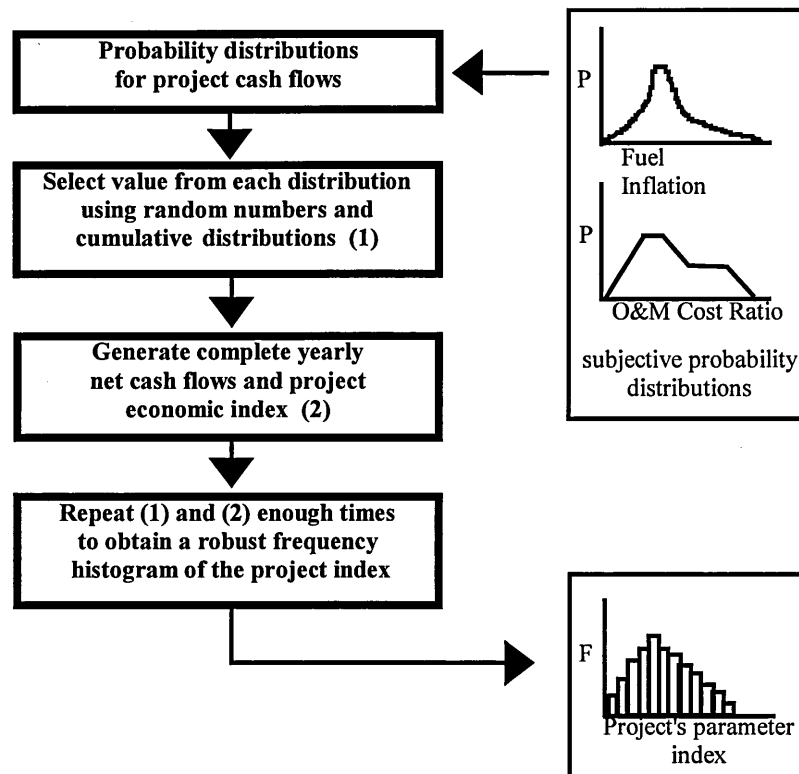


Figure 103: Procedure for economic risk analysis using Monte-Carlo simulation

At each iteration a different random number is generated for each stochastic input and this random number is matched with the input distribution to select a particular value from the distribution. These selected values, along with any other fixed input data, are used in the project cash flow and evaluation model to calculate a value for the project NPV, which is classified within predetermined steps of an overall NPV range.

As iterations are repeated more and more times, the frequencies with which particular values are selected from the input distributions approach more and more closely to their probabilities and the frequency with which the resulting NPV falls within the specified NPV steps approaches a stable distribution.

If sufficient iterations have been performed to simulate the output distribution adequately, the shape of the histogram should be insensitive to changes in the number of iterations and to changes in the sequence of random numbers used.

Risk evaluation can be represented by the cumulative distribution curve, as it will be shown in chapter 9.

Chapter 8

DECISION SUPPORT SYSTEM PROGRAMMING

8.1. Excel and Visual Basic

Excel 7.0 is the first Microsoft's spreadsheet, designed especially for Windows 95 environment. From a programming viewpoint, nothing has been added in Excel 7.0 compared to the previous version 5.0, except a larger memory capacity for Visual Basic "modules". That explains why the DSS which counts several thousands statements, more than 40 window-oriented objects, can not be run by Excel 5.0.

Microsoft developers basically increased by one third the programming capacity when they designed the version 7.0.

Even if Lotus 1-2-3 is commonly used in industry, there is no doubt that the teachers and students have accepted Excel as the best spreadsheet available in the market, not only in terms of its user-friendliness, but also in terms of its flexibility. Excel is more than a spreadsheet. Linked with all Microsoft software and many other types of external applications, it can deal with several data base configurations and import a wide range of external documents. In addition, versions 5.0, 7.0 and 97 offer the opportunity to use a Visual Basic compiler in order to build and personalise any application in a Windows-frame.

However this particular new powerful tool is generally underestimated and then partially exploited. To illustrate how powerful Visual Basic programming can be compared to usual "cells programming", to perform exactly the same task, a Visual Basic program requires one quarter of hard disk memory of a usual "cells programming".

The use of Microsoft's software in a Window operating system without Visual Basic language can be very tricky (Mansfield, 1995).

First of all, most of Microsoft's software are programmed in Visual Basic. The main reason is, that Visual Basic offers a radically new, radically easier way to create programs for the Window operating system. Now that Windows 95 will become the standard operating system for personal computers, Visual Basic must respond to the new structures and techniques involved in writing programs for Windows 95 instead of Window 3.1: 32-bit vs.16-bit, object-oriented programming versus "procedural" programming, and so on.

It is clear that Visual Basic represents one of the most adapted programming language in a Windows 95 environment and surely the easiest to be quickly learnt.

8.2. DSS Structure

The structure of the DSS is such that each simulation model can be run as a separate task. It is important that the DSS always keeps its usability; therefore by using built-in menus, the user can run any type of calculation at any time, as shown is Figure 104.

On top of that, it is always possible to run Pythia by simply entering its location path in a dialog-box displayed for that purpose.

Once any type of calculation is performed, a new menus-bar is displayed enabling the user to modify technical and economic input data, and then run further simulations, or to display results, as well as all sorts of plots and charts.

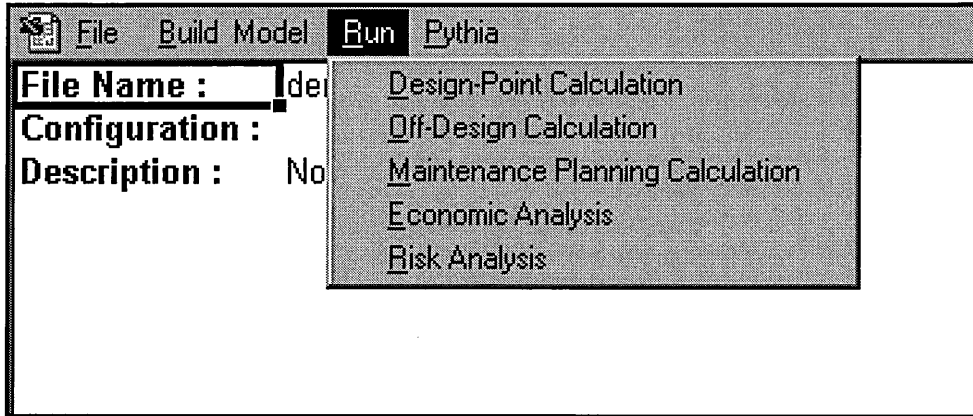


Figure 104: Run Menu

Details of this result menu are illustrated in the following figures:

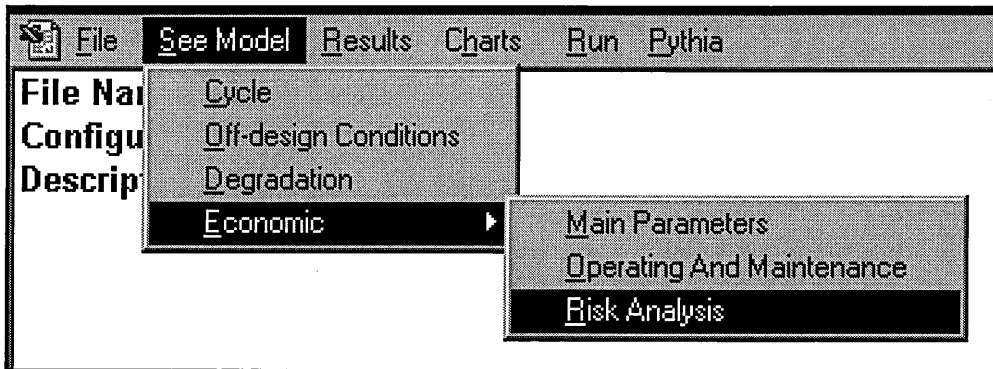


Figure 105: Input Data Modification

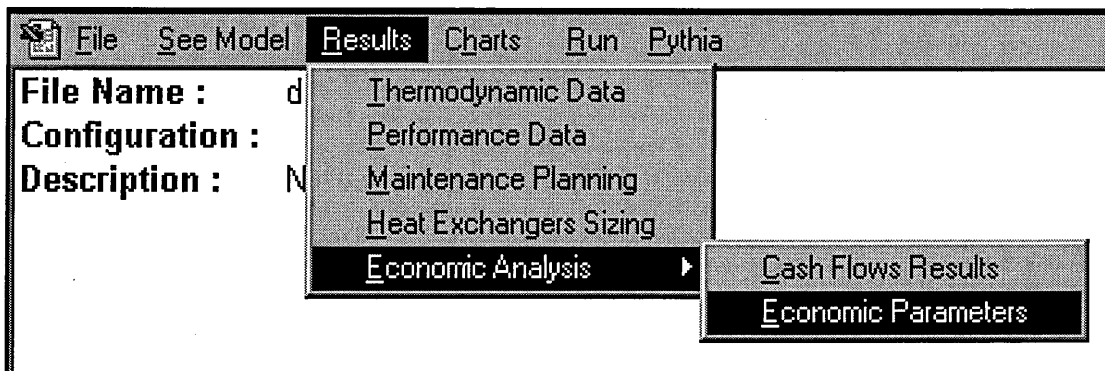


Figure 106: Results Display

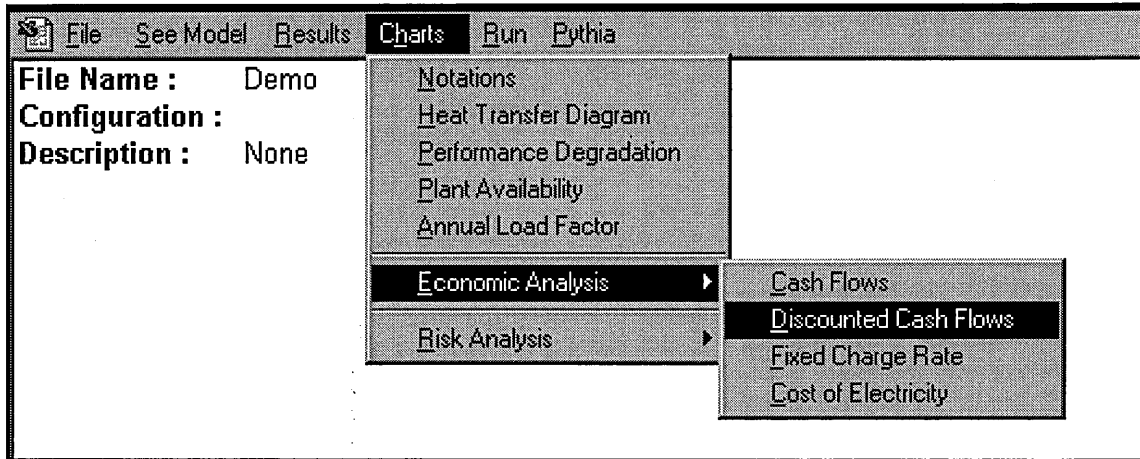


Figure 107: Charts Display (1)

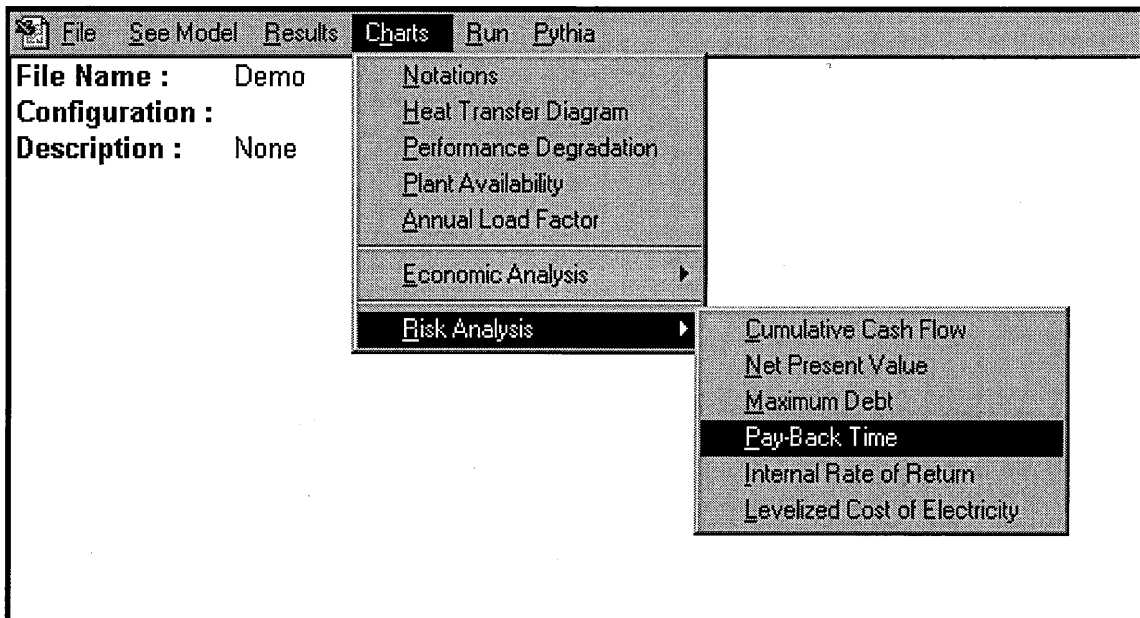


Figure 108: Charts Display (2)

It is clear that the combined utilisation of menus and windows, with all sorts of dialog-tools enhances the user-friendliness of the software and creates an effective computational environment, always sought by commercial package designers.

As computer programs are always perfectible, software designers must ensure a certain level of "survivability". This means that "open doors" must be kept in mind during the design, in order to ease further developments.

In the case of the DSS, it was known from the beginning that further developments would be undertaken. Therefore the user-interface, composed by menus and dialog-boxes is totally independent of a specific "worksheet" that contains all input data that have been stored after being written through the windows user-interface and which is

used as reference for all types of calculation. This means for instance that the user can always go to this worksheet before performing any calculation and modify the input data. This is a powerful structure, since it enables anybody to add another model that would directly modify some input data into this worksheet.

The overall structure is summarised in Figure 109, and it shows that everything is centralised around this "Data Worksheet".

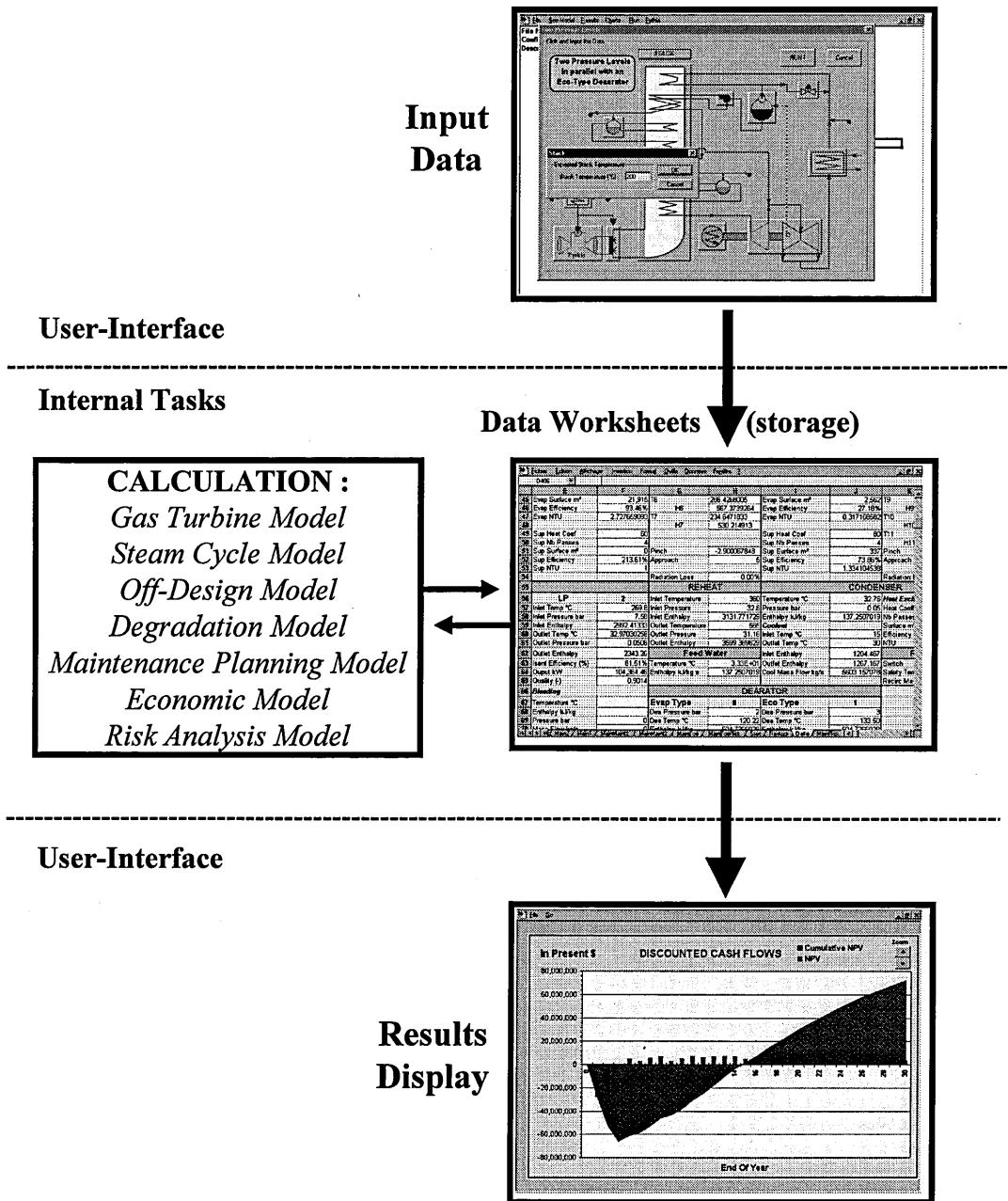


Figure 109: DSS Structure

Chapter 9

CASE STUDY

9.1. Case Study Definition and Presentation

9.1.1. Project Definition

The aim of the following case study is to illustrate one possible use of the DSS. One could imagine an IPP or a major power generation company that could be in the process of assessing bids sent by plant manufacturers. Bids usually provide all necessary data to perform plant performance assessment under common off-design conditions. Planners must closely study plant deterioration since the latter is greatly related to the type of application, location and maintenance planning strategy.

As far as the economic appraisal is concerned, plant manufacturers provide capital cost, payment scheduling and delivery times of the plant. The IPP or the large power generation company then, according to its financing policy and strategy, perform economic predictions and risk analyses in order to assess the level of short and long-term profitability.

This case study deals with a two-pressure (parallel) GE combined cycle power plant, in a 2-2-1 configuration. Design output is estimated at 657 MW and stack temperature around 99°C.

Both GE MS9001-FA, rated at 226.5 MW (natural gas) each, provide to the HRSGs the heat capable to produce 160 kg/s steam at 80 bar/560°C and 32 kg/s steam at 14 bar/300°C. The steam turbine net output is then 213 MW. The expected net efficiency of the plant is around 55%.

Steam turbine back-pressure and condensing pressure are 50 mbar, and deaeration pressure is set at 3 bar.

Generator efficiency is 99% and radiation loss through the HRSGs is set at 1% of the whole duty.

The plant is located in central Europe (at sea level) in countryside where air pollution is said to be minimum, but where pollen and other particulates provide relatively high fouling rates.

Maintenance actions are concurrently carried out by three 8hrs shifts.

As far as GSA contract is concerned, a 20 years fixed fuel price contract has been agreed with natural gas supplier, and the upward trend is based upon American price and fuel inflation and hence fixed at 1.5 % per year. Starting price is 2.5\$/GJ and natural gas LHV is 47 MJ/kg. It is assumed that fuel inflation after 20 years project will remain at 1.5% /year. But due to this lack of pricing knowledge, planners intend to conduct risk analysis to ensure feasibility and profitability for the last ten years of the scheme.

PPA requirements include a minimum annual load factor of 90% for 30 years, calculated from clean design conditions.

Besides the initial electricity retail price is at 55 \$/MWh and follows the annual price inflation rate set up at 3% per year. This pricing can be assessed using the cost of

electricity plot or the levelised cost of electricity, both provided by the DSS as illustrate in Figure 110 with regard to operating and maintenance costs including fuel and preliminary evaluations of plant availability and capacity factor.

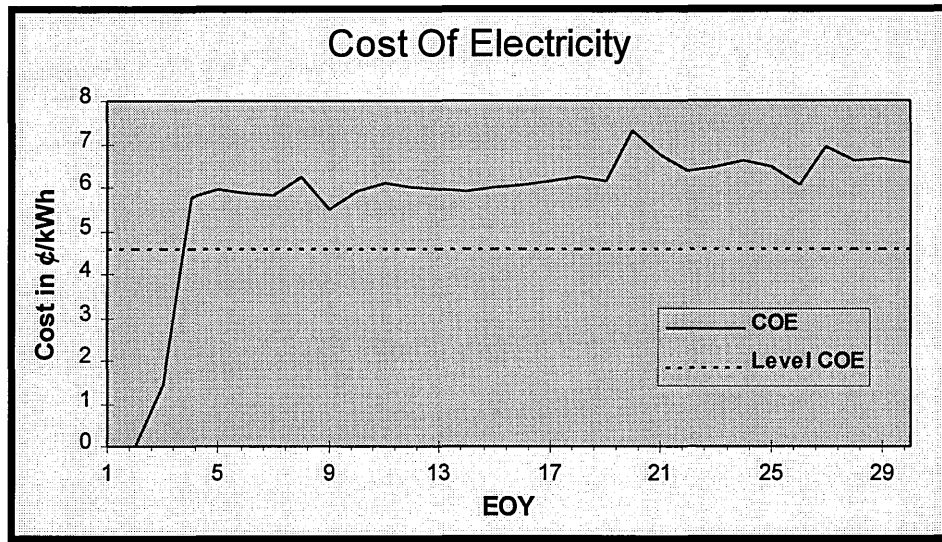


Figure 110: Electricity Pricing Curve

Operating and maintenance cost (without fuel) are assumed to be constant at 20\$/MWh per year with extra costs varying in accordance with the type of maintenance action as shown in the following table:

Table 7: Maintenance Planning

Inspection type	Interval (fired hour)	Duration (in 8hr shift)	Additional cost (in \$/kWh)
Combustion	8,000	6	0.1
Hot-gas Path	24,000	42	0.25
Major Overhaul	48,000	168	0.5

Plant financing (summarised in Table 8) provides a composite cost of capital of 11.6% and an after-tax CCC of 9.6%, which is the discount rate applied in DCF calculations. The capital cost is estimated at 400\$/kW and bank loans have a 20 years lifespan.

Table 8: Plant Financing

Financing type	% of each type	Interest rate
Long-term Bonds	50	10
Preferred Stock	20	15
Common Stock	30	12

9.1.2. Off-Design Input Data

9.1.2.1. Ambient conditions

French Normandy has a rather cold climate all through the year. Monthly ambient temperatures are extracted from 1991-1992 statistics. Input temperature profile is the following:

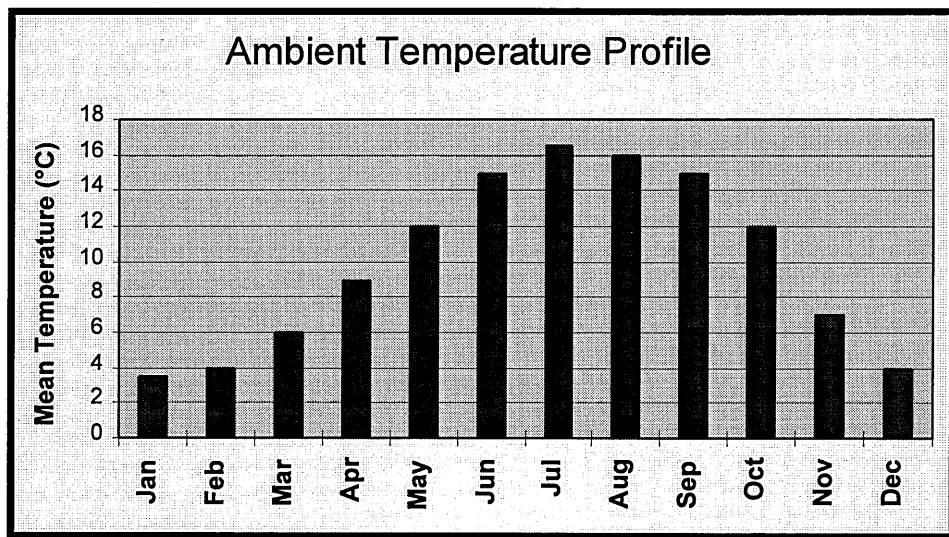


Figure 111: Ambient Temperature Profile (French Normandy)

Very cold conditions like in January for instance, increase the power output by about 8%. But the effect on the efficiency is less due to the decrease in compressor delivery temperature that leads to a slight increase in fuel flow to maintain the TET at its design level.

9.1.2.2. Part-load conditions

Even so the type of duty is base-load, planners intend to run the plant at 90% clean capacity in order to avoid power output shortages due to high degradation rates or very adverse ambient temperatures.

It must be pointed out that gas turbines can be used for peaking up to 110% of their design rating, but only for short periods. Peaking duty for combined-cycle plants is rather new and has only been successfully implemented in Japan by TEPCO with Frame 9FA engines. In this case study, one of the prime goal is to be capable of delivering power during 30 years, therefore anticipating degradation rates up to 10%, plant operators can prevent engines from peaking by lowering the capacity factor. The result is an annual load factor (Figure 112), which is well below 100% and which provides the power output required by PPA contract for years without major overhauls being carried out.

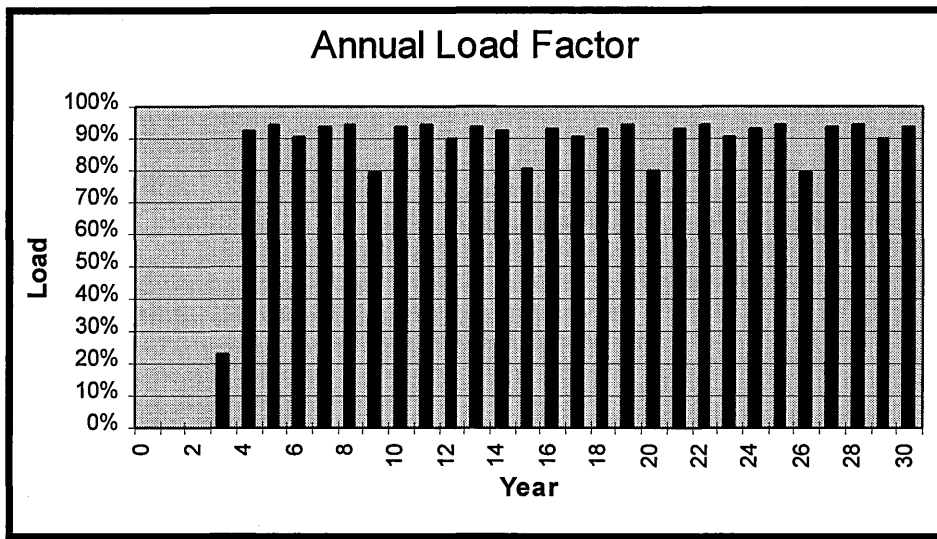


Figure 112: Plant Annual Load Factor

There are two ways available to plant operators in order to maintain a power output equals to 90% of design clean output. First way would be the decrease of the TET, increasing therefore hot-gas path components, but affecting overall plant efficiency by several tenths of percent. The second way would be the use of VIGV to decrease compressor inlet mass flow by about 10%. This method does not increase the life of hot-gas components, but it provides a better efficiency as illustrated in Figure 83. Frame 9FA as an up-to-date engine, based on latest turbine blade materials and cooling techniques for a longer life, fuel saving due to a higher efficiency appears to be more economically attractive than savings in spares and maintenance actions in the case of a lower TET.

9.1.2.3.Engine Deterioration

According to the default model providing by the DSS, two sets of degradation curve parameters must be input. The following values are based on baseline degradation curves advised by GE Power System. They have been modified in order to take into account the type of duty, the location and specific levels of components degradation such as compressor fouling or turbine erosion. Referring to Equation 34, input data are presented in Table 9.

Table 9: Performance Deterioration Curve Input Data

	A_0	τ
Average Loss During Operation	4	1625
Non-Recoverable Degradation	2	9225

It has been assumed that the recovery factor after major overhauls is 100%, since such maintenance actions carried out during several months implies new blading and other new hot-gas path components, providing thus almost clean design performance. Engine degradation rates based on Escher (1995) and illustrated in Chapter 4.3.2, have been computed into Turbomatch and using mathematical expressions stored in the engine library, deteriorated engine exhaust conditions have been calculated to perform steam cycle simulation.

It is clear that another maintenance planning strategy would have provided different performance loss curves and thus different plant availability and cash flows distributions.

9.2. Datum Case Results

The datum case deals with the scenario that contains predicted economic and operating parameters.

However because forecast is not an absolute science, planners usually conduct scenario studies in order to deal with unexpected events, such as forced outages or new windfall taxes. This type of analysis is very closed to the sensitivity risk analysis, and is often regarded as the preliminary step before running a more detailed risk analysis. The most common simulation technique is the Monte Carlo method, previously presented in Chapter 7.3.2.

9.2.1. Plant Simulation Results

Very few input data usually are missing from a bid provided by plant manufacturers in order to perform the whole thermodynamic calculation. HRSG pressure drops, heat exchange coefficients and radiation losses as well as generator efficiency have to be assumed. Thermodynamics results are summarised in reference to specific notations available by calling and displaying the reference map illustrated in Figure 113.

Planners usually look at pinch points and stack temperatures to validate the simulation. Steam quality at the steam turbine exit is always checked and must be greater than 0.88. In this particular simulation LP pinch point is 7°C and HP pinch point 16°C.

Optimised HRSGs usually provide pinch points around 10°C, thus pressure drops assumed previously turn out to be close enough to the reality.

As far as stack temperature is concerned, the simulation gives a value of 100.4 °C, which is very similar to manufacturer specifications.

Steam moisture is 0.884 proving that superheated steam has expanded as much as possible.

After a design point simulation, the user can have access to heat exchanger sizing results, which will enable him to assess HRSG cost related to total heat exchange

surface. He could even try to optimise it in order to claim a better HRSG internal arrangement from plant manufacturers.

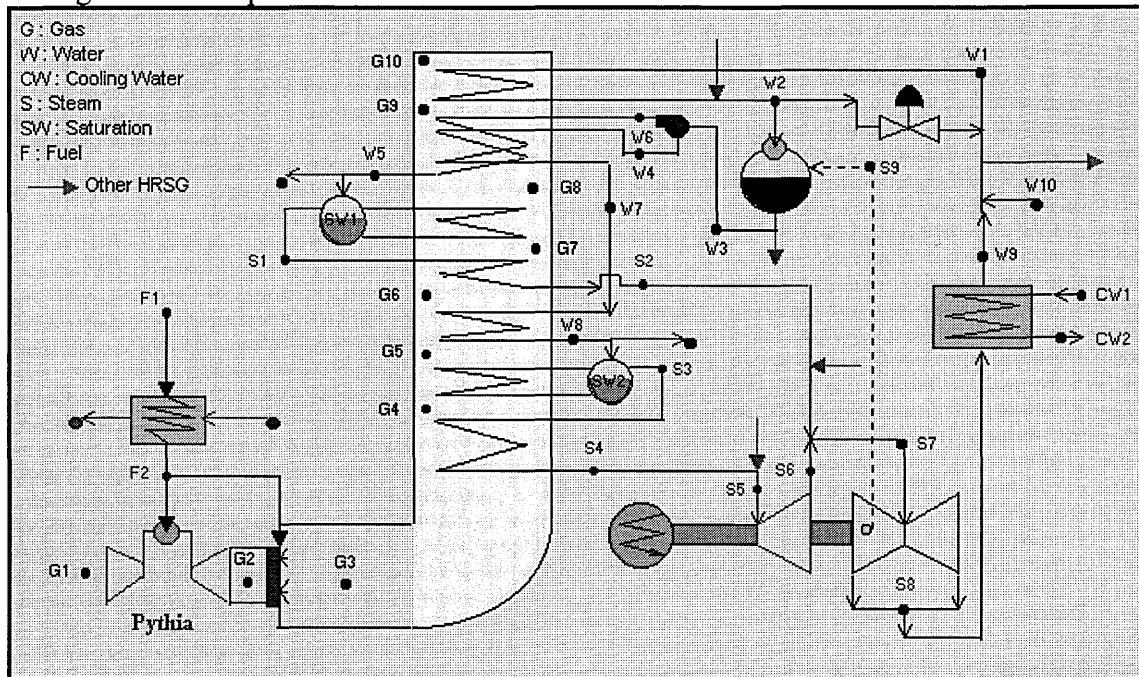


Figure 113: Plant Diagram and Notations

As far as plant performance, they are summed up and displayed in the following type of window:

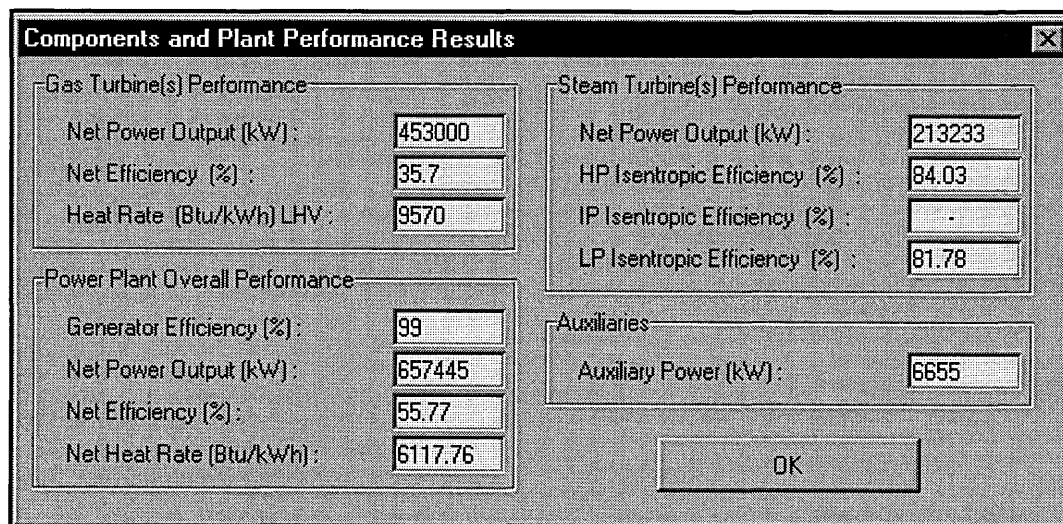


Figure 114: Plant Performance Results

These results are consistent with the 56% efficiency promised by this type of combined cycle plant, implementing up-to date engines and optimised HRSG arrangement.

Concerning maintenance planning, concurrently carried out by three shifts, the DSS provides a detailed calendar of all types of maintenance action. Therefore planners can

optimise it if necessary by changing inspection intervals, degradation rates, or maintenance work duration, and run the subroutine again.

9.2.2. Economic Results

Planners are mainly interested in three economic parameters and one operating parameter namely:

1. The payback time.
2. The net present value.
3. The internal rate of return.
4. And the cost of electricity or the levelised cost of electricity.

All these key parameters are displayed as follows:

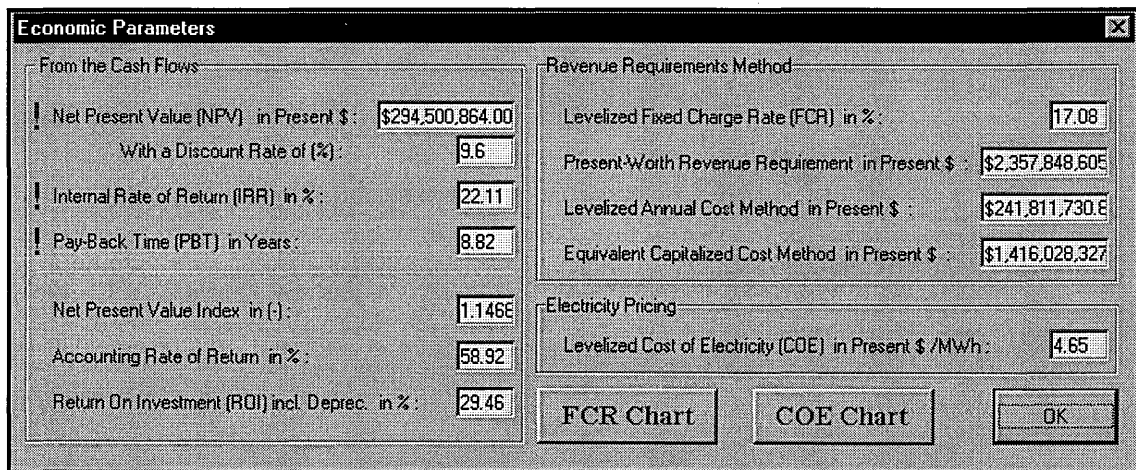


Figure 115: Economic Results

According to them, the scheme seems to generate good profits but with a rather late payback time, usually about 5-6 years. It must be pointed out that the payback time is always calculated from the non-discounted cash flows which means that in present currency planners would have to wait for 11 years before recovering the investment, as Figure 116 shows.

COE curve as well as the levelised COE figure give a good evaluation of the profit that can be expected with regard to retail pricing negotiated in PPA contracts. Fuel contracts or purchase agreements could be reviewed if simulations showed weaknesses in the NPV or IRR generated by the project or showed some risk of bankruptcy. That is why planners usually run scenario studies and try to assess risk attached to specific high impact factors with regard to defined economic targets such as a payback time or a rate of return. In that particular project, it could be useful to have the risk pattern attached to the PBT.

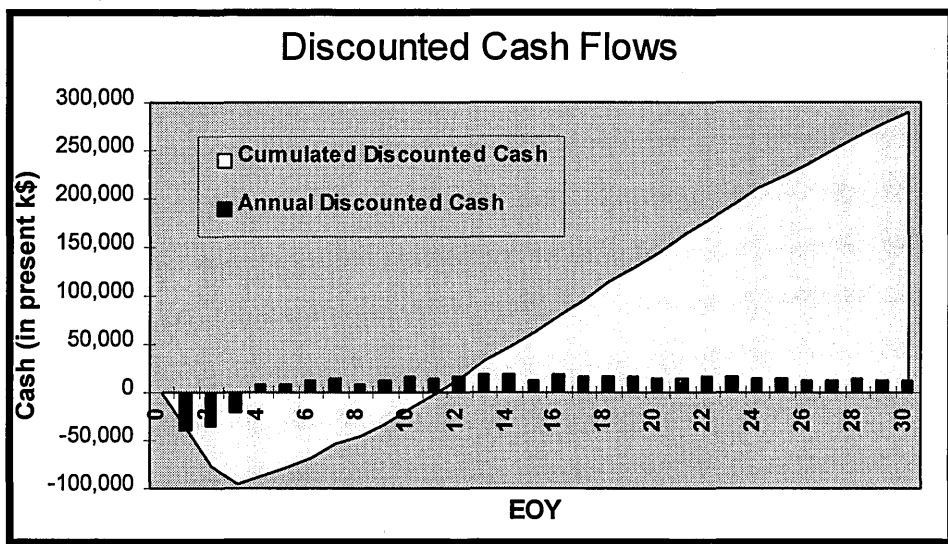


Figure 116: Project's Discounted Cash Flows

9.3. Scenario Study

Scenario studies are a quick and effective way of assessing high impact factors on the economics of the project. It is eventually up to the planners to decide whether or not a high impact factor contains also high level of uncertainty.

The following instance of scenario study must reveal worse scenarios possible. For that purpose, seven different scenarios have been run and their economic performance normalised with regard to the datum case, described previously.

The following parameters have been recognised as potential high impact factors for the likelihood of the project:

1. The electricity retail price
2. The fuel inflation rate
3. Operating and maintenance costs
4. The accounting procedure
5. The tax depreciation method
6. Financing sources
7. Loans life

Some of them, especially the economic factors should only guide planners to define basic accounting input data, and should require no prediction whatsoever.

On the contrary, operating parameters must be defined and predicted in order to ensure plant profitability over the years. Besides, it is of paramount importance to accurately

estimate the cost of electricity in order to negotiate a good PPA contract. Similarly with fuel pricing regarding GPA contract.

In Table 10 are summarised input and output data of this scenario study.

Table 10: Scenario Description

Change	Datum	Case A Elec. Price -0.5 \$/MW	Case B Fuel Inflation x 2	Case C O&M cost x 1.50	Case D Flow-through	Case E Linear Tax Deprec. Method	Case F Cost of Capital 12.5 %	Case G Loans Life = 10 years
NPV in present k\$	289,318	127,758	155,093	-7,284	289,318	290,735	278,258	280,889
IRR in %	21.91	14.69	17.24	9.37	21.91	21.79	21.63	20.01
PBT in years	8.88	13.16	10.62	18.81	8.88	9.02	9.08	10.03
Lev COE in ¢/kWh	4.61	4.61	5.08	5.51	4.64	4.64	4.7	4.61
Lev FCR in %	17.08	17.08	17.08	17.08	17.81	17.68	19.65	17.08

Figure 117 shows normalised deviations from the datum case for every variable parameter.

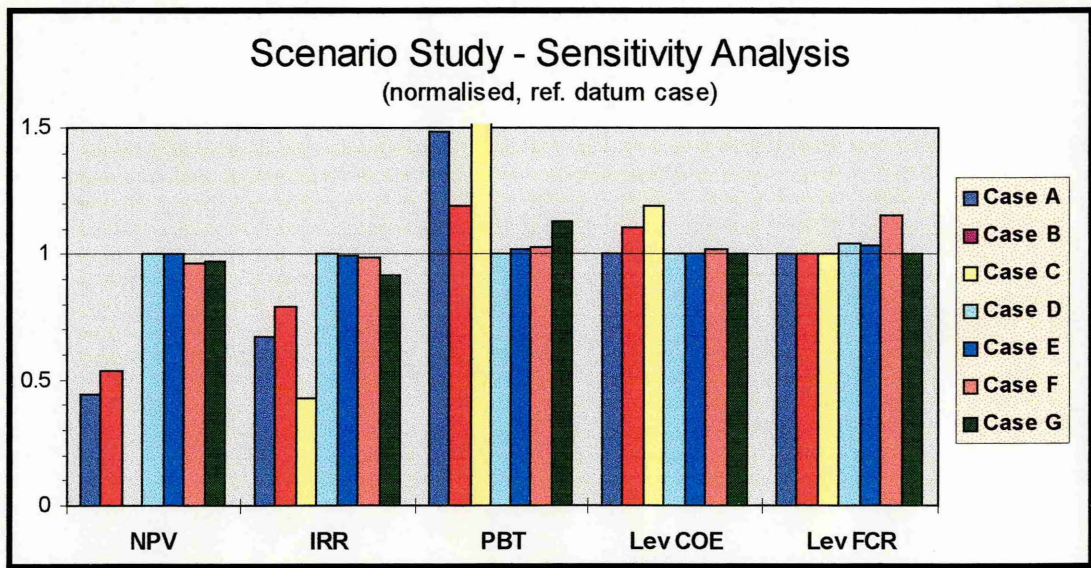


Figure 117: Sensitivity Analysis

According to the table and the plot, cases D, E, F and G, which refer to economic and financing parameters, clearly do not challenge the success of the project. But this type of study highlights important points. For instance accelerated tax depreciation methods provide better long-term economic results. Besides, bank loan life must be long enough to ensure reasonable financing charges in early years, thus an acceptable PBT.

As a matter of fact, loans are usually negotiated once GSA contracts signed. This emphasises the importance of a fixed fuel price agreement for the likelihood of such a scheme.

As far as operating parameters are concerned, huge differences can be noticed. First of all, case C is a bankrupt scenario type, since after 30 years, the NPV is negative and the IRR less than the discount rate. This scenario is more striking when plotting as follows:

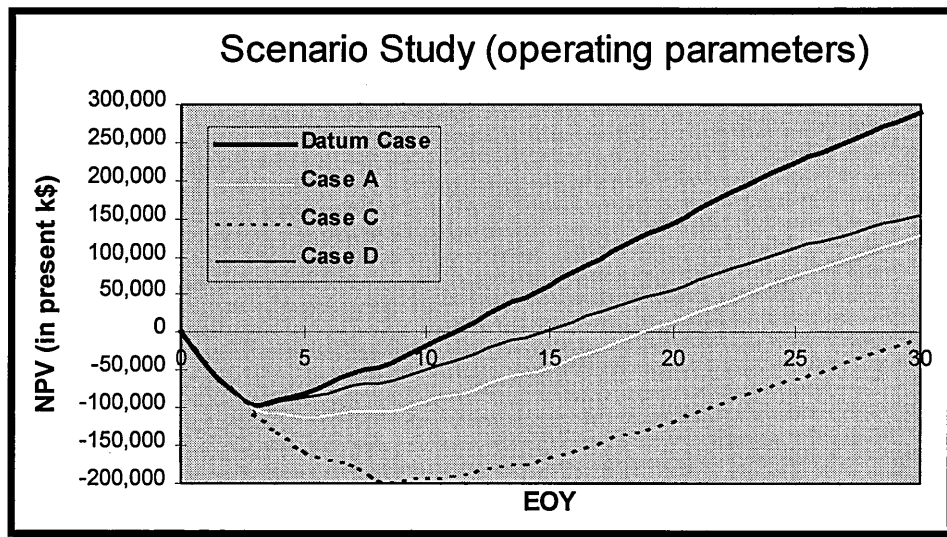


Figure 118: Discounted Cash Flows

Operating and maintenance costs without fuel must be carefully handled in the early years because plant-financing charges are very high and tend to dramatically postpone the payback time when unexpected costs add up. An increase in O&M could be well reflect new emission taxes that could be implemented by new governments or even by international environment protection agreement.

In addition to O&M costs, electricity retail pricing turns out to be one of the most important parameters since a slight decrease cuts profits by almost a half. Therefore plant owners and investors usually assess PPA contracts very carefully by running extensive and very complete risk analysis. PPA contracts are sometimes signed after several years of intensive negotiation and constitute a puzzle for investors, plant owners and lawyers.

Finally and without surprise, fuel inflation is a parameter that has to be well predicted since fuel costs are accounted for about 60% of overall production cost. Due to the fact that fixed fuel contracts do not exceed 15 or 20 years, planners must deal with high level of uncertainty concerning long-term profitability of the plant. Therefore, the use of a simulation tool to conduct risk analyses is almost compulsory, since fuel price trend can have any type of probable distribution. For the same reasons, with the annual price inflation rate, which provides electricity pricing trend, a high level of uncertainty in the last ten or even twenty years of the scheme must be considered for the decision making.

Owing to all these areas of uncertainty, it is wise to run a Monte Carlo risk analysis, whereby hundreds of different scenarios will provide risk patterns and direct readings of the likelihood for achieving specific economic targets, such as a PBT less than ten years or an IRR greater than 20%.

9.4. Risk Analysis

With regard to scenario study results and conclusions, three parameters have been regarded as high risk factors and thus a standard deviation applied to a Normal curve have been computed in order to reflect the level of uncertainty attached to them:

1. Fuel cost (annual fuel inflation): +/- 1%
2. Electricity pricing (annual price inflation): +/- 1%
3. O&M costs: +/- 1 ¢/kWhr

600 iterations have turned out to be sufficient to provide robust results and accurate materials, when 200 just had only given a good picture of the range of all economic outcomes.

The DSS provides NPV, IRR, PBT, CCF, maximum debt and LevCOE risk curves. Figure 119 and Figure 120 show possible outcomes related to the IRR and the PBT, and give the risk pattern attached to them.

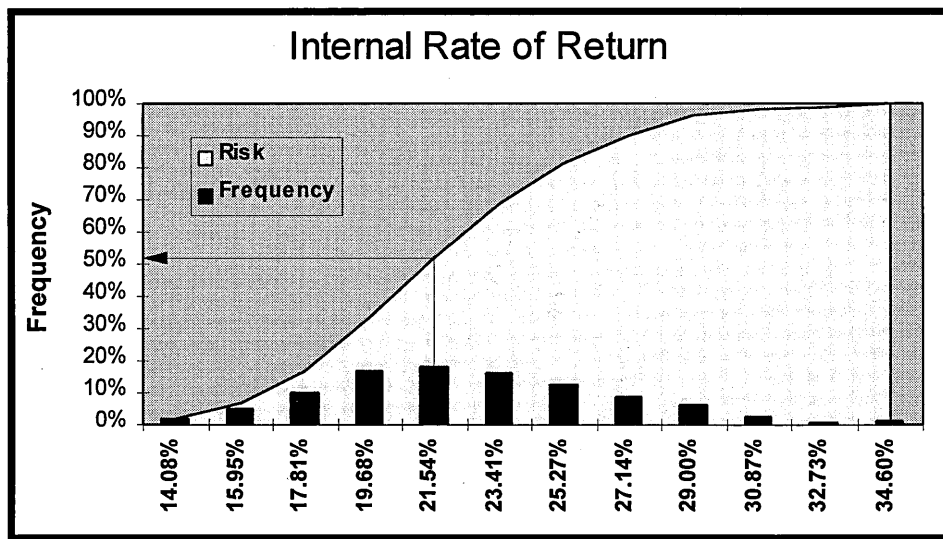


Figure 119: IRR Monte Carlo Simulation Result

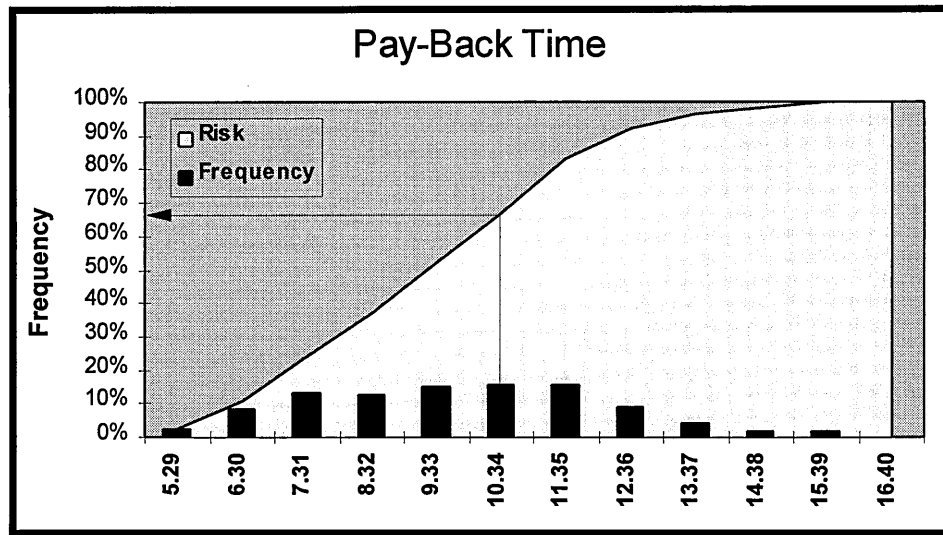


Figure 120: PBT Monte Carlo Simulation Result

To illustrate the decision making process related Monte Carlo simulations, it will be assumed that different bids are in competition and the selection will be based upon different criteria, two of them being the IRR and the PBT.

Planners have decided to set limiting values for both economic parameters, namely 20% for the IRR and 10 years for the PBT. These disqualifying values are illustrated in Monte Carlo plots and the level of risk attached to them is pointed out, 50% and 33% (=100 - 67) respectively.

It can be noticed that the shape of Normal curves computed for every risky variable is fully reflected by the above frequency plots. They are rather flat because the standard deviations for fuel and price inflation rates are very high compared to their mean values. That explains why ranges are wide and mean values give high cumulative frequency (or risk level).

These risk assessments provide materials for the decision making process since they are direct measurements of the risk attached to a particular project under a particular set of uncertainty models that must be compared to other competitive alternatives. Final decision will be a combination of different factors, some of which can not be modelled and simulated.

Chapter 10

FURTHER REQUIREMENTS

10.1. Technical Models

Hestia is a DSS for combined cycle plant schemes but it can be used for simple single or dual-pressure combined heat and power plant project evaluations. However, the lack of steam process plant modelling and the lack of load and heat demand curves, which are usually plotted in a daily basis make further works necessary. CHP schemes would be a very straightforward improvement and would dramatically increase the versatility of Hestia, without modifying any of the existing degradation, maintenance planning, economic and risk analysis modules.

Overall up-grades of the different steam cycle components could be implemented in order to up-date HRSG and steam turbine new technologies for instance. Similarly, since Hestia has been designed in such a way that it is simple to add new cycles and any other type of simulation model, users can always add advanced cycles like these based on supercritical steam for instance.

Steam cycle deterioration must be included since they constitute performance losses that can affect economic outputs. Combined with the existing degradation model provided by Pythia and Turbomatch, this improvement would enhance the overall accuracy of thermodynamic, performance and economic calculations.

Besides new maintenance models could add more flexibility for plant availability calculation; sequencing loading could be more investigated for instance.

Finally, a next big step could be the implementation of dynamic modelling in order to cope with peaking-duties that imply long transient periods. But it must be pointed out that this would require heavy numerical analysis, recognised in literature for being highly computational time consuming. It is not proven that dynamic modelling, even for peaking would give much better accuracy in economic appraisals of such long-term projects.

10.2. Economic Model

As far as the economic model is concerned, very few improvements can be added to the existed model. Economic appraisals usually require both accounting and finance analysis features such as tax depreciation, cost of capital, cash flows modelling, etc. Hestia provides a very robust model, which has been recognised as the most detailed and versatile since it can easily deal with different fiscal systems without large modifications. Incomes and outcomes are calculated in a monthly basis, increasing the accuracy of annual cash flows.

Besides all commonest economic techniques and parameters can be produced by the DSS, covering thus a wide range of different decision making requirements.

Finally an accounting table, summarising project cash flows and formatted for bookkeeping purposes is provided and can be saved with all other results.

However the life cycle cost model could be enhanced if new techno-economic factors were added. Fuel handling and treatment when liquid fuel is used, could improve the accuracy of production cost calculation. A more detailed overheads and maintenance cost model, divided into fixed and variable cost would reflect more effectively the reality. Hestia offers the opportunity to introduce new LCC models. O&M cost per kilowatt is directly read from a special spreadsheet interface that can be available to any other types of application or internal subroutines.

10.3. Risk Analysis Model

Risk analysis model provided by Hestia is limited to six parameters affecting: O&M fixed cost, O&M extra-cost due to maintenance action, fuel cost, down-time ratio, capacity factor and electricity retail price. These variables can be escalation factors, inflation rates, etc, depending on the type of trending method selected by the user.

Other parameters could be added to the existing list to make the risk analysis module more versatile and maybe more applied to specific types of simulation. The user could run risk analysis with engine component characteristics as variable parameters, such as compressor non-dimensional mass flow or turbine isentropic efficiency to assess the level of risk attached to some performance deterioration factors, presenting some uncertainty in a long-term prospective.

On top of that, new algorithms could be implemented in order to increase the number of possible probabilistic distributions that could be selected or new simulation techniques like Latin-hypercube could provide exactly the same type of results with a shorter computational time.

10.4. Programming

As far as software programming, Visual Basic turned out to be easy, quick and effective. Window oriented programming is nowadays becoming the least of software standard for evident usability and user-friendliness reasons.

Hestia like any other software has been designed so that new developers could up-grade it and up-date the existing code as easily as possible.

The user-interface design is directly reflecting personal design choices that can only be accounted to programming experience and aesthetic feelings. Therefore new developers are free to fully or partially reshape the program like plastic surgeons, by keeping the code alive in linking already built subroutines to new interface features.

It is also important to stress the fact that, Excel is becoming a very powerful tool integrating more programming capacity and capability, enabling thus very high standard software programming.

CONCLUSION

Entering the third millennium, combined cycle technology will probably be one of the most important power generation technologies because of its high flexibility with respect to fuel, type of duty and plant configurations in the uncertain world of competitive market. Making decision in a risky business implies the need of tools in order to reach a certain level of confidence especially when long term large capital are invested.

The main purpose of this work has been to lay the basis of an effective DSS for combined cycle schemes appraisals under uncertainty. Because usability and user-friendliness are of prime importance when designing such a complex computational program, Visual Basic 4.0 and Excel 7.0 have been designated for being the programming language and software framework. Such windows-oriented program offer a very efficient user interface, providing all sorts of communication tools such as buttons, dialog boxes, plots menus, etc ...

The main challenge of a DSS is to tie up plant simulation results such as efficiency, power output, availability with operating and economic parameters such as O&M costs, cash flows and pay-back time... in a risky environment, implying all sorts of uncertainty sources like price inflation, forced outages...

In those conditions, a steam cycle modelling including HRSG, ST, condenser... models have been linked with existing engine simulation programs, Pythia and Turbomatch in order to provide plant simulation results under any type of off-design conditions and deterioration.

Given that 1% in plant availability or 1% in efficiency can modify cash flows by millions of dollars or pay-back time by years, it is of paramount importance to ensure an accurate and effective way of assessing plant performance as well as the maintenance planning strategy, which usually sets planned outages and deterioration curves throughout the life span of the plant.

Hestia owns a default maintenance planning strategy based on actual industrial behaviours. This provides an easy and very quick way of running relevant case studies plant operating and maintenance knowledge.

Decision making under uncertainty is nowadays one of the most difficult challenge that long term capital investment in a competitive market has to face up. Simultaneously, risk analysis based on Monte Carlo method has been recognised as the most effective way for assessing risk, attached to a set scheme for a particular economic target. This methodology is easy to use but requires minute preliminary studies in order to recognise high risk impacts parameters and to estimate probabilistic distribution attached to them.

Finally; because the likelihood of making the right decision is after all a combination of factors, the economic model must provide large amount of data and plots, which have been seen as relevant and reliable. Hestia, in addition to a very versatile risk analysis tool, includes a robust and evolved economic and finance modelling capable of providing tens of key parameters such as internal rate of return, net positive value, payback time, return on investment, cost of electricity, fixed charge rate and which

deals with different economic analysis techniques such as discounted cash flows, revenue requirement method ...

Hestia is another member of Cranfield's Olympus, who highlights three major and promising concepts for research applied to the power generation industry: performance simulations, life cycle cost and risk analysis.

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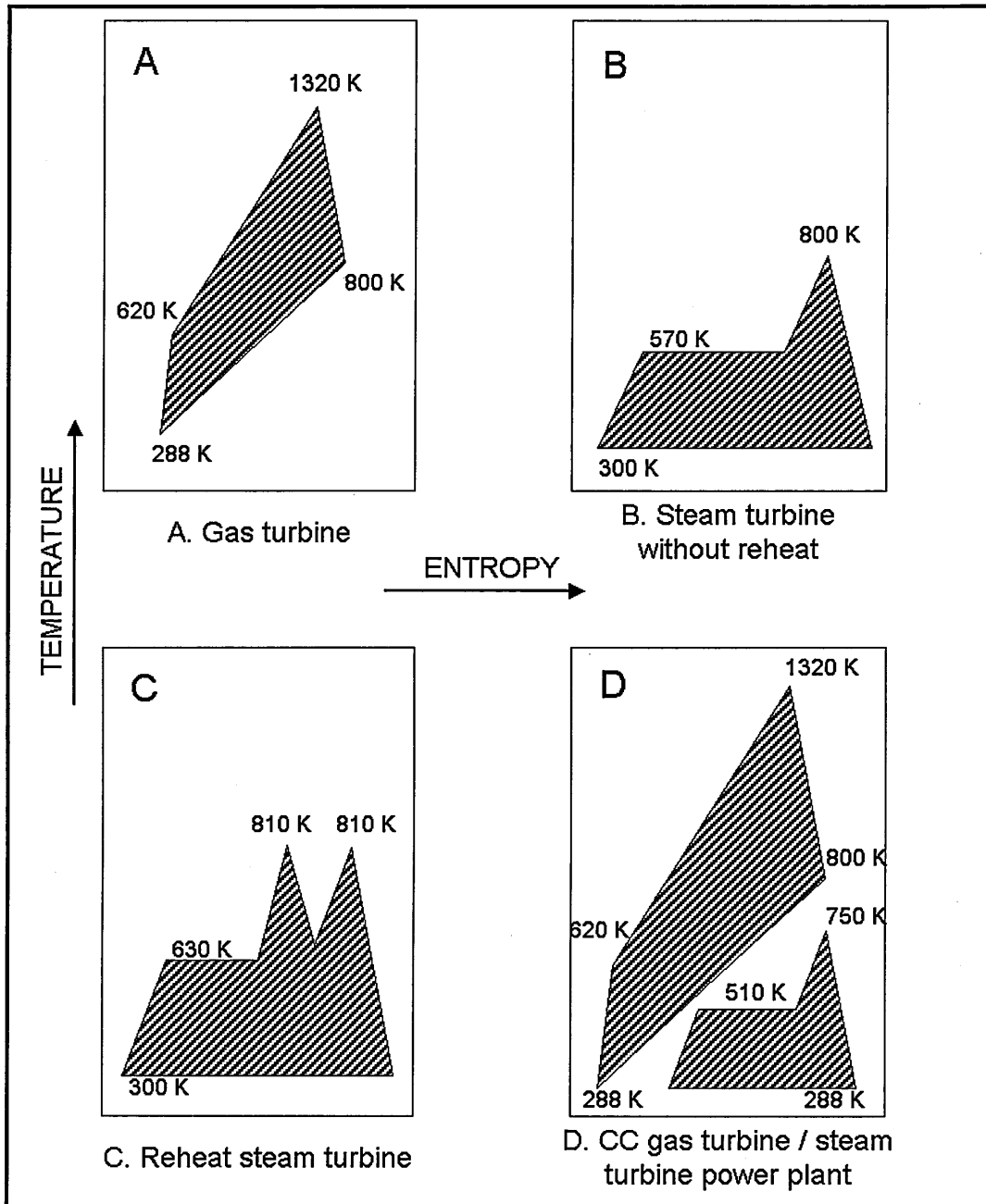
APPENDICES

APPENDIX A

Largest Energy Users and populations by country, 1996			
Largest energy users (MTOE)		Largest populations (millions)	
1 - US	2,130	1 - China	1,170
2 - China	870	2 - India	910
3 - Russia	610	3 - US	260
4 - Japan	500	4 - Indonesia	190
5 - Germany	350	5 - Russia	150
6 - India	250	6 - Brazil	150
7 - France	240	7 - Japan	130
8 - UK	230	8 - Pakistan	120

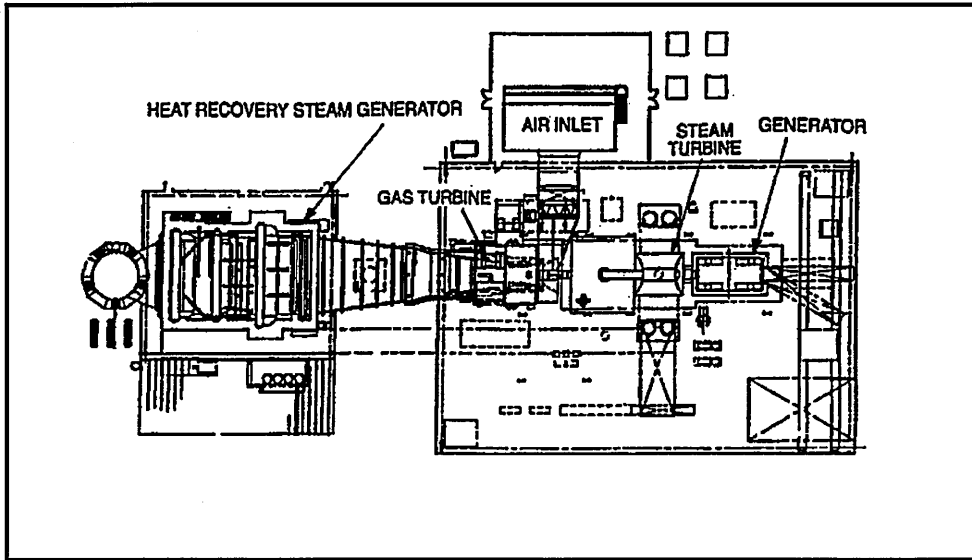
The Fastest Growing Energy Users (minimum consumption 50MTOE per annum)			
Country	1986 consumption	1996 consumption	Per cent increase
Thailand	16.1	56.3	250
South Korea	59.4	164.7	177
Indonesia	37.5	78.2	109
Taiwan	36.5	69.2	90
India	138.0	246.7	79
Iran	57.5	93.4	62
China	566.0	874.0	54
Turkey	39.3	60.1	53

APPENDIX B

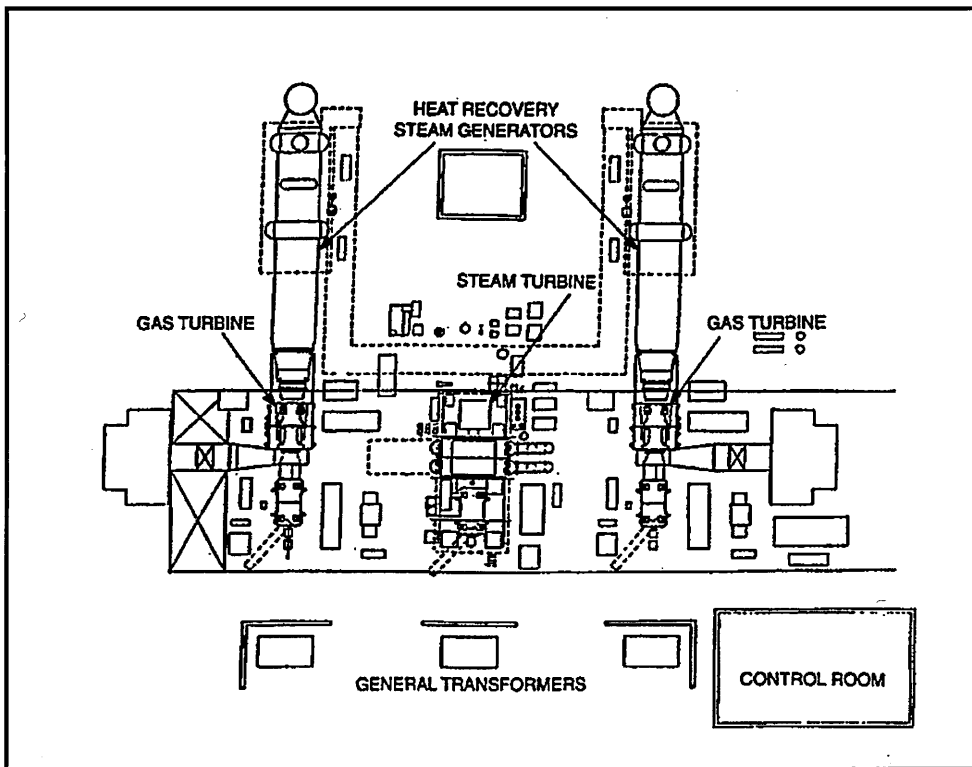


Gas/Steam Cycles

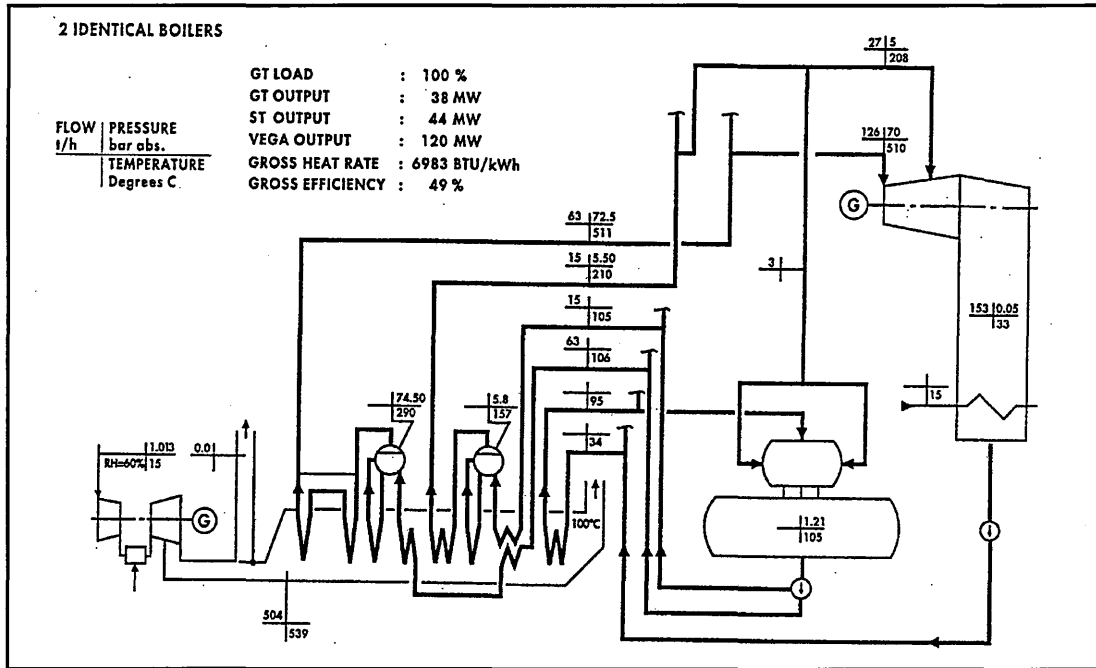
APPENDIX C



STAG 107FA Equipment Arrangement Plan



STAG 207FA Multi-shaft Combined-Cycle Equipment Arrangement Plan



Natural Gas Fired VEGA 206-2P Combined Cycle

APPENDIX D

```

Turbomatch simulation////
This is the GE MS9001FA. Very simple model.
Designer: Gayraud Stéphane (MPhil., 1997-12-09)
This file can run off-design calculations in loop with TET and IGV
angle as variables///
OD SI KE VA FP
-1
-1
INTAKE S1,2 D1,2,3,4 R300
COMPRES S2,3 D5,6,7,8,9,10,11 R301 V5 V6
PREMAS S3,4,9 D12,13,14,15
BURNER S4,5 D16,17,18 R302
MIXEES S5,9,6
TURBIN S6,7 D19,20,21,22,23,24,25,26,301,0 V20
NOZCON S7,8,1 D27 R303
ARITHY D412-418
ARITHY D700-706
PLOTBD D11,708,419 !Output File Format (For003.dat)
PLOTSV D600-607 ! Output File Format
PERFOR S1,0,0 D19,28,29,30,303,300,302,0,0,0,0,0
CODEND
Turbomatch simulation////
1 0.000000
2 0.000000
3 0.000000
4 0.980000
5 -1.000000
6 -1.000000
7 15.000000
8 0.850000
9 0.000000
10 2.000000
11 0 ! Design VIVG angle(=0°)
12 0.905000
13 0.000000
14 1.000000
15 0.000000
16 0.050000
17 1.000000
18 -1.000000
19 22700000.0
20 -1.000000
21 -1.000000
22 0.904000
23 -1.000000
24 1.000000
25 1.000000
26 1000.0
27 -1.000000
19 227000000.0
28 -1.000000
29 0.000000
30 0.000000
429 1000000
412 4
413 -1
414 419
415 -1
416 19
417 -1
418 429
707 2.2075
700 4

```

```
701 -1
702 708
703 -1
704 302
705 -1
706 707
600 8
601 6
602 8
603 2
604 8
605 4
606 5
607 6
-1
1 2 602.0
5 6 1561.0 ! Design TET
-1
11 10      ! VIGV angle fixed for the loop
-1
5 6 1555   ! TET loop
-1
-1
5 6 1550
-1
-1
5 6 1545
-1
-1
5 6 1540
-1
-3
-3
```

APPENDIX E

Fuel properties

Extracted from: Chemical & Process - Thermodynamics - 2nd edition, Prentice Hall, New Jersey, 07632 by B.G. Kyle

$$C_p [\text{cal/g.mol.K}] = a_0 + b_0 * T + c_0 * T^2 + d_0 * T^3 \quad 273 \text{ K} < T < 1500 \text{ K}$$

	a_0	$b_0 * 10^2$	$c_0 * 10^5$	$d_0 * 10^9$	Error (%) max/average
CH ₄	4.75	1.2	0.303	-2.63	1.33 / 0.57
C ₂ H ₆	1.648	4.124	-1.53	1.74	0.83 / 0.28
C ₃ H ₈	-0.966	7.279	-3.755	7.58	0.4 / 0.12

Specific heat capacity of the exhaust gases:

$$C_p(\text{exhaust gases}) = K_1 * [\text{O}_2] + K_2 * [\text{CO}_2] + K_3 * [\text{N}_2] + K_4 * [\text{H}_2\text{O}]$$

With:

$$K_1 = [0.035721771 + 0.060662382 * (\ln T) + 0.039804195 * (\ln T)^2 - 0.12159203 * (\ln T)^3 + 0.023750345 * (\ln T)^4 - 0.0012455555 * (\ln T)^5] / [1 - 0.10400777 * (\ln T) - 0.13520367 * (\ln T)^2 - 0.047263758 * (\ln T)^3 + 0.019843886 * (\ln T)^4 - 0.0021709747 * (\ln T)^5]$$

$$K_2 = (16.362761 - 0.017081933 * T + 1.019409 * T^{0.5} * (\ln T) + 1.6176342 * (\ln T)^2 - 9.4483842 * T^{0.5})$$

$$K_3 = [0.062654781 + 0.11563222 * (\ln T) + 0.12711384 * (\ln T)^2 - 0.028025904 * (\ln T)^3 + 0.0015019508 * (\ln T)^4] / [1 - 0.12498161 * (\ln T) - 0.11469881 * (\ln T)^2 + 0.089781528 * (\ln T)^3 - 0.018393236 * (\ln T)^4 + 0.0013868857 * (\ln T)^5]$$

$$K_4 = (0.74259637 - 0.0021761709 * T + 0.000011255032 * T^2 - 0.0000014064974 * T^2 * (\ln T) + 262.78101 / T^2)$$

(T in °C)

Saturated liquid

(and subcooled liquid at the same temperature with the ideal liquid assumption)

$$h(T) = 2.788e-19*T^9 - 3.987e-16*T^8 + 2.39894e-13*T^7 - 7.857e-11*T^6 + 1.521311e-8*T^5 - 1.76274e-6*T^4 + 1.208711e-4*T^3 - 4.45397e-3*T^2 + 4.25348*T$$

domain: 0°C < T < 370°C - average rel. error: 0.6%

$$s(T) = 8.73065e-13*T^5 - 7.078056e-10*T^4 + 2.341782e-7*T^3 - 4.794256e-5*T^2 + 1.619232e-2*T - 8.387074e-3$$

domain: 0°C < T < 370°C - average rel. error: 0.2%

Saturated steam

$$h(T) = -7.35167e-6*T^3 - 2.33298e-3*T^2 + 2.43725*T + 2491.695 + 6349.4/(T-387.449)$$

domain: 0°C < T < 370°C - average rel. error: 0.2%

$$s(T) = -1.18467e-7*T^3 - 7.9544e-5*T^2 - 0.024623*T + 9.13$$

domain: 0°C < T < 370°C - average rel. error: 0.4%

Saturation relations

$$P(T) = \exp(-5.09132 + 0.0718934*T - 2.72751e-4*T^2 + 7.65835e-7*T^3 - 1.29359e-9*T^4 + 9.5642e-13*T^4)$$

domain: 0°C < T < 370°C - average rel. error: 0.2%

$$T(P) = 0.019523*\ln(P)^4 + 0.2438*\ln(P)^3 + 2.388*\ln(P)^2 + 27.834*\ln(P) + 99.69$$

domain: 0.01b < P < 220b - average rel. error: 0.2%

Superheated steam

$$h(T,P) = 3.04331e-4*T^2 + 1.81867*T + 2503.63 - 21492.63*W*(1.93115e-2/V^3 + W^2/V^{14.7866}*(1.35956e-2 + 4.06747e-3*W^2))$$

with $V = (T + 276.158)/647.719$ and $W = P/219.345$

domain: $0.01\text{b} < P < 220\text{b}$

saturation $< T < 800^\circ\text{C}$ - average rel. error: 0.1%

$$s(T,P) = 0.907643*T''^3 - 3.64635*T''^2 + 6.57334*T'' + 2.13856 - 0.461853*(\ln(P'')) + P''/T''^4 + P''^7$$

with $P'' = P/219.1936$ and $T'' = (T + 271.8659)/645.9763$

domain: $0.01\text{b} < P < 220\text{b}$

saturation $< T < 700^\circ\text{C}$ - average rel. error: 0.2%

APPENDIX F

HEI Condenser Calculation Inputs

Tube Outer Diameter: This is used in the design case for calculating the correction factor C1. The correction factor is calculated using the following equation (in English units - diameter in inches, U in Btu/hr--ft²).

$$C1 = -15.6643 * O.D. + 277.56$$

This is a linearised version of the values found in “Standards for Steam Surface Condensers”.

Velocity Inside Tube: This is used in the design case for calculating the overall heat transfer coefficient U using the equation:

$$U = C1 * C2 * C3 * C4 * \sqrt{V}$$

Note that for using the HEI equation in design mode, you will have already specified all the correction factors in the equation. Normally, the design velocity is above 5 ft/sec for good heat transfer, and below 8 to minimize erosion. It is usually in the range of 7-8 ft/s.

Correction Factor for Tube Material and Gauge: This factor C3 is used only if using the HEI method in design mode.

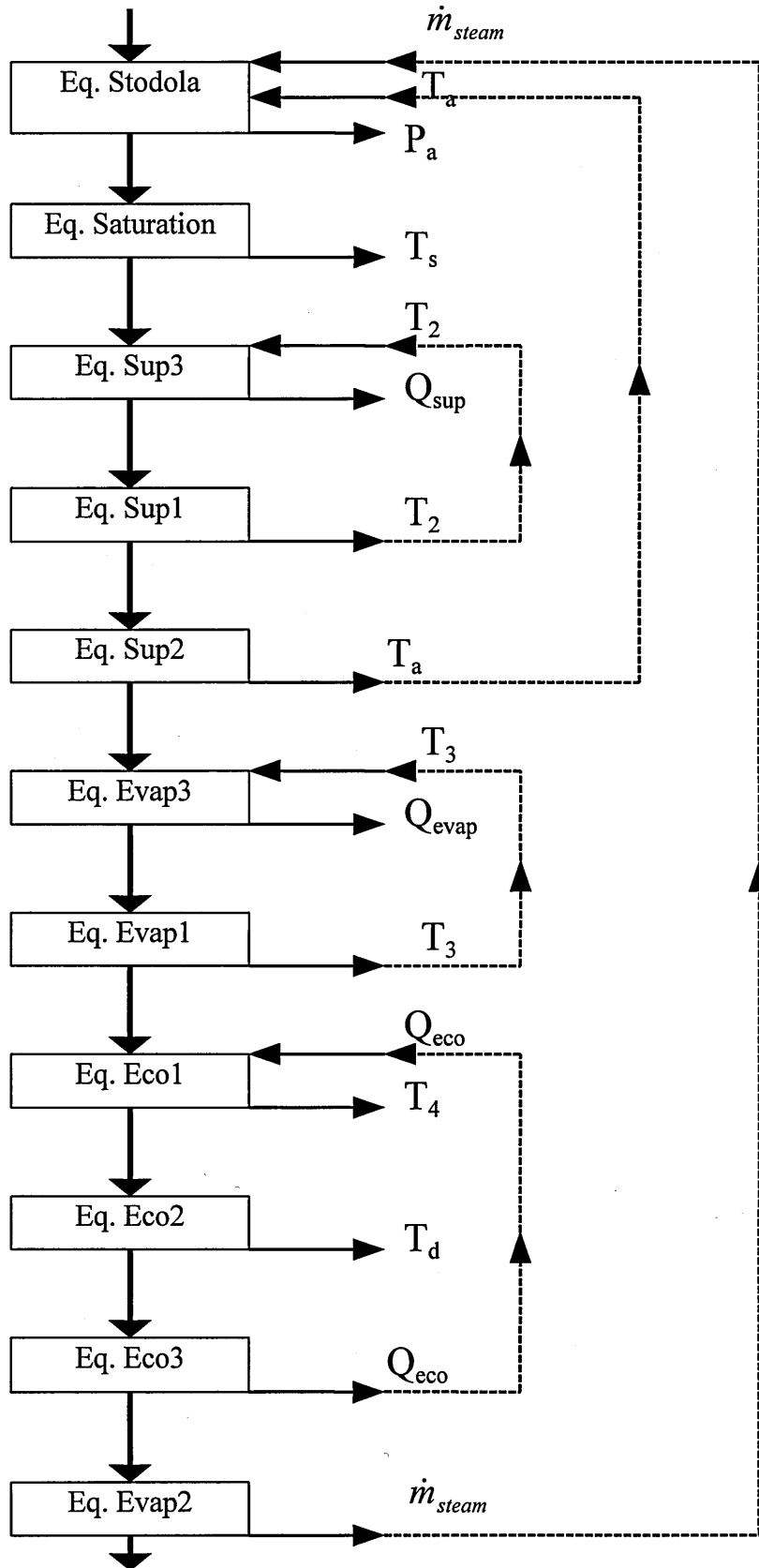
Tube materials	Tube Wall Gauge - BWG						
	24	22	20	18	16	14	12
Admiralty Metal	1.06	1.04	1.02	1.00	0.96	0.92	0.87
Arsenical Copper	1.06	1.04	1.02	1.00	0.96	0.92	0.87
Copper Iron 194	1.06	1.04	1.02	1.00	0.96	0.92	0.87
Aluminium Brass	1.03	1.02	1.00	0.97	0.94	0.90	0.84
Aluminium Bronze	1.03	1.02	1.00	0.97	0.94	0.90	0.84
90-10 Cu-Ni	0.99	0.97	0.94	0.90	0.85	0.80	0.74
70-30 Cu-Ni	0.93	0.90	0.87	0.82	0.77	0.71	0.64
Cold-Rolled Low Carbon Steel	1.00	0.98	0.95	0.91	0.86	0.80	0.74
Stainless Steels Type 304/316	0.91	0.87	0.83	0.76	0.70	0.63	0.55
Titanium	0.91	0.87	0.83	0.76	0.70	0.63	0.55

Cleanliness Factor: This factor C4 is used by the HEI calculations both in design and off-design modes. It simulates the change in the heat-transfer of the condenser with usage. Brand new condensers have a cleanliness factor of 1, but quickly develop film on the tubes that changes the heat-transfer mechanism. Usually manufacturers quote performance using a cleanliness factor of 0.85 or less.

Cooling Flow Water Temperature Correction Factor: This factor C2 is calculated by the HEI calculations both in design and off-design modes. It is calculated using spline fits of the following table:

Water Temperature (F)	30	40	50	60	65	70	75	80	90	100	110	120
Correction Factor C2	0.65	0.743	0.835	0.922	0.967	1.00	1.026	1.045	1.073	1.10	1.121	1.14

APPENDIX G



APPENDIX H

Elapsed Operating Time - Hours	Engine Components	Maintenance Work
-	Compressor blading	Wash on-line and soak wash at cranking speeds at intervals determined by condition monitoring
1,000	Combustion chambers plus compressor & turbine blading	Visually inspect using borescope
2,000 and every	Lubricating oil system	Send oil sample for analysis and check level
2,000 thereafter	Intake air systems	Check intake air filters
7,000	Combustion chambers plus compressor and turbine blades	Visually inspect using borescope
8,000	Combustion chambers	Dismantle and inspect
	Controls	Check correct operation of all control and safety systems
	Starter, auxiliary gearbox, turbine, main gearbox and alternator	Check alignments
	All ancillary equipment	Visual inspect
15,000	Lubricating oil system As 7,000 hours	Change intake air filters as necessary
16,000	As 8,000 hours	
23,000	As 7,000 hours	
24,000	As 8,000 hours plus	
	Compressor section	Dismantle and inspect all components and check clearances
	Turbine blades	Remove sample blades for off-site condition assessment
	Auxiliary gearbox	Dismantle and inspect all components and check clearances
31,000	As 7,000 hours	
32,000	As 8,000 hours	
39,000	As 7,000 hours	
40,000	As 24,000 hours plus	
	Turbine section	Dismantle and inspect all components and check clearances
	Reduction gearbox	Dismantle and inspect all the component and check clearances
	General	Replace ancillary equipment as necessary

Indicative maintenance requirements - GT operating on natural gas