Klein Woud

Rolls-Royce

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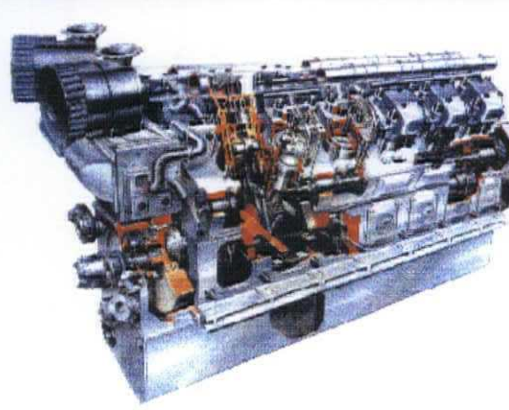

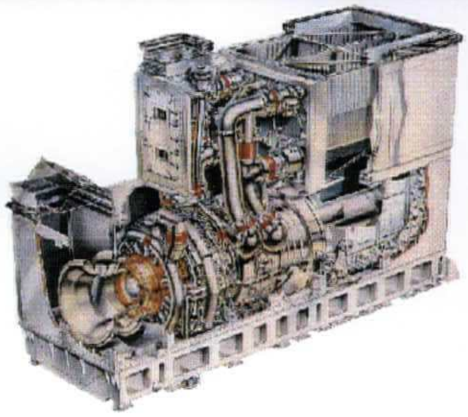

R Kok, Delft University of Technology, December 1998

Report No: OvS 98/04

A comparison of advanced cycle gas turbines with medium speed diesels in cruise ship applications

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Study conducted at Rolls-Royce plc,
Marine Power in
Ansty, Coventry, UK

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Rolls-Royce

Acknowledgements

Dear reader,

With a great deal of pleasure I look back upon the year I spent in England, the period during which I conducted my graduate project at Rolls-Royce Propulsion Power in Ansty.

Before you take notice of this report I would like to thank the people, without whom this report could never have been established.

I thank Roger Tooke for giving me the opportunity to conduct my graduate project at Rolls-Royce. It has been an extremely instructive year for me, with respect to the knowledge I have gained on both the subject of this report and my English. Furthermore I would like to express my thanks for your guidance and support, when you were in the office.

Gordon Price, thank you for providing me with answers to most diverse subjects. Do not despair, there will be other students to bother you with annoying questions.
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Furthermore the people of outside Rolls-Royce who provided me with the information I required for this report: Simon Yeo and Alex Brown of P&O Cruises, Hans Klein Woud, Hugo Grimmelius, Jakob Pinkster and Dave Boëtius of DUT, Esa Ylinen of Kvaerner Masa Yards, Douwe Stapersma of the Royal Netherlands Naval College/DUT, Lars Nordin of Princess Cruises, Markku Kanerva of Deltamarin, Norman Pearce of Alfa Laval Desalt, Paolo Daniotti of Marinco, Hans van Herwerden of Wärtsilä NSD Nederland, Jeff Buckley of Cegelec and Mr. Miller of the Carnival Corporation.

Ansty, 10 December 1998

Rutger Kok

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Summary

This year for the first time a cruise line has opted for a (combined) cycle gas turbine plant as opposed to a traditional diesel plant for its newbuildings. Among the generic advantages offered by gas turbines are high power density, low emissions, low noise and vibration levels, low lubricating (lub) oil consumption and reduced engine room manning.
For the switch from diesels to gas turbines to be commercially beneficial, these advantages should offset the less favourable initial cost and higher fuel and maintenance costs. A study previously conducted has demonstrated that technically there are no disadvantages involved in applying gas turbines as prime movers in cruise ships.

This report describes a commercial, technical and environmental comparison of a proposed advanced cycle gas turbine-electric power system with the existing diesel-electric power systems in large cruise ship applications.

The recognised trends in the cruise liner industry include an increase in fleet and vessel size, as well as maximum speed. The order book comprises vessels up to 138,000 gross tonnes and maximum ship speed up to 26 knots. Nowadays more attention is paid to environmental issues such as global warming and air pollution. Emission regulations such as the IMO NO_X emission limits, will become more restrictive. Starting from the late 80's, cruise lines made the switch from diesel-mechanical to diesel-electric, mainly because of the improved plant flexibility.

The minimum vessel gross tonnage suitable for a gas turbine fit <u>in this case</u>, is 50,000 tonnes. The associated power demand is 30 MW: to be delivered by one WR-21 and one 601-R. Three modern vessels tonnes were 'selected' for the comparison: 77,000 tonnes *Sun Princess*, 109,000 tonnes *Grand Princess* and 75,000 tonnes *Capricorn*. Subsequently, these ships are described in more detail. Subjects illustrated include the power-speed curves, the electric load and heat load, the diesel generating sets (gensets) and auxiliaries and finally the steam related systems.

Three different cruise itineraries have been considered: a 7-day Caribbean cruise, a 7-day Alaska cruise and a 92-day World cruise. The itinerary descriptions are converted to ship operating and speed profiles. The adopted ambient temperatures are 30°C, 10°C and a combination of 15°C and 25° C, respectively. The relative humidity for every itinerary is 90%.

The Rolls-Royce history and the current organisation is briefly discussed. Subsequently the two advanced cycle gas turbines in this study are introduced: the InterCooled and Recuperated (ICR) WR-21 and the recuperated only 601-R. The output power under ISO (International Standards Organisation) no loss conditions is 25 MW and 6.5 MW, respectively. Also, the WR-21 ICR cycle is illustrated and maximum installed power output is investigated. A preliminary 601-R genset arrangement including a spiral recuperator is presented in the last section.

Next , a number of suitable combinations of prime movers are generated. For each vessel this results in three possibilities: the electric power is delivered by either WR-21 gas turbines only, a combination of WR-21 and 601-R gas turbines, or 601-R gas turbines only. This yields a total of 11 gas turbine configurations, which this results in a total number of 14 configurations which will be used for the comparison.

The main body of the report comprises the comparisons of the diesel with gas turbine power plant configurations with regard to the following ten issues.

Electric load and heat load. Due to the lack of diesel auxiliaries such as combustion air fans and cooling water pumps, the electric load is reduced by typically 2%. The domestic steam or heat load decrease of 20-30% is caused by the lack of fuel preheating systems. With gas turbines, there is plenty of exhaust gas waste heat available, especially at high engine load. As a result the oil fired boiler capacity required to generate steam, can be reduced significantly.

Initial cost. Because the gas turbine acquisition cost is based on a total package including lub oil, cooling and control systems these auxiliary costs were added to the diesel genset cost. As a result, the specific diesel genset acquisition cost amounts to $ 360/kW. The gas turbine specific acquisition cost, at ISO no loss conditions, is $ 400/kW (WR-21) and $ 575/kW (601-R), respectively. The installation cost for gas turbines on the other hand is lower due to a more simple auxiliary layout. The extra investment cost varies from US $ 1 to US $ 12 million.

Fuel costs. Although the overall fuel consumption does not differ a great deal, the difference in annual fuel cost is significant. This is caused by the fact that diesels can burn heavy fuel, typically costing $ 80 per tonne while gas turbines require more expensive distillates with a bunker price varying from $150 to $ 200 per tonne. As a result the annual fuel cost increase varies from US $ 4 to US $ 6 million.

Weight and dimensions. With the high power density of gas turbines, the weight and space savings with a gas turbine plant are significant. For example a 60 MW plant can result in 1,500 tonnes weight reduction. The overall machinery space can be reduced by as much as 4,500 m^3 on *Grand Princess*. This potentially creates more than 10 million dollars of added revenues per year. Clearly this requires a complete redesign of the machinery spaces and the ships superstructure.

Emissions. Nowadays great attention is paid to ships emissions when selecting the machinery concept. The main concern regarding pollution is for NO_X, SO_X and lately also for CO_2. The CO_2 emission for gas turbines is higher because it is proportional to fuel consumption. The NO_X and SO_X emissions on the other hand are significantly lower: the annual NO_X emission can be reduced by typically 70% and the SO_X emission even by 90%. This reduction in SO_X emission is caused by the fact that medium speed diesels typically burn heavy fuel which has a higher sulphur content. Several emission regulations are expected to come into force in the near future. The section is concluded with a brief survey on emission abatement costs and a summary of the several types of noise emissions from prime movers.

Lubricating oil costs. Gas turbines only require lub oil for lubricating and cooling the various bearings. With diesels on the other hand, a significant amount of lub oil is burnt in the combustion process. Specific lub oil consumption of a modern diesel in practice is about 0.9 g/kWh, whereas the gas turbines consume 0.004 and 0.015 g/kWh, respectively. Gas turbines however require more expensive synthetic oils as opposed to mineral oils used in diesel engines. The annual cost savings vary from US $250 to US $ 425 thousand, depending again on the configuration in question.

Maintenance and spares. This section commences with a description of the WR-21 maintenance schedule and typical diesel component lifetimes. A survey of diesel and gas turbine maintenance costs has resulted in annual maintenance cost increase between from US $ 0.6 to US $ 1.5 million for the gas turbine plants. Due to a modular gas turbine design, the number of spares though can de reduced.

Engine room personnel. A list of present engine room personnel under usual duty conditions on *Grand Princess* and *Capricorn* is presented. Reduced manning levels are facilitated by full automation of gas turbines. The predicted savings on personnel costs however are relatively low: 68 thousand dollars on *Sun Princess* and *Capricorn* and 154 thousand dollars on *Grand Princess*.

Life Cycle costs. In this section the extra investment cost and annual operating costs are summarised. An investment appraisal is conducted by means of the net present value method (interest rate 6%) and the pay-back period method. The investment cost for each gas turbine configuration is higher. Net positive cash flows result in a pay back period being less than four years for the majority of the configurations. A summary of the economic benefits with advanced cycle gas turbines for a 30-year life cycle as derived in this section are presented below. For each ship the 'WR-21 only' configuration is displayed. The effect of a change of interest rate is alos investigated in the report.

'WR-21 only'	***Sun Princess***	***Grand Princess***	***Capricorn***
Initial cost	**+ $ 2.1m**	**+ $ 3.4m**	**+ $ 7.0m**
Operating costs	**+ $ 118m**	**+$ 148m**	**+$ 107m**
Added revenue	**+ $ 205m**	**+ $ 310m**	**+ $ 195m**
Total differential	**+ $ 85 m**	**+ $ 158m**	**+ $ 81m**
Pay-back period	**0.8 years**	**0.7 years**	**2.4 years**
Net present value	**$ 37.7m**	**$ 70.8m**	**$ 33.6m**

In conclusion, the effect of a number of variations in operating costs and first cost on the net present value of the investment is investigated. As was to be expected, the most critical element in the calculation is the extra revenue created by the additional cabins. The effect of a change in initial cost is relatively small.

Availability, Reliability and Maintainability. As not enough information is available on these statistical quantities, the effect of prime mover failure and compartment flooding on the electric power availability is investigated in this section. Apart from two cases with a rather unfavourable prime mover layout, even in case of compartment flooding the maximum attainable ship speed will be over 14 knots.

Furthermore number of possibilities to enhance the overall power plant performance were investigated. The waste heat available from the WR-21 intercooler for example can be used as heating medium for among others fresh water generation. Furthermore alternative fresh water makers which only require electric energy are briefly investigated. An advantage in applying these desalination plants is that the steam load is reduced significantly. Possibly the vessel will no longer need steam assistance from the maintenance intensive oil fired boilers if WR-21 engines are used.

The penultimate chapter deals with the combined heat and power plant as ordered for a number of cruise ships: General Electric's COGES system. By comparing this power plant with a similar sized WR-21 plant, it is found that the performance and first cost of the systems are similar, but the emissions and maintenance costs for a COGES plant are slightly more favourable. One of the objectives of the COGES system is the high pressure steam system required to feed the steam to a steam turbine driving a generator. This is thought to be a vulnerable and therefore a maintenance intensive system.

In conclusion can be said that especially with 'WR-21 only' configurations, significant cost benefits are achievable. A good alternative to increase the prime mover redundancy is offered by a power plant comprising both WR-21 and 601-R gas turbines. However, the operating costs involved will increase and the potential cabin revenues will decrease. This clearly results in a reduction of profits that can be made with a switch from diesels to a gas turbines.

It has to be emphasised that the promising figures as shown above depend highly on the extra cabin revenues. Consequently a careful redesign of the ship and possibly a ticket sales analysis should be conducted to arrive at a accurate life cycle cost comparison.

Glossary

Abbreviations

AC	Alternating Current
ARM	Availability, Reliability and Maintainability
CA	Capricorn
CF	Cash Flow
COGES	COmbined Gas and Steam turbine integrated Electric drive System
Cyl.	Cylinder(s)
DLE	Dry Low Emissions
DWI	Direct Water Injection
FO	Fuel Oil
FW	Fresh Water
g	Gauge
Genset	Generating set
GOSTA	Global Ocean Surface Temperature Atlas
GP	Grand Princess
GRT	GRoss Tonnage
HC	HydroCarbon
HFO	Heavy Fuel Oil
HP	High Pressure
HVAC	Heating Ventilation and Air Conditioning
HVAC	Heating, Ventilation and Air Conditioning
I	Investment
ICR	InterCooled Recuperated
IMO	International Maritime Organisation
ir	Interest Rate
ISO	International Standards Organisation
LCV	Lower Calorific Value
LN	Low NO_X
LO	Lub Oil
LOM	Lub Oil Module
Lub oil	Lubricating oil
MCR	Maximum Continuous Rating
MDO	Marine Diesel Oil
MGO	Marine Gas Oil
MSF	Multi Stage Flash
MTBR	Mean Time Between Repairs
NO_X	Carbon monoxide and dioxide
NPV	Net Present Value

PS	Pound Sterling
PV	Present Value
RO	Reverse Osmosis
SCR	Selective Catalytic Reduction
SECA	SO_X Emission Control Area
sfc	Specific Fuel Consumption
SO_X	Sulphur monoxide and dioxide
SP	Sun Princess
SSF	Single Stage Flash
UPS	Uninterruptable Power Supply
VAN	Variable Area Nozzles
vppm	Volume Parts Per Million
VVC	Vacuum Vapour Compression
WHB	Waste Heat Boiler
WI	Water Injection

Symbols

$\propto$	Constant	kg/m
ρ	Exhaust gas density	kg/m^3
η_P	Propulsion Efficiency	-
$\eta_{Thermal}$	Thermal efficiency	-
$c(v)$	Polynomial (function of ship speed)	-
$C_{P,water}$	Specific heat value	kJ/kgK
k_1	Constant (gross tonnage)	-
n	Certain period	years
P_D	Delivered shaft power	kW
P_E, $P_{E,Max}$	(Maximum) Effective propeller power	kW
$P_{Installed}$	Total installed power	kW
$P_{Propulsion}$	Propulsion power	kW
q	Specific energy	kJ/kg
r	Evaporation heat	kJ/kg
R_{Ship}	Ship resistance	N
$T_{Feedwater}$	Feedwater temperature	K
T_{Steam}	Steam temperature	K
V	Volume of all the enclosed ships spaces	m^3
v, v_{ship}, v_{Max}	(Maximum) Ship speed	m/s

1. Introduction

In the marine world, gas turbines have only been widely used in the naval sector. The cruise market has moved from diesel-mechanical propulsion to a predominance of diesel-electric propulsion and power. Medium speed diesel engines dominate this market due to better fuel economy, lower maintenance costs and more favourable initial cost over the gas turbines. Earlier this year (1998) however, Royal Caribbean Cruises ordered up to six General Electric gas turbine/steam turbine sets for its newbuildings. Another notable change is the switch from 'traditional' propulsion electric motors inside the hull to so called 'Azipods'. These propulsors are mounted underneath the ship. This results in machinery deck space savings and propulsion efficiency increase.

Gas turbines however do possess a number of advantages over diesels. Additional cabin revenues due to high power density and low lub oil consumption and engine room personnel costs should compensate for the increase in first cost, fuel costs and maintenance costs. Low noise and vibrations levels result in increased passenger comfort. Furthermore, the impact of cruise ships on the environment is reduced in consequence of the low emissions. The 'green ship' aspect will become increasingly important since nowadays more and more attention is paid to environmental issues. A complete overview of the advantages and disadvantages is presented in Appendix 1.

A feasibility study[1] has demonstrated that, from a technical point of view, advanced cycle gas turbines are suitable prime movers for large cruise ships. However, the implications on life cycle costs have not been investigated yet. A detailed comparison of diesel-electric and gas turbine-electric power systems will give Rolls-Royce information on the future possibilities of advanced cycle gas turbines for cruise ships.

In this report a commercial, technical and environmental comparison of existing diesel-electric and proposed advanced cycle gas turbine-electric power systems for large cruise ship applications is made.

The study involves the intercooled recuperated Northrop Grumman/Rolls-Royce WR-21, rated at 25 MW and the 6.5 MW recuperated Rolls-Royce/Allison 601-R. The smaller gas turbine is involved because it was recognised by Rolls-Royce that by using 25 MW engines only, it was difficult to effectively match the electric power demand, and redundancy as required by cruise operators.

[1] [van Lier, 1997]

A limitation when conducting this study was the availability of information. For instance: although it is recognised that the advanced cycle concept is feasible for ships over 50,000 gross tonnes, the smallest vessel used in this study is 75,000 tonnes. Furthermore it was difficult to obtain accurate information on costs, because of the sensitive nature of this information. Despite this and thanks to the co-operation and assistance given by cruise lines, manufacturers and ship yards it was possible to produce a reliable and objective report.

The structure of the report is as follows. In chapter 2 current trends in the cruise liner market and the main features of modern cruise ships are discussed. Subsequently three suitable cruise ships are selected for the comparisons. These vessels are illustrated in chapter 3. Chapter 4 deals with a set of three existing cruise itineraries and the associated ambient conditions. Next, the two advanced cycle gas turbines are illustrated in chapter 5. Based on the findings in the previous chapters, the possible gas turbine configurations are shown in chapter 6.

The main comparisons are dealt with in chapter 7. In this chapter, comparisons are made of initial cost, fuel and lub oil costs, weight and dimensions, maintenance and spares, emissions, and engine room personnel. The chapter concludes with a life cycle cost comparison.

Chapter 8 investigates the possibilities of advanced cycle gas turbines in more detail from a 'total energy' point of view. Subsequently a brief comparison is made of General Electric's COGES plant and WR-21 plant. The report concludes with conclusions and recommendations in chapter 10.

2. Cruise ships

Before a technical, commercial and environmental comparison can be made the cruise liner industry will be investigated. Data was obtained from cruise trade magazines, but they did not contain sufficient information. Additional information was received from P&O Cruises and Carnival Cruises.

As found in cruise trade magazines, the current state of the cruise business can be described as follows. The economic growth is expected to be high in Europe, the Far East and America. This will create a good basis for the cruise business and ship design development. The cruise business has been growing at an average annual rate of 7.6 % since 1980. Consequently it is expected that the whole cruise market will expand further and big strong operators will continue setting the design trends. The key equation on their business forecast is that growth equals survival.

For this reason the ships and fleets are getting larger. Vessels with a gross tonnage (GRT) over 100,000 tonnes or so called Post-Panamax ships will become the new standard. Economics of scale explain recent orders of Post-Panamax vessels. The first vessel in the category 'too wide for the Panama Canal', Carnival's 101,000 GRT *Carnival Destiny*, has been in service since 1996. Her sister ships *Carnival Triumph* and *Carnival Victory* are scheduled for delivery in 1999 and 2000, respectively. P&O's 109,000 tonnes *Grand Princess* made her maiden voyage in May this year and Royal Caribbean Cruises have placed an order for three 138,000 tonnes ships: "Project Eagle". These vessels will be built by Kværner Masa Yards at the Turku New Shipyard in Finland.

The maximum speed has also increased significantly over the last decade. New destinations and replacement trips require higher speeds: from 18 - 23 knots a few years ago to 20 -26 knots or more nowadays. This can be achieved neither technically nor economically without drastic changes. For instance an increase of maximum ship speed from 20 to 25 knots, requires an increase of propulsion power by roughly 100%, or a factor 2: namely $(25/20)^3$.

The third order can be explained as follows: the first assumption is that the resistance of a ship (R_{ship}) is proportional to ship speed (v_{ship}) squared:

$$R_{ship} = \alpha * v_{ship}^2 \text{ [N]},$$

with α = constant [kg/m]

The power required to tow the ship (P_E or the effective power) equals resistance times speed:

$$P_E = v_{ship} * R_{ship} \text{ [W]}$$

This yields:

$$P_E = \alpha * v_{ship}^3 \text{ [W]}$$

The shaft power delivered to the propeller (P_D) equals:

$$P_D = \eta_P * P_E \text{ [W]},$$

with η_P = the propulsion efficiency

Although the propulsion efficiency is not constant with varying ship speed it will be considered constant over a certain speed range. This yields the third order relationship between delivered shaft power and ship speed: the cube law.

Furthermore cruise lines tend to pay more attention to ecological requirements. The ships impact on the environment has become an important issue nowadays with regard to subjects such as air pollution and global warming. At the moment no global regulations on exhaust gas emissions are in force. There are however definite intentions rather than proposals for the limitation of the oxides of nitrogen and sulphur; some regionally based and some of an international character. The text of the diesel engine NO_X emission controls of Annex VI to MARPOL 73/78 will increasingly focus the attention of many sectors of the marine industry. MARPOL 73/78 is the term given to the protocol to the International Convention for the Prevention of Pollution from Ships, which took place in 1973 and was agreed in 1978. The proposed limits will apply to all newbuilt-engines larger than 130 kW which are to be installed on ships that are constructed after 1 January 2000[2].

2.1 The world cruise fleet

As mentioned before, the size of cruise ships has increased significantly over the last decades, as shown in Figure 1. The figure focuses on cruise ships of 1,000 tonnes and above, used for ocean going activities and does not include regular passenger traffic, coastal/river cruising vessels or catamarans[3]. The four names mentioned in the figure are the names of the vessels of exceptional size for the year they were or will be built.

It is expected that further growth will be limited due to port restrictions like draught, air draught under bridges, quay length and terminal capacity. This expected trend is shown in Figure 1. Moreover the existing infrastructure restricts the number of passengers coming onshore. If large vessels were to anchor offshore then smaller vessels would be used to transport passengers and supplies to shore so there would be no restriction on size.

[2] [Verkley, 1998]
[3] [Cruise Review, 1998]

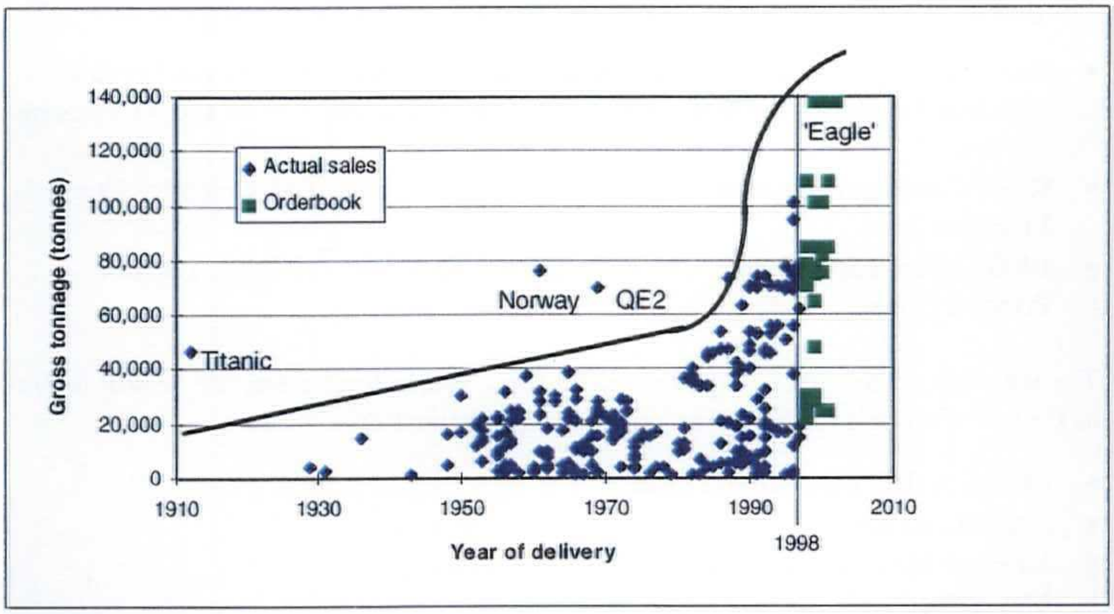

Figure 1: Cruise ship size increase against year of delivery

Since gross tonnage is the quantity most widely used to characterise a cruise ship's dimensions, Figure 1 displays the gross tonnage against the year of delivery. The gross tonnage does not indicate the weight of a ship, but rather the volume and is therefore a measure of capacity.

The 15 major cruise operators[4] which own the majority of the ships shown in Figure 1 are listed Appendix 2. The gross tonnage-market shares of the three largest operators which own a fair amount of different sized cruise ships are displayed in Figure 2. The three cruise lines had a gross tonnage market share, of over 56 %.

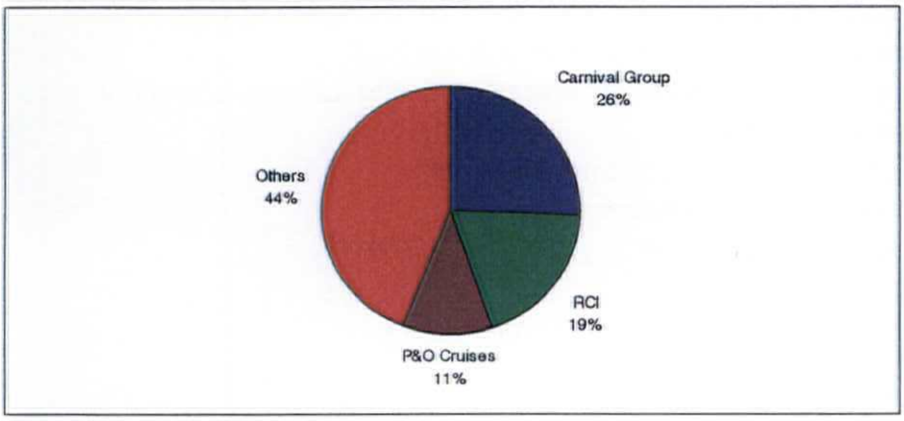

Figure 2: Gross tonnage-market shares

[4] [ISL, 1997]

The parent companies shown in Figure 2 comprise the following subsidiaries:

- Carnival Corporation: Carnival Cruise Lines, Holland America Line-Westours, Windstar Cruises, Seabourn Cruise Line, Costa Crociere (50 %) and Airtours plc (30 %).
- Royal Caribbean International: Royal Caribbean Cruises Ltd, Celebrity Cruise Lines Inc.
- P&O Group: P&O Cruises Ltd, with its subsidiary Swan Hellenic Ltd and Princess Cruises.

The majority of the cruise ships owned by these cruise lines were, are or will be built in Europe. Europe's leading ship yards (in alphabetical order) are:

- Chantiers de l'Atlantique (France)
- Fincantieri (Italy)
- Kværner Masa-Yards (Finland)
- Meyer Werft (Germany)

2.2 Cruise ship selection

In this section a cruise vessel selection regarding gross tonnage and year of delivery will be made. From this selection a number of ships will be selected for the comparison.

Most modern cruise vessels incorporate diesel-electric power systems as opposed to the diesel-mechanical systems as found in older vessels. Operational advantages as illustrated in case studies[5] are:

- a more flexible operation
- more flexible design possibilities; offers freedom in location of main engines
- permits running diesel engines at a more efficient load
- standard equipment and thus easy maintenance due to uniform machinery
- less engine room personnel
- permits running diesel engines at constant speed
- lower noise and vibration on board
- improved safety through redundancy and better reliability

The disadvantages, irrespective of the ship type are:

- initial cost or price
- transmission losses
- weight

[5] see for instance [Stapersma en Wilgenhof, 1997] and [Henriksson, 1998]

Moreover it was demonstrated in [Stapersma, 1997] that in general no fuel savings are can be achieved with adopting a diesel-electric power plant. The emissions for both options are similar and not necessarily lower for diesel-electric plants.

Diesel-mechanical power systems were not replaced by diesel-electrical power systems from the late 1980's. Consequently vessels built after 1988 are of interest for this study.

Secondly the minimum ship size, or gross tonnage has to be determined. This is carried out as follows: from availability and redundancy considerations a minimum of two gensets is required. This can be derived from a rule requirement from Solas, chapter II-1 regulation 41:

"....... *The capacity of these generating sets shall be such that in the event of any generating set being stopped it will still be possible to supply those services necessary to provide normal operation conditions of propulsion and safety.* ...."

Also, within the International Maritime Organisation (IMO) proposals have been discussed as to operational requirements to maintain at least two generators in operation in areas where navigation requires special attention.

One option would be to install 601-R engines only. However, compared to the WR-21 both the specific fuel consumption and the specific acquisition cost are considerably higher, as will be shown later. These disadvantages should be offset by a higher power density (in kW/m^3 installed volume) for instance, resulting in extra cabin space. As illustrated later this is not the case. The only advantage is improved redundancy and flexibility. To demonstrate both acquisition cost and running costs increase the '601-R only'-configurations will still be investigated.

As a result, the 'minimum' engine configuration will comprise one WR-21 and one 601-R which are rated at 24 and 6 MW (25°C ambient temperature, no losses). Thus the cruise vessel should require at least a total installed power of approximately 30 MW. These ratings are only used as a starting point, but as shown later, the maximum power output is largely dependent on the ambient air conditions and pressure losses in the intake and uptake.

The total installed power can be divided between: power required for propulsion purposes and power required for all other services (hull and deck, safety, engine, air conditioning, galley, accommodation and lighting). The latter is referred to as 'hotel load'. For cruise ships the hotel load is relatively high compared to other types of ships. The relationship between total installed power and installed propulsion power is illustrated in Figure 3. The actual values are also shown in Appendix 2 (Table 2).

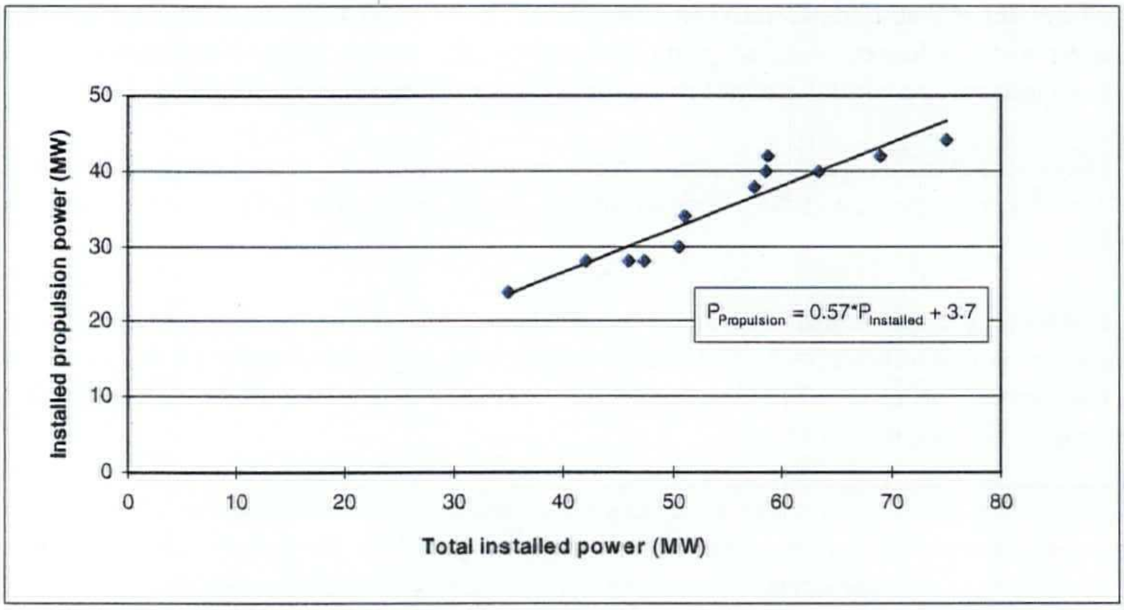

Figure 3: Installed propulsion power versus total installed power

The best linear fit equation for the total installed power-propulsion power relationship is given by:

$$P_{Propulsion} = 0.57 * P_{Installed} + 3.7 \text{ [MW]}$$

So the minimum propulsion power this study focuses on is about 20.8 (0.57*30+3.74) MW which leaves 9.2 MW for the hotel load. It is assumed here that the engine load is 100% at maximum electric load. In practice however, the maximum service factor (maximum electric load divided by the total installed power) varies typically from 85 % to 95 %. The minimum hotel load of interest consequently becomes 8.3 MW , with a service factor of 90 %.

Now a relationship between gross tonnage (GRT) and installed power must be found. Figure 4 shows the data obtained from cruise trade magazines and cruise lines (see also Table 2, Appendix 2).

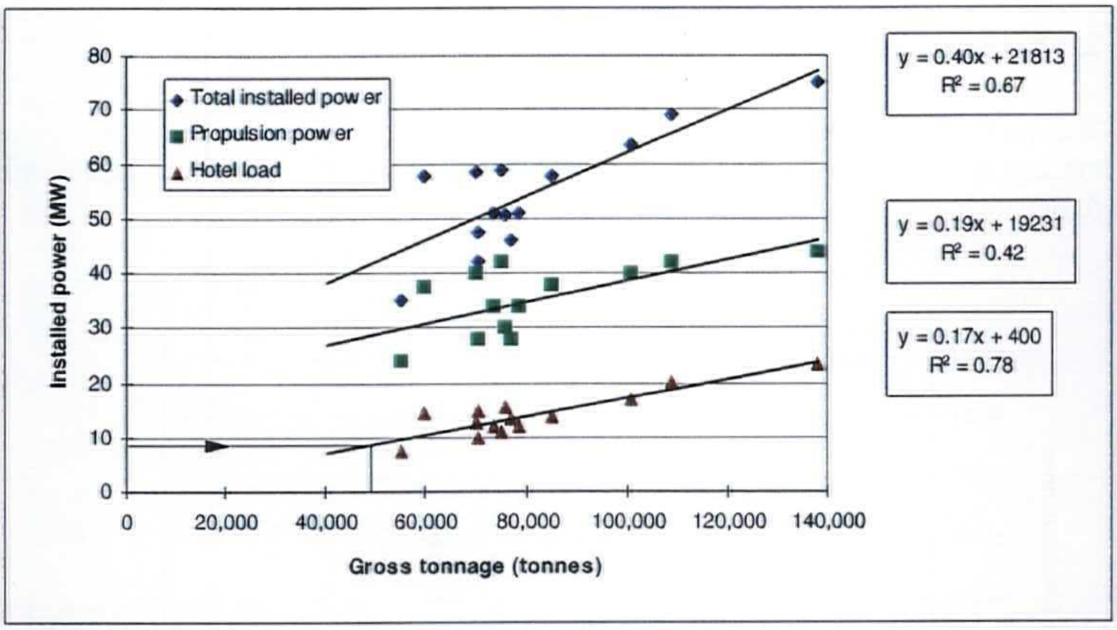

Figure 4: Cruise ship installed power against gross tonnage

As opposed to propulsion power, the hotel load is much less dependent on the ship's maximum speed. Only power required for engine service depends on maximum speed (higher maximum speed ⇒ more propulsion power required ⇒ higher auxiliary system power demand). For a generic cruise ship, the power requirement for engine service is 20 - 25 % of the total hotel load. Therefore the gross tonnage-hotel load relationship should give the most accurate fit line. This is confirmed in Figure 4, where the lines of best fit and their R^2-values are plotted. This R^2-value is a statistical measure for the accuracy of a best fit line[6]. The value for a perfect fit is 1; the (theoretical) minimum is 0 (no correlation). As a result the lower limit of interest is about 50,000 tonnes, as can also be seen in Figure 4.

There is no upper limit to the size of the ship. Gas turbines are available at a higher power output than medium speed diesels and will be smaller at the same power rating. For example: the maximum power output of a four-stroke medium speed diesel nowadays is typically 20 MW (18 cylinder models), whereas the WR-21 is rated at 25 MW (ISO, no losses). From a redundancy point of view a high electric load demand is definitely a prerequisite. For example 90 MW installed power or more would require at least four WR-21 engines, which would result in relatively low first cost and through life costs (compared to 601-R configurations) while retaining current diesel plant redundancy and flexibility.

[6] [Lepuhaä/van Zomeren, 1996]

2.3 Selected vessels

The previously described selection of ships (year of delivery: after 1988, gross tonnage over 50,000 tonnes) from Figure 1 is displayed in Figure 5. The ships in this figure are listed in Appendix 2.

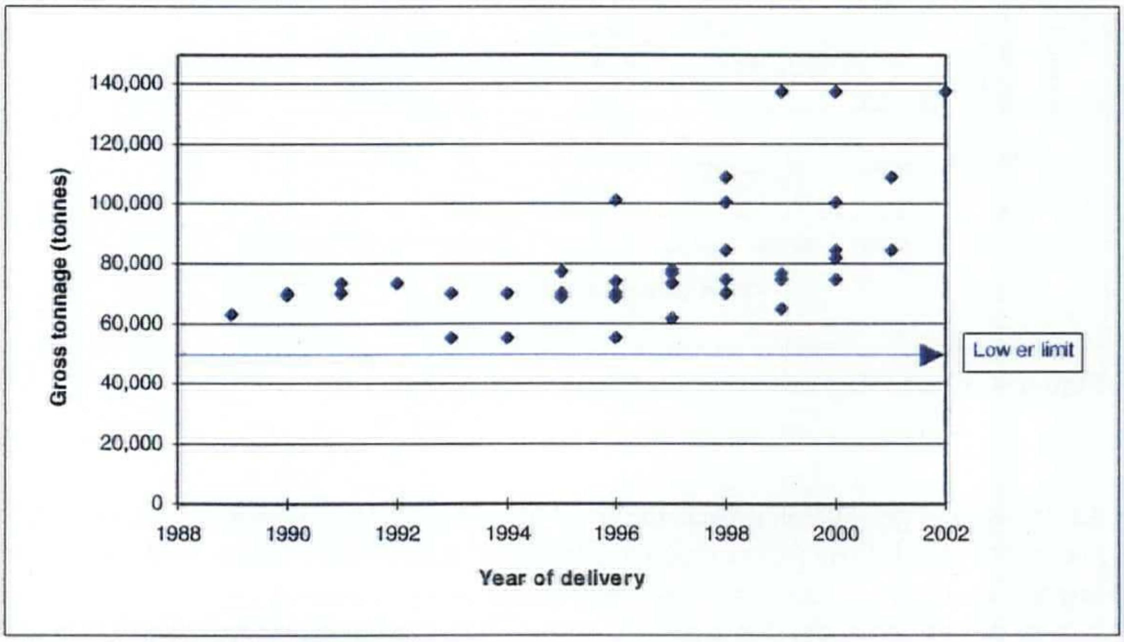

Figure 5: Selected cruise vessel range

The maximum ship size has increased from approximately 75,000 tonnes before 1994 up to the 138,000 tonnes for the future "Project Eagle" ships, due to be delivered in 1999, 2000 and 2002, respectively.

Based on the available information, the following ships will be used for further investigation and comparison: *Sun Princess*, *Grand Princess* and *Capricorn*, owned by Princess Cruises and P&O Cruises.

3. The selected cruise ships and their main features

In this chapter the selected cruise ships will be further examined.

The main characteristics of the three ships are shown in Table 1.

Ship	*GRT*	V_{Max}	$P_{Installed}$	$P_{Propulsion}$	*Year*	*Pax*[7]	*Shipyard*
	tonnes	*knots*	*kW*	*kW*	-	-	-
Sun Princess	77,000	22.4	46,800	28,000	1995	2000	Fincantieri
Grand Princess	109,000	23.9	69,100	42,000	1998	2600	Fincantieri
Capricorn	75,000	25.5	58,800	42,000	2000	2000	Meyer Werft

Table 1: Selected cruise ships and their main features

Sun Princess was the first vessel of its class to be built by the Italian shipyard Fincantieri. The Sun Princess class consists of the following vessels: *Sun Princess* (1995), *Dawn Princess* (1997), *Sea Princess* (1998) and *Ocean Princess* (1999). The vessels spend from September to April in the Caribbean and the remaining four months in Alaska. The reported price is \$ 275m[8].

Grand Princess is the largest cruise ship currently in service. Two *Grand Princess* sister ships are due to be delivered in 2001. These ships are or will also be built by Fincantieri. The Grand Princess class will cruise the Caribbean from September to April. The remaining four months will be spent in the Mediterranean. Alaska is not an option for this ship, because she is Post-Panamax size. *Grand Princess* is 36 metres wide while the maximum width for the Panama Canal is 32.2 meters. Obviously, two transatlantic cruises are required to sail from one continent to the other. The reported vessel price is \$ 450m[7].

Capricorn, also referred to as *Oriana II*, is currently under construction at the German shipyard Meyer Werft. As opposed to *Oriana*, *Capricorn* will be diesel-electric. Just like *Oriana*, *Capricorn* will cover most of the world with her various cruises. Next to a 90 day world cruise she will make various cruises all over Europe. The world cruise usually starts in Southampton early January and ends in the same port early April. Therefore April to December will be spent in Europe. The reported price for *Capricorn* is \$ 300m[7].

[7] Pax = passengers (lower berth); the amount of cabins is half this number. The certified passenger figure is 2,270 on *Sun Princess* and 3,100 on *Grand Princess*.
[8] From cruise trade magazines

3.1 Power-speed curves

The propulsion power on existing diesel-electric cruise ships is delivered by usually two synchronous motors. The two propellers are each directly driven by these motors. *Sun Princess* incorporates two 14,000 kW electric motors whereas both *Grand Princess* and *Capricorn* comprise two 21,000 kW motors.

Figure 6 below shows the required propulsion power with ship speed. The propulsion power mentioned here represents the electric motor shaft power output. Due to the losses in the electrical systems (propulsion electric motor, cables, rectifier, transformer and generator) the actual engine load will be higher, typically about nine percent. Six percent in the propulsion system and three percent in the generator.

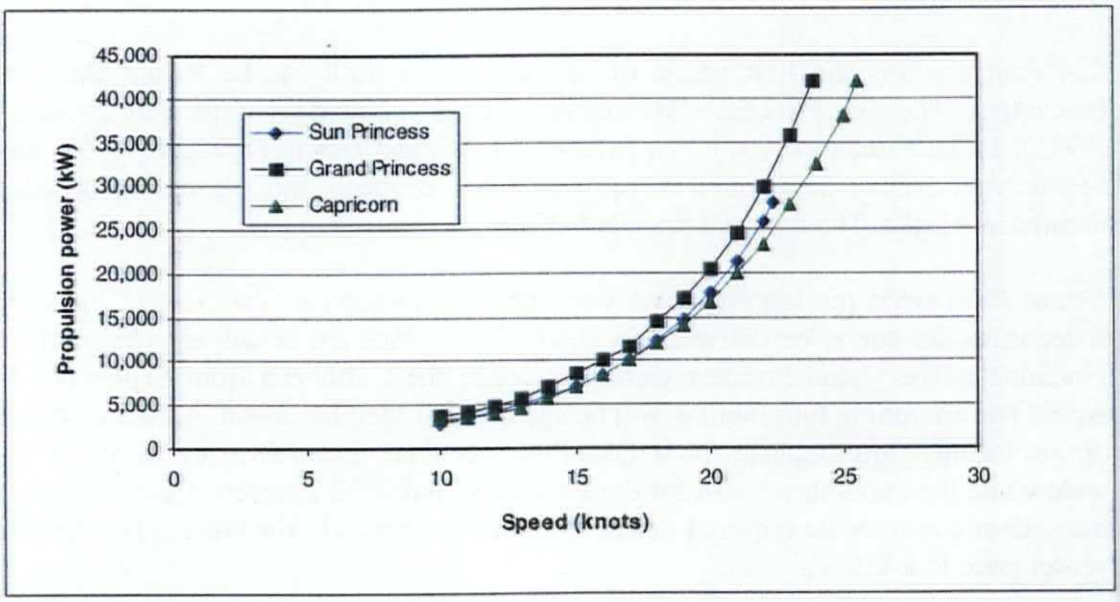

Figure 6: Propulsion power - speed curves

It has to be noted here that the curves in Figure 6 are valid for a calm sea and no hull fouling. When the hull becomes foul the drag or frictional resistance (a component of the total resistance) increases. Consequently the propulsion power required to attain a desired speed increase by as much as 10%. As a result the maximum speed will decrease with roughly 0.5 knots.

As discussed in chapter 2, the required propulsion power can be roughly estimated according to the cube law. A more accurate estimation can be obtained according to the following equation:

$P_E = c(v)*v^3$ [kW],

where $c(v)$ is a function of ship speed.

The $c(v)$ curves for each ship can be determined by dividing the curves from Figure 6 by the associated ship speed to the third order. $C(v)$ is displayed in Figure 7 against relative ship speed. As can be seen in this figure a 3rd order polynomial best fit line represents a good estimation for $c(v)$. Thus, when the maximum ship speed and the maximum propulsion power are known, the propulsion power curve can be estimated by:

$$\frac{P_E}{P_{E,Max}} = \underbrace{[-550*(\frac{v}{v_{Max}})^3 + 1524*(\frac{v}{v_{Max}})^2 - 1316*(\frac{v}{v_{\max}}) + 441]}_{\text{by } \frac{v}{v_{max}} = 1 \;\rightarrow\; c(v) = 100} * (\frac{v}{v_{Max}})^3 \rightarrow \text{dues } \frac{P_E}{P_{E_n}} \text{ in } \%$$

where,

$P_{E,Max}$ = the maximum propulsion power (100% motor load)
v_{Max} = the maximum ship speed

This curve gives an fairly accurate estimation for ship speeds over 30 % and for cruise ships similarly shaped to *Sun Princess*, *Grand Princess* and *Capricorn*.

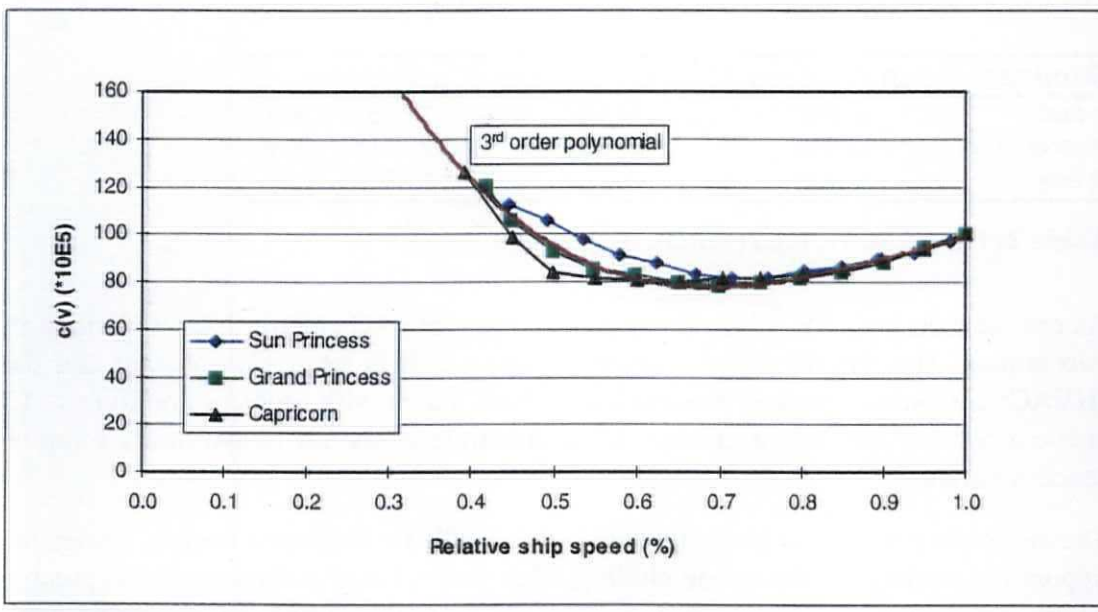

Figure 7: $c(v)$ with ship speed

The propulsion power demand when manoeuvring is assumed to be 6 MW. This value is an arbitrary average since the propulsion power is highly variable because the ship has to accelerate and decelerate several times. The same can be said for the bow- and stern thruster load.

3.2 Electric and steam load balances

Appendix 3 presents the electric load balances based on the information provided by P&O Cruises. The day and night variation is neglected here, because this varies only slightly. This is caused by the fact that a relatively large part of the electric load is represented by 'engine service' and 'Heating, Ventilation and Air Conditioning' (HVAC), as can be seen in Appendix 3. The 'engine service' electric load at night will be at maximum, since the ship then usually cruises at high speed. The lower temperature at night causes the HVAC load to decrease, however this load will not decrease significantly as the humidity is remains around 90%. The temperature may be lower, but reducing the humidity to a desired 50-60% will still require a large amount of energy. During the day the vessel is in harbour, with a low number of engines running. Especially for summer conditions the HVAC load will be relatively high compared to night conditions. Clearly also the load from accommodation, galley and laundry services will increase in the daytime. The net result is a 5-6% electric load decrease at night[9].

The values from Appendix 3 are summarised in Table 2.

Summer	*Sun Princess*	*Grand Princess*	*Capricorn*	
In port	8,266	14,986	11,280	kW
Manoeuvring	13,654	23,586	20,517	kW
At sea	10,738	16,733	11,567	kW

Table 2: Hotel electric load (summer)

As can be seen from the tables in Appendix 3 the HVAC electric load is divided in two groups. The 'HVAC: fans' - electric load, which is relatively constant and the 'HVAC: compressor/pumps' electric load which varies with ambient conditions. To create a comfortable indoor climate the temperature of the air blown into a room or space is varied.

The air cooling process is brought about by a so called compressor cooling system, as opposed to cooling on absorption chilling. This process uses compressors and pumps, which are driven by electric motors. The air heating process on the other hand requires steam to heat the air through heat exchangers. Electric energy is then only required for pumps.

In 'winter' conditions (temperature -5 °C) the electric load demand is as follows:

Winter	*Sun Princess*	*Grand Princess*	*Capricorn*	
In port	4,649	10,163	7,663	kW
Manoeuvring	10,037	18,763	16,900	kW
At sea	7,121	11,910	7,950	kW

Table 3: Hotel electric load (winter)

[9] As measured by P&O Cruises

The 'HVAC: compressors/pumps' electric load for various ambient conditions is shown in Figure 8 below. The figure is based on a simple HVAC model for *Grand Princess*.

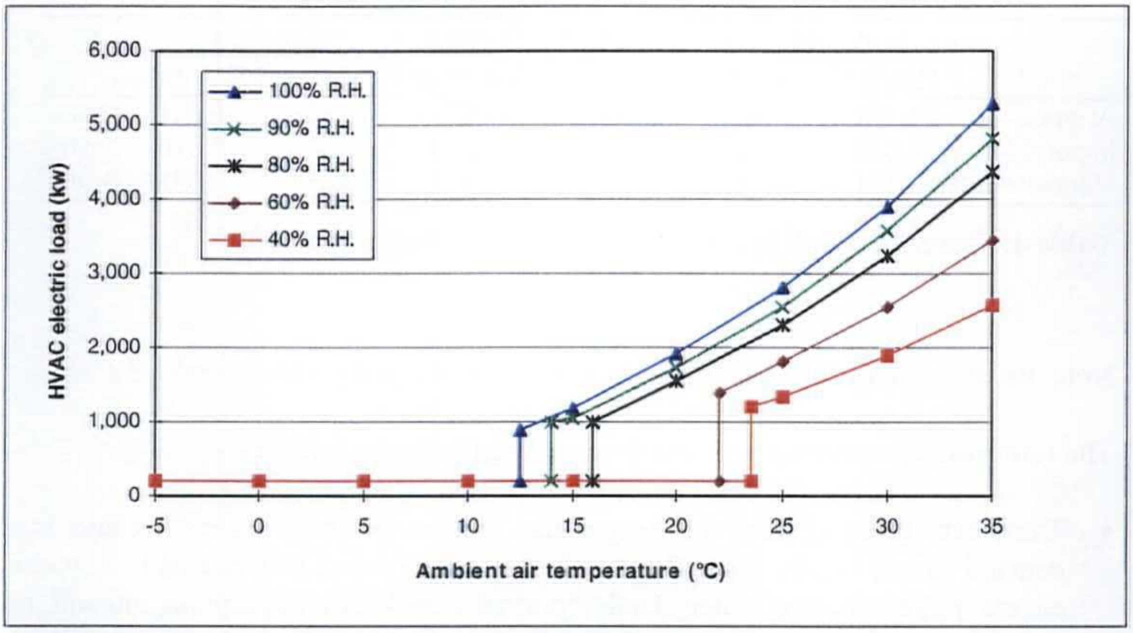

Figure 8: *Grand Princess* 'HVAC: compressors and pumps' electric load

Note: the -5 °C and +35 °C are the extreme temperatures as adopted by P&O Cruises.

The HVAC electric load curves for *Sun Princess* and *Capricorn* are assumed to be similar to the curves in Figure 8. Since both vessels are significantly smaller than *Grand Princess* the power requirements will be lower. The gross tonnage ratio is 0.71 for *Sun Princess* and 0.69 for *Capricorn*. A pessimistic HVAC load ratio of 0.75 is assumed for both ships and all curves will be multiplied by this ratio to determine the actual electric load. The assumption made here is that the HVAC electric load is proportional to the volume of all rooms and spaces in the ship.

In conclusion: the electric load will be higher for summer than for winter conditions, whereas the steam load will be higher for winter conditions compared to summer conditions.

Also shown in Appendix 3 are the steam load balances. The numbers are summarised in Table 4 below. The steam generated by the waste heat or oil fired boilers is saturated steam at 9 bar (g), 180°C.

	Sun Princess		Grand Princess		Capricorn		
	Winter	Summer	Winter	Summer	Winter	Summer	
At sea	21,131	14,655	28,320	14,970	13,556	9,841	kg/h
In port	18,013	11,124	18,150	10,830	12,374	7,875	kg/h
Manoeuvring	21,131	14,655	28,320	14,970	13,556	9,841	kg/h

Table 4: Steam load with heavy fuel preheating (existing situation)

Note: the steam load during manoeuvring is assumed equal to the steam load at sea.

The following assumptions were made regarding the steam balances:

- The steam demand from the evaporators heaters is omitted here because this demand varies, largely depending on the amount of waste heat available from the engine jacket cooling water. Under normal conditions (maximum amount of engines running) the majority of the required heat for the evaporators will be provided by this jacket cooling water. However when engine load falls, so does the amount of waste heat available from the cooling water and additional steam is needed to meet the heat demand from the evaporators.

- At intermediate temperatures the steam load varies linearly between the extremes as mentioned in Table 4.

3.3 Machinery deck

A part of a typical machinery deck arrangement of a large cruise vessel is shown in Appendix 4. This machinery space deck is divided in 14 (watertight) compartments (see below). Those printed bold will be affected by the choice of prime mover.

- Bow thruster room
- Potable water room
- Pool treatment room
- Heeling pump room
- Fresh water pump room
- Stabiliser and Air Conditioning compressor room
- Domestic refrigeration compressor room
- **Mid-ships auxiliary room**
- **Evaporators room**

- **Forward diesel generators room**
- Propulsion Electronics Motors (PEM) room
- **Aft diesel generator room**
- **Aft auxiliary room**
- Aft thruster room

The last six compartments which apply to this study are shown in Appendix 4.

Although the machinery arrangement will differ for every ship, this list mentions all the main systems that will be installed on a generic diesel-electric cruise ship.

Not all systems mentioned above are of interest. The systems that will be considered for further investigation are those systems which differ with gas turbine-electric power systems: the auxiliary systems, the evaporators and of course the gensets.

3.3.1 Diesel gensets

Existing large cruise liner prime movers all comprise four stroke medium speed diesel gensets, with a power range from 4,000 to 19,000 kW. A wide choice of medium speed engines is available on the international market, but most cruise ship projects have favoured large bore models from MAN B&W Diesel and Wärtsilä NSD (the latter is a merger of Wärtsilä Diesel and New Sulzer Diesel Ltd). The engines and their main features are listed below in Table 5:

Manufacturer	*Engine type*	*Bore (mm)*	*Speed (rpm)*	*Cyl No (-)*	*Power (kW/cyl)*	*Power range (kW)*
Wärtsilä NSD	ZA40S (Line+Vee)	400	510	6 - 18	720	4,320 - 12,960
	38 (Line+Vee)	380	600	6 - 18	660	3,960 - 11880
	46 (Line+Vee)	460	500/514	6 - 18	1050	6,300 - 18,900
MAN B&W	48/60 (Line+Vee)	480	500/514	6 - 18	1050	6,300 - 18,900
	58/64 (Line)	580	428	6 - 9	1390	8,340 - 12,510

Table 5: Medium speed four stroke diesel engines for large cruise vessels

The ZA40S can among others be found in the: *Rotterdam VI*, *Statendam* class (4), ***Sun Princess*** *class (3)*, ***Grand Princess***, *Carnival Destiny class (3)*, *Fantasy* class (6), *Disney Magic, Disney Wonder and Crystal Symphony*, the 46 in the *Legend of the Seas*, *Splendour of the Seas*, and in the future "Project Eagle" ships, the 38 in the Azipod propelled *Elation* and *Paradise*, the 48/60 in the ***Capricorn***, *Mercury* class (3), *Grandeur of the Seas*, *Enchantment of the Seas* and *Costa Victoria* and the 58/64 in the *Arcadia*, *Regal Princess*, *Crown Princess* and *Crystal Harmony*.

Table 6 shows the prime mover configuration for each ship.

Ship	*Engines (-)*	*Configuration (-)*	*Total installed power (kW)*
Sun Princess	Sulzer ZAV40S	4x16 cyl	46,100
Grand Princess	Sulzer ZAV40S	6x16 cyl	69,120
Capricorn	MAN B&W V48/60	4x14 cyl	58,800

Table 6: Cruise ship prime movers

The specific fuel consumption curves according to the project guides are shown below.

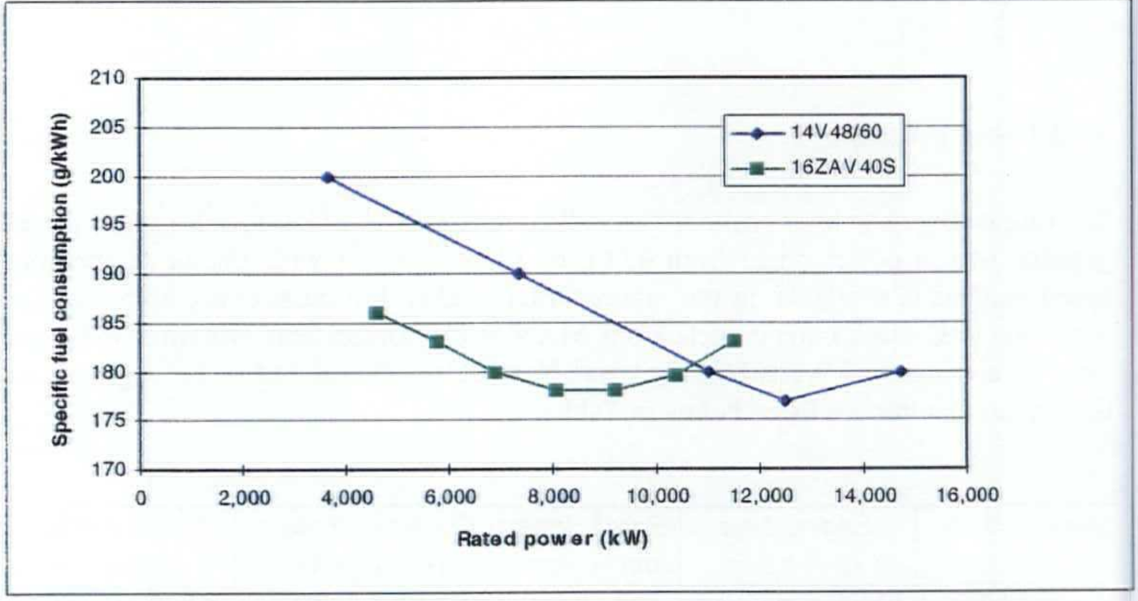

Figure 9: Specific fuel consumption curves at ISO conditions, standard losses

Note: the 14V 48/60 curve appears to be strange, however, these are the actual values as obtained from the project guide.

The curves are valid for the following conditions:

- Ambient air temperature: 25 °C
- Ambient pressure: 1,013 mbar
- Standard intake and exhaust losses (maximum exhaust gas back pressure: 250 mm wg)
- Fuel lower calorific value: 42,700 kJ/kg (Marine Diesel Oil)
- No engine mounted pumps

The fuel consumption for off-design conditions on a cruise ship will be investigated in section 7.3.1: **Fuel consumption**.

3.3.2 Auxiliary systems

As mentioned before medium speed diesel engines can burn both heavy and distillate fuels whereas aeroderivative gas turbines require distillate fuels. The fuel treatment process for heavy fuel being more complex, so the diesel engine auxiliary system also becomes more complex. In this section a comparison between both systems will be made in order to obtain an auxiliary system general comparison.

For further investigation, the auxiliary systems will be subdivided in the following categories:

- Cooling water systems
- Lub oil systems
- Fuel treatment systems
- Air intake and exhaust gas uptake systems
- Engine starting systems
- Engine room ventilation systems

The function and the main components of the systems mentioned above are described in Appendix 5. The actual comparisons will be made in the following chapters.

All components after the generator (rectifier, inverter, electric motor, propeller shaft and propeller) are considered similar for the comparison made in this report since they remain the same for both diesel-electric and gas turbine-electric.

3.3.3 Evaporators and oil fired boilers

The selected cruise ships all use flash evaporation systems for fresh water generation. Both *Sun Princess* and *Grand Princess* incorporate three evaporators whereas *Capricorn* has two. The fresh water generating capacity for each ship is shown in Table 7.

Evaporators	*No. of units* -	*Generating capacity (tonnes/day)*	*Steam cons. (tonnes/day)*	*Energy cons. (kW)*
Sun Princess	3	2x640+1x320=1,600	461	12,600
Grand Princess	3	3x740=2,220	648	17,700
Capricorn	2	2x640=1,280	374	10,200

Table 7: Cruise ship evaporator capacities and energy consumption

The heating medium for evaporators in this case is either (high temperature) cooling water or steam, in case the heat from the cooling water is insufficient. For example: the plant from *Sun Princess* is configured to use waste heat from the engine jacket cooling system. Operation at 85% Maximum Continuous Rating (MCR) on two engines is sufficient to operate one 640 tonnes per day unit. Table 7 shows the steam consumption (see also steam balances in Appendix 3) in case no heat is available from the engine jacket cooling water.

Gas turbines do not use a central fresh water cooling system for engine cooling. Instead, the engine is cooled internally with bleed air from the compressor and externally with ventilation air. As a result the waste heat available from engine cooling is negligible. However, for a WR-21, a significant amount of waste heat is rejected through the combustion air intercooler. This is further investigated in chapter 8.

To create 1 kg of saturated steam of 9 bar (g) and 180 °C the following amount of energy is needed:

$$q = c_{P,water}(T_{steam} - T_{feedwater}) + r$$

q	=	specific energy [kJ/kg]
$c_{P,water}$	=	specific heat of water [kJ/kgK]
T_{steam}	=	steam temperature [K]
$T_{feedwater}$	=	feedwater temperature [K]
r	=	evaporation heat [kJ/kg]

It is assumed that the feedwater temperature is about 100°C. For 9 bar steam, the evaporation heat is 2,015 kJ/kg. Consequently the energy required to create 1 kg steam becomes:

$$q = 4.2 * (453 - 373) + 2{,}015 = 2{,}352 \text{ [kJ/kg]}$$

As a result the energy consumption (Table 7, column 5 'Energy cons.') becomes:

$$Energy\ consumption = \frac{1{,}000 * 2{,}352 * Steam\ consumption}{24 * 3{,}600} \text{ [kW]}$$

Clearly the evaporators require a large amount of energy in proportion to the electric load.

The actual daily fresh water consumption and the percentage of the maximum generating capacity is shown below.

	Daily FW cons. (tonnes)	Relative cons. (%)
Sun Princess	1,000	62.5
Grand Princess	1,100	49.5
Capricorn	620	48.5

Table 8: Fresh water consumption

The reason for this apparent over-capacity is that fresh water can not be produced while being in port or manoeuvring because of possible water pollution. Typically the ships start generating water when more than 10 miles off the coast. The fresh water consumption on *Sun Princess* is relatively high compared with the other vessels. This is simply caused by the fact that Americans tend to use more water during a cruise than Europeans. The column 'Relative cons.' in Table 8 thus indicates the percentage of time that the evaporators have to be operated (at 100% capacity).

Additional oil fired boilers are installed to create steam when the heat available from the diesels is insufficient. *Sun Princess* incorporates two oil fired boilers each capable of producing 12,000 kg steam at 9 bar (g) per hour. On *Grand Princess* two oil fired boilers produce a total of 30,000 kg per hour, and on *Capricorn* two systems produce 20,000 tonnes in total.

The oil fired boiler capacities for gas turbine cruise ships will be determined in section 7.1.3: **Meeting the steam demand**.

4. Operating profiles and associated ambient conditions

The comparisons in this study will be based on three different itineraries: a 7-day Caribbean cruise, a 7-day Alaska cruise and a 92-day World cruise. The reason for this choice is the variety in ambient conditions and operating profiles. The ambient conditions have to be taken into account, since they determine the actual prime mover performance, the steam demand and the electric demand. The influence on both fuel consumption and maximum prime mover power output will be investigated in sections 5.1: **The Northrop Grumman/Rolls-Royce WR-21** and 7.3.1 **Fuel consumption**.

From the itinerary descriptions provided by P&O Cruises operating profiles will be constructed for every itinerary.

4.1 7-Day Caribbean cruise

Four Caribbean operating profiles are shown in Appendix 6. These profiles will be joined in one itinerary, simply by regarding all the periods at sea as if they belonged to one cruise. Then the total number of hours is divided by four to create a seven day operating profile. Of course the total amount of hours from the 11-day Caribbean cruise is multiplied by 7/11 before that, to obtain the generic 7-day Caribbean cruise operating profile.

The itineraries mentioned in Appendix 6 belong to *Arcadia* (formerly known as the *Star Princess*), *Sun Princess*, *Crown Princess* and *Regal Princess*. The itinerary with a 20.5 knots maximum speed (average speed during that particular interval) belongs to the *Arcadia*. Since her maximum speed is about 22 knots, this will be adopted as the maximum speed for the profiles to create a generic operating profile valid for every ship cruising in the Caribbean.

The corresponding generic speed profile is shown in Figure 10, which displays the total time at sea at a certain speed range against the relative speed as a percentage of the maximum speed.

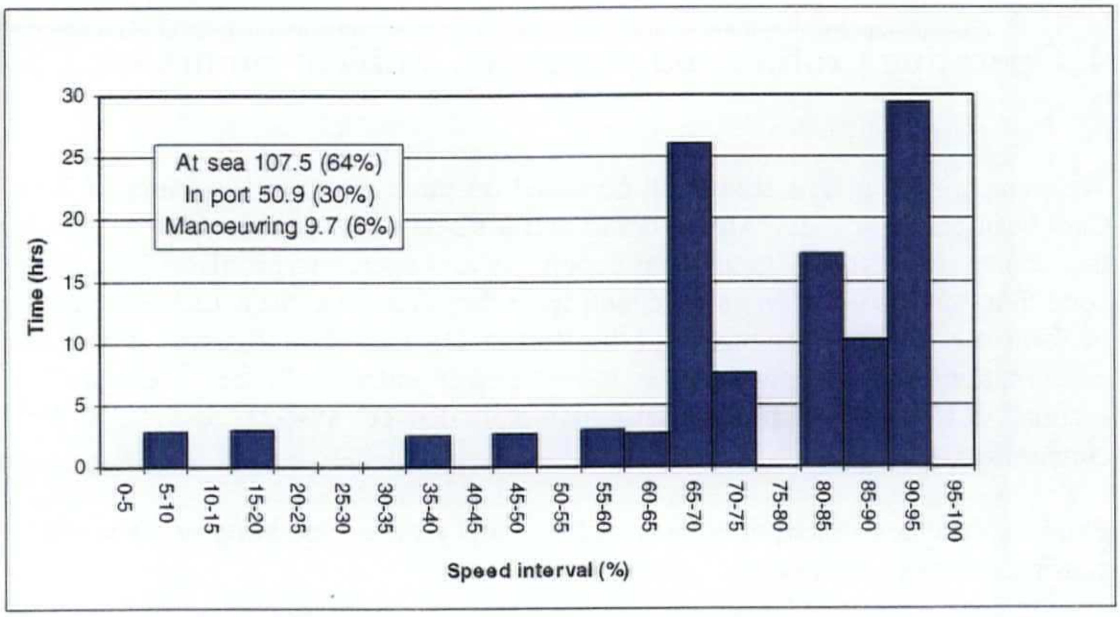

Figure 10: Generic 7-day Caribbean cruise operating profile

Table 9 below presents the minimum and maximum air temperatures at sea level during the year[10].

Period	Ambient air temperature(°C)	
	Minimum	Maximum
January	**24.9**	**26.6**
February	**24.6**	**26.5**
March	**24.7**	**27.1**
April	**25.4**	**27.9**
May	25.9	28.3
June	26.8	28.6
July	27.2	28.7
August	27.4	28.9
September	**27.3**	**29.3**
October	**27.1**	**28.8**
November	**26.5**	**28.1**
December	**25.7**	**27.3**

Table 9: Ambient air temperatures in the Caribbean

As can be seen in the table, the air temperatures vary from 24.6 °C in February up to 29.3 °C in September. A worst case design point temperature of 30°C will be

[10] Original data from the Global Ocean Surface Temperature Atlas (GOSTA) from the UK Meteorological Office

adopted. Since the temperature variations are only about 5 °C, the minimum temperature will not be considered as a separate case.
Onshore temperatures tend to be more extreme. In this study however, 30 °C is assumed maximum because this temperature will be exceeded for relatively short periods only. During this short period the electric (HVAC) load will be at a maximum and the engine maximum power output will be minimum.

Also, according to the UK Meteorological Office the relative humidity at sea varies from 80% to 95%, globally. These values are independent of the ambient temperature. The absolute humidity (kg water per kg air), on the other hand, is less for lower temperatures because 'cold' air can contain less water than 'warm' air. The relative humidity is defined as the ratio of the amount of water contained by the air and the maximum amount of water air can contain before condensation occurs. Consequently, 90% relative humidity is adopted for every itinerary, regardless of its climate.

4.2 7-Day Alaska cruise

The 7-day Alaska operating profile will be regarded as a generic operating profile for the Alaska region and is shown below in Table 10.

Itinerary	*Speed (knots)*	*Interval (hours)*
7 day Alaska	19.8	24
	17.5	15.5
	15.9	35.5
	15.3	9
	12.2	9
	7.9	12
	Man.	9.5
	In port	53.5

Table 10: 7-day Alaska operating profile

The corresponding speed profile, again with a 22 knots maximum speed is shown in Figure 11.

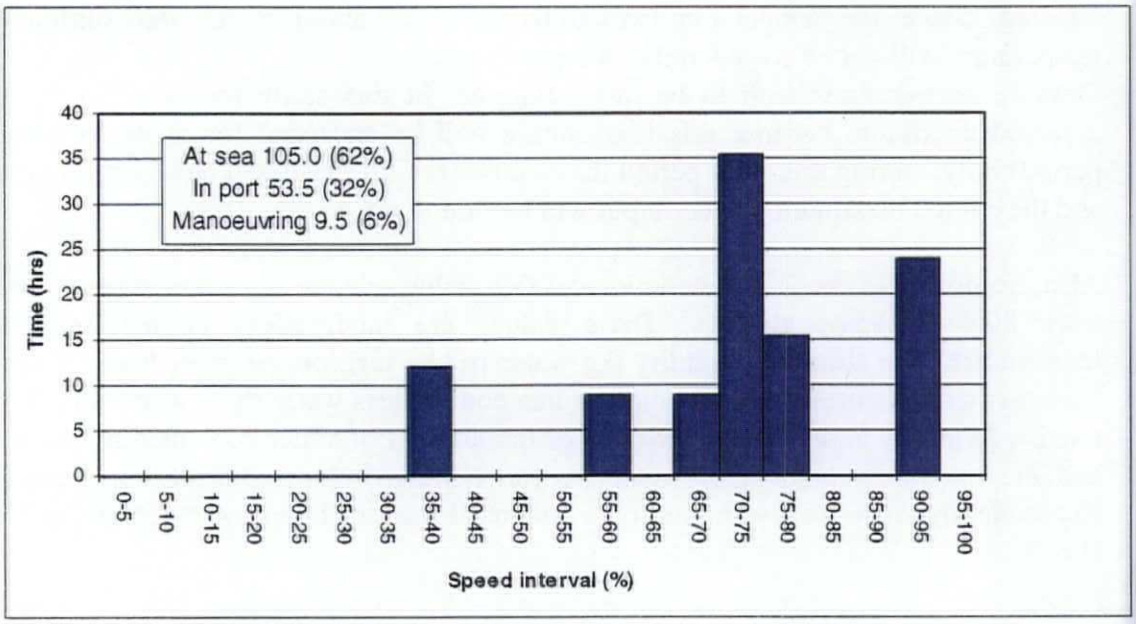

Figure 11: 7-Day Alaska speed profile

The Alaska climate, with a typical 20 °C lower air temperature, is more favourable for prime mover performance. The steam consumption on the other hand will increase while the electric load decreases.

The mean and maximum temperatures (GOSTA) are shown in Table 11.

Period	*Ambient air temperature(°C)*	
	Minimum	*Maximum*
January	2	6
February	0.9	5.9
March	0.6	5.6
April	3.4	7.2
May	**4.4**	**10.4**
June	**5.4**	**12.4**
July	**8.8**	**14.2**
August	**9.7**	**15.6**
September	**9.2**	**14.2**
October	6.5	11
November	3.1	9.5
December	1.7	6.7

Table 11: Ambient air temperature in Alaska

As can be seen above, the monthly average air temperatures vary from 0.6 °C in March up to 15.6 °C in August. None of the ships spend their time in Alaska from October till April. The minimum temperature they encounter is 4.4 °C. The average temperature during the period the ships cruise in Alaska is therefore 10 °C. This will be taken as the design point temperature rather than the 15.6 °C maximum in order to investigate the ambient condition influence more clearly.

P&O Cruises' design point minimum is -5 °C, rather than 0 °C according to the above mentioned table to account for more severe conditions. These conditions may occur when cruising more land inward during the Alaska itinerary or in harbour.

4.3 92-Day World cruise

This cruise is the 92-day *Oriana* World cruise. A similar cruise will be made by *Capricorn* and therefore this operating profile will be used in this study. The ship departs from Southampton on 6^{th} January and returns to the same port on 8^{th} April. The ports of call are displayed in Appendix 6.

The ambient temperatures during this cruise varies from 10 °C to 30 °C, except for the first day, leaving Southampton. The temperature will be divided in two groups: 10-20 °C and 20-30 °C: 'moderate' and 'summer' condition. The ambient temperature for each part of the journey can be found again from the GOSTA data. It is assumed that the temperature for the trips is either moderate, summer or equally spread over both conditions. The maximum temperature for the moderate condition is assumed to be 15 °C and for the summer condition 25 °C. Therefore, summer conditions are the same as the ambient conditions as defined for the Caribbean. The relative humidity is constant at 90% again. Both conditions are also displayed in Appendix 6. The World cruise speed profile is shown in Figure 12 .

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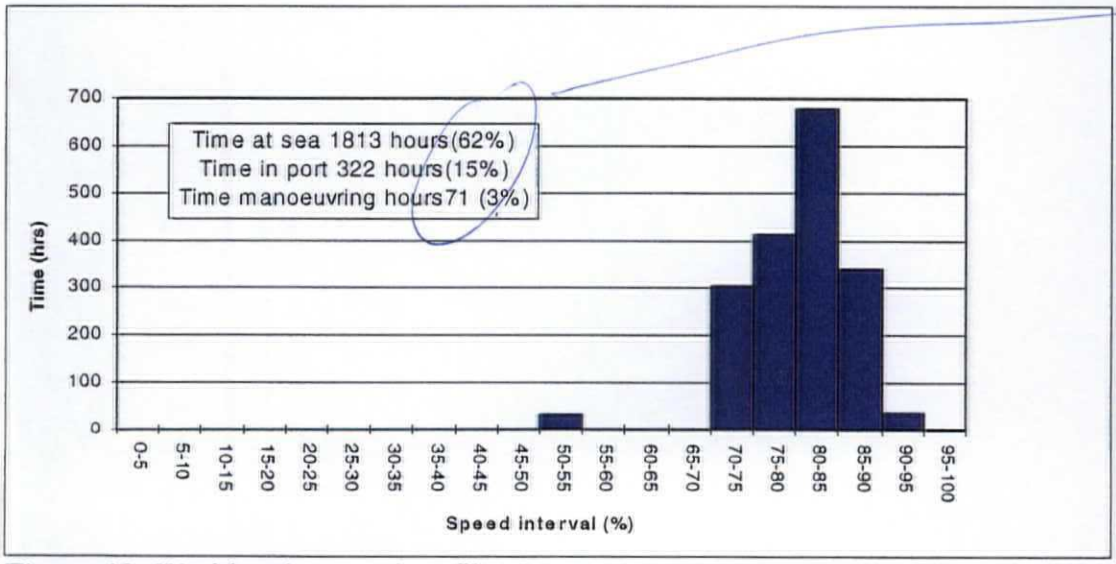


Figure 12: World cruise speed profile

5. The Rolls-Royce gas turbines

Rolls-Royce Ltd. was founded in 1906 by Charles Rolls and Henry Royce. The company then manufactured cars only. During the first world war Rolls-Royce started the development of aero engines. Due to the success of these engines the company quickly expanded from a relatively small company to a company which supplied the majority of engines for aeroplanes used in the second world war.

During the second world war Rolls-Royce introduced the gas turbine. The first gas turbine, the W1, had to demonstrate that gas turbines could be reliable power sources for aircraft. The engine ran for 35 hours. The next engine, the Welland was more successful and was used in the Gloster Meteor fighters.

After the war the company focussed its efforts on developing gas turbines and the piston aero engines were put on a sideline. For the first time in 1953 a gas turbine was used in civil aviation: the Dart propjet.

In 1961 Napier Aero Engines and in 1966 Bristol Siddely were acquired. These were the last two remaining British aero engine manufacturers. In the late 60's Rolls-Royce launched a new engine, the RB211. Initial problems with this engine forced the company to be taken into State ownership. Moreover the motor car division was split off. Since 1987 Rolls-Royce has been a private company again.

In 1989 Northern Engineering Industries was acquired to expand the industrial power division. The Allison Engine Company which produces small gas turbines was acquired in 1995. Since 1997, large steam turbines are no longer part of the Rolls-Royce portfolio; this division was sold to Siemens.

The current Rolls-Royce organisation is shown below.

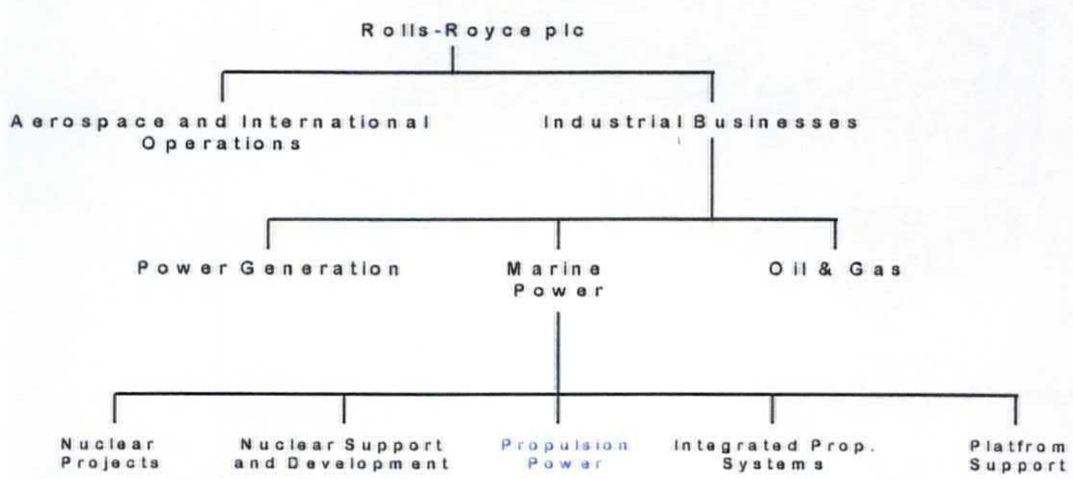


Figure 13: Rolls-Royce Industrial Businesses-organisation chart

Currently, Rolls-Royce Marine Power has a portfolio of gas turbines, including the Allison 501-KF5/KF7, Allison 601-KF9/KF11, Marine Spey, WR-21 and Trent. Furthermore they provide refurbishment services for the Tyne, Olympus and Proteus.

Although still in full development the WR-21 and the 601-R are the gas turbines used for the comparison in this study. The reason for this is their low fuel consumption due to advanced cycle technology. The first WR-21 is scheduled for delivery in 2001 whereas the 601-KF9 will be available in the year 2000. A recuperated version is expected to appear on the market a few years later.

5.1 The Northrop Grumman/Rolls-Royce WR-21

The Westinghouse Electric Corporation and Rolls-Royce plc 21st century gas turbine, or WR-21 (see Figure 14) is an advanced cycle aeroderivative gas turbine. This engine is the US Navy's only propulsion engine development programme for ships and is their base engine choice for the twenty-first century. The WR-21 uses proven reliable components from current Rolls-Royce RB211 and Trent aero engines as its core.

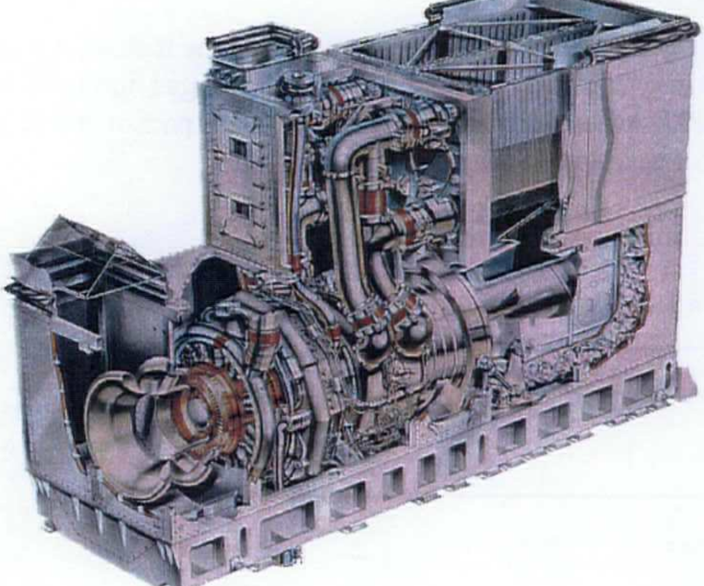


Figure 14: WR-21 module

Significant elements of the Westinghouse Electric Corporation were bought by Northrop Grumman Marine Systems in 1997 and as a result the engine is now referred to as the Northrop Grumman/Rolls-Royce WR-21.

The WR-21 incorporates an intercooler, recuperator and Variable Area Nozzles (VAN) at the inlet of the power turbine to increase overall performance. The recuperator and VAN improve both part load and overall efficiency, while the intercooler increases the specific power of the engine.

Figure 15 shows schematically the InterCooled Recuperated (ICR) cycle.

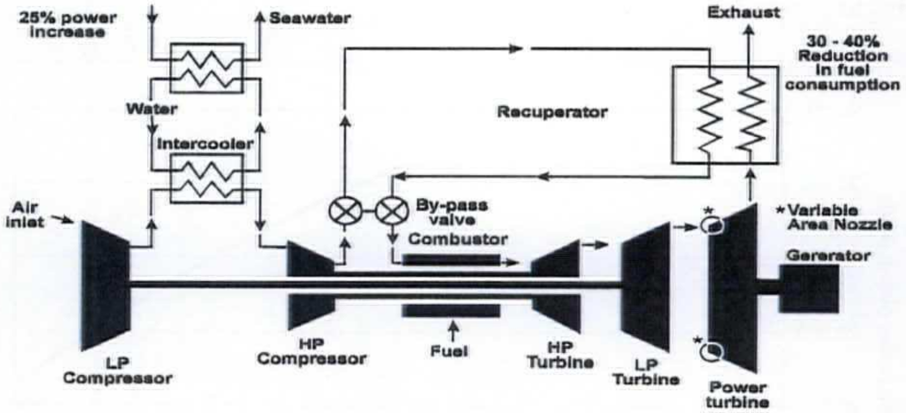


Figure 15: WR-21 Intercooled Recuperated cycle

The intercooler increases the specific power by reducing air temperature entering the High Pressure (HP) compressor. By reducing inlet temperature and therefore air density, the work done by the compressor is reduced. However, since the HP compressor exit temperature is also reduced, additional energy is required to reach the desired turbine entry temperature. Part of the recuperator-won-energy is used for this.

The recuperator transfers heat from the exhaust gases to the compressor delivery air, raising its temperature and therefore reducing the amount of fuel required to achieve the required turbine entry temperature. There is a small loss in specific power due to the increased pressure losses.

The Variable Area Nozzles (VAN) at the inlet of the power turbine control the flow area and therefore the air flow rate. The VAN operate by reducing the gas flow as power falls, which maintains the high exhaust gas temperature and allows the recuperator to be used to its full extent even at part load. This results in a flat specific fuel consumption curve compared to a simple cycle curve.

Combining the intercooler, recuperator and VAN in one engine leads to both a high thermal efficiency (and therefore low fuel consumption), especially at part load and high specific power.

The WR-21 estimated performance programme

Ambient condition variations influence both fuel consumption and maximum power output. The actual engine performance for off-design conditions will be investigated by using the WR-21 performance programme[11]. The fuel consumption for off-design conditions will be investigated in section 7.3.1: **Fuel consumption**. The input parameters for the programme are:

1. Ambient air temperature (°C): the maximum power output variation is displayed in Figure 16 below.

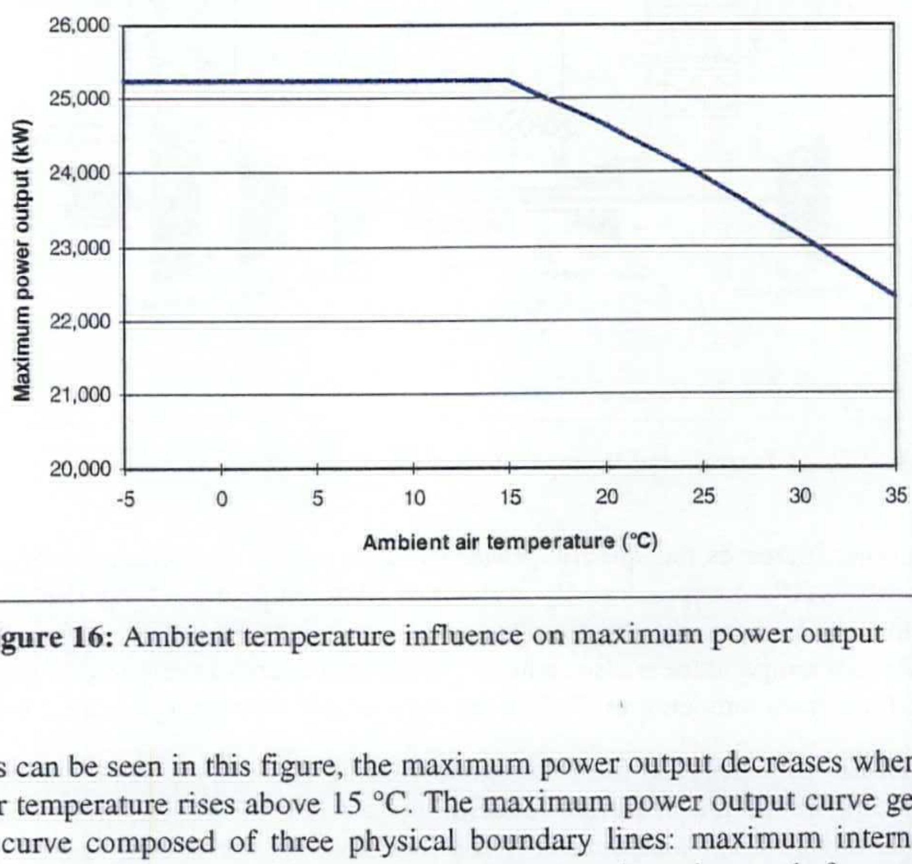


Figure 16: Ambient temperature influence on maximum power output

As can be seen in this figure, the maximum power output decreases when ambient air temperature rises above 15 °C. The maximum power output curve generally is a curve composed of three physical boundary lines: maximum internal engine pressure, maximum turbine entry temperature and maximum shaft speed. This is demonstrated more clearly in Figure 17 below.

[11] Westinghouse Rolls-Royce WR-21 Marine gas turbine estimated performance programme

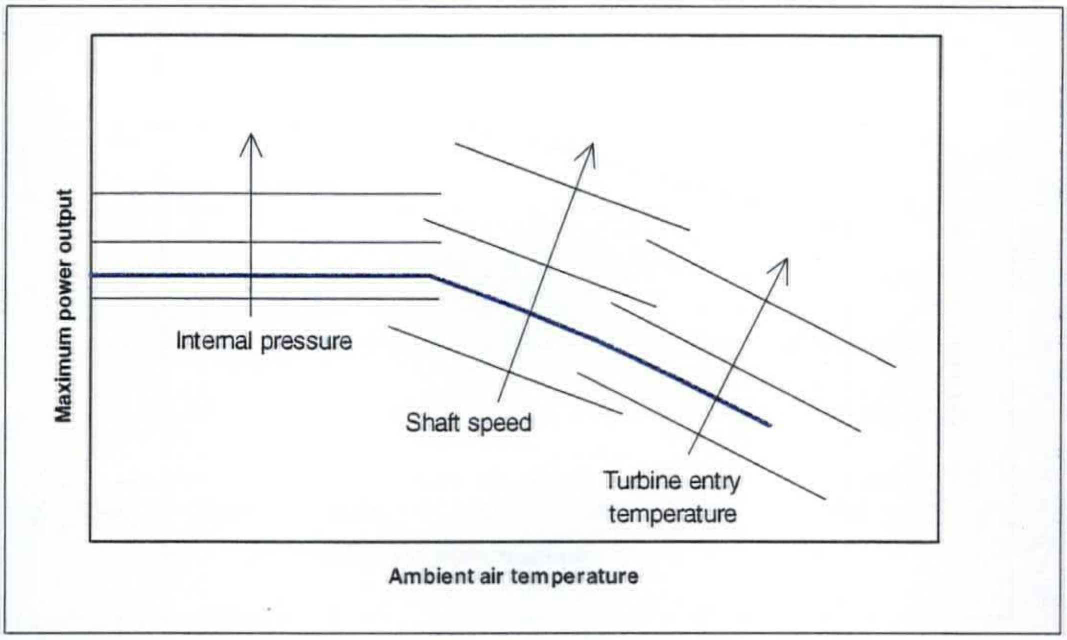


Figure 17: Gas turbine maximum power limitations

The three sets of boundary lines determine the actual maximum power output curve. The internal pressure lines are horizontal, because the power output is directly proportional to internal pressure (ambient temperature, compressor speed and turbine entry temperature assumed constant). Therefore the maximum power output below 15 °C (Figure 16) is constant and does not increase with decreasing ambient air temperature. The maximum power output at this speed is 25,240 kW (ISO, no losses).

2. Ambient pressure (mbar): this has a significant influence on maximum power output, as can be seen in Figure 18. From 1,013 mbar ISO, the curve is flat again, which is again because of the restrictions as explained previously.

3. Relative humidity (-): the relative humidity influence at low temperatures is minimal. At higher temperatures however, the influence becomes significant. This is displayed in Figure 19. The slope of the curves, starting at a certain humidity, depending on ambient temperature is brought about by a condensation control system. This system controls a bypass arrangement from the off-engine sea water/coolant heat exchanger. In this way the intercooler air outlet temperature can be controlled. This temperature must not fall below the dew point temperature, to avoid condensation and associated erosion of the HP compressor blades.

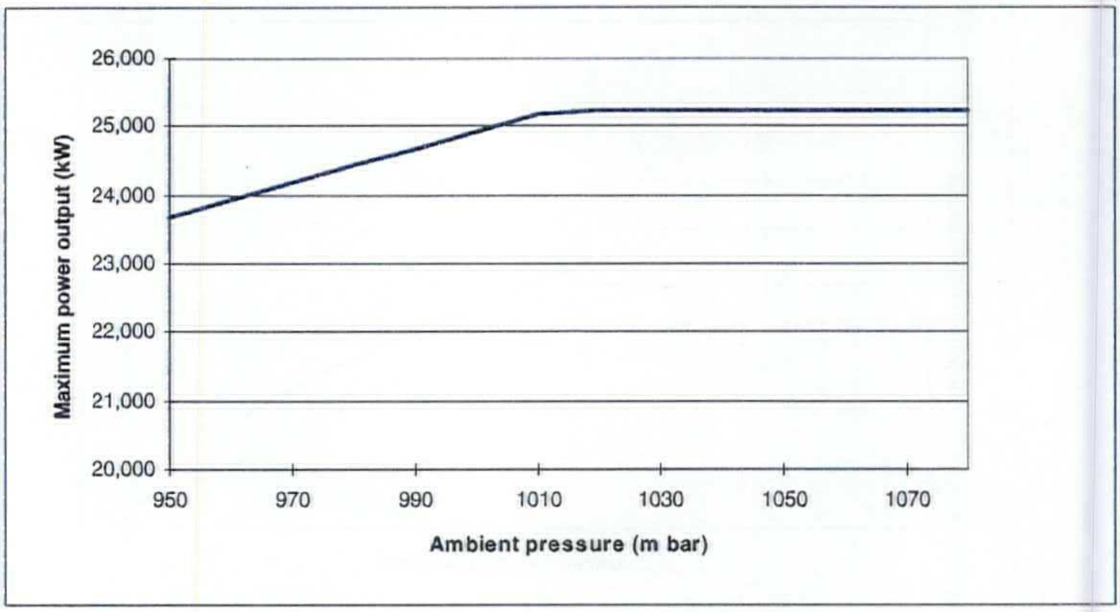


Figure 18: Ambient pressure influence on maximum power output

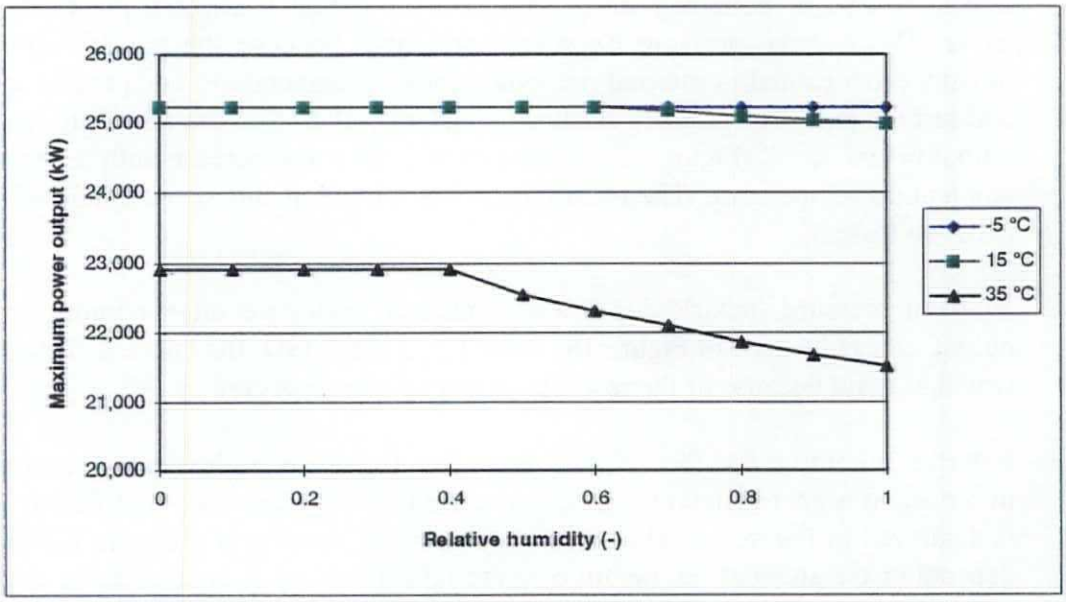


Figure 19: Relative humidity influence on maximum power output

As discussed before, an average 90% humidity will be used in this study.

4. Fuel lower heating value (kJ/kg): 42,700 for MDO and 43,125 for MGO
5. Specified shaft power (kW): variable.
6. Power turbine shaft speed (rpm): constant at 3600 rpm

7. Customer bleed flow (kg/sec): the customer bleed flow is 0, because there is no application for this in cruise ships.
8. Inlet duct pressure loss at datum point (mm wg): typically 145 mm wg[12]. The effect of the installation losses on maximum power output is displayed in Figure 20.
9. Exhaust duct pressure loss at datum point, (mm wg): typically 185 mm wg[12]. These calculations are also shown in Appendix 16.

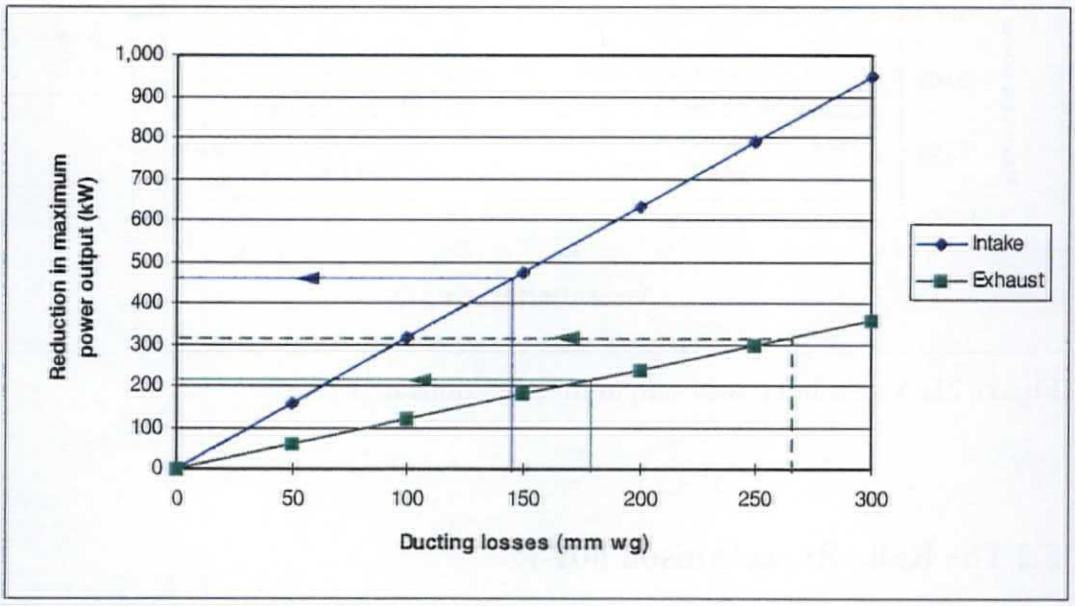


Figure 20: WR-21 installation loss effects on maximum power[13]

10. Recuperator bypass switch (-): operative or bypassed; this determines the exhaust gas temperature and therefore the amount of waste heat available.
11. Intercooler inoperative switch (-): the intercooler is kept operative at all times.
12. Reference power (kW): 25,240 kW.
13. Reference speed (rpm): 3,600 rpm (power turbine speed).

The power output according to the descriptions above for four cases in practice are shown in Figure 21.

[12] [van Lier, 1997]
[13] [WR-21 IRD, 1997]

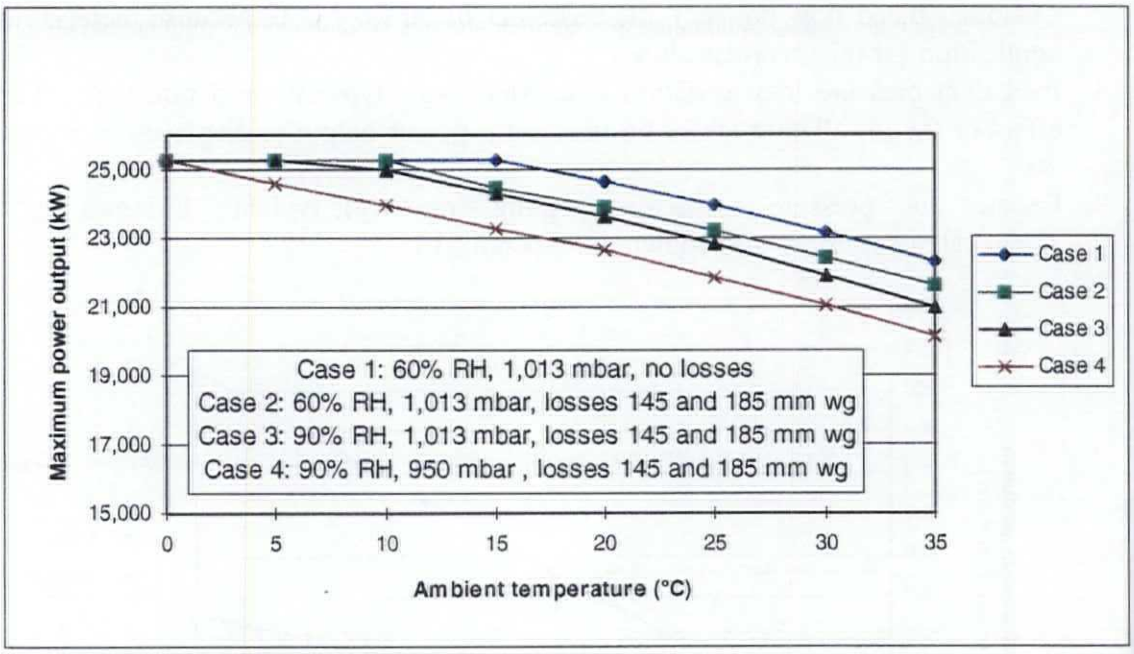


Figure 21: Maximum power output for conditions in practice

5.2 The Rolls-Royce/Allison 601-R

The relatively small gas turbine in this study is the 601-R. Although the recuperated engine has not been designed yet the performance was modelled by Allison. An estimation of the overall dimensions based on an existing module for the 601-KF9 is given below.

The 601-KF9 (see figure 22) is a simple cycle gas turbine capable of delivering about 6,500 kW for ISO no loss conditions.

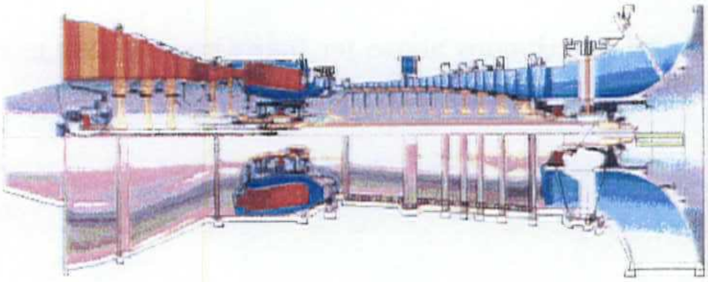


Figure 22: The 601-KF9

Key features of the Rolls-Royce/Allison 601-K (KF9 and KF11) development programme are:

- 6,500 to 9,000 kW class
- 34+% thermal efficiency
- extensive cost reduction
- 601-KF9/KF11 available in 2000

As opposed to the WR-21 recuperator which is designed and manufactured by Allied Signal, the 601-R recuperator will be developed by Rolls-Royce. The Allied Signal recuperator is a plate fin heat exchanger. The 601-R recuperator on the other hand will be a spiral-wound type heat exchanger. It is claimed that this type can achieve higher efficiencies at lower weights. A spiral recuperator consists of a number of spiral units as displayed in Figure 23. These cylindrical units have 670 mm diameter and are approximately 1,100 mm high.

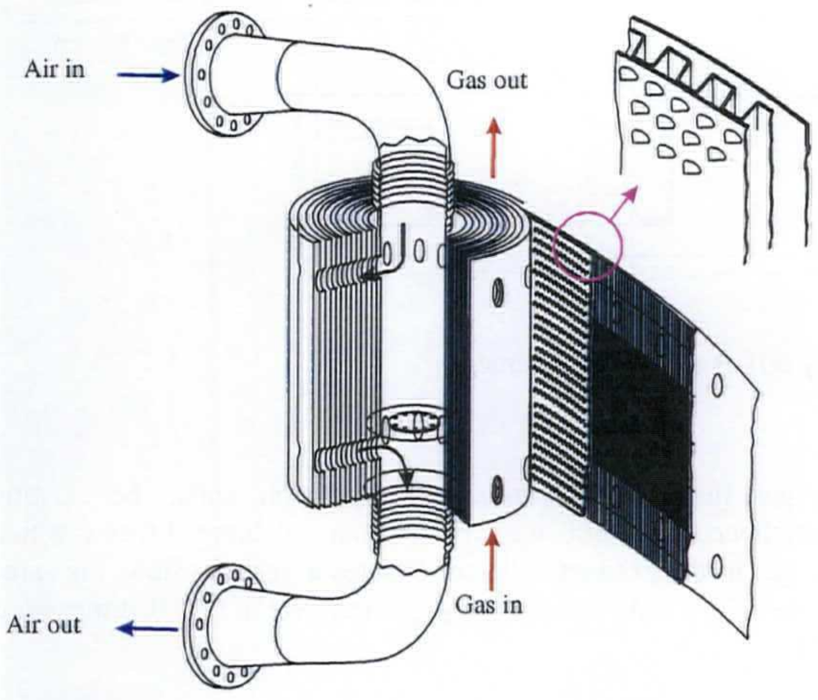


Figure 23: Spiral recuperator unit

The number of required units depends on the exhaust gas properties which of course depend on the engine power. For example: a 1,500 kW engine would require a recuperator embodying 6 units whereas a 22,000 kW gas turbine would require 16 to yield an optimum compromise between high efficiency and small size.

The amount of units suitable for the 601-R engine varies from 5 to 12. The corresponding effectiveness varies from 69 to 78%, which can be further increased by increasing the amount of pins in each unit. By doing this, the heat exchanging surface is increased. This improves the efficiency, but also increases pressure losses.

A total amount of 8 units is chosen regarding the assembly length in comparison with the gas turbine enclosure dimensions. A 2x4 configuration yields the following recuperator dimensions: 4.2x1.6x2.0 metres. This includes space required for the support structure, bellows and piping. A preliminary arrangement, including a salient pole 1,800 rpm generator is shown below.

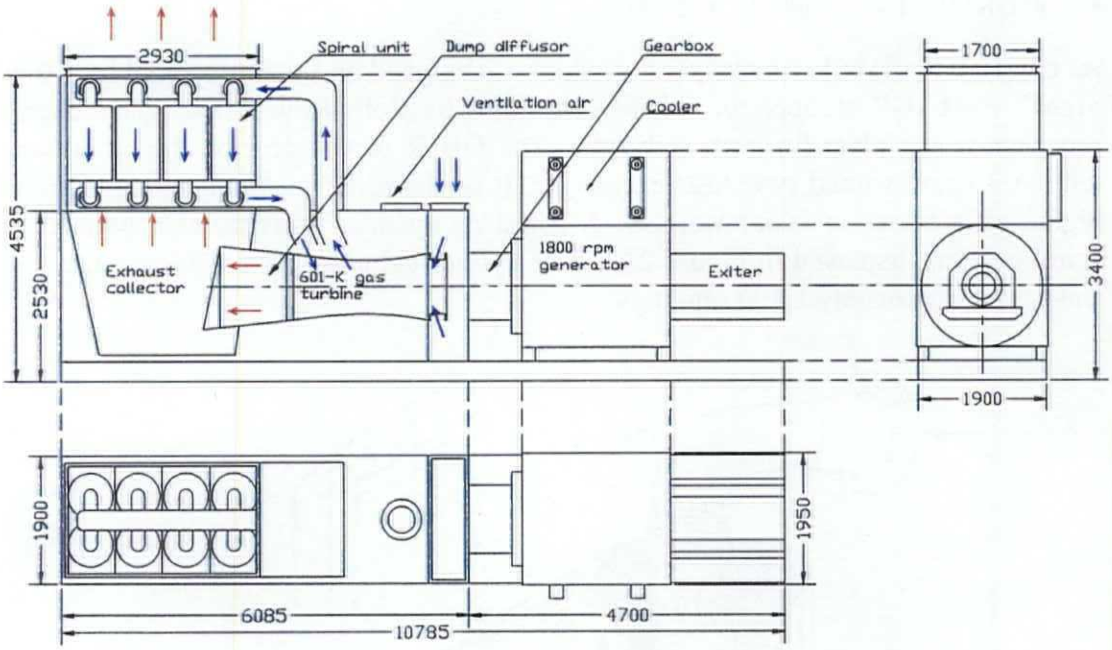


Figure 24: Preliminary 601-R genset arrangement

To obtain a uniform gas flow distribution across the spiral units, the existing enclosure has been lengthened by 1,500 mm. Furthermore a dump diffuser which discharges the exhaust gas in the exhaust collector ensures a uniform flow, the same principle as applied in the WR-21. As a result the predicted overall 601-R dimensions are: 6,085x1,900x4,535.

6. Prime mover configuration selection

The sea trial results were shown in section 3.1: **Power -speed curves**. This section shows the required shaft power with varying ship speed. With the losses in the electric propulsion system, the power demand from the generators will be higher. The efficiency curve from the electric propulsion system[14] is shown in Figure 25 below. The curve takes into account losses in cables, transformers, synchronous converters and motor. Although this curve belongs to the electric propulsion system from *Sun Princess*, it is assumed to be similar for the other ships and will therefore be used as generic in this study.

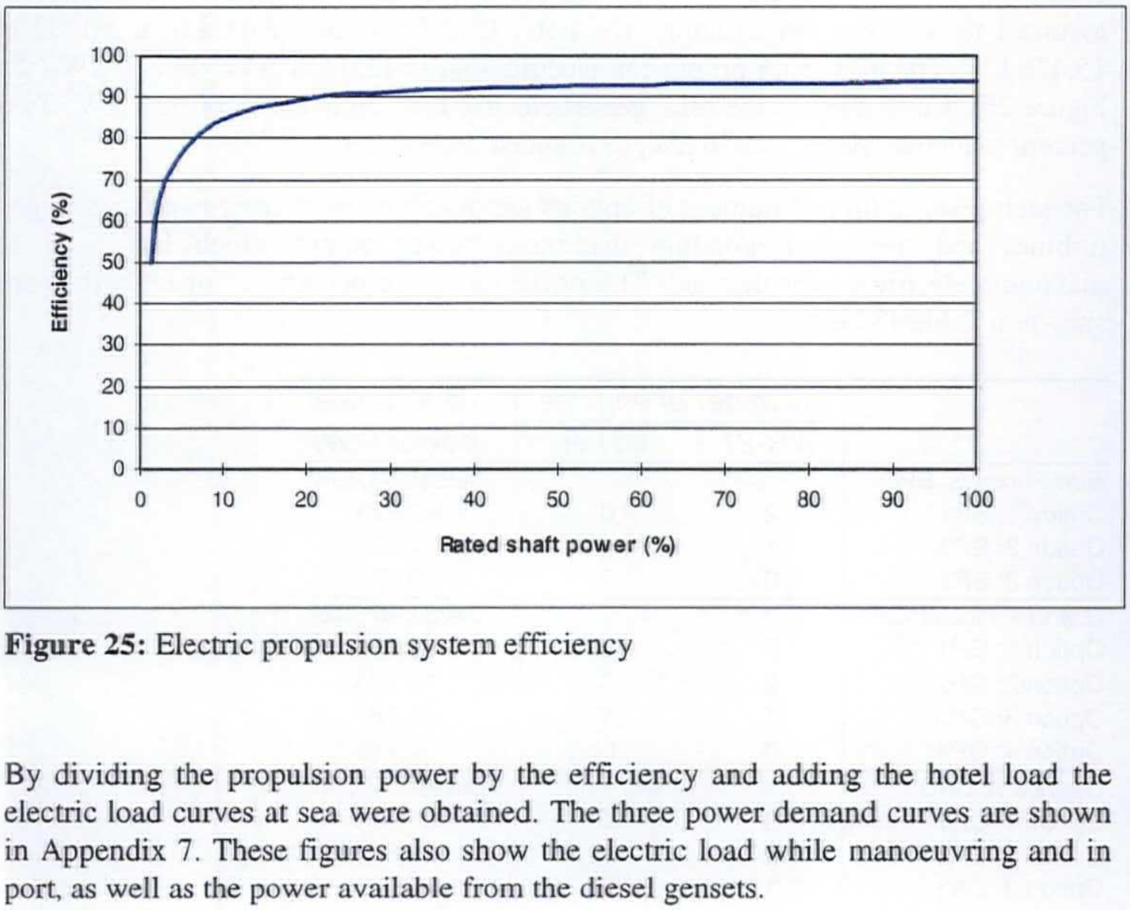


Figure 25: Electric propulsion system efficiency

By dividing the propulsion power by the efficiency and adding the hotel load the electric load curves at sea were obtained. The three power demand curves are shown in Appendix 7. These figures also show the electric load while manoeuvring and in port, as well as the power available from the diesel gensets.

At this point a gas turbine configuration selection has to be made in order to compare this power plant with the existing diesel-electric plants. In Table 12 the maximum electric load and the maximum gas turbine power at the generator terminals (generator efficiency of 97%) for conditions in practice are summarised.

[14] [van Lier, 1997]

	Sun Princess	*Grand Princess*	*Capricorn*	
Maximum electric load	38,800	59,000	54,200	kW
WR-21 genset power	21,240	kW		
601-R genset power	5,700	kW		
Ambient temperature	30	°C		
Relative humidity	90	%		
Ambient pressure	1,013	bar		

Table 12: Key properties for a gas turbine configuration determination

A two percent electric load reduction (see section 7.1.1: **Electric load reduction**) due to the lack of diesel auxiliaries is taken into account here. The WR-21 generator efficiency is assumed to be 97% whereas the 601-R generator/gearbox efficiency is assumed to be 95%. For example: the hotel load for *Grand Princess* at 30 °C is 15,476 kW. The maximum propulsion electric load is 42,000/0.94 = 44,680 kW (see Figure 25). Consequently the total genset electric load amounts to 60,156 kW. Two percent reduction yields 58,976 kW, or rounded 59,000 kW.

For each vessel a limited number of options are possible considering two types of gas turbines and their corresponding maximum power output which has to meet maximum electric power demand. The prime mover compositions for each ship are shown in Table 13 below.

	Number of engines		*Total genset*
	WR-21	*601-R*	*power (kW)*
Sun Princess: SPD			Diesel: 44,698*
Option 1: SP1	2	0	42,523
Option 2: SP2	1	4	44,076
Option 3: SP3	0	7	39,925
Grand Princess: GPD			Diesel: 67,046*
Option 1: GP1	3	0	63,784
Option 2: GP2	2	3	59,634
Option 3: GP3	1	7	61,187
Option 4: GP4	0	11	62,740
Capricorn: CAD			Diesel: 57,036*
Option 1: CA1	3	0	63,784
Option 2: CA2	2	3	59,634
Option 3: CA3	1	6	55,483
Option 4: CA4	0	10	57,036

Table 13: Gas turbine configuration options

*: For most cases the installed diesel power is higher than the gas turbine power. It should be emphasised that the installed diesel engine power displayed here is valid for ISO standard conditions. Strictly, this value will be slightly lower for the above mentioned conditions. This is not taken into account here because this effect is negligible compared to gas turbines. Moreover the maximum electric load is two percent lower .

The 11 different gas turbine configurations form the basis of the comparisons to be made in this study.

The annual running hours based on the operating profiles in chapter 4, the associated propulsion power and the hotel load (see chapter 3) are shown in Figure 26. Also displayed in this figure is the amount of engines. For example: configuration SP2 (*Sun Princess,* configuration 2) consists of one WR-21 running 7,700 hours per year and four 601-R gas turbines each running on average 1,600 hours per year.

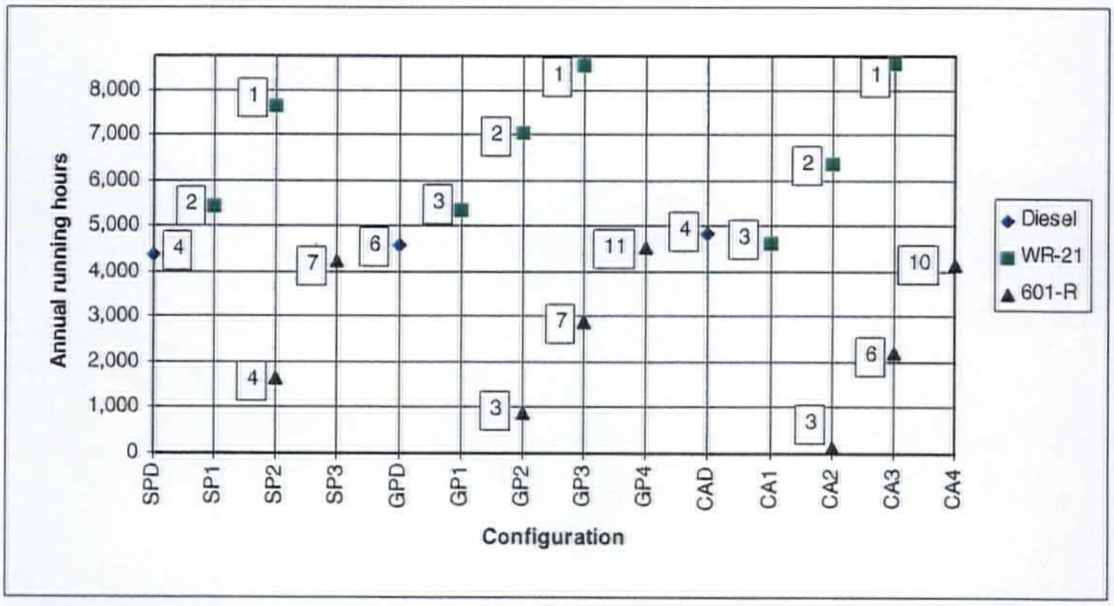


Figure 26: Annual engine running hours

7. Diesel - gas turbine power plant comparisons

This chapter presents the comparisons of the 3 diesel-electric and the 11 gas turbine-electric prime mover configurations, including relevant auxiliaries. The comparisons are subject to the following restrictions and assumptions:

- On several occasions the diesel-gas turbine comparison will only involve the WR-21. This is due to the fact that the development of the 601-R has not yet reached the same stage as the WR-21 development.
- Although a few simple prime mover layouts are presented in this chapter, it was clearly not possible for the author to overview the implications on the entire ship layout and superstructure. Obviously the results of the comparisons differ for every (re-)design of the ship layout.
- The costs and prices are converted according to the following exchange rates:

Pound Sterling ⇒ US Dollar	1.60
Dutch Florin ⇒ US Dollar	0.48
Deutschmark ⇒ US Dollar	0.56
Italian Lire (100) ⇒ US Dollar	0.06
Finnish Mark ⇒ US Dollar	0.19
French Franc ⇒ US Dollar	0.18

7.1 Electric load and heat load

In this section a comparison of both electric and heat load demand will be evaluated. As mentioned before the electric load will be lower due to the lack of diesel auxiliary systems. The steam load is lower due to the lack of heavy fuel preheating systems. With a diesel plant, the steam load is reduced by using engine jacket cooling water as heating medium. This heat is not available from a gas turbine. An intercooled engine however rejects a significant amount of heat through its intercooler. Utilisation of this waste heat if further investigated in chapter 8.

7.1.1 Electric load reduction

The following diesel auxiliary electric motors from *Grand Princess'* electric load analysis were taken into account:

- Fuel oil purifiers (6 x 29.6 kW)
- Diesel genset fresh water nozzle pumps (6 x 4.3 kW)

Voor 1 motor

- FW central pumps (3 x 50.5 kW)
- Diesel genset HT FW pumps (6 x 70.1 kW)
- Diesel genset LT pumps (6 x 75.3 kW)
- Engine room main fans (2 x 322 kW)
- Diesel genset LO pumps (6x23.9 kW)
- Total: 2,014 kW

The main engine room fans are taken into account here, because of the basic difference in air intake systems for diesels and gas turbines. Gas turbines do not need fans to provide the combustion air.

The WR-21 requires independent AC power supplies for the engine/enclosure, the Uniterrruptible Power Supply (UPS), the intercooler off engine module and enclosure ventilation:

Voor 1 gast

- Engine/enclosure: lights (770 W)
- Fire control system (150 W)
- UPS (2 kW, maximum)
- Intercooler off engine module: FW pump motor (37 kW)
- Intercooler off engine module: bypass valve motor (600 W)
- Axial flow fan for enclosure ventilation (40 kW)
- Total: 79.3 kW

The total amount of WR-21 engines to meet the total power demand is three. Consequently the electric load reduction amounts to $2{,}014 - 3 * 79.3 = 1{,}776$ kW. The electric load for a 601-R engine is assumed to be far less since it does not incorporate an intercooler which represents the majority of required electric power (approximately 95%). The lub oil pump is engine mounted.

Therefore *Grand Princess'* electric load reduces by 1,780 kW when using gas turbines for electrical power supply. The total installed electric power (users and motors) is 89,000 kW, so the reduction amounts to 2%. It is assumed that this is a generic figure for the selected cruise ships and that the hotel load decreases by the same amount.

7.1.2 Steam load implications

From the steam load balances shown in Appendix 3, can be seen that the percentage of the total steam demand used for heavy fuel preheating is on average 20% in summer and 28% in winter conditions. Referring to the advantages for gas turbines as listed in the Introduction it can be concluded that the heat or steam load, due to the lack of a fuel preheating system for various conditions is reduced by the same amounts.

The steam required for heavy fuel preheating for summer and winter conditions is displayed below.

	Sun Princess		Grand Princess		Capricorn		
	Winter	Summer	Winter	Summer	Winter	Summer	
At sea	4,030	3,531	5,660	4.305	2,041	1,846	kg/h
In port	3,012	2,657	3,760	3,115	1,399	770	kg/h
Manoeuvring	4,030	3,531	5,660	4,305	2,041	1,846	kg/h

Table 14: Steam load for heavy fuel preheating

The steam load balances without heavy fuel preheating steam consumption and therefore valid for a gas turbine ship, are displayed in Table 15.

	Sun Princess		Grand Princess		Capricorn		
	Winter	Summer	Winter	Summer	Winter	Summer	
At sea	17,101	11,124	22,660	10,665	11,515	7,995	kg/h
In port	15,001	9,024	15,380	7,715	10,975	7,105	kg/h
Manoeuvring	17,101	11,124	22,660	10,665	11,515	7,995	kg/h

Table 15: Steam load without heavy fuel preheating and evaporators

When the steam production is higher than the steam demand, the amount of surplus steam will be used to feed the evaporators. Whether this is enough to meet the fresh water demand will have to be determined by examining the engine load profiles and the associated steam availability. It can then be determined whether additional oil fired boilers are necessary. This is conducted in the next section.

7.1.3 Meeting the steam demand

Obviously dependent on the specific design of a Waste Heat Boiler (WHB) a certain amount of (saturated) steam can be generated from the energy left in the exhaust gases.

The WR-21 and 601-R feature a recuperator bypass mode, which can be used to increase the steam production. When the recuperator is bypassed, the exhaust gas temperature rises which allows the WHB to produce more steam as efficiency increases. The current design admits either recuperator fully operative or recuperator fully bypassed. If a continuously variable bypass ratio was practical, implying the mechanical and thermal stress was engineered down to acceptable levels, then a variable steam demand could be easily met.

The only problem with intermittent operation of the recuperator is the fact that the recuperator has to deal with more thermal stresses than it is designed for. To meet this problem a 90% maximum steam production limit is introduced. In that case the recuperator is only 90% bypassed and a small amount of compressed air flows through as a coolant. The maximum waste heat boiler steam production will be based on this 90% recuperator bypassed.

The effect on the steam production is demonstrated in Figure 27. This figure shows the steam production with engine load[15] for conditions in practice. The maximum steam production from several suitable boilers are shown in Appendix 8.

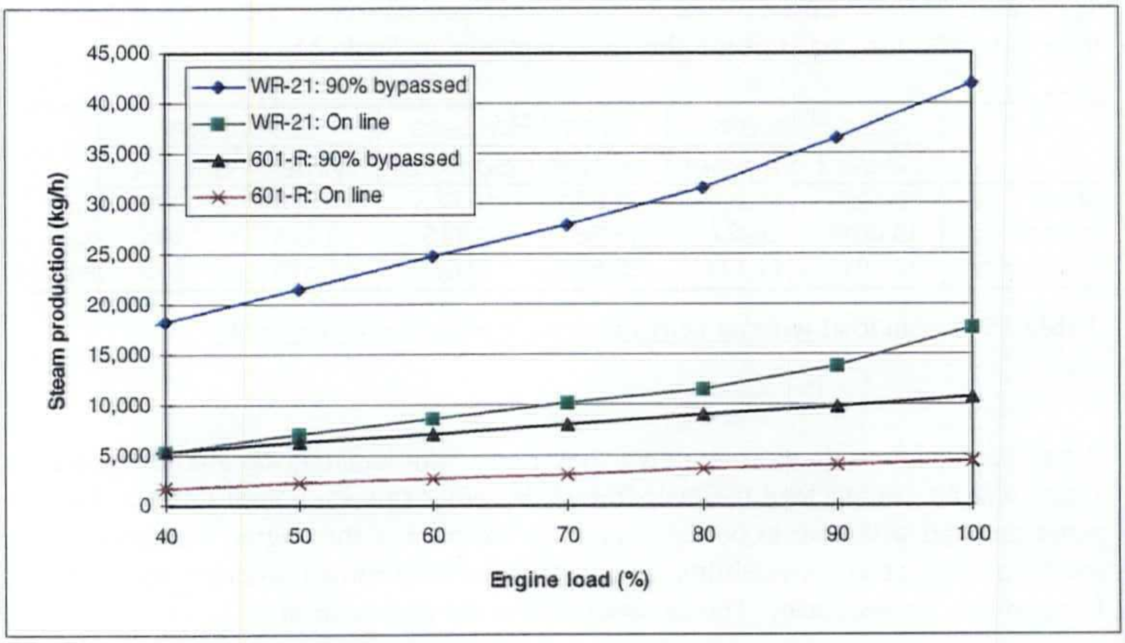


Figure 27: Gas turbine WHB steam generating capacity

The associated exhaust gas properties are shown in Appendix 9.

Other circumstances for which the engine will be operated with the recuperator bypassed are in case of recuperator cleaning and recuperator damage.

The effects of bypass operation on engine performance are:

- increase in fuel consumption
- increase in exhaust gas temperature
- no restriction on power capability

[15] As calculated by Thermal Engineering International Ltd, a marine boiler manufacturer

The increase in specific fuel consumption is shown in Figures 28 and 29 below. Each figure displays two specific fuel consumption curves: one for recuperator on-mode and one for recuperator bypassed-mode. The 601-R curve was predicted by Allison.

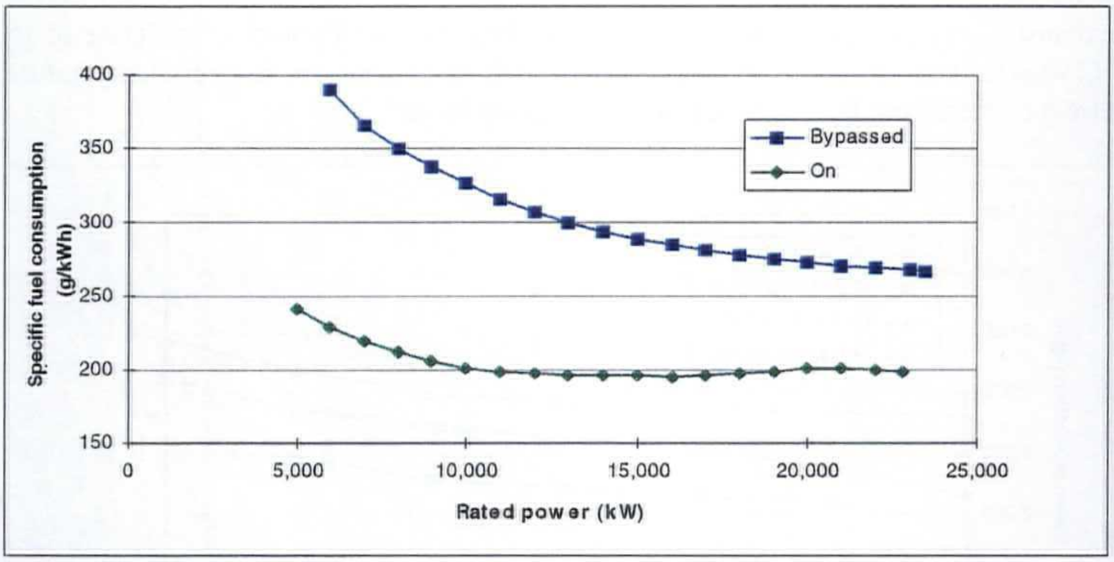


Figure 28: WR-21 specific fuel consumption: recuperator on and 90% bypassed

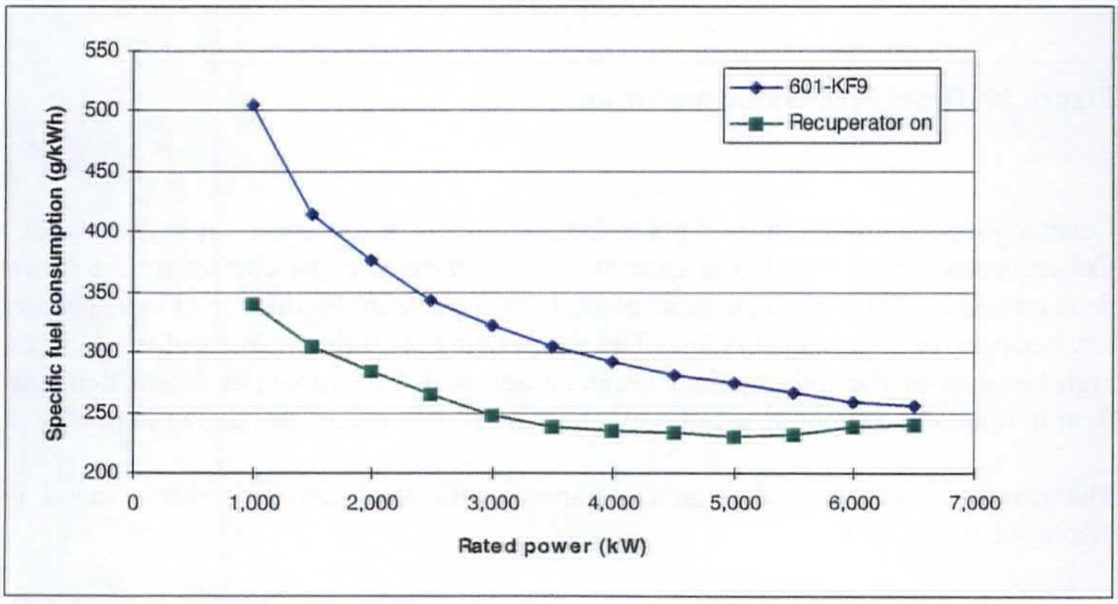


Figure 29: 601-R specific fuel consumption: recuperator on and bypassed

The steam production curves for the 16 ZAV40S and the 14V 48/60 (see Figure 30) are displayed for 25 °C and 45 °C ambient temperature. Again, the associated exhaust gas properties are shown in Appendix 9.

The ambient air temperature influences the exhaust gas temperature which in its turn influences the waste heat boiler efficiency and therefore the steam generating capacity. A 10 °C ambient temperature rise typically increases the exhaust gas temperature with 15 °C for diesels and only 6 °C for gas turbines. Moreover the exhaust gas properties for the gas turbines are based on an ambient temperature of 15 °C whereas this value is 25 °C for diesels. For these reasons, the ambient temperature effect on steam production is neglected for gas turbines.

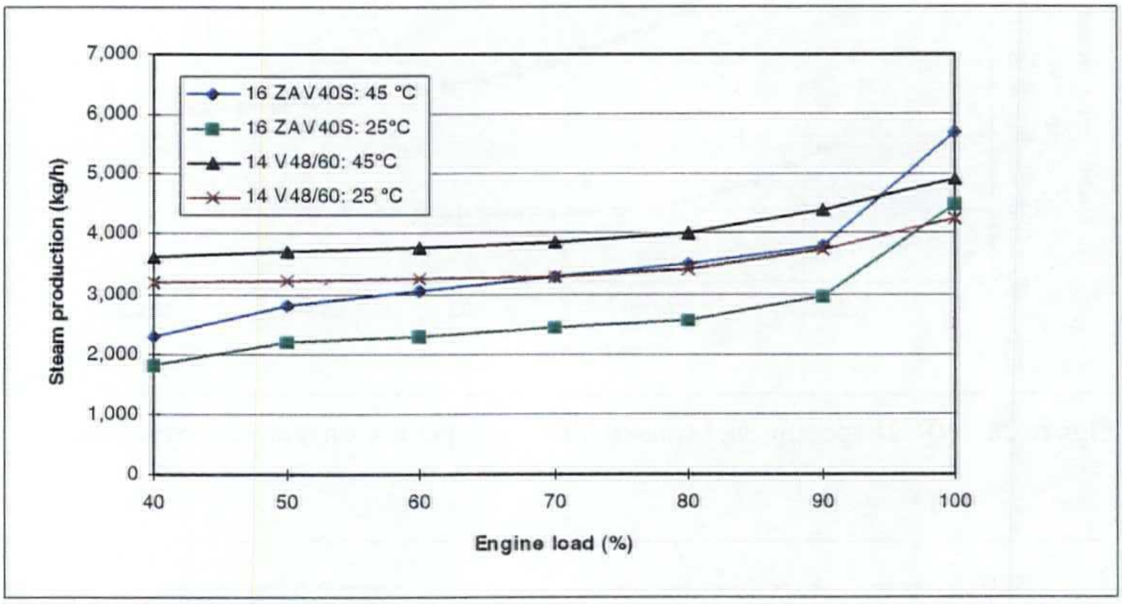


Figure 30: Diesel WHB steam production

Generally the maximum oil fired boiler load will occur at low speed (at sea) in winter. Consequently the oil fired boiler capacity can be determined by comparing the steam demand and availability under these conditions. The steam production at low speed is low because the engine load is low. The steam demand on the other hand is relatively high because of the low ambient temperatures and the evaporator steam demand. Bear in mind that evaporators can only produce fresh water at sea and not in port.

The maximum steam production comparison with total engine power demand is displayed in Figure 31.

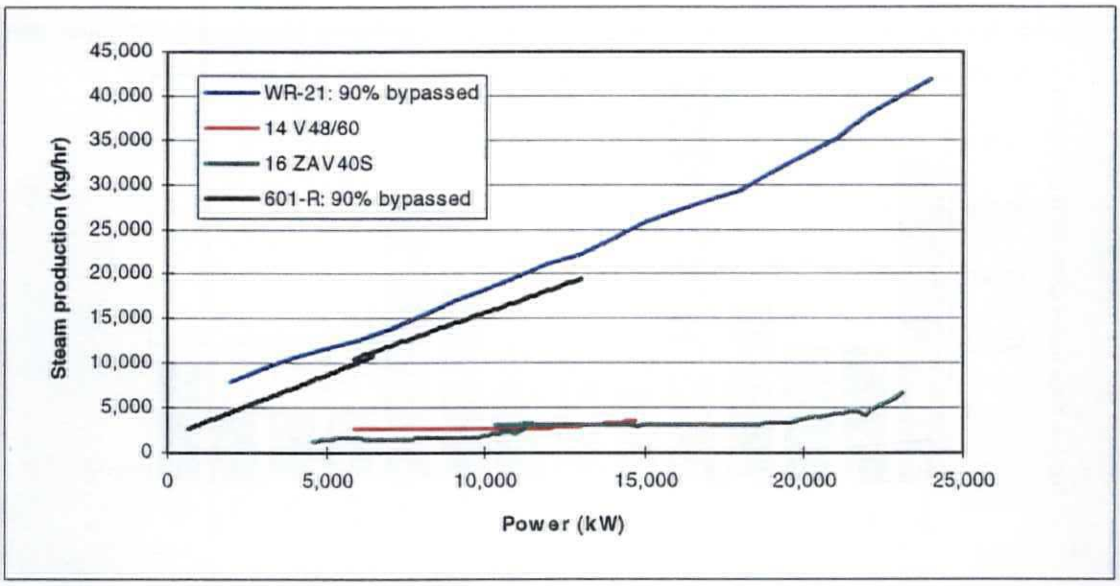


Figure 31: Maximum steam production comparison (recuperator 90% bypassed)

At low speed in winter the electric load is typically 7,800 kW on *Sun princess*, 12,900 on *Grand Princess* and 8,600 on *Capricorn*. As a result, the waste heat available from the engines is relatively low. For this condition it is assumed that the amount of waste heat from the cooling water is negligible. Thus the heating medium for the evaporators is steam only. The steam productions at the low engine loads are presented in Table 16 below. The gas turbine steam production is valid for 'recuperator bypassed' mode. This will increase the fuel consumption.

	Steam production (kg/h)			*Difference (kg/h)*	
	Diesel(s)	*WR-21*	*601-R(s)*	*WR-21/Diesel*	*601-R/Diesel*
Sun Princess	1,600	16,000	13,000	14,400	11,400
Grand Princess	3,100	24,000	19,000	20,900	15,900
Capricorn	2,800	18,000	14,000	15,200	11,200

Table 16: Maximum steam production comparison at low speed in winter

As a result the oil fired boiler maximum capacity can be reduced according to the steam production differences as displayed above. This yields oil fired boiler capacities as displayed in Figure 32. The implications on fuel consumption, boiler dimensions and first cost are investigated in the relevant sections later in this chapter. It has to be emphasised that this is an extreme condition, only adopted to estimate the oil fired boiler capacity. Summer conditions usually will not require oil fired boiler assistance, with gas turbines. This was investigated with a spreadsheet. On a number of occasion however, the recuperator has to be bypassed to meet the stem demand.

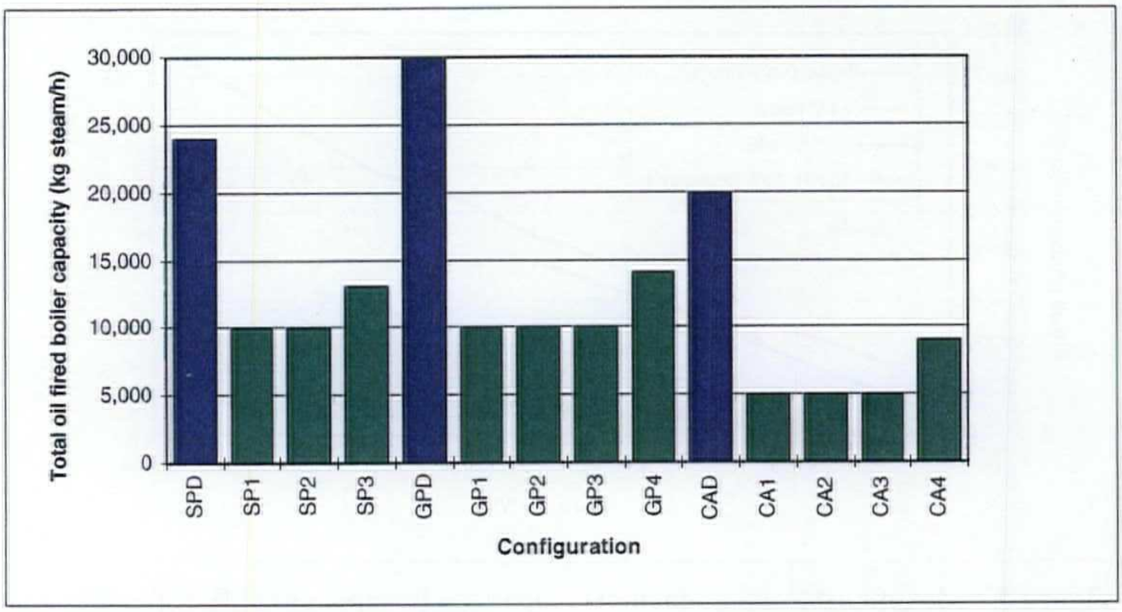


Figure 32: Oil fired boiler total capacity

7.2 Initial cost

In this chapter a comparison of diesel and gas turbine initial cost will be made. Initial cost, also referred to as capital, first or investment cost, includes acquisition and installation. The prices or costs mentioned in this chapter should be regarded as average budget prices. Even acquisition costs quoted by manufacturers, should still be regarded as an indication, simply because of the fact that the costs depend upon the scope of plant equipment, geographical area, special site requirements, etc.

7.2.1 Genset acquisition cost

An overview of the obtained diesel and gas turbines acquisition cost is given in Figure 33 and 35. It should be noted that the acquisition cost is based on total a package, including enclosure, baseframe, lub oil systems and controls. Compared to advanced cycle, simple cycle engines are typically 25-30% cheaper which is demonstrated in Figure 35.

Diesel genset acquisition cost will be split up in three: engine cost, generator cost and auxiliary cost. Diesel engine specific acquisition cost tends to increase with cylinder diameter. This is demonstrated in Figure 33. The price of a 320 mm bore engine is typically only 70% of an equal powered 500 mm bore engine[16]. This is confirmed by Figure 33.

[16] [Nurmi, 1998]

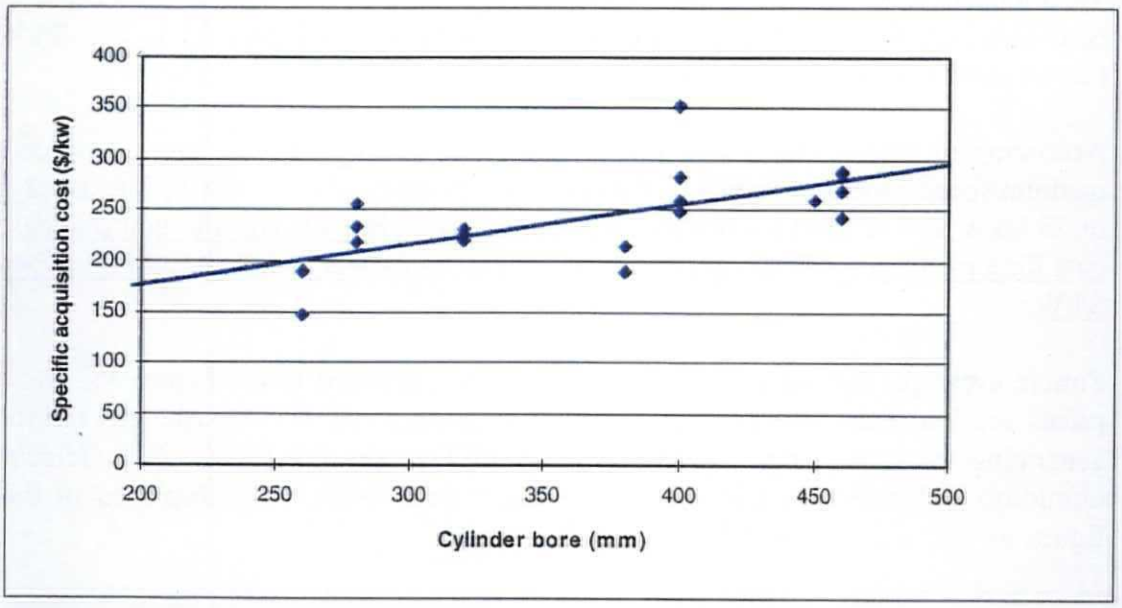


Figure 33: Diesel specific acquisition cost against cylinder bore

Figure 34 shows the specific unit purchase cost for several medium speed, large bore diesel engines with cylinder bore ranging from 380 to 460 mm.

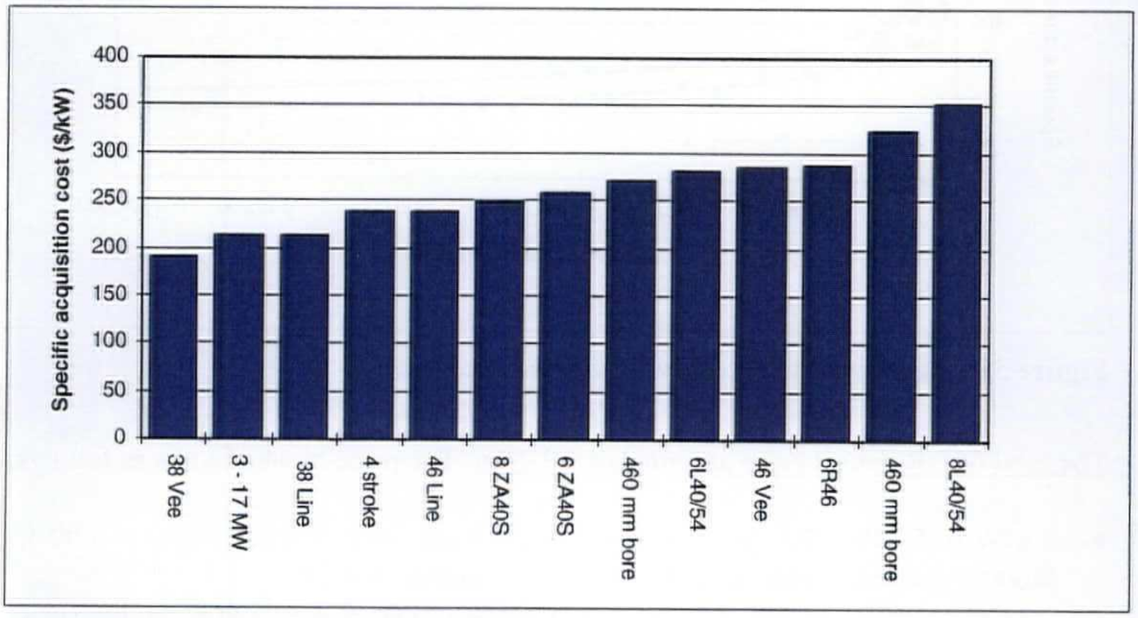


Figure 34: Diesel engine specific acquisition cost

The numbers were obtained from several different sources, not mentioned because of confidentiality reasons. The average prime mover price in Figure 34 is 265 $/kW (diesel auxiliaries and generator excluded).

According to among others manufacturer quotations, the unit purchase costs for a medium speed generator (typically 500 rpm) varies from 55 to 95 $/kW. The average of 75 $/kW will be used for further calculations. Therefore the specific unit purchase cost for a medium speed diesel genset, no auxiliaries (power range: 5-19 MW) is 340 $/kW.

Simple cycle gas turbine genset price levels[17] are presented in the Figure 35. These prices are budgetary average equipment-only price levels for a basic gas turbine generating package. Engine ratings are at ISO conditions (15°C, 60% relative humidity) and based on zero inlet and exhaust duct losses. Also displayed in this figure are prices for the WR-21 and 601-R.

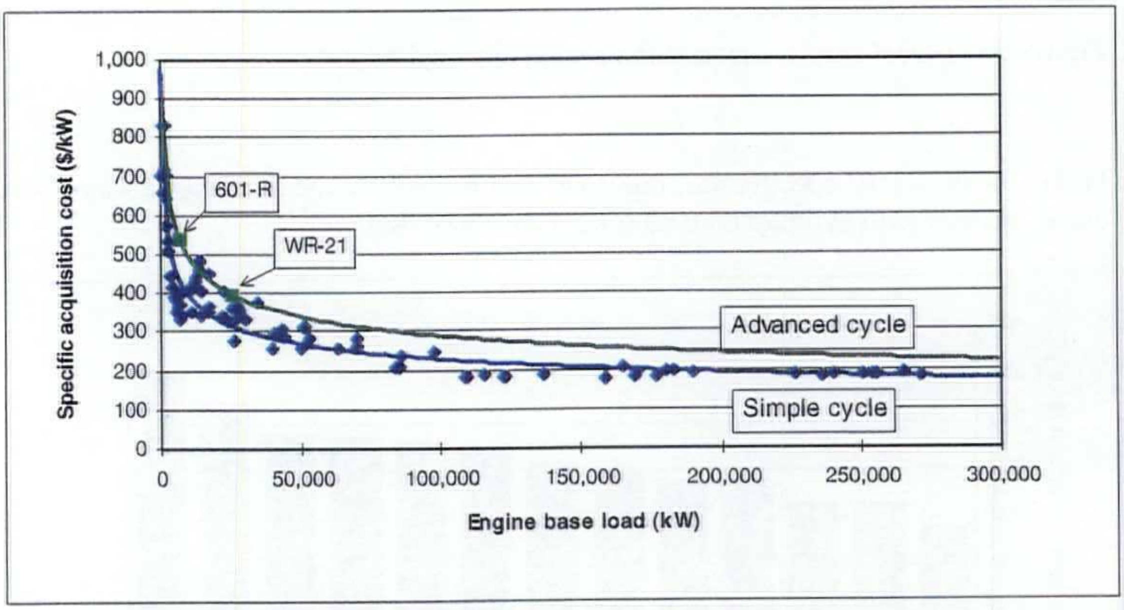


Figure 35: Gas turbine first cost with engine base load

The total 601-R genset price amounts to $ 3.73m. The price breakdown is as follows:

- 6 MW (6.5 MW ISO, no losses) 601-R package, including gas turbine, Lub Oil Module (LOM), controls, enclosure and baseframe: $ 3.21m.
- 7 MW gearbox, 11,500 to 1,800 rpm (i = 6.39): £ 80k - 90k, budget price. Assume worst case: £ 90k, or $ 144k
- 7 MW high speed generator, 1,800 rpm: £ 190k or $ 300k[18].
- gearbox/generator baseframe: roughly $ 76k (maximum).

[17] [Gas Turbine World, 1997]
[18] Quotation form Brush, May 1998

The total WR-21 genset price amounts to approximately $ 10m, with a price breakdown as follows:

- 22,000 kW (25,240 kW ISO, no losses) WR-21 module, LOM, off engine heat exchanger and control systems included: £ 5.5m, or $ 8.69m.
- 25.5 MVA high speed generator, 3600 rpm: £ 785k or $ 1.24m[19].
- generator baseframe: roughly $ 70k.

7.2.2 Auxiliary systems acquisition cost

The price breakdown for a 60 MW diesel-electric power plant, provided by Kvaerner Masa Yards includes a breakdown for the diesel auxiliaries. The cost in this table for diesels and generators are significantly lower compared to the values as found in the previous section. This may be possibly caused by exchange rate fluctuations.

60 MW diesel plant	*Cost ($m)*	*Specific cost ($/kW)*
Diesels	13.4	223.2
Generators	2.7	45.0
Machinery space ventilation	0.7	12.4
Fuel supply systems	0.3	5.6
Fuel + lub oil transfer and separating systems	0.5	7.8
Lubricating oil systems	1.2	20.2
Engine cooling systems	0.4	6.2
Compressed air systems	1.0	17.1
Uptake ducting	0.5	7.8
Machinery automation	0.2	3.7
Total	20.9	348.8

Table 17: Initial cost breakdown for diesel-electric plant

According to this price breakdown the total diesel auxiliary costs are 80.6 $/kW, or 30 % of the total plant price.

7.2.3 Installation cost

As will be illustrated in section 7.4.2: **Power plant dimensions**, the space required for engine auxiliaries is less for gas turbines. In consequence, the number of work hours for machinery installation is lower. As calculated by Kværner Masa Yards, on average 70 work hours per m^3 machinery deck space are required for installation. An indicative value for the average cost of one work hour is $ 40. The reduction in installation cost is shown in Figure 36.

[19] Quotation from Cegelec, September 1998

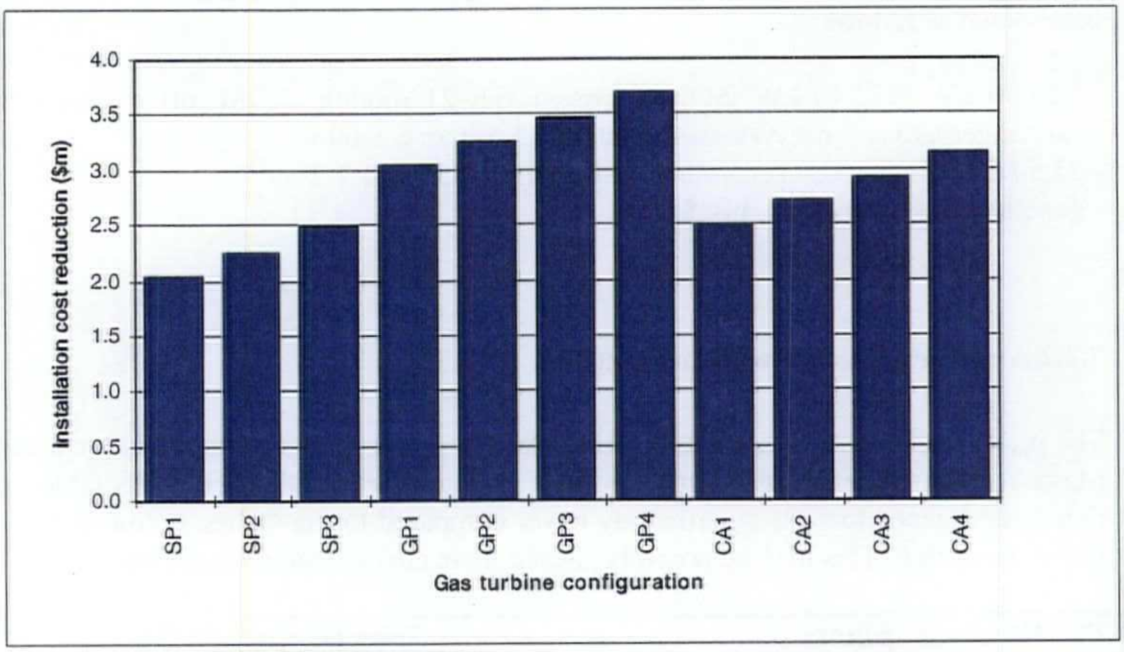


Figure 36: Installation cost reduction for the gas turbine configurations

7.2.4 Conclusions

As discussed before, the WR-21 package includes a lub oil module and an off engine heat exchanger/intercooler. Furthermore no additional internal cooling systems are necessary for engine cooling, except for the enclosure ventilation. However, this is considered a part of the machinery space ventilation system.

Therefore the specific costs for the lub oil and cooling systems will be added to the specific diesel engine costs. From Table 17 above the costs for the lub oil systems are 20.2 $/kW and the cooling systems add another 6.2 $/kW to the total.

As a result, the total specific acquisition cost for a medium speed diesel genset, with lub oil and cooling systems (power range: 5-19 MW) adds up to 360 $/kW.

For the WR-21 this figure is $ 400/kW whereas for the 601-R this is $ 575/kW; both at ISO no loss conditions.

Figure 37 compares the total genset first cost. This figure displays the eleven gas turbine options and the three existing diesel options from Table 13. The exact numbers are shown in Appendix 10.

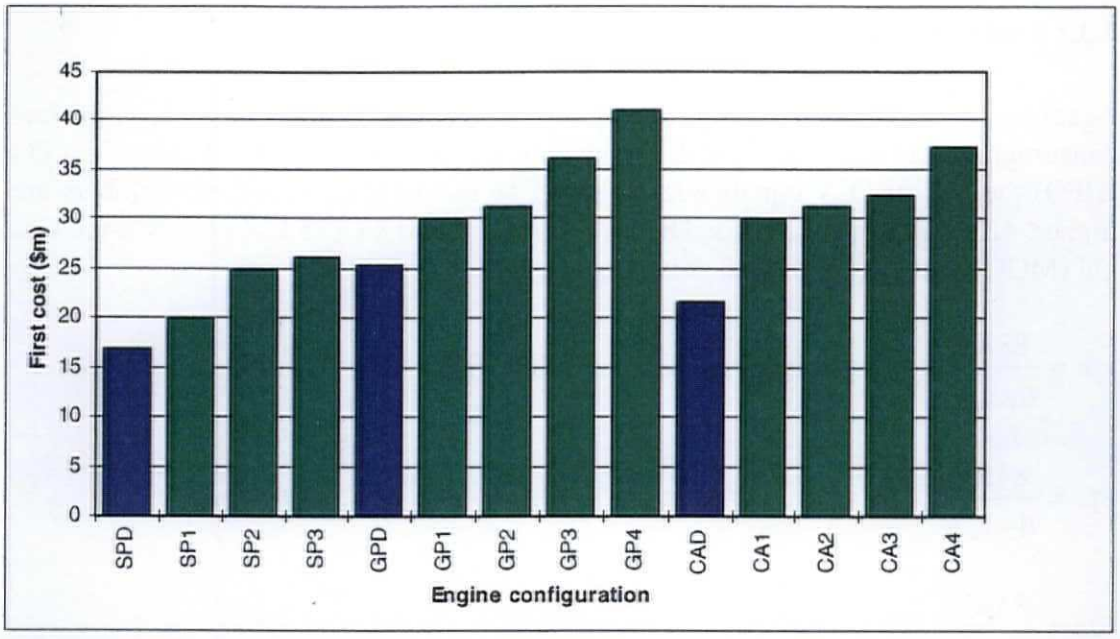


Figure 37: Genset acquisition cost comparison

7.3 Fuel costs

To make a proper specific fuel consumption/thermal efficiency comparison, the conversion from specific fuel consumption to thermal efficiency is shown below.

$$sfc = \frac{3{,}600 * 1{,}000}{LCV * \eta_{Thermal}}$$

where,

sfc = specific fuel consumption [g/kWh]
LCV = Lower Calorific Value [kJ/kg]
$\eta_{Thermal}$ = thermal efficiency [-]

The starting point of this comparison is that every engine has a more or less fixed efficiency, independent on the fuel type. This assumption is necessary to compare the specific fuel consumption when the engine uses different types of fuel. For example: instead of distillate fuel, diesels burn heavy fuel. The LCV-ratio is about 1.05 (42,700/40,500). The decrease in combustion efficiency is neglected here. The combustion efficiency will drop when burning heavy fuel, because the combustion temperature and pressure will generally be lowered slightly to protect the engine against the more aggressive fuel.

7.3.1 Fuel consumption

According to the conversion described above, the relationship between specific fuel consumption and efficiency for different fuel oils is as follows: for Heavy Fuel Oil (HFO) the typical LCV equals 40,500 kJ/kg, as for distillate fuel these numbers are higher: 42,700 kJ/kg for Marine Diesel Oil (MDO) and 43,125 kJ/kg for Marine Gas Oil (MGO). This yields the following equations:

$sfc = \dfrac{88.88}{\eta_{Thermal}}$, for diesel engines that burn HFO (FO 380),

$sfc = \dfrac{84.31}{\eta_{Thermal}}$, for diesel engines or gas turbines which burn MDO and

$sfc = \dfrac{83.47}{\eta_{Thermal}}$, for gas turbines that burn MGO

Table 18 compares the fuel consumption for optimum ISO conditions (engine load typically 80-85%, no losses) for several power systems according to the conversion described above.

Machinery type	*η (%)*	*sfc (g/kWh)*
Low speed diesel (HFO)	50 - 52	171 - 177
Medium speed diesel (HFO)	**46 - 48**	**185 - 193**
High speed diesel (MDO)	42 - 44	190 - 200
Gas turbine: aeroderivative simple cycle (MGO)	32 - 36	234 - 263
WR-21 (MGO)	**44**	**191**
601-R (MGO)	**38**	**219**

Table 18: Diesel - gas turbine efficiency comparison

As can be seen in table 18 the sfc of the medium speed diesel is still considerably lower than the WR-21 and the 601-R fuel consumption. The specific fuel consumption with rated power curves for these optimum conditions are shown in Figure 38 below.

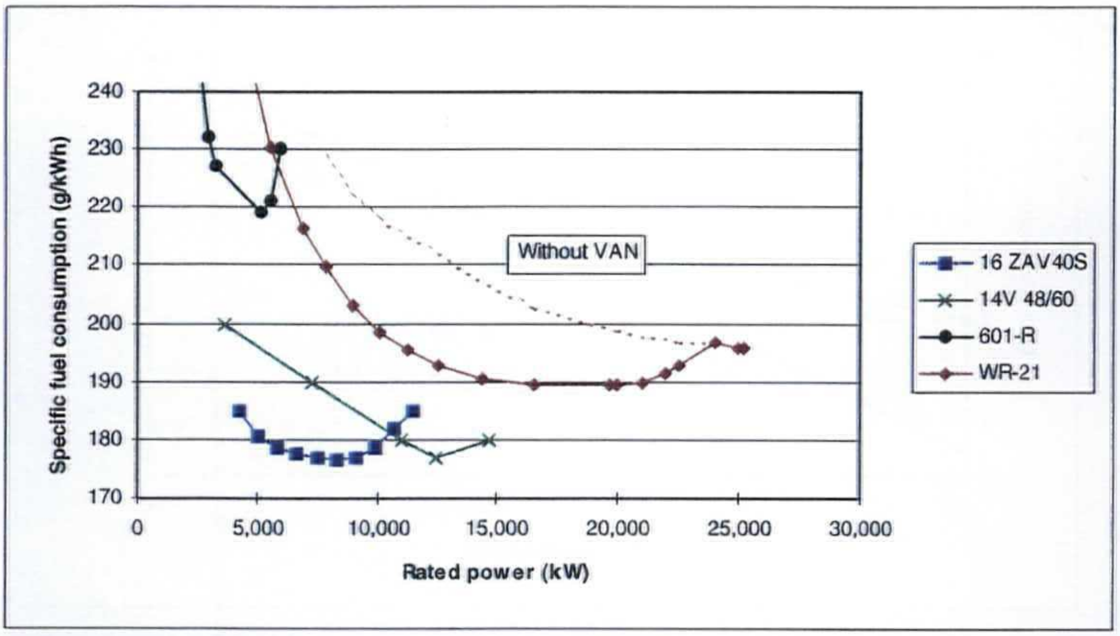


Figure 38: Specific fuel consumption curves at optimum conditions

The 'odd' shape of the WR-21 curve is caused by the operation of the VAN. The point of deflection at 24,000 kW represents the power at which the VAN are fully open. As a result, the sfc curve changes to the 'normal' curve for an engine without VAN. A gas turbine without VAN would typically have a fuel consumption curve as shown by the dotted line.

The curves are only valid for design or ideal tested conditions. Because we are interested in the consumption in practice, these curves will have to be adjusted for off-design conditions: varying ambient conditions (temperature) and the introduction of ducting losses.

For the diesel engines this is generally done by adding or subtracting a certain amount or percentage to the fuel consumption curves as obtained from the project guides. The addition/subtraction accounts for:

- Addition/subtraction: different ambient air and charge air cooling water temperature: according to the project guides from the ZA40S and the 48/60 the specific fuel consumption changes as displayed in Figure 39.
- Engine mounted pumps: according to the project guides the addition should amount to 1% (2 g/kWh) for each lubricating oil pump and 0.5% (1 g/kWh) for each water pump.
- Heavy fuel instead of distillate fuel burn: 5% addition, as explained in the introduction of this chapter.
- Tolerances in the project guide-curves: ± 3% according to the project guides.

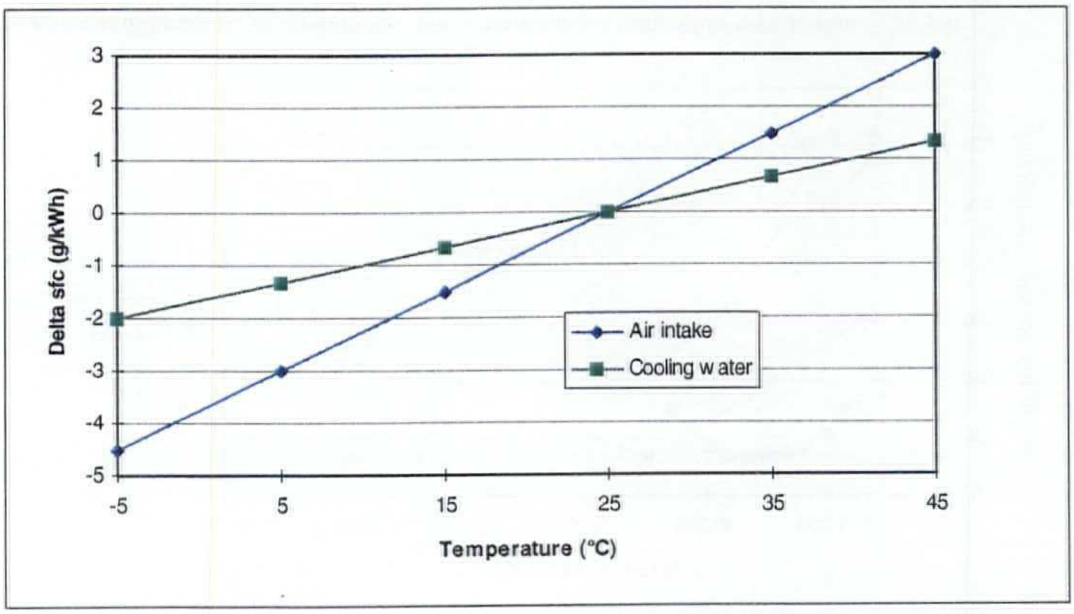


Figure 39: Specific fuel consumption variation with ambient temperature

The specific fuel consumption curves for the 16V ZA40S, as fitted in *Sun Princess* and *Grand Princess* and the 14V 48/60 as fitted in *Capricorn* according to the adjustments described above are displayed in Figure 40.

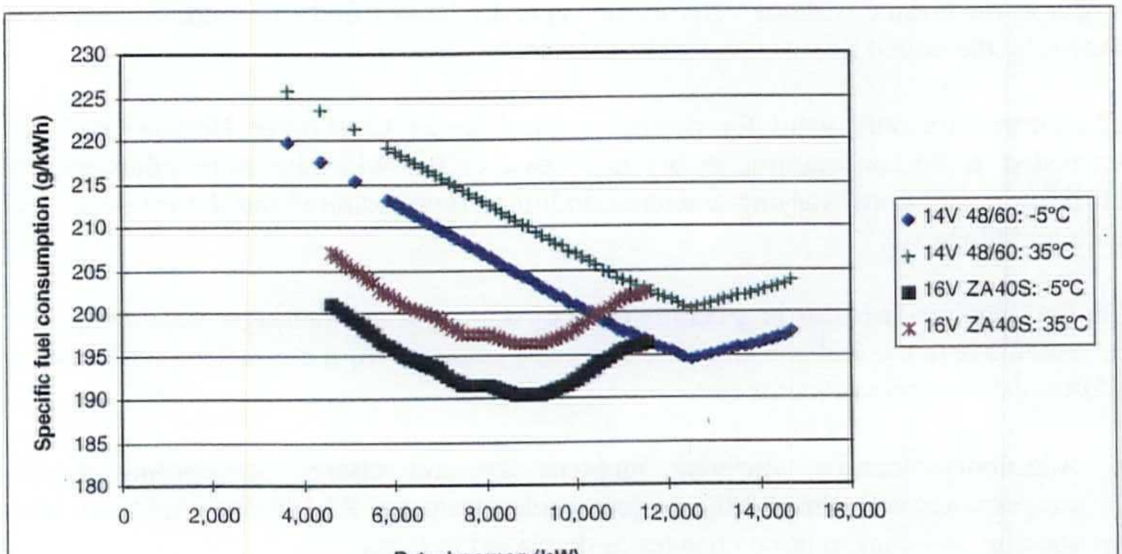


Figure 40: Diesel specific fuel consumption curves in practice

The WR-21 fuel consumption in practice was again obtained from the Westinghouse/Rolls-Royce WR-21 programme. The input parameters for this programme that determines the actual fuel consumption are illustrated below:

- Ambient air temperature (*°C*): the air intake temperature influences both specific fuel consumption and maximum power output (see also section 5.1: **The Northrop Grumman/Rolls-Royce WR-21**). This is also shown in Figure 41, which shows the specific fuel consumption curves for different air intake temperatures (60% relative humidity, 1,013 m bar, no losses)

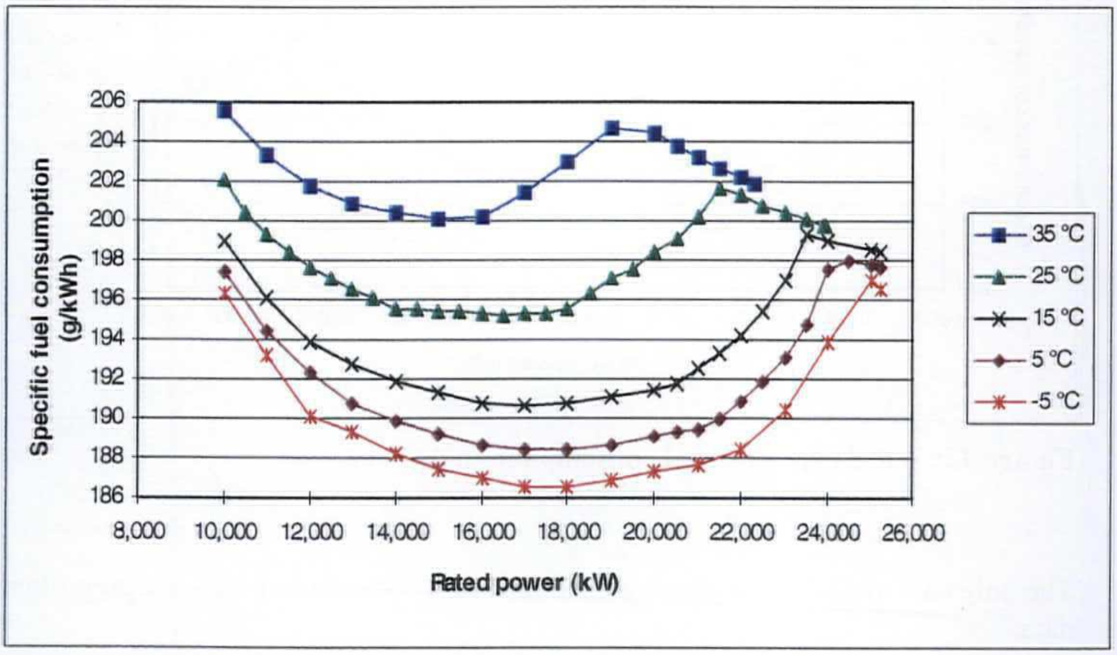


Figure 41: Air intake temperature influence on sfc and power output

- Ambient pressure (*m bar*): The specific fuel consumption, at maximum power varies only 2 g/kWh over/on the total range (950 up to 1,080 mbar), with the average fuel consumption at 1,013 m bar. It is assumed that the time the ambient pressure is either higher or lower is equally distributed.
- Relative humidity (-): Relative humidity influence is marginal. For example, the fuel consumption corresponding to a relative humidity of 0, 0.5 and 1: 198.5, 198.5 and 198.4 g/kWh. As discussed before 90% will be used in this study.
- Fuel lower heating value (*kJ/kg*): 42,700 for MDO and 43,125 for MGO, which will be used for the fuel consumption comparison.
- Inlet duct pressure loss at datum point (*mm wg*): 145 mm wg[20].
- Exhaust duct pressure loss at datum point, (*mm wg*): 185 mm wg[20].

[20] [van Lier, 1997]

At this point the specific fuel consumption curves for the actual cruise condition (ambient conditions as described above) can be constructed, according to the climates defined in the previous chapter. Five curves with temperature varying again from -5°C to 35°C., plus 3% tolerance are shown in Figure 42.

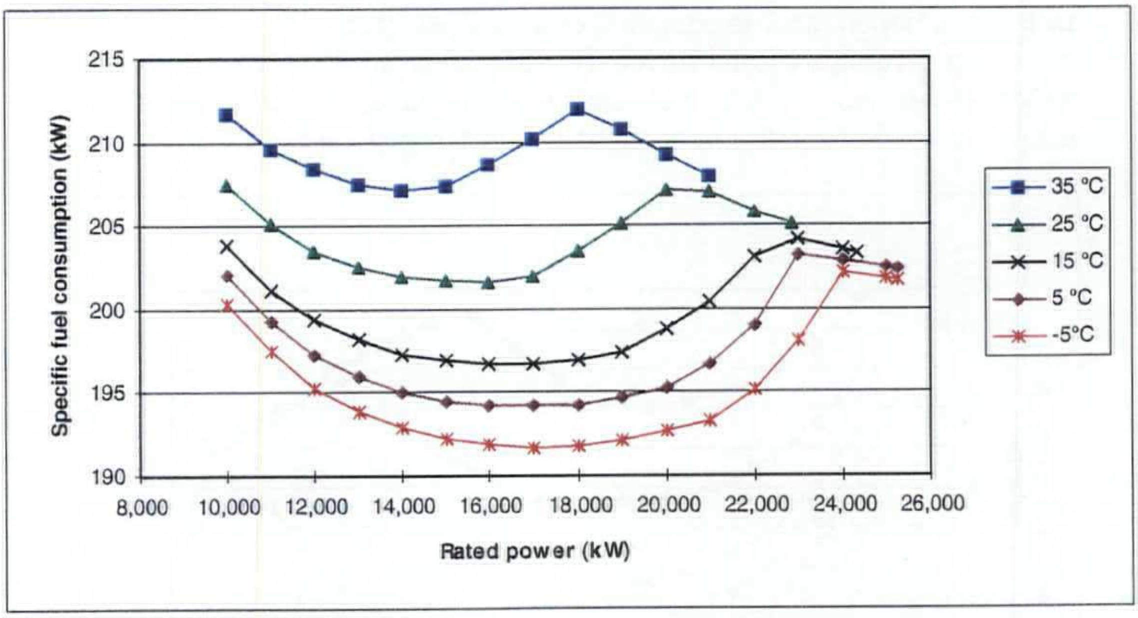


Figure 42: WR-21 specific fuel consumption in practice

The only data available for plotting the 601-R fuel consumption curve is preliminary data.

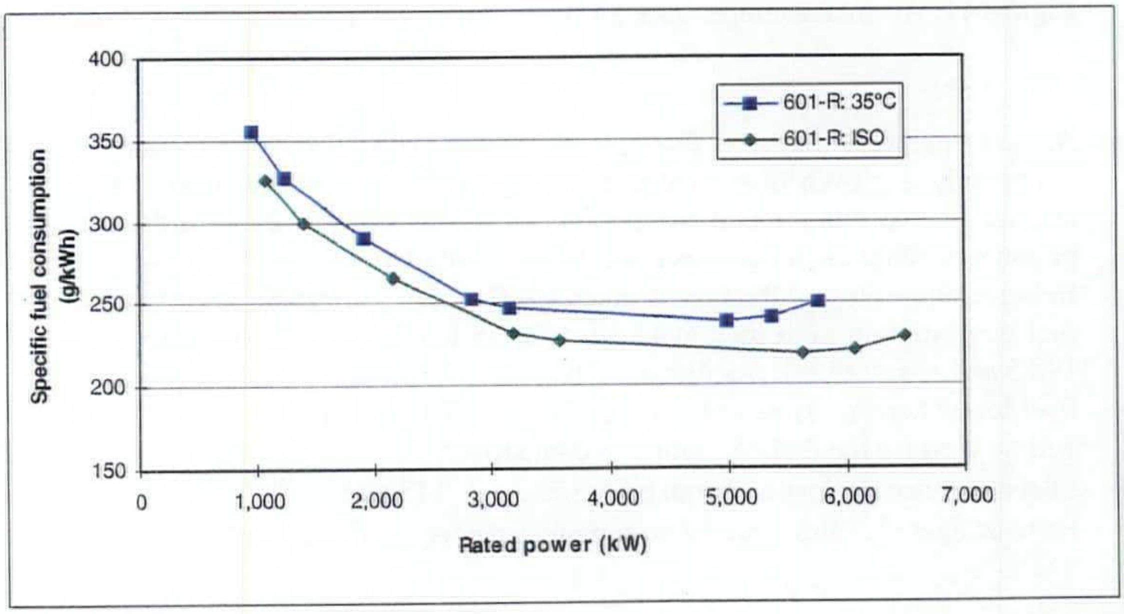


Figure 43: 601-R fuel consumption curves

Figure 44 compares the fuel consumption curves as described above. These curves will be used to the actual fuel consumption in practice.

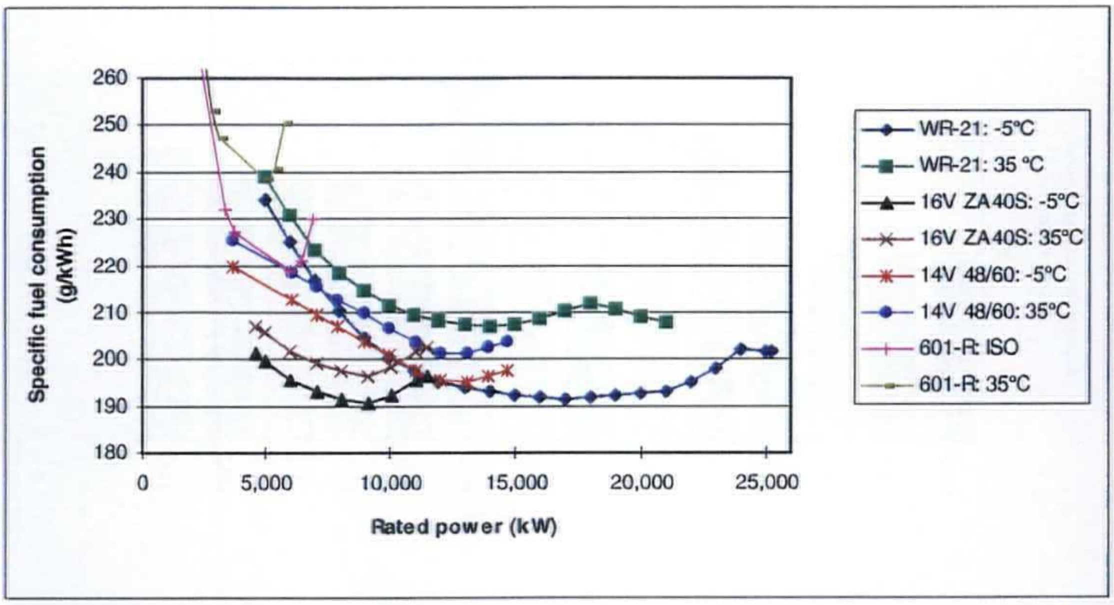


Figure 44: Specific fuel consumption comparison in practice

The vessel's cruise area and the associated ambient air temperatures as determined in chapter 4 are summarised in Table 19 For each ship ten days a year is reserved for maintenance purposes. Typically a dry dock period is scheduled once every two years.

	Period	*No. of weeks*	*Ambient air temperature*
Sun Princess			
Caribbean	Sep-Apr	34	30 °C
Alaska	May-Aug	17	10 °C
Grand Princess			
Caribbean	Sept-Apr	34	30 °C
Mediterranean	May-Aug	17	30 °C
Capricorn			
World	Jan-Mar	13	30% 20 °C, 70% 30 °C
Europe	Apr-Dec	38	70% 20 °C, 30% 30 °C

Table 19: Cruise areas during a year with associated ambient temperatures

Appendix 11 describes the annual prime mover fuel consumption calculation. The results are shown in Figure 45 below.

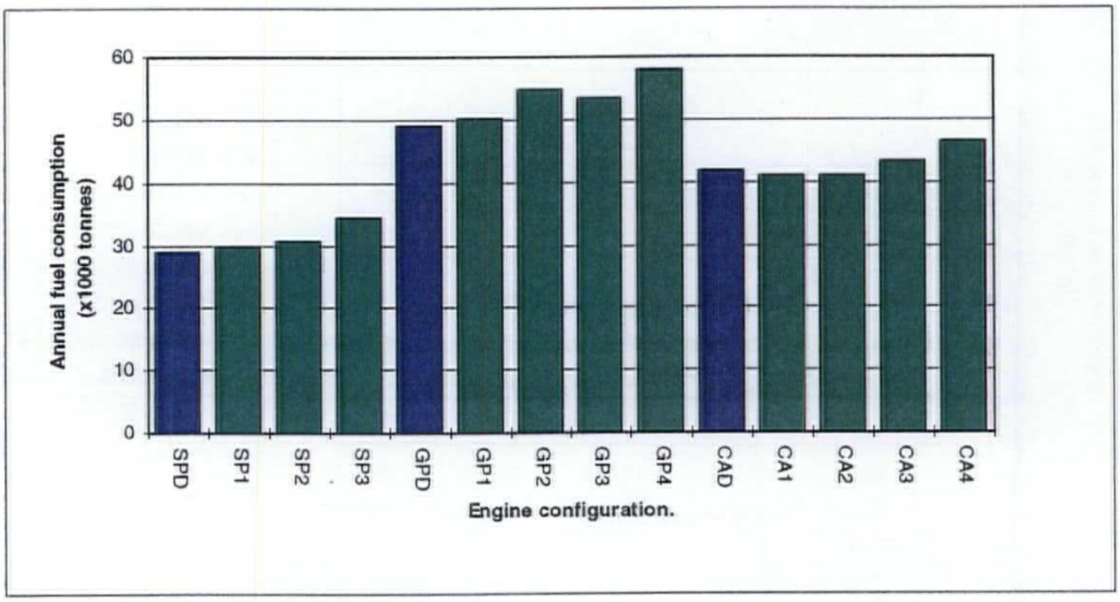


Figure 45: Annual prime mover fuel consumption

The fact that the fuel consumption of GP3 is actually lower than GP2 is caused by more favourable engine loading. As can be seen in Figure 44, the WR-21 optimum load regarding fuel consumption is less than 85%. Typically between 65 and 70%

As can be seen in the figure, the annual fuel consumption from gas turbine configurations CA2 and CA3 (see Table 13) are actually lower than the diesel fuel consumption. This caused by two things: first of all the electric load is two percent lower due to the lack of diesel auxiliaries (see section 7.1.1: **Electric load reduction**). A second effect is demonstrated in Figure 46. This figure displays the specific fuel consumption curves of three WR-21 and four 14V 48/60 engines. When the engine load becomes higher than 90% another engine is started and the load is equally shared. As can be seen in the figure, in the range from 13 to 17 MW and 26 to 29 MW, the WR-21 specific fuel consumption is actually lower due to load sharing.

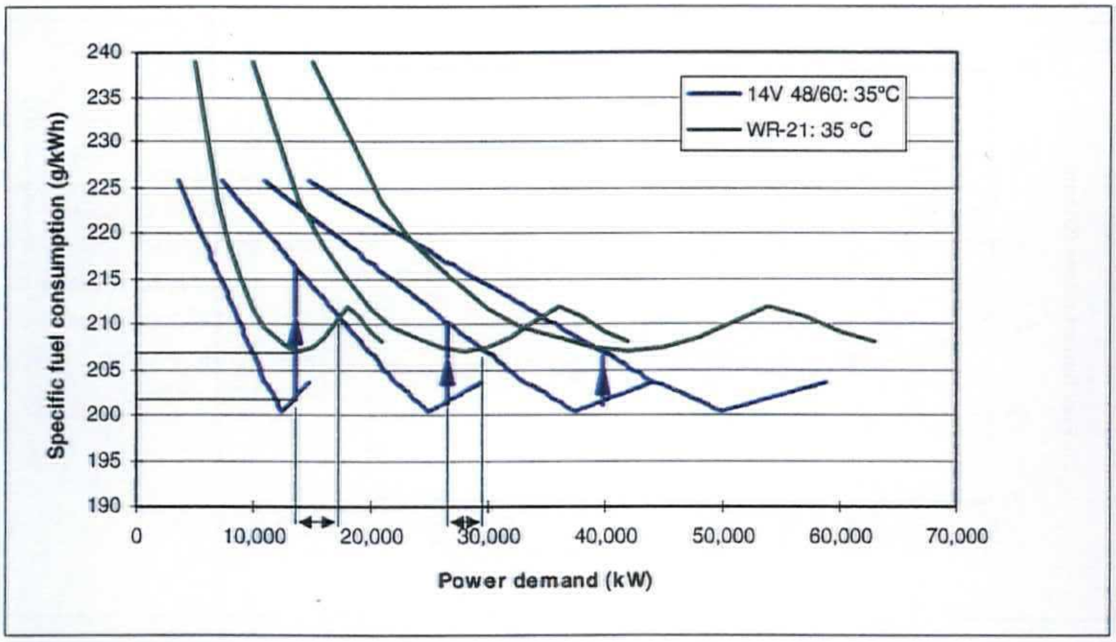


Figure 46: Specific fuel consumption comparison (load sharing)

7.3.2 Fuel bunker prices

Medium speed diesel engines can burn both residual and distillate fuel. Gas turbines however do not have the ability to burn residual fuel; they can burn MDO for short periods, but MGO or better grade fuel is recommended. Appendix 12 gives a detailed fuel specification for both diesel engines and gas turbines.

Heavy fuel is considerably cheaper than distillate fuel (see Figure 47), so current practice for medium speed diesels is to burn heavy fuel. The type of HFO that is usually used for large medium speed engines is FO 380.

There are however, several disadvantages involved in burning heavy fuel: extensive fuel treatment systems are required and maintenance costs increase significantly (time between overhauls decreases, more spares required, more maintenance hours). The MGO/MDO fuel treatment systems are less extensive and therefore the required auxiliary systems are less expensive.

International fuel oil bunker prices[21], over a period of ten months (1998) are presented in Figure 47. This figure shows Fuel Oil (FO) 380 and MDO bunker prices.

[21] [MER, 1998]

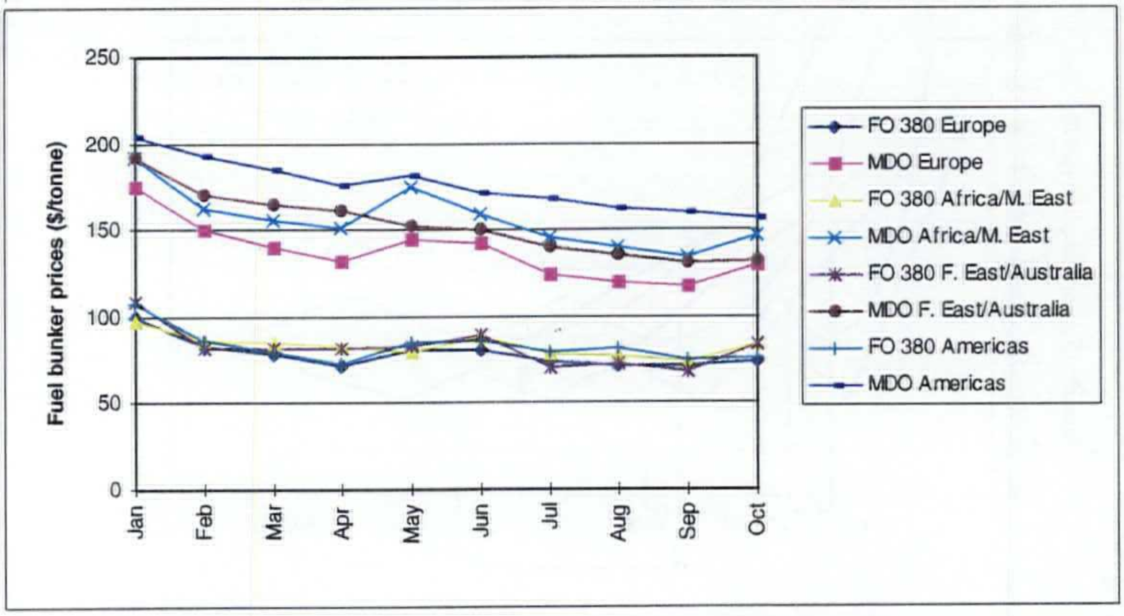


Figure 47: Fuel bunker prices (January 1998-October 1998)

The average MDO/FO 380-price gap during these ten months varies from $ 60/tonne in Europe to $ 95/tonne in the Americas. The trend demonstrated in Figure 47 though, indicates a decreasing price gap over the last year.
MGO bunker prices are typically about 10% higher than MDO prices[22]. Considering the variety in itineraries, the following prices will be adopted for the specific cruise areas:

	FO 380	MGO	Price Gap
Americas	85	195	110
Europe	80	150	70
World	85	175	90

Table 20: Average fuel oil prices ($/tonne)

7.3.3 Annual fuel costs

A certain amount of saturated steam can be generated without extra fuel consumption by using the available waste heat from exhaust gases. This was investigated in section 7.1.3: **Meeting the steam demand**. Moreover waste heat from cooling water can be used in addition to steam to feed the evaporators. This is demonstrated in chapter 8: **Alternatives for optimum plant usage**.

[22] According to A/S Dan Bunkering Ltd, Copenhagen

To avoid a complex and irrelevant calculation the oil fired boiler and increased recuperator-bypass fuel consumption will be roughly calculated. This is conducted in Appendix 13. The additional fuel costs are shown in Figure 48.

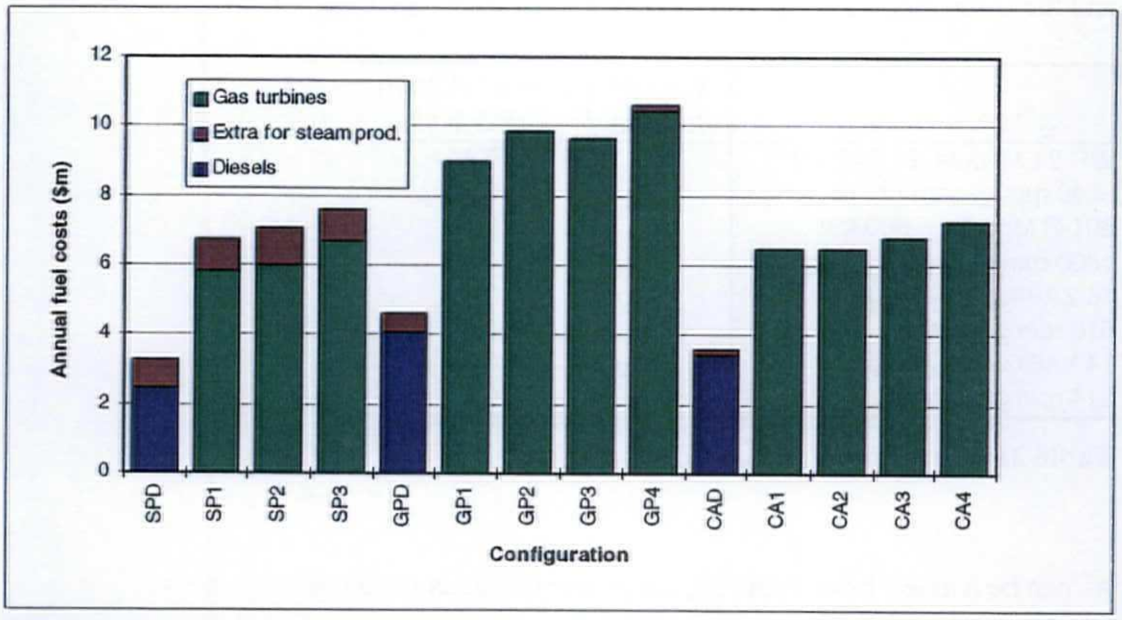


Figure 48: Annual fuel costs

7.4 Weight and dimensions

One of the most beneficial features of gas turbines is their low weight and modest dimensions, resulting in a higher power density. This potentially creates space which can be used to install extra cabins.

Another possibility to create space for extra cabins is simply to construct a longer ship. A rough estimation for the costs involved will be made at the end of this chapter.

The extra cabins will only generate extra revenues if cabin occupancy is 100%. For example: suppose a ship embodies a 1,000 cabins. During one specific cruise 800 cabins are occupied, resulting in a 80% occupancy. By adopting a gas turbine-electric power system, say 5% or 50 additional cabins can de installed within the same hull. In spite off the extra available cabins there is no reason to assume that more than 800 cabins will be booked for the next cruise. The occupancy will therefore become 800/1,050 or 76%.

This is not a problem in practice because as discussed with P&O Cruises, they always ensure 100% occupancy. If necessary, the cabins will be offered at lower prices although at the moment this generally is not necessary.

7.4.1 Power plant weights

The prime mover/generator weights from project guides and manufacturer quotations and the associated power density in kW/kg are shown in Table 21.

	Weight (tonnes)	Power density (kW/kg)
WR-21 Module: 25,240 kW	63	0.40
3600 rpm generator	50	0.50
601-R Module: 6,500 kW	16	0.41
1800 rpm generator + gearbox	26	0.25
16 ZAV40S: 11,520 kW	132	0.09
510 rpm generator	68	0.17
14 V48/60: 14,700 kW	209	0.07
514 rpm generator	89	0.17

Table 21: Genset power density comparison

As can be derived from Table 21, the power densities for the gensets are:

- WR-21: 0.22 kW/kg
- 601-R: 0.16 kW/kg
- 16 ZAV40S: 0.06 kW/kg
- 14 V48/60: 0.05 kW/kg

This yields the following genset weight comparison figure:

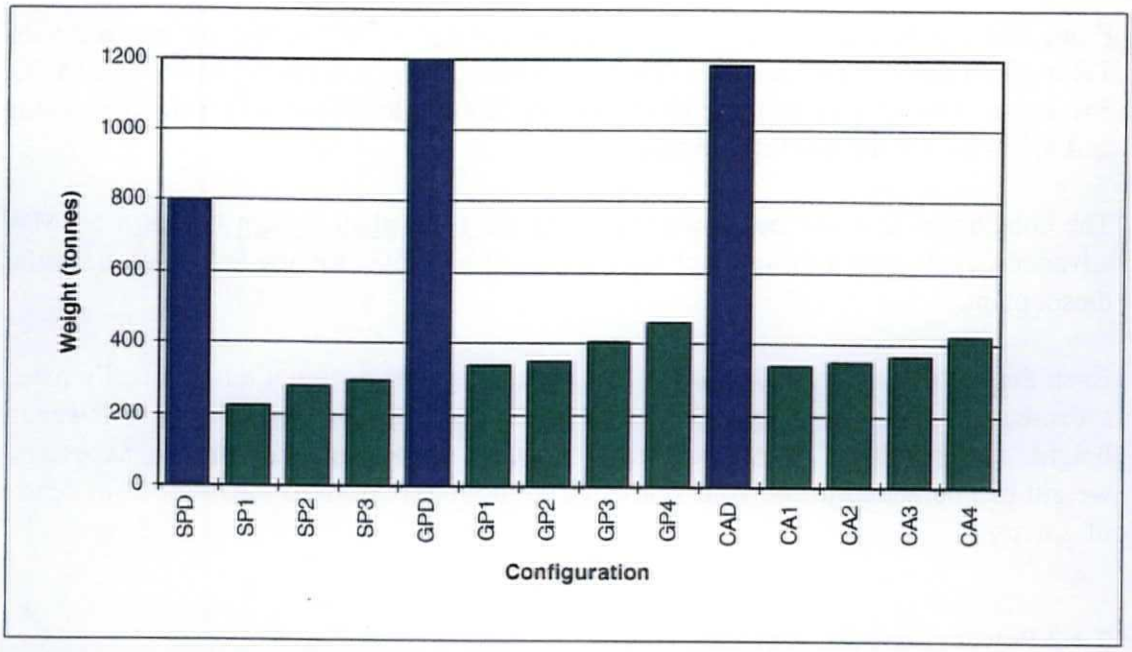


Figure 49: Total genset weight comparison (without auxiliaries)

Appendix 14 presents a weight breakdown for the individual diesel components plant (without electric systems), based on the information provided by a shipyard. Also shown in this appendix is an estimated weight breakdown for a gas turbine-electric plant (66 MW, three WR-21 engines). Information on the total weight of a 601-R plant is not available at the moment, but it will be assumed that the weight of the auxiliary systems is less for the same installed power. Less because there are no off-engine intercoolers.

Another thing that has to be taken into account is the fuel and lub oil bunker weights. According to P&O Cruises typical bunker capacities (for a *Sun Princess* class ship) are:

- HFO: 2,500 tonnes
- MDO: 250 tonnes
- Lub oil: 250 tonnes

In order to maintain the existing cruise range for a gas turbine ship, a larger amount of fuel is required because of the higher fuel consumption. According to Figure 45 typically 10% extra fuel is required. This amounts to 250 tonnes. However, no dual fuel bunkering and less lub oil is required. A WR-21 Lub Oil Module contains approximately 137 kg lub oil. In case of a *Sun Princess* class ship this would amount to 2 x 137 = 274 kg. This is approximately 0.1% of the diesel lub oil bunker capacity and therefore it will be neglected. Consequently, the gas turbine plant bunker capacities will be as follows:

- MGO: 2,750 tonnes
- Lub oil: negligible

From this can be concluded that the total oil bunker weight is less for gas turbines. Taking into account the densities (average values: HFO: 950 kg/m^3, MDO and MGO: 840 kg/m^3, LO: 850 kg/m^3), the total volumes amount to: 3,200 m^3 for the diesel ship and 3,250 m^3 for the gas turbine ship.

The conclusion that can be drawn regarding the total plant weight is that a 60 MW advanced cycle gas turbine plant typically weighs 1,500 tonnes less than a similar diesel plant.

Even though the weight reduction is low in the ship this does not automatically mean a decrease in the ship stability[23], because the gas turbine engine room is lower in height. This results in a lower centre of gravity for the superstructure. If necessary, weight can be added to the void spaces in the double bottom to lower the ship centre of gravity.

7.4.2 Power plant dimensions

For the comparisons in this section the power plant will be split up in two parts: genset and auxiliary systems.

- Genset dimensions

The basic genset dimensions are shown in the Table 22. This excludes space required for maintenance and overhaul purposes, which is described in the next section.

	Length (mm)	*Width (mm)*	*Height (mm)*	*Volume (m^3)*	*Area (m^2)*
WR-21 Module: 22 MW	8,300	2,640	5,800	127.1	21.9
3600 rpm generator	4,800	3,600	3,550	61.3	17.3
601-R Module: 6 MW	6,085	1,900	4,535	52.4	11.6
1800 rpm generator + gearbox	4,700	1,950	3,400	31.2	9.2
16 ZAV40S: 11.5 MW	9,790	3,750	6,470	237.5	36.7
510 rpm generator	5,610	2,760	4,440	68.8	15.5
14 V48/60: 14.7 MW	10,870	4,640	5,950	300.1	50.4
514 rpm generator	4,500	4,000	5,400	97.2	18.0

Table 22: Basic genset dimensions

[23] [Nurmi, 1996]

Gas turbine maintenance and overhaul clear space

The WR-21 required maintenance clear space is 914 mm (36 inches) on each side, 610 mm (24 inches) on the far end opposite to the coupling side and 762 mm (32 inches) below the enclosure[24].

Next to maintenance clear space a certain amount of space is required for element (sideways) removal during overhaul. Removal through the intakes is not considered regarding the size of the ship and the impact on the intake ducting. Huge dock side cranes would be required to remove the elements. Moreover the intake ducting dimensions would have to be increased, the splitter-silencer would have to be removed during element removal and no bends would be allowed in the ducting.

The largest WR-21 elements for removal are:

- Gas generator, including on engine intercooler assembly: 2,769x2,388x2,464 mm, 7,200 kg
- Power turbine: 819x1,550x1,550 mm (circular), 4,320 kg
- Recuperator module (two modules in a recuperator assembly): 3,454x1,270x1,803 mm, 4,850 kg.

Consequently 3,400 mm (with respect to the 2,388 mm width of the gas generator) of overhaul space is required directly next to an engine, or in between two engines. In order to move this element towards the hull of the ship, a total of approximately three metres at the end of the genset is required to turn and remove the gas generator (see Figure 50). It is assumed here that all components will be moved to one side of the ship for discharge through a hatch in the ships hull. This hatch is typically located on deck four.

Figure 50 displays a simple top view drawing including two WR-21 gensets. The large-dashed lines indicate the maintenance clear space, whereas the small-dashed lines represent the space required for element sideways removal. The lifting equipment has to be installed above the engine for recuperator module removal. This requires roughly one meter.

Although the 601-R is considerably smaller, it will be assumed that the same amount of maintenance clear space is required. The largest element for sideways removal is the entire gas turbine: 2,413x1,143x1,016 mm, 1,360 kg. Consequently roughly 1,600 mm overhaul clear space is required next to the 601-R module. No extra space is added to the length, since maintenance clear space for generator rotor withdrawal (as discussed below) already allows for gas turbine sideways removal. This can be seen more clearly in the plan view drawing below. Again lifting equipment for the recuperator parts require one metre of space above the engine. A prediction of the overall plant enclosure dimensions was shown in section 5.2: **The Rolls-Royce/Allison 601-R**.

[24] [WR-21 IRD, 1997]

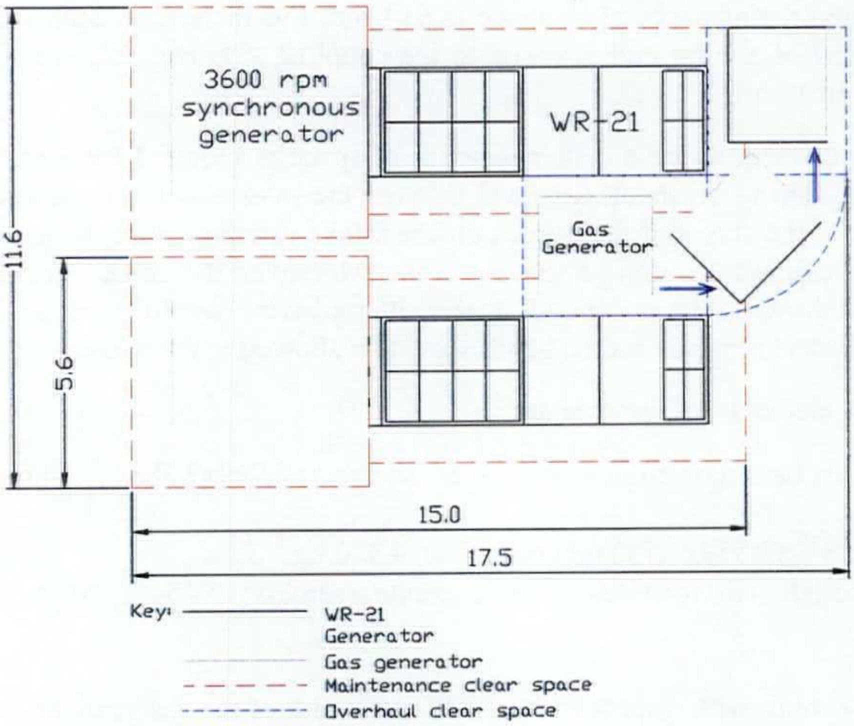


Figure 50: WR-21 gensets (plan view)

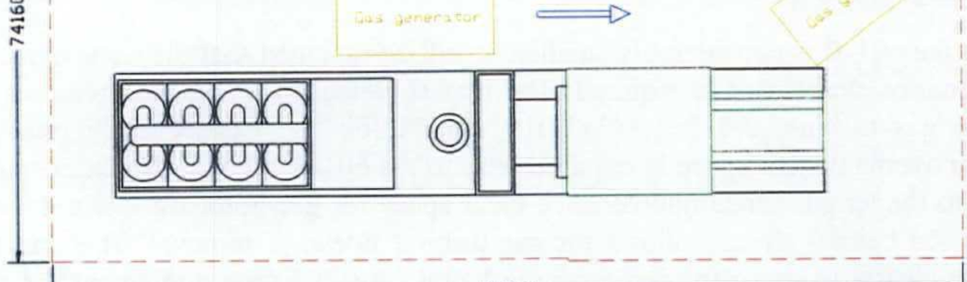


Figure 51: 601-R gensets (plan view)

Diesel engine maintenance and overhaul clear space

Maintenance clear space for diesel engines was obtained from several machinery arrangement drawings. The space on each side varies from one to two metres. An average of 1.5 (as applied on *Grand Princess*) will be taken as generic. The required space at the far end again varies from one to two metres. Therefore again 1.5 metres will be taken as an average.

Moreover the engines require typically two metres of space above the engine. The lifting equipment is used for transporting heavy engine parts from their storage area to the engine and vice versa during overhaul. Because the elements have to be lifted above the engine, typically one metre is required for the actual parts and one metre for the lifting equipment. Component removal can take place over the engine and therefore no extra space is required next the engine.

Generator maintenance and overhaul clear space

Maintenance clear space for generators is typically 1 metre on all sides. The WR-21 Brush generator also requires space above the generator to remove parts and to gain access to the rotor parts. The required space will be about one third of the height and this will also be applied to the diesel generators.

The rotor from the 601-R generator on the other hand can only be removed at the far end. According to the manufacturer, 2.1 metres is required for rotor removal during overhaul.

Individual prime mover plan view drawings are presented in Appendix 15.

When two engines are installed directly next to each other, the maintenance and overhaul clear space can be shared. The effect on the average genset volume is illustrated in Figure 52, which presents the volume per genset against the amount of engines installed next to each other.

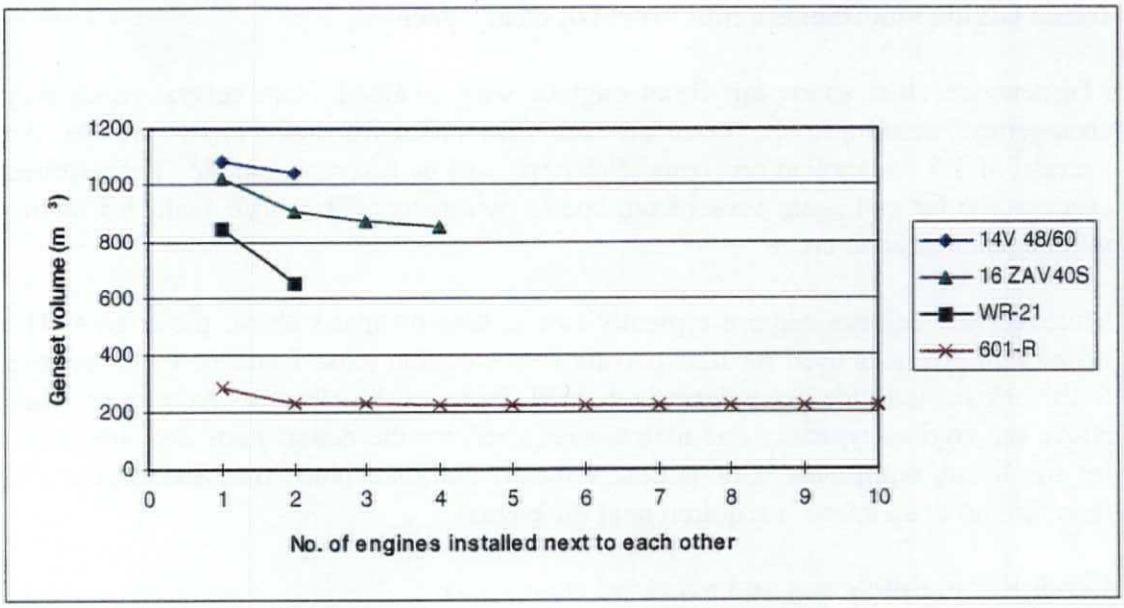


Figure 52: Genset volume as a function of the configuration

At this point a preliminary gas turbine layout will be chosen. For some configurations the 'right' layout is obvious forward whereas other options results in a lot of different possibilities.

It has to be taken into account here that the prime movers have to be divided between at least two watertight bulkheads, according to the existing safety regulations. Another point that has to be taken into account is the removal route. One possibility is a central rail track on the machinery deck, as applied on *Grand Princess* and *Capricorn*. This however eliminates the possibility to install a prime mover in the middle of the ship. Since it is generally preferable to install the prime movers symmetrically in the ship, this may cause problems with the layout because of the sometimes odd amount of gas turbines. A possible unequal weight distribution however can be compensated by placing for example the evaporators or separators on the 'opposite side' of the central line of the ship. Another possibility for component removal is by using lifting equipment only instead of a rail track. This allows for prime mover installation on the central line.

A possible layout for each gas turbine configuration and the three existing diesel layouts are shown in Appendix 15. This is conducted without changing the existing superstructure of the ship (bulkheads). An attempt has been made to equally distribute the power over the two engine rooms. In some cases however, this was difficult to achieve. Clearly the internal structure for each preliminary layout will have to redesigned for an optimum utilisation of the machinery deck space.

The total genset volumes according to the preliminary layouts are shown in Figure 53 (see also Table 26, Appendix 15). Also shown in this figure is the machinery deck space required for lub oil, cooling and control systems. This is further illustrated below.

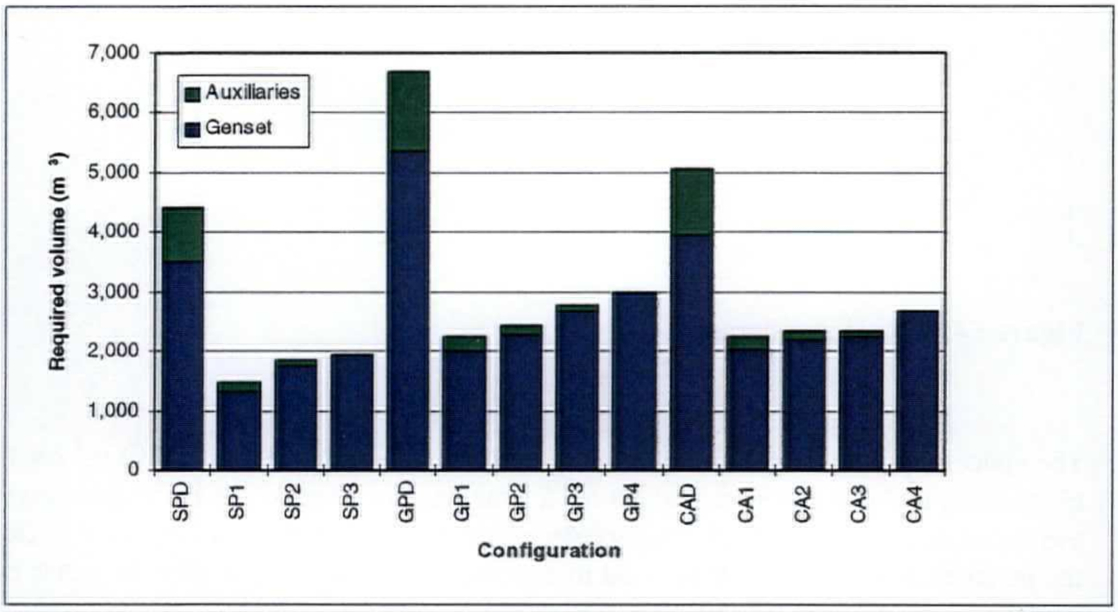


Figure 53: Total genset and auxiliary (lub, cooling and controls) volume comparison

- Auxiliary system dimensions

Auxiliary systems that are of interest for this comparison are fuel systems, lub oil systems, cooling systems and intakes and uptakes. Moreover waste heat boiler and oil fired boiler dimensions will be considered. A more detailed investigation will be made for the 'WR-21 only' gas turbine option GP1 comprising three WR-21 engines. The results from this evaluation will be used to estimate the auxiliary area and volumes on the other vessels.

Machinery deck auxiliaries

The auxiliaries included in a WR-21 package are a lub oil module, an off-engine intercooler and an engine controller. These systems take up 1.1, 5.6 and 0.15 m^2 of machinery deck space, respectively. The total area (see below), including maintenance clear space is roughly 26 m^2 for this configuration. The total area is therefore 78 m^2. The 601-R design will be such that the lub oil and control system are integrated in the enclosure. Therefore no extra space is required for these auxiliaries.

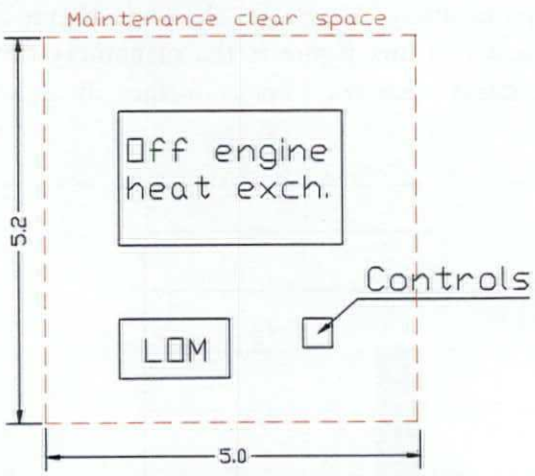


Figure 54: WR-21 auxiliaries

The space required for the diesel cooling systems is roughly 250 m^2, 130 m^2 for lub oil systems including storage and 60 m^2 for the control systems. These numbers were estimated according to the machinery deck drawings (*Grand Princess*). In this case, the installed power-ratio will be used to estimate the required space for the other two vessels: 295 m^2 (46,100/69,100*(250+130+60) for *Sun Princess* and 375 m^2 (58,800/69,100*250 = 130+60) for *Capricorn*.

As a result between 360 and 440 m^2 of machinery deck space can be saved *on Grand Princess*. This estimation is rather pessimistic than over-estimated because the fuel systems have not been taken into account (see Appendix 5). The 360-440 m^2 represents about 7-8.5% of the total machinery deck area. With the engine room height typically being 3 metres, the volume saving for this case is 1,090-1,320 m^3. These numbers are also shown in 53.

Engine casing auxiliaries and boilers

An engine casing typically embodies the following systems and auxiliaries:

1. exhaust gas uptakes and waste heat boilers (WHB)
2. combustion air intakes
3. ventilation air intakes
4. oil fired boilers

1. Exhaust gas uptakes and waste heat boilers

The existing diesel exhaust duct diameter is 1.3 metres. WR-21 uptake ducting designs[25] have resulted in an inner duct diameter of 2 metres (outer diameter 2.2 metres). A preliminary WR-21 uptake arrangement including waste heat boiler were presented in Appendix 8. The diesel waste heat boiler indicative dimensions are: 3x1.6x5.5.

[25] Conducted by AAF International

Grand Princess is 16 decks high (deck 13 does not exist), whereas *Sun Princess* and *Capricorn* are 14 decks high. This results in a distance from engine outlet to the upper deck of roughly 36 metres on *Grand Princess* and an estimated 30 metres on *Sun Princess* and *Capricorn*. The ducting from upper deck to funnel top is not taken into account, because it is basically located outside the ship. This ducting length has of course to be taken into account for duct loss calculations (see Appendix 16). For this purpose an additional 15 metres is added.

As opposed to gas turbines diesel engines require silencers in the uptake to reduce the noise produced by the engine. The dimensions of the silencers are shown in both project guides and engine casing drawings (as provided for *Grand Princess*). The dimensions for a 16V ZA40S silencer (with spark trap, 35 dB(A) silencing) are: length 9,700 mm, diameter 2,500 mm. The volume including support structure and cleaning ports is roughly 78 m^3.

This results in the following uptake volume comparison:

	Total uptake volume (m^3)	
	Diesel	*Gas turbine*
Waste heat boiler	6x26=158	3x44=132
Transition ducting	-	3x135=405
Straight duct	6x35=208	3x126=378
Silencers	6x78=468	-
Total	834	915

Table 23: Total uptake volume comparison

2. Combustion air intakes

Figure 55 shows an indicative drawing of the existing diesel intake arrangement, located on deck 15. This sectional view in longitudinal direction shows the intake ducting as it goes down the engine casing. The rectangular duct dimensions are 2x1 metres (the figure shows the '1 metre view'). The air is discharged directly above the turbochargers, approximately 30 metres below deck 15. The overall width of this intake arrangement is about 3 metres and consequently the overall volume is roughly 78 m^3. The duct takes up another 60 m^3.

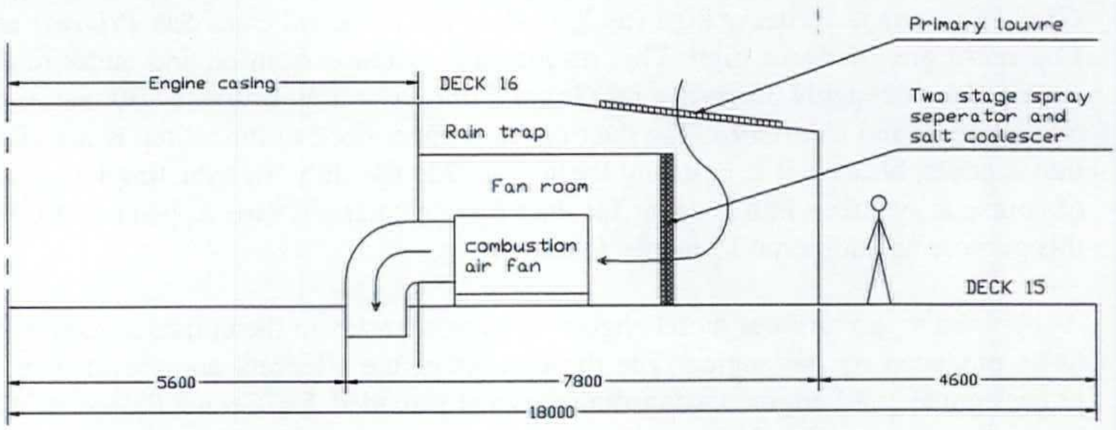


Figure 55: Diesel intake arrangement on *Grand Princess*

A similar preliminary arrangement for a WR-21 is shown below. Another position for air intakes is on the side of the ship. The intake filters are then typically located on deck six. However, because of the existing engine casing an arrangement on deck 15 is chosen for a comparison. It is assumed that the ducting between deck 15 and the engine inlet boot is straight.

The rectangular intake WR-21 intake duct cross section is 2x2 metres. As opposed to diesels, gas turbines require intake silencers to attenuate the high-frequency compressor noise. These silencers however do not increase the cross-sectional area of the duct.

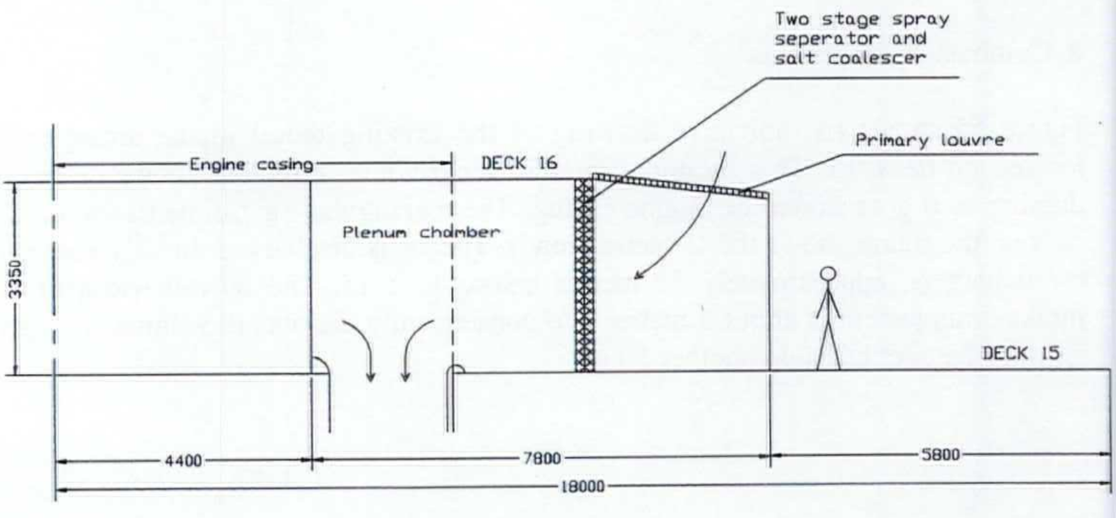


Figure 56: Preliminary WR-21 air intake arrangement

Normally the intake filter requirements are more strict for gas turbines, especially when they are mounted on the side of the ship (in view of possible spray). Due to the proposed arrangement and the position of the filters (more than 30 metres above sea level) two-stage filters will be sufficient. The width is now determined by the filter width which is 3,950 mm. The total volume estimation of this arrangement is 104 m^3. Another 120 m^3 is required in the engine casing. An additional advantage is that the available deck space for the passengers is increased.

As a result, the overall uptake volume comparison yields 828 (6x138) m^3 for the existing diesel intakes and 672 (3x224) m^3 for the gas turbine intakes. An prediction of the associated intake losses is shown in Appendix 16.

3. Ventilation air intakes

The diesel engine room ventilation air for the diesels is included in the combustion air. The WR-21 however requires a separate forced draft system for enclosure ventilation. The air can be taken in via the existing plenum chamber for the combustion air. The ducting diameter is approximately 800 mm and thus the ventilation duct takes up another 19 m^3 per engine.

4. Oil fired boilers

As demonstrated in section 7.1.3: **Meeting the steam demand**, the oil fired boiler capacity can be reduced from 30,000 to 10,000 kg steam production per hour. The dimensions of the vertical down oil fired boiler can be estimated from the engine casing drawings from *Grand Princess*. The cylinder shaped boiler is about four metres in diameter while the length is about nine meters (144m^3). The volume for a 10,000 kg/h unit is estimated at 100 m^3. As a result a space saving of 190 m^3 can be achieved.

A summary is shown in Table 24.

	Total volume (m^3)	
	Diesel	*Gas turbine*
Uptakes + WHB	834	915
Comb. air intakes	828	672
Vent. air intakes	-	57
Oil fired boilers	288	100
Total	1,950	1,744

Table 24: Total required engine casing volume comparison

The budget price for two vertical down oil fired boilers, including a burner suitable for 700 cSt oil, an oil pumping and heating unit, valves and mountings and water level controls is £ 450k[26] ($ 720k). The overall efficiency varies from 88 to 96 %, burning heavy fuel.

[26] Quotation from Thermal Engineering International, April 1997

In conclusion can be said that the required engine casing and intake arrangement volumes are similar.

7.4.3 Extra cabin revenues

The total space savings according to the previous section are shown in Figure 57. As can be seen in the figure, the space saving varies from 2,300 to 4,400 m^3. To express this amount as a percentage, the total volume of the enclosed ship spaces is calculated. This volume is defined as follows[27]:

$$V = \frac{GRT}{k_1} \text{ [m}^3\text{], where}$$

$$k_1 = 0.2 + 0.02 * \log V,$$

where GRT = gross tonnage [tonnes]

This yields a ship volume of 250,000, 351,000 and 244,000 m^3, respectively. Consequently the space savings vary from 0.9 to 1.2 % of the total volume of the enclosed ship spaces.

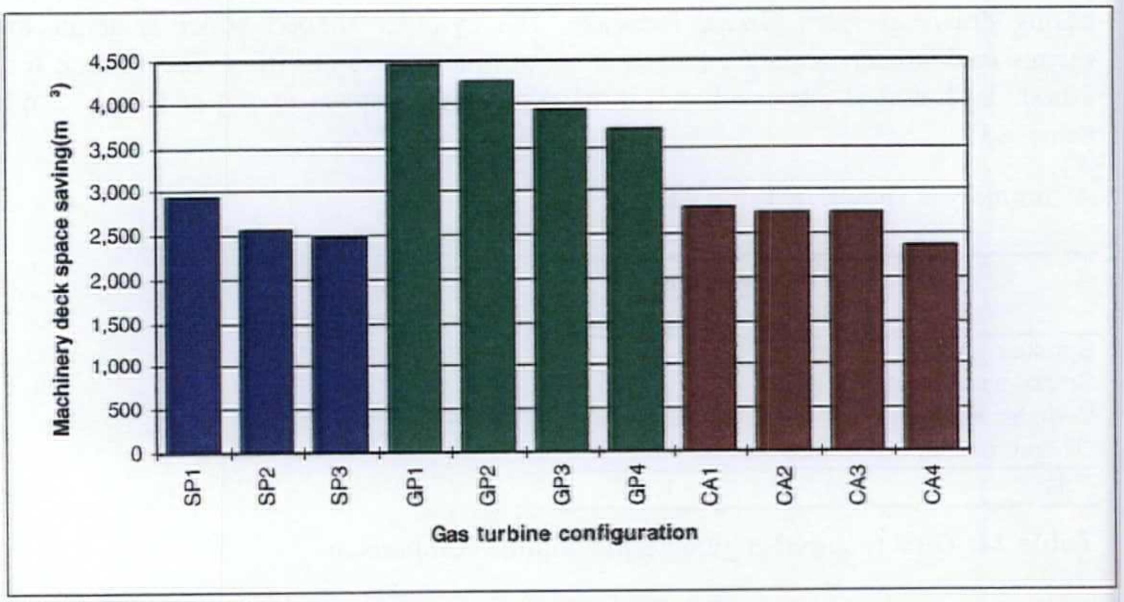


Figure 57: Machinery deck space savings with gas turbines

[27] According to the IMO-convention, London 1969

At this point it is assumed that 100% of this space saving can be used effectively as extra space for cabin construction on the lower decks. Clearly this requires a total redesign of the entire machinery layout. It is however not unthinkable that only 75% can be used effectively. On the other hand this percentage could also be more than 100% due to more favourable layout possibilities. The effect of these changes will be investigated in section 7.9.3: **What if scenario**.

The volume of a twin room on *Grand Princess* vary from 45 to 58 m^3. On *Sun Princess* this is 41 to 48 m^3. The average of 48 m^3 will be adopted as generic. As a result an amount of extra cabins displayed in Table 25 can be constructed within the same hull.

The extra cabin revenues were obtained from various cruise brochures. The fares in these brochures are all based on two persons sharing a stateroom (cabin). Extra machinery deck space can only be converted to low grade cabins, situated on the lowest passenger decks. Consequently the fares used for revenue calculation will be for 'Twin Rooms Inside'. The daily extra revenues per cabin for a large amount of P&O/Princess cruises are presented in Figure 58. To account for transfers and possible overnight stays on shore, the generic cabin revenues that will be used for further calculations is $ 300 per cabin per day. As can be seen in Figure 58 this is a minimum rather than an average. The potential additional cabin revenues are also displayed in Table 25 (based on 355 revenue generating days a year).

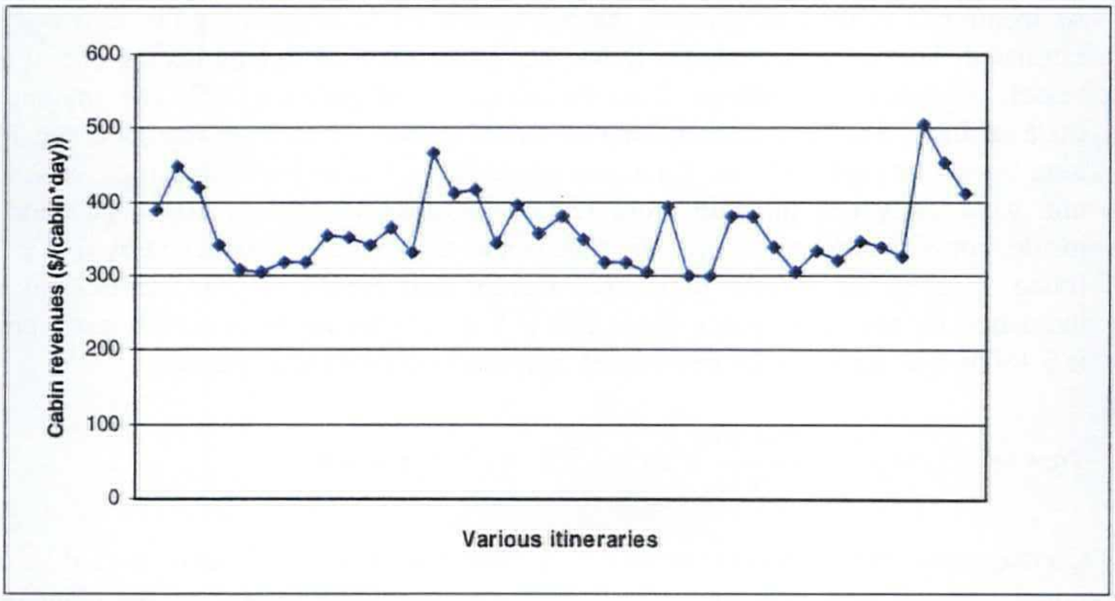


Figure 58: Daily cabin revenues

Typical cabin construction costs vary form $ 24,000 to $ 36,000 for the more luxurious cabins (suites). As discussed above, mostly low grade cabins will be constructed and therefore $ 24,000 will be adopted for extra construction costs determination.

	Additional cabins*	Percentage of total	Extra annual cabin revenue potential	Extra cabin construction costs
SP1	62	6.2	$ 6.60m	$ 1.48m
SP2	54	5.4	$ 5.75m	$ 1.30m
SP3	52	5.2	$ 5.54m	$ 1.25m
GP1	93	7.2	$ 9.90m	$ 2.23m
GP2	89	6.9	$ 9.48m	$ 2.14m
GP3	82	6.3	$ 8.73m	$ 1.97m
GP4	78	6.0	$ 8.31m	$ 1.87m
CA1	59	5.9	$ 6.28m	$ 1.44m
CA2	57	5.7	$ 6.07m	$ 1.42m
CA3	58	5.8	$ 6.18m	$ 1.39m
CA4	50	5.0	$ 5.33m	$ 1.20m

Table 25: Additional cabin revenues and construction costs

The items that are not included in the ticket fares and consequently will create extra revenues for the operator are tips, drinks, personal purchases (in the various shops on board) and tours. It is assumed that these revenues largely compensate for the extra food, beverages and other items that have to be stocked.

As mentioned in the introduction, the costs involved in lengthening the ship will be estimated. This is conducted as follows. The costs involved in lengthening one of the vessels is estimated as follows. Take for example configuration GP2. The amount of extra cabins is 89. The frame spacing on Grand Princess varies between 680 and 715 mm, say on average 700 mm. Each low grade cabin is four frame spacings, or 2,800 mm wide. A typical midship cross section contains 16 cabins, 10 outside and 6 inside. For 89 additional cabins, the ship would then have to lengthened by 6x4 = 24 frame spacings or 16,800 mm. The overall ship length is 290 metres and an indication for the power plant initial cost is $ 35m whereas the reported vessel price is $ 450m. The price for the new vessel can thus be estimated as follows:

$$\textit{New vessel price} = \frac{290+16.8}{290} * (450-35) \; + \; 35 = \$\, 474m$$

Consequently, the extra cost amounts to $ 24m, which is significantly more than the initial or investment cost increase when opting for a gas turbine power plant (see Figure 37, GPD compared to GP1: a difference of nearly $ 5m).

* Additional cabins due to reduced manning not included

7.5 Emissions

Most of the worlds fleet is continuously operating relatively close to the shoreline. Thus the ships emissions have an impact on both acidification and air pollution in cities as well as global warming. Consequently great attention today is paid on ships emissions when selecting the machinery concept.

Emissions from prime movers and oil fired boilers are CO_2, CO, SO_X , NO_X , HC and particulate matter. The main concern regarding pollution is for NO_X , SO_X and lately also for CO_2.

7.5.1 Origin from and reduction methods for exhaust gas emissions

The information in the next section is partially based on a paper from Deltamarin: *Environmental aspects - fuel emissions* by Jari Nurmi.

CO_2, carbon dioxide and CO, carbon monoxide

CO_2 is a greenhouse gas causing global warming and is therefore subject to wide interest. CO_2 emission is related to the carbon content in fuel. In a stochiometric burning process all carbon is turned into CO_2 in reaction with oxygen. In reality, the burning is not so efficient and some of the carbon is not completely burned and emitted as carbon monoxide, CO. As the carbon content of liquid fuels is quite constant,
85-88 % of weight, the emission is directly proportional to the fuel consumption. Thus in spite of slight fuel type differences, every tonne of fuel creates about 3.1-3.2 tonnes of CO_2, according to the following reaction equation:

$$C + O_2 \Rightarrow CO_2$$

Taking into account the molecular weights (16 for oxide and 12 for carbon): 12 tonnes of carbon reacts with 32 tonnes of oxide, resulting in 48 tonnes of carbon dioxide. Or alternatively: 1 tonne of carbon results in 4 tonnes of carbon dioxide. Carbon content in fuel is 85-88% and therefore 1 tonne of carbon creates:

$$0.87 * \frac{32+12}{12} = 3.2 \text{ tonnes of carbon dioxide}$$

Consequently the only effective way to reduce CO_2 emission is to reduce fuel consumption by applying higher efficiency engines or propulsors. Because gas turbine fuel consumption is higher, the same thing can be said about its CO_2 emission.

The specific CO_2 emissions in g/kWh can be estimated as follows: assume an average fuel consumption of 190-200 g/kWh for diesel engines, 205 g/kWh for WR-21 engines and 240 g/kWh for 601-R engines (see Figure 44). Multiplying these numbers by 3.2 yields the specific CO_2 emissions: 610-640 g/kWh, 660 g/kWh and 770 g/kWh, respectively.

NO_X, nitrogen oxides

NO_X is one reason for acid rain and ozone depletion. NO_X has a 20 times bigger relative impact on global climatic change than CO_2[28], hence NO_X has got more attention on ships emissions than CO_2.

The most common nitrogen oxides in exhaust gases are NO and NO_2, which are denoted as NO_X. NO_X builds up either by reaction between nitrogen and oxygen of combustion air (thermal NO_X), by reaction between exhaust gas hydrocarbon and combustion air oxygen (prompt NO_X) or by reaction between nitrogen bindings in fuel (fuel NO_X).

Thermal NO_X is decisive for the total emission and all the abatement methods are targeted to reduce that component. High temperature and free oxygen molecules are prerequisites for thermal NO_X. At temperatures above 1,500 °C NO_X formation rises very sharply. Abatement methods like retarded injection or water injection in the fuel or the burning process reduces peak temperatures, lowering NO_X emissions.

NO_X emission reduction can be carried out either by affecting the burning process, or by applying a separate exhaust gas catalysator (Selective Catalytic Reduction). A SCR unit can reduce the NO_X level from a medium speed engine typically down to 2 g/kWh. A water based method on the other hand is applicable only down to 6 g/kWh. Clearly these abatement methods increase investment cost and running costs as will be shown in section 7.5.3: **Exhaust gas emission abatement costs**.

Basically the same abatement methods can be applied to gas turbines. Another abatement method which drastically reduces NO_X emissions is brought about by incorporating a pre-mix lean burn combustion process. The so called Dry Low Emissions (DLE) engines demonstrate low emissions (both NO_X and CO) over a wide operating range.

A DLE combustion system for aeroderivative gas turbines consists of a number of separate combustion zones. This so called pre-mix lean burn combustion process eliminates the high peak temperatures which occur near stochiometric conditions that create high NO_X concentrations. Since NO_X emissions are dominated by the combustion temperature, their production can be minimised by reducing the combustion peak temperature. The average combustor air outlet temperature has to be maintained to ensure that the engine performance does not decrease.

[28] According to Det Norske Veritas, a Norwegian classification company

Typical NO_X emission curves for medium speed diesels with varying engine load are presented in Figure 59 below. In practice the specific NO_X emission at 100% engine load is typically 10-14 g/kWh when no extra measures are taken for emission reduction.

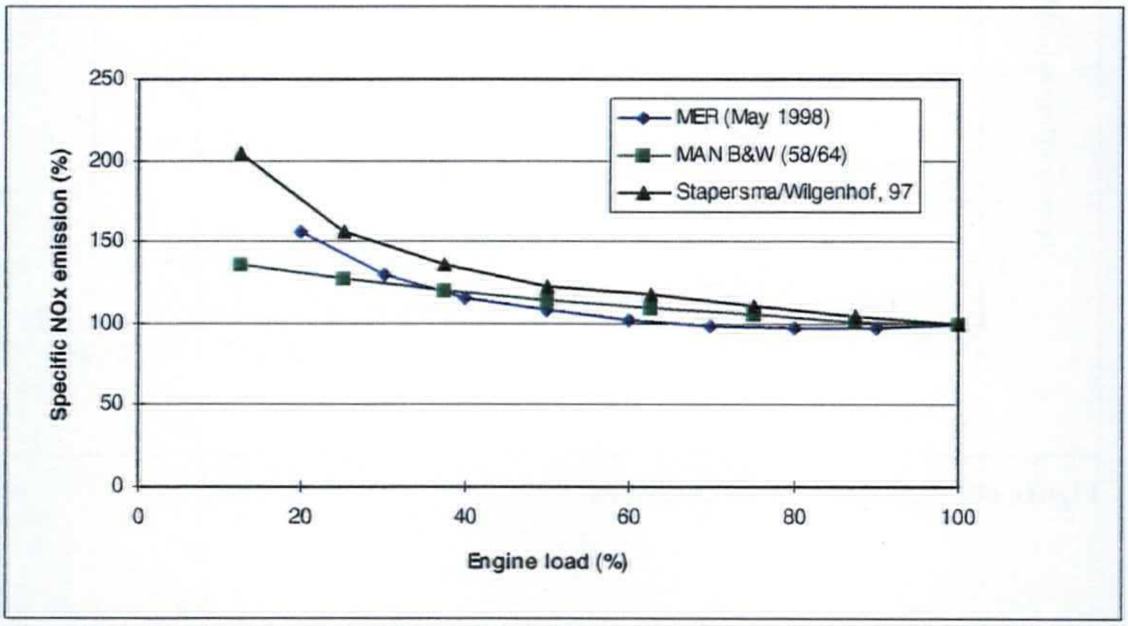


Figure 59: Diesel engine specific NO_X emission curves

The curve from the *Marine Engineers Review*(MER), issue May 1998 is a measured curve. The tolerance for this curve is ± 1.5 g/kWh. The measurements are taken on four nominally identical medium speed diesel engines installed on a ferry under in-service conditions.

The predicted emissions for the WR-21 are displayed in Figure 60. These curves are based on the emissions as shown in Appendix 17. The emissions in Appendix 17 are displayed in vppm (volume parts per million) and the conversion to g/kWh is also elucidated in this Appendix.

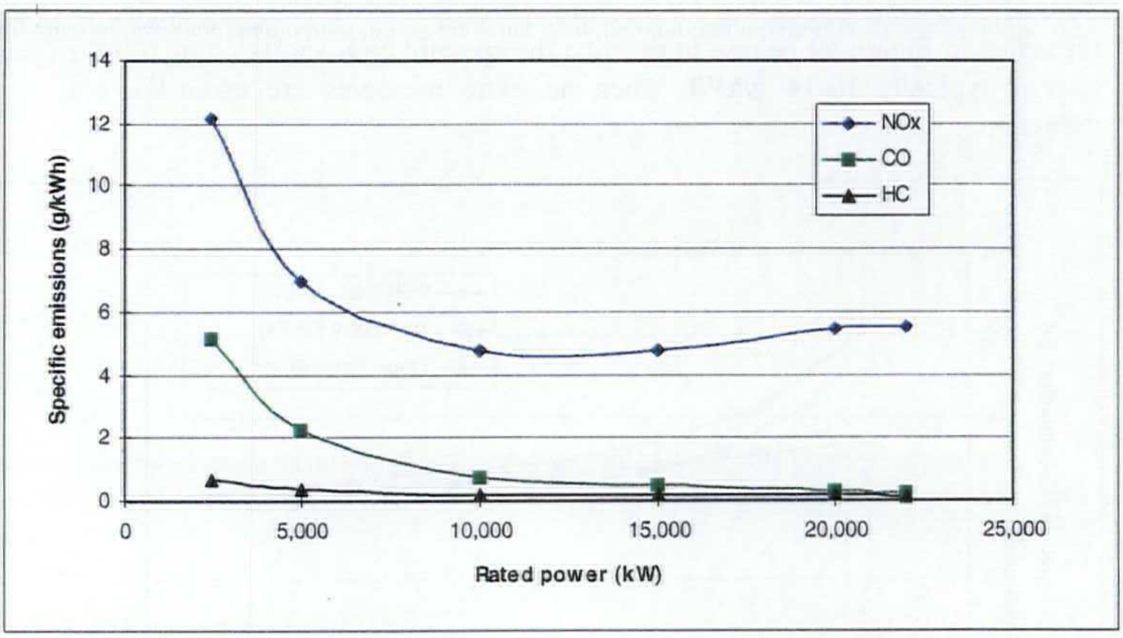


Figure 60: Predicted WR-21 emissions

The average values over the power range from 8,800 to 22,000 kW (40 to 100%) are: NO_X: 5 g/kWh, CO: 0.3 g/kWh and HC: 0.18. Present-day tests however show that these values are rather pessimistic. The estimated 601-KF9 NO_X emission is 3.7 g/kWh. The 601-R target NO_X emission with DLE combustion is far less: 0.18 g/kWh.

SO_X, sulphur oxides

Like NO_X, the effect on environment is acid rain and ozone depletion.

With SO_2 emissions, there is actually only one feasible method to apply: reduction of the sulphur content in marine fuel. Because all sulphur in fuel will remain in the exhaust gases, one tonne of sulphur in fuel equals two tonnes of SO_2 in exhaust gas. The reaction equation is as follows:

$S + O_2 \Rightarrow SO_2$, with the associated molecular weights are 32 (S) and 32 (O_2).

Low sulphur fuel is already available on selected markets and there are no technical problems regarding pollution prevention through use of low sulphur fuel. Although it is possible to desulphurize the residual fuel at the refinery, it requires a lot of energy and investments. This results in a price increase of $ 15-30/tonne when reducing the heavy fuel sulphur content from 3% to 1%.

As can be seen in Appendix 12 the maximum sulphur contents for (heavy) diesel fuels is 5% and 1.3% for gas turbine fuels. Typical average values however are 3% and 0.3%. Again taking into account the average specific fuel consumption as mentioned before, this yields the following specific SO_2 emissions: 11-12 g/kWh (200x2x0.03) for diesels, 1.2 g/kWh (205x2x0.003) for WR-21's, and 1.4 (235x2x0.003) for 601-R engines.

A summary of the emissions (average over the engine load range from 40% up to 100%) as discussed above is shown in Table 26 below.

Pollutant	*Diesel (g/kWh)*	*WR-21 (g/kWh)*	*601-R (g/kWh)*
CO_2	610-590	660	770
CO	1.6-1.8	0.3	0.3 ?
NO_X	12-15	5.0	3.7
SO_2	11-12	1.2	1.4
HC	0.5-0.6	0.2	0.2 ?

Table 26: Typical prime mover emissions

The annual emissions in tonnes per year for the two most important pollutants are shown in Figure 61 below.

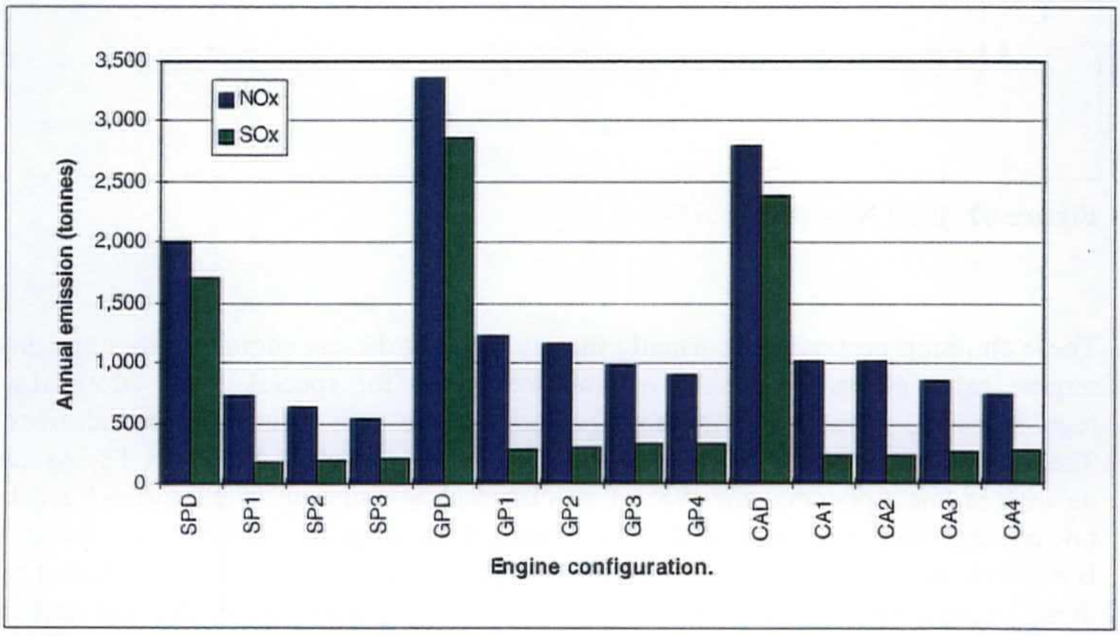


Figure 61: Annual NO_X and SO_2 emissions

The conclusion that can be drawn from this figure is that the NO_X emission can be reduced by on average 70% and the SO_2 (or SO_X) emission even by 90%.

7.5.2 Exhaust gas emission limit regulations

The International Maritime Organisation (IMO) NO_X emissions limit proposals (see Figure 62) are related to diesel engine speed and represent a reduction on existing levels. Even more drastic reductions are considered by individual states for their national waters. After six years of discussion with delegations of maritime committees, engine manufacturers and environmental organisations, a protocol on emission limits has been drawn up. This protocol will become law when it has been signed by at least 15 countries which represent at least 50% of the worlds merchant fleet. The proposed limits will apply to all newbuilt-engines larger than 130 kW which are to be installed on ships that are constructed after 1 January 2000[29].

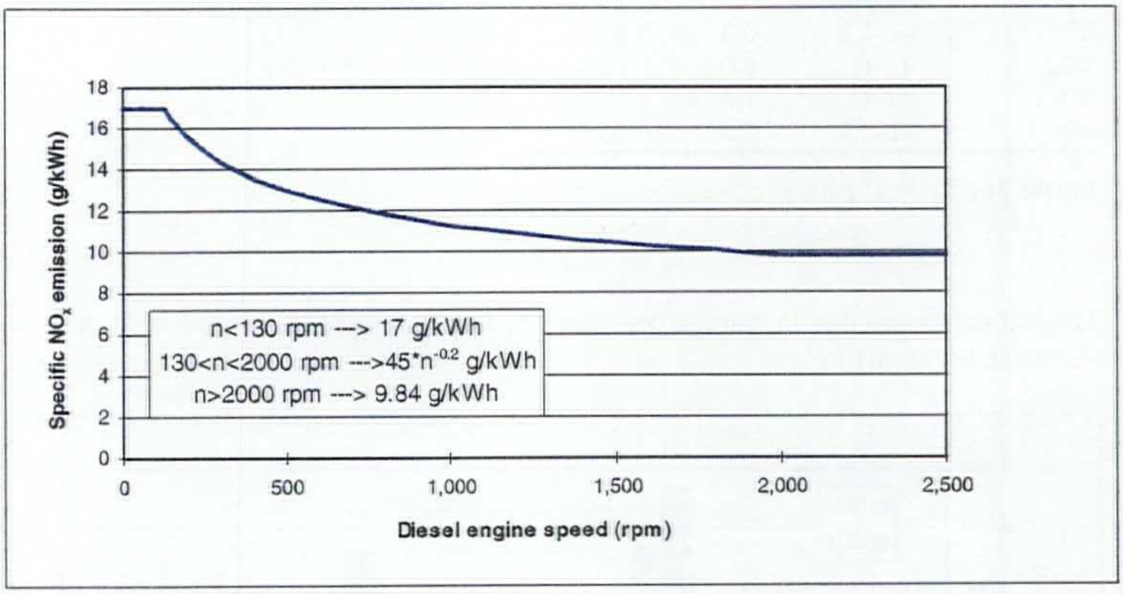


Figure 62: IMO NO_X emission limit

These emission proposals are usually met by existing diesels engines, although some require extra measures to achieve emission limits for special areas. More strict regulations are already into force or expected in the near future in 'special areas'. These present-day and projected emission limits are shown in Appendix 18. As can be seen in this Appendix, the WR-21 predicted NO_X emission levels throughout the power range will meet both 'global' and 'special areas' projected near term limits. It is however open to interpretation as to how a gas turbine will be classified against the diesel engine speed rating. This has particular relevance since the gas turbine emission falls within the 2-6 g/kWh limit range for 'Special areas' in the near future.

[29] [Verkley, 1998]

A recent article[30] discusses the proposed limits on the sulphur content of bunker fuel. These limit the sulphur content of bunker fuels world-wide to 4.5%; but also allow the IMO to designate SO_X Emission Control Areas (SECAs). The Baltic Sea is the first candidate to be nominated and the North Sea, Irish Sea Channel and part of the Atlantic to the west of Ireland are likely to be designated SECAs in the future. Within SECAs, ships must burn fuel with a sulphur content of 1.5% maximum or adopt equivalent measures, such as exhaust gas scrubbing.

7.5.3 Exhaust gas emission abatement costs

As discussed in the previous sections, the applied commercially available methods at the moment for diesel engines in ship use are:

- Retarded injection
- Direct water injection
- Water-emulsification
- SCR-catalyst with urea as agent

Quite often these methods are compared with each other by just calculating the first cost and running costs. However, this would lead to false conclusions. The vital variable to be considered in the comparison is the cleaning capability. Figure 63 shows the situation[31]. Displayed in this figure are the abatement methods and the associated minimum achievable NO_X levels against the annual costs. The numbers are based on systems for a 1,500 kW auxiliary engine on a cargo ship. The system is only in use in harbour. A SCR system for a 14 MW main engine in continuous use can easily create $ 1m per year additional costs for the owner. Below 6 g/kWh SCR is the only commercially viable option for a medium speed diesel engine.

NO_X abatement also affects the ships systems. In case a water based method is selected, the fresh water production must be enlarged by 30 tonnes/day for a 14 MW engine installation. On cruise ships this is a relatively small percentage.

If SCR is selected, the catalysator is just one of the many components of the system to be considered. The specific volume of an SCR plant is about 1 m^3/MW whereas the specific weight amounts to 1 tonne/MW for a medium speed engine. Additionally dedicated urea storage tanks, pumps and injection and control systems must also be provided. Urea consumption is about 20 g/kWh for a medium speed engine when the target NO_X value is 2 g/kWh. With urea costing typically $ 200/tonne the urea costs can easily rise up to one fourth of the fuel costs.

[30] [Verkley, 1998]
[31] [Nurmi, 1998]

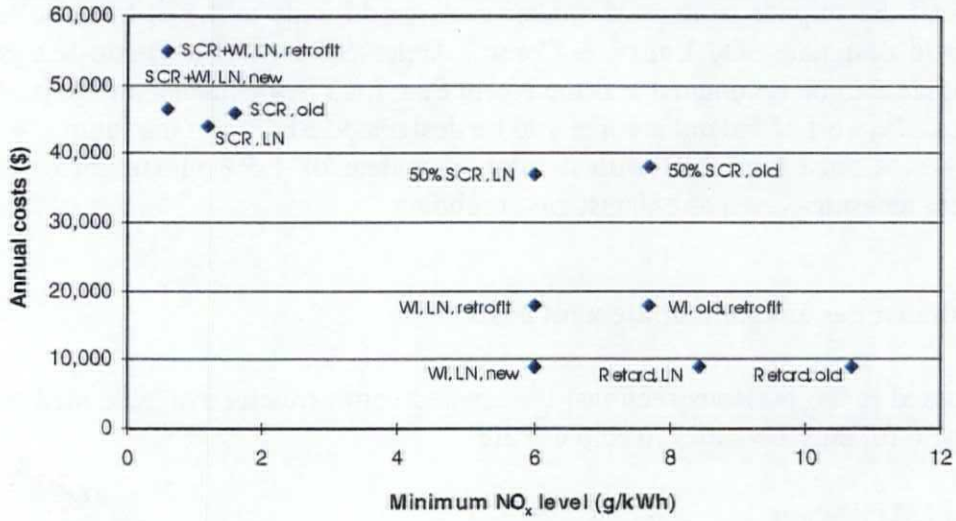


Figure 63: Annual costs of NO_X emission abatement for a 1,500 kW auxiliary diesel

Key: SCR = Selective Catalytic Reduction, WI = Water Injection, LN = Low NO_x engine, Retrofit = Retrofitted engine, New = New engine, Old = Old engine, Retard = Retarded injection

Typical investment costs for a DWI system are \$ 20-30 per kW and \$ 30-50 per kW for SCR[32]. The operating costs for the same systems are \$ 0.25-0.35 and \$ 0.4-0.5 per kg NO_X reduction, respectively. A NO_X emission limit of 6 g/kWh, would result in the following investment cost and annual operating costs increase for the three vessels:

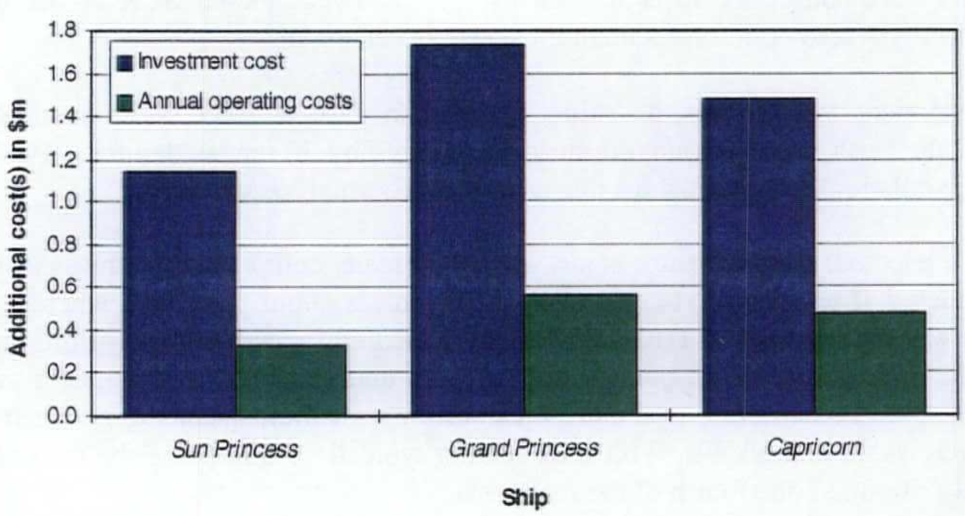


Figure 64: Extra cost(s) associated with NO_X reduction to 6 g/kWh: DWI

[32] According to Wärtsilä NSD

7.5.4 Noise emissions

The misunderstanding that gas turbines are very noisy typically arises from the association with gas turbines installed in aircraft. However, there are significant differences between installation of gas turbines in a ship and installation in an aircraft. These differences result in shipboard gas turbine installations having low noise levels, substantially lower than typical diesel engine installations. Weight and volume restrictions in an aircraft installation limit the noise reduction measures that can be taken.

There are four common sources of noise associated with an engine whether it is a gas turbine or a diesel engine. Direct airborne noise radiated into the engine room from the engine casing, structureborne noise from engine vibrations transmitted through the engine mounts into the ship structure, intake noise and exhaust noise.

1. **Airborne engine** room noise, radiated into the engine room by a gas turbine is attenuated by installing the engine in an acoustically insulated enclosure which reduces the noise level. This is generally lower than the noise level from a large medium speed diesel (see Figure 65).The WR-21 sound pressure levels represent the pressure levels in accordance with MIL-STD-740-1[33].

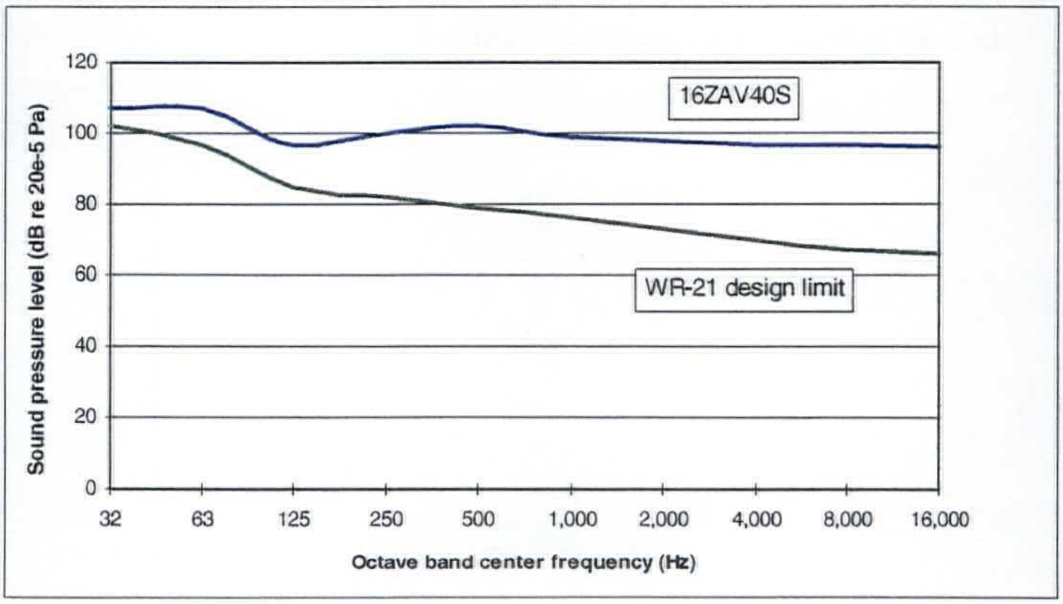


Figure 65: Air borne noise comparison

[33] Military standard: *Airborne sound measurements and acceptance criteria of shipboard equipment*

2. Gas turbine **structureborne noise** is also much lower than that of a diesel engine. The lightweight gas turbine with balanced rotating shafts operating in a continuous combustion process is inherently less vibration-inducing than a heavy multiple cylinder reciprocating diesel engine. In addition the resilient mount system is much simpler and easier to maintain because of the lighter weight of the gas turbine. This results in cabins near the engine room in a gas turbine ship being of higher value.
3. Gas turbine **intake noise** is composed primarily of high frequencies which are relatively easily attenuated using low density noise absorbing and attenuating materials. Generally intake silencers are used to attenuate the intake noise. An intake silencer can usually be accommodated in the inlet duct with very little, if any, increase in cross-sectional area of the duct (see also section 7.4.2: **Power plant dimensions**)
4. Most gas turbine **exhaust noise** is concentrated in the very low frequency end of the spectrum where the relative response of the human ear is low. Gas turbine exhaust ducts are normally internally lined with a thermal insulation which also serves as acoustic insulation. This usually provides sufficient attenuation and most shipboard gas turbine installations do not use exhaust silencers, The final determination as to whether an exhaust silencer is required is made once the height and orientation of the exhaust stack is determined with respect to noise level requirements on open deck areas. The open deck - funnel top distance varies from 15 to 20 metres for the selected ships.

7.6 Lubricating oil costs

Gas turbines only use lub oil for lubrication and cooling of the various gas generator and power turbine bearings and external gear cases of the accessory drive. Lub oil in diesel engines on the other hand is used for lubrication and cooling of: various bearings, crankshaft, camshaft, turbocharger and pistons/cylinders. A certain amount of lub oil is burnt in the combustion process and discharged with the exhaust gases when lubricating pistons and cylinders. Consequently the lub oil consumption for diesel engines is higher.

7.6.1 Specific lubricating oil consumption

Lub oil consumption for medium speed diesel engines, as obtained from the project guides varies from 0.7 up to 1.4 g/kWh (average 1 g/kWh). However, in practice a 50% addition is necessary[34] to account for:

- tolerances in the project guides
- leakage
- the age of the engine

Consequently the lub oil consumption will add up to 1.5 g/kWh.

Recently manufactured engines however demonstrate a lower lub oil consumption due to an anti-polishing ring in the upper part of the cylinder liner. This anti-polishing ring eliminates the risk of bore polishing. The purpose of this ring is to 'calibrate' the carbon deposits on the piston top land to a thickness small enough to prevent contact between the liner inner wall and the deposits on the piston top land. The so-called bore polishing which is avoided here, can lead to local liner wear and increased lub oil consumption.

The generic lub oil consumption for engines without this measure is 1 g/kWh, whereas the lub oil consumption for engines with anti-polishing ring reduces to 0.6 g/kWh. Again taking into account the 50% addition, this yields a specific consumption of 0.9 g/kWh. This value will be used for evaluation of the annual lub oil costs.

A typical gas turbine lub oil consumption is less than 0.1 litre per hour[35] , regardless of the engine type and load. Converted to kg/h (lub oil density at 70 °C: 850 kg/m^3) this amounts to 0.085 kg/h. The 0.1 litre per hour will be adopted for both WR-21 and 601-R engines. The annual lub oil consumption are presented in Appendix 19. Converted this amounts to 0.004 g/kWh for a WR-21 and 0.015 g/kWh for a 601-R, for conditions in practice at 100% MCR.

[34] According to data from P&O Cruises
[35] According to Rolls-Royce

7.6.2 Lubricating oil costs

In general two types of lub oil can be used: mineral oils or synthetic oils. Diesel engines use the cheaper mineral oils whereas gas turbines use synthetic oil. Generic bunker prices are: $ 2,000 per tonne[36] for mineral oil and $ 9,300 per tonne for synthetic oil. This results in an annual lub oil cost comparison as shown below.

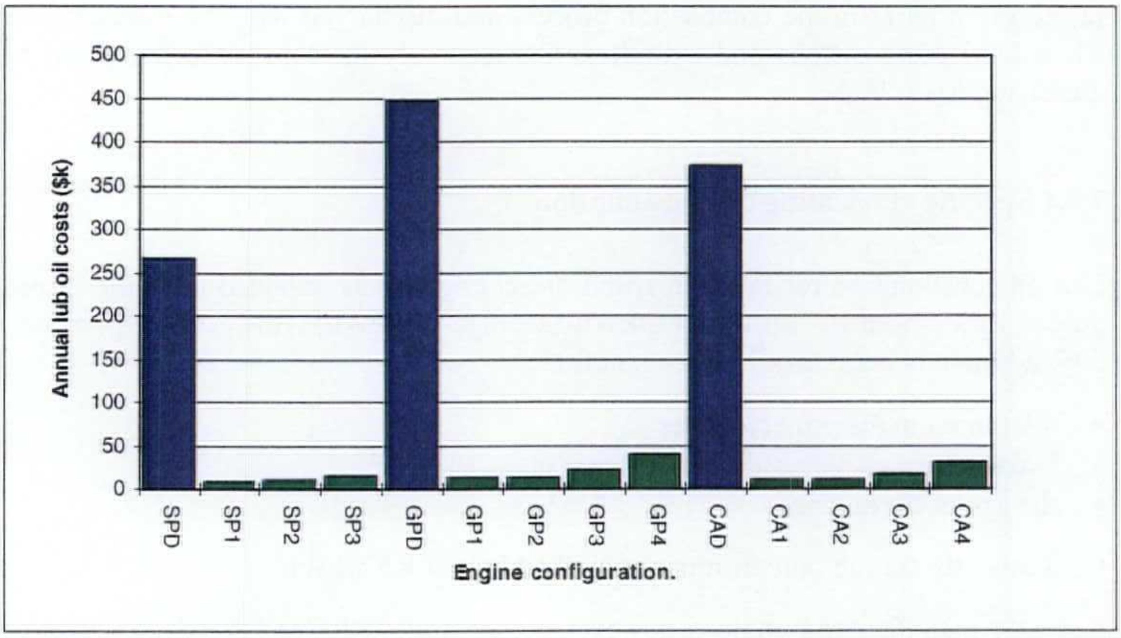


Figure 66: Annual lub oil costs

One can infer from Figure 66 that an average of $ 250k for *Sun Princess*, $ 425k for *Grand Princess* and $ 350k for *Capricorn* can be saved annually on the lub oil bill. It should be emphasised here that the lub oil consumption is valid for an engine that has been running for a few years and has not experienced cylinder liner wear problems.

7.7 Maintenance and spares

Maintenance, overhaul and associated spare parts requirements are based on expected lifetimes of engine components. Overhaul intervals and component lifetimes in practice however will vary and are subject to:

- environmental and operating conditions
- fuel and lubricating oil qualities
- engine load factor
- fuel and lubricating oil care

[36] Quotation from British Petrol

- overhaul according to engine manuals
- genuine spare parts used

The lifetimes for the major components are predicted in order to be able to create maintenance and overhaul schedules. The specific maintenance contracts are based on these schedules.

The predictable part is the scheduled maintenance. Scheduled maintenance consists of a number of checks and inspections at fixed times. This however will represent only a part of the total maintenance costs.

Gas turbines maintenance and overhaul

WR-21 maintenance is divided in three groups:

- All scheduled maintenance (crew on board)
- Organisation level (crew on board) unscheduled change-outs including:
  - Starter
  - Fuel pump
  - Lub oil pump
  - Combustor cans
- Intermediate level (I-level team, dockside facility and on-board) change-outs including:
  - Gas generator
  - Power turbine
  - Intermediate pressure compressor
  - Tail bearing
  - Recuperator module*
  - Intercooler segment*

A summary of scheduled maintenance or preventive maintenance tasks is shown in the figure below.

* Performed in factory

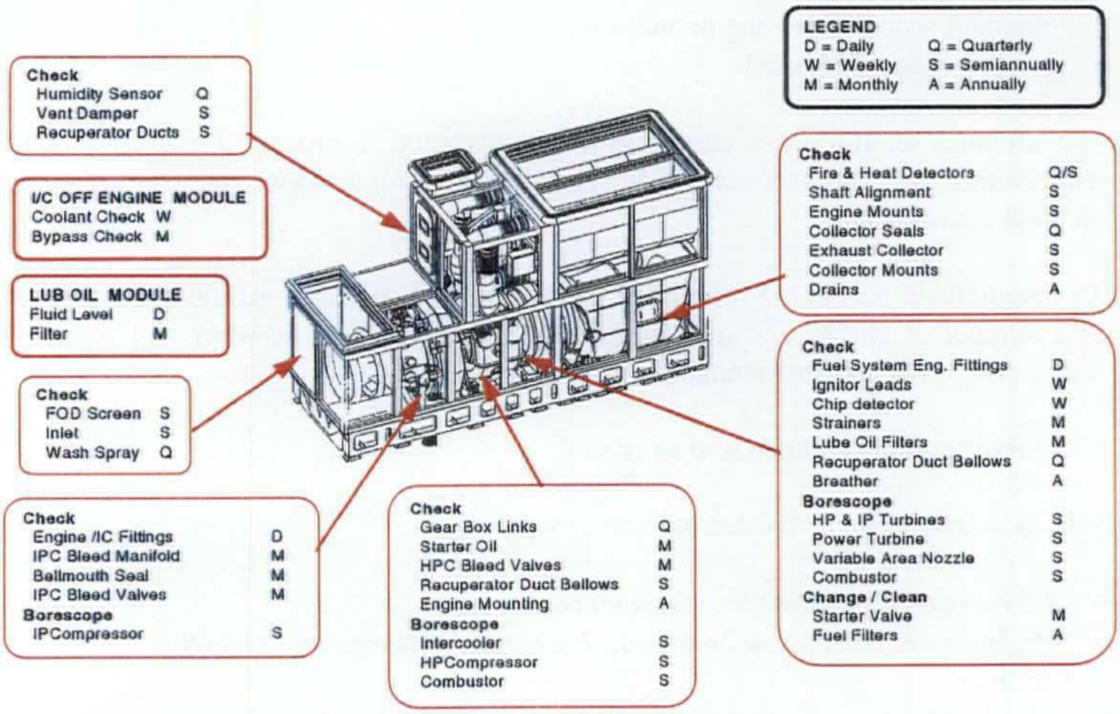


Figure 67: WR-21 scheduled maintenance summary

The entire WR-21 system has been designed to allow all scheduled maintenance to be performed by the crew on board. All scheduled maintenance averages fewer than 4.2 hours/week, including crank and on-line water washing. Gas generator sideways removal can be performed within 36 hours.

The WR-21 water wash system provides gas path cleaning for the compressors and intercooler for removal of salt deposits and dirt from internal surfaces. Water washing is recommended every 24 to 36 hours of engine operation. This requires approximately 80 litres of water. On line water washing can only be carried out while the engine load is low (possibly idle running).

The predicted Mean Time Between Repairs (MTBR) of the gas generator is 12,600 hours and 21,300 hours for the power turbine.

The maintenance envelope as described above has resulted in an average maintenance and overhaul cost of £ 70 ($ 112) per fired hour plus a £ 50,000 fixed fee for depot maintenance by Northrop Grumman. This is calculated with a maintenance model that is currently being developed at Rolls-Royce.
The maintenance model makes a distinction between scheduled and unscheduled events. The latter represents corrective maintenance for failures which are likely to occur with the current information available.

The results of this maintenance model should be used with care. The cost per fired hour should only be used as an indication because this depends on the actual engine load profile. Furthermore, the model is still being developed and is based on a WR-21 running according to the cube propeller law. Finally, the model is based on the information about the predicted failure rate of the engine parts. This information is for generally given within a safety margin because at this moment the actual failure rate can not be guaranteed.

The 601-R total maintenance and overhaul budget costs are estimated at $ 50 per fired hour.

Diesel engine maintenance and overhaul

The inspection and overhaul intervals for a ZA40S are shown in Table 27 below. These are numbers as presented in the project guides. In practice, these intervals are probably longer.

Component	*Inspection or overhaul*		*Lifetime (x 1,000 hours)*
	Interval (hours)	*Work time (mins.)*	
Fuel nozzle	4,000	20	8
Cylinder head	12,000	90	60
Inlet valve		5	24-36
Exhaust valve		5	24-36
Piston		50	48-60
Piston rings		10	12
Scraper ring		5	12
Piston cooling space	24,000	45	N/A
Piston ring groove		45	36-48
Rotating mechanism		45	48-60
Top - end bearing		45	48-60
Bottom - end bearing	Checks at random	50	24-36
Main bearing		70	24-36
Fuel pump plunger		40	24-36
Valve seat		20	24-36
Cylinder liner		45	48-60

Table 27: ZA40S inspection intervals and component lifetimes

The spares requirements for a 14V 48/60 engine are shown in Appendix 20. The table mentioned in this Appendix displays the spare parts volume to be supplied with the engine in addition to the classification spare parts volume for the first 30,000 running hours. This amount of running hours typically represents a period of 6 to 7.5 years. The total spares cost for this period would amount to $ 1.7m for *Capricorn*, or $ 225k to $ 280k annually. A modular gas turbine design results in a reduction of the number of spare parts.

Indicative values for scheduled maintenance, including spares and work hours as obtained form various quotations and articles are listed below.

1. 15-20 % of the annual fuel costs, depending on age and condition of the engine
2. $ 250 k for a 14 MW engine (burning HFO)
3. 4-6 % diesel engine first cost annually, depending on age and condition of the engine
4. Dfl 5/MWh, or $ 2.4/MWh for a Wärtsilä 38 engine (burning HFO)
5. $ 28 per running hour for an 11,500 kW diesel engine

The table below shows the diesel-electric properties for each vessel associated with the points listed above.

	Sun Princess	*Grand Princess*	*Capricorn*
Annual fuel costs	$ 2.31m	$ 3.64m	$ 3.50m
Installed diesel power	46,100 kW	69,199 kW	$ 58,800m
Diesel engine first cost	$ 12.2m	$ 18.3m	$ 15.6m
Annual energy consumption	1.33E+05 MWh	2.17E+05 MWh	2.13E+05 MWh
Acc. diesel running hours	17,500	27,600	19,400

Table 28: Properties for indicative maintenance costs calculations

This yields the following annual maintenance costs for the vessels:

	Sun Princess	*Grand Princess*	*Capricorn*	
1	350-460	550-730	530-700	$k
2	820	1230	1050	$k
3	490-730	730-1100	620-940	$k
4	320	520	510	$k
5	490	770	690*	$k
Average	530	820	730	$k

*: the increased power output per engine has been taken into account

Table 29: Indicative annual diesel plant maintenance costs

Information on maintenance costs in practice[37] are shown in Figure 68. This figure contains information of four Princess ships: *Dawn Princess*, a sister-ship of *Sun princess* which is delivered in 1997, *Sun Princess*, and two sister-ships *Regal Princess* and *Crown Princess*, both delivered in 1990.
As opposed to the 12,000 hours piston overhaul time as quoted by the manufacturers, cruise lines overhaul the pistons every 10,000 hours.

[37] Data obtained from Princess Cruises

All four ships are diesel-electric with four diesel engines and two electric motors. The difference in costs from one year to the next is due to the fact that one year only one engine will be overhauled and the next year maybe three. Also as the engines get older there are bearing changes, flexible coupling changes, etc. that come into play.

The values for *Sun* and *Dawn Princess* are considered not to be representative because they are only known for the first few years in service. The average annual maintenance costs for *Regal* and *Crown* Princess is \$ 400k. The installed power on these ships is 39,000 kW, which is delivered by four 7L 58/64 MAN B&W engines.

The relation between spare parts consumption and cylinder size and number can be estimated by using the empirical equation for the engine 'wear rate':

No. of cylinders x cylinder diameter x mean effective pressure x piston speed[38]

Consequently, the 16 ZAV40S/7L 58/64 maintenance cost ratio becomes:

$$\frac{16}{7} * \frac{400}{580} * \frac{24.1}{21.9} * \frac{9.5}{9.1} = 1.8$$

Therefore an indication for the annual maintenance costs for *Sun Princess* after four years in service is \$ 700k. The fact that this value is higher can be caused by the fact that the values in Table 29 are scheduled maintenance costs, while Figure 68 displays the actual costs including unscheduled events.

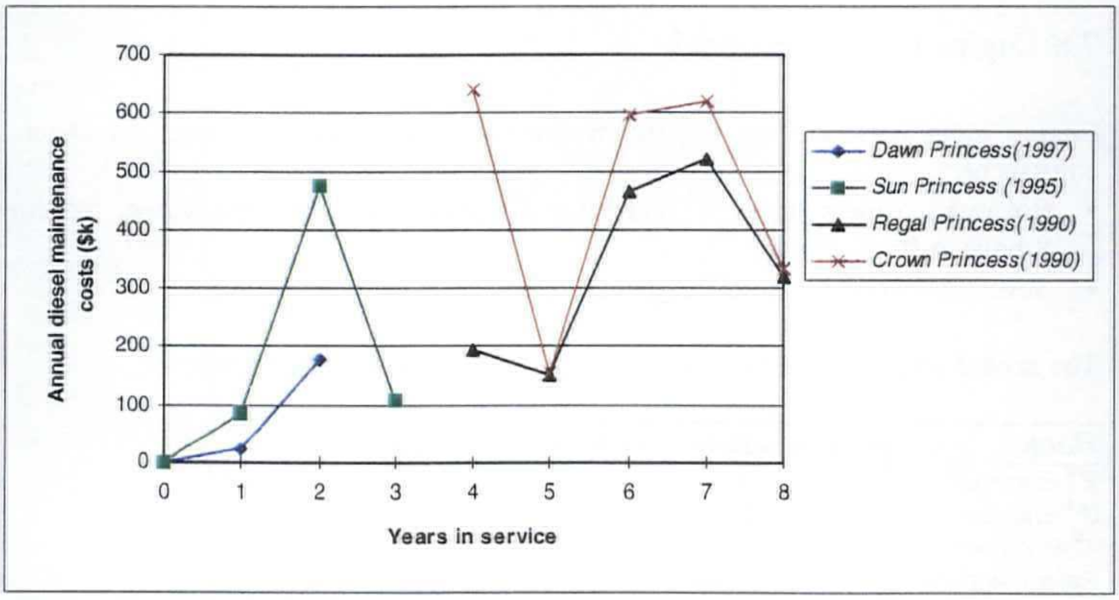


Figure 68: Annual diesel maintenance costs in practice

The annual maintenance costs for the other two vessels is estimated at \$ 1080k and \$ 960k. These values represent the averages from Table 29 multiplied by a factor

[38] [Nurmi, 1998]

700/530. Figure 69 below displays the annual maintenance costs. The associated annual gas turbine running hours are displayed in Figure 26. Converted to cost per running hour this amounts to \$ 36 for 16 ZAV40S and \$ 45 for 14V 48/60.

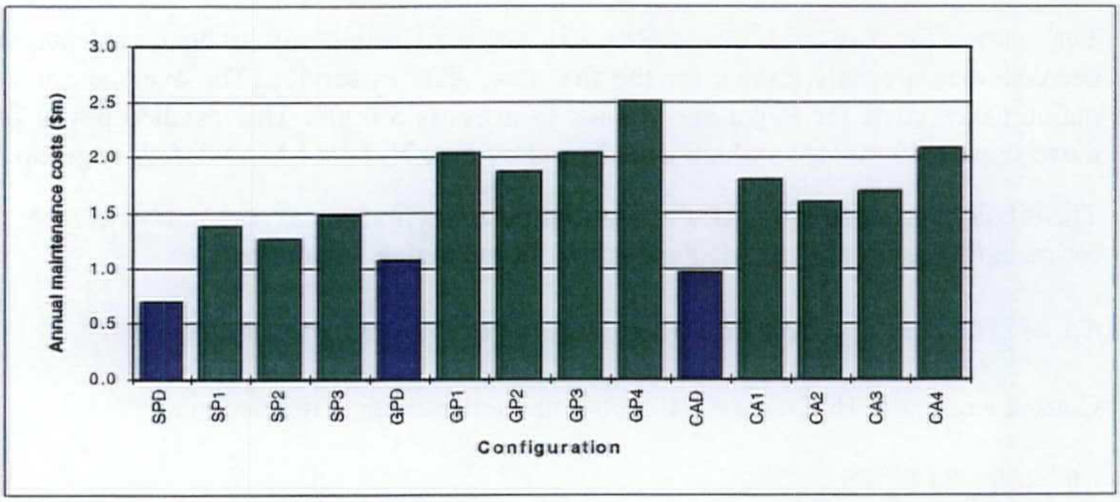


Figure 69: Annual maintenance costs comparison

The fact that the maintenance costs for SP2 are actually lower than SP1 is caused by the fixed depot fee of \$ 80k.

7.8 Engine room personnel

Engine room personnel on *Capricorn* (based engine room personnel on *Oriana*) consists of:

- Six watchkeepers (three 3rd and three 4th engineers) and three wipers for three 8-hour shifts.
- Seven day workers: one 2nd engineer, one 4th engineer and 5 fitters.

The annual salaries (based on 243 working days a year) are shown below.

Rank	*Annual salary*	*Number*	*Subtotals*
2nd Engineer	\$ 57,000	1	\$ 57,000
3rd Engineer	\$ 41,800	3	\$ 125,400
4th Engineer	\$ 28,800	4	\$ 115,200
Fitter/mechanic	\$ 12,800	5	\$ 64,000
Wiper	\$ 12,000	3	\$ 36,000
Total	-	**16**	**\$ 397,600**

Table 30: Engine room personnel and salaries for *Capricorn*.

It is assumed that the engine room personnel on *Sun Princess* is similar. The engine room personnel onboard *Grand Princess* consists of:

Rank	*Annual salary*	*Number*	*Subtotals*
1st Engineer	$ 61,900	1	$ 61,900
2nd Engineer	$ 57,000	1	$ 57,000
3rd Engineer	$ 41,800	3	$ 125,400
4th Engineer	$ 28,800	4	$ 115,200
Fitter/mechanic	$ 12,800	6	$ 76,800
Wiper	$ 12,000	4	$ 48,000
Total	-	**19**	**$ 484,300**

Table 31: Engine room personnel and salaries for *Grand Princess*

As discussed with P&O Cruises, the engine room personnel for a gas turbine power plant can be reduced by:

- *Sun Princess and Capricorn*: one 3rd engineer and two fitters.
- *Grand Princess*: one 1st engineer, one 3rd engineer, three fitters and one wiper.

This yields a reduction on engine room personnel costs by $ 68k and $ 154k respectively. In addition, a certain number of crew cabins can be replaced by revenue generating passenger cabins. This amount is two on *Sun Princess* and *Capricorn* and four on *Grand Princess*. The associated annual extra revenues are: $ 215k and $ 430k respectively.

7.9 Life cycle costs

Due to the higher genset initial cost and extra cabin construction costs, the total investment costs for a gas turbine electric plant are higher. This is compensated slightly by the reduction in oil fired boiler capacity. Obviously, the net cash flows have to be positive to pay back the investment. The viability of the investments will be investigated according to two economic methods: the pay back period and net present value method[39].

7.9.1 Investment cost and operating costs summary

The extra investment cost for the gas turbine configurations is displayed in Figure 70. The values are also presented in Appendix 21. The initial cost for the gas turbine configurations include a reduction for the installation cost.

[39] See for example [Blommaert&Blommaert, 1995]

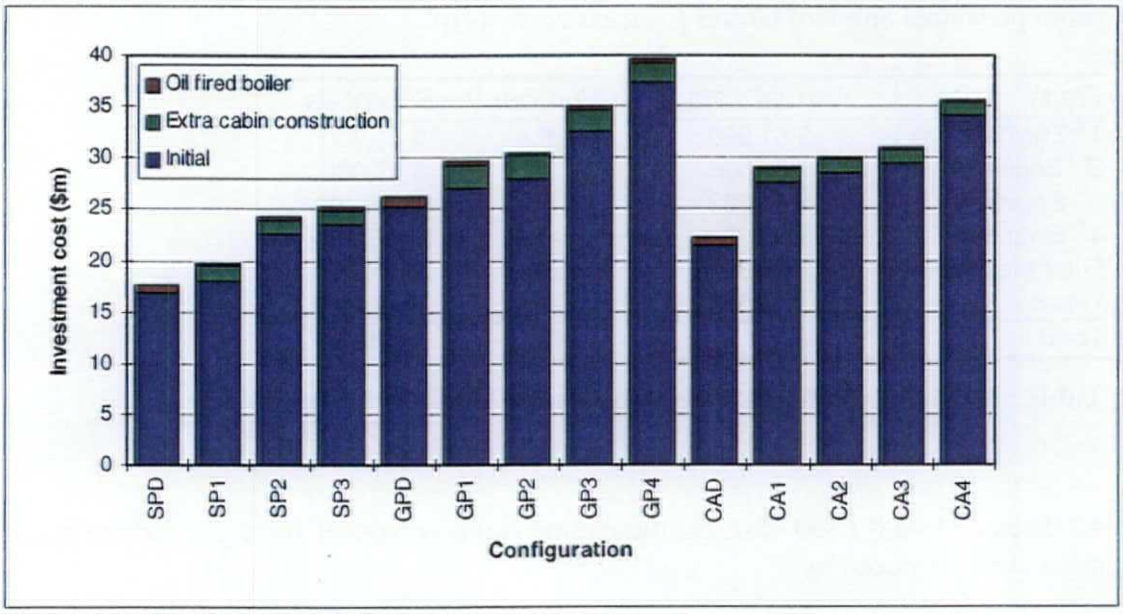


Figure 70: Investment cost comparison

A summary of the annual operating costs is presented below (see also Appendix 21).

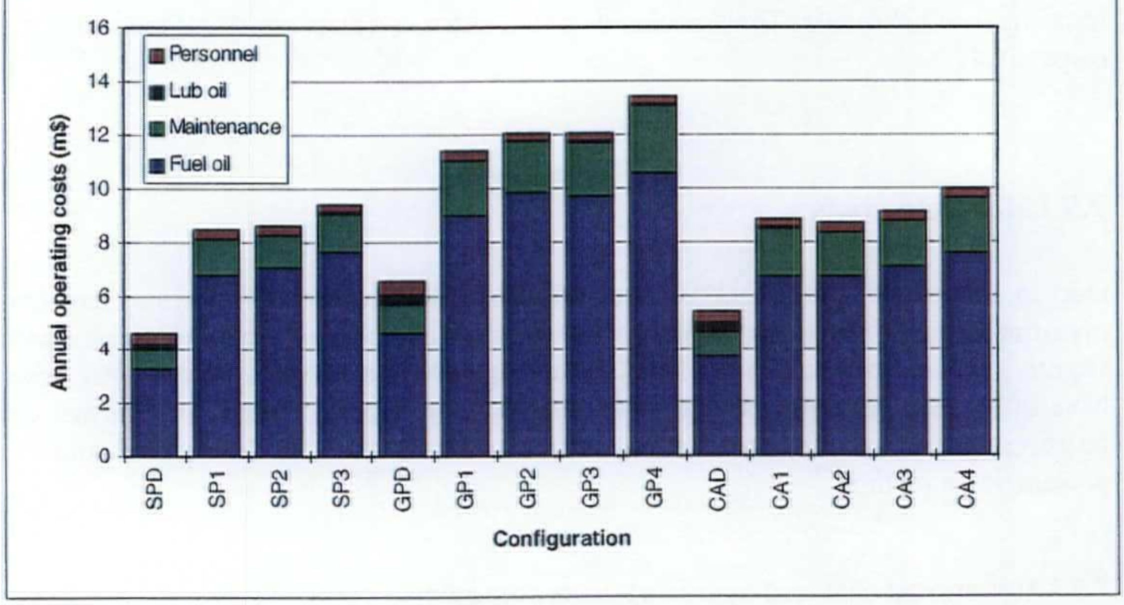


Figure 71: Annual operating costs comparison

The higher operating costs from Figure 71 should be compensated by the additional cabin revenues (Table 25) to generate a net positive cash flow. As can be seen in Figure 72 this is the case for all configurations.

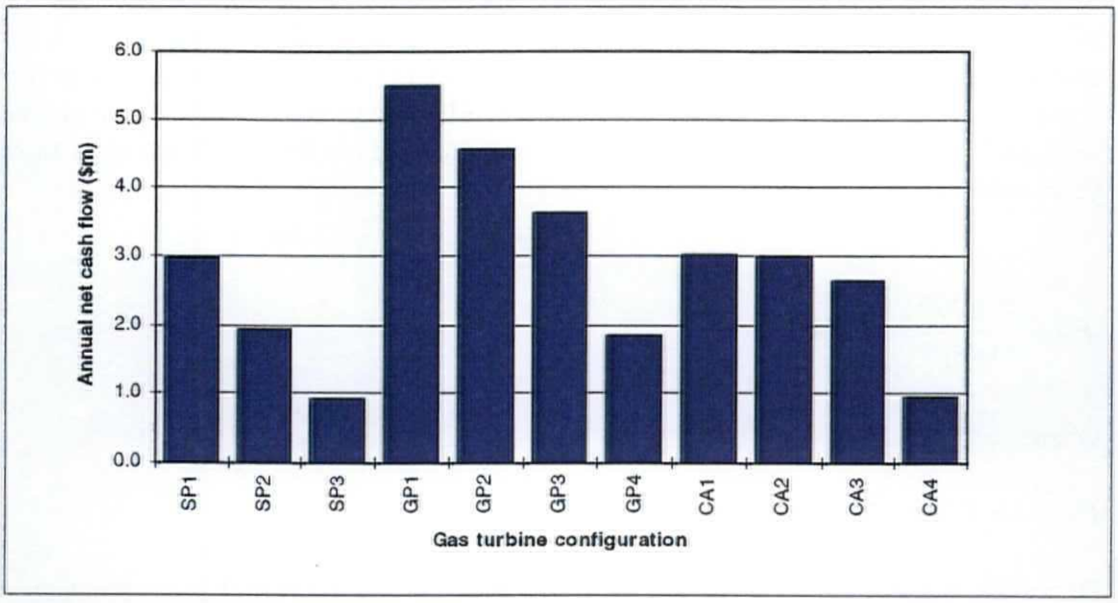


Figure 72: Annual net cash flow

7.9.2 Investment appraisal

The pay back period is the number of periods (in this case years) which are required to recover the present investment by means of positive net cash flows resulting from the investment. This simple method does not take into account the space of time over which the cash flows come in. It should be emphasised here that the investment only compares the configurations. Consequently, the pay back period for a gas-turbine as calculated below is actually a relative pay back time. For example, if a cruise line would opt for gas turbine power systems instead of the traditional diesel systems, it would take an x amount of years to pay back the investment involved in this switch. After that period, the investment is paid for and the net cash flow turns into profit. The pay back period for the 11 gas turbine configurations are presented below.

Pay-back period (years)					
SP1	0.8	GP1	0.7	CA1	2.4
SP2	3.5	GP2	1.0	CA2	2.7
SP3	8.3	GP3	7.3	CA3	3.4
		GP4	7.8	CA4	14.3

Table 32: Pay back period

The net-present value or capital-value method calculates the present value of an investment over a certain period; in this case the service lives of the vessels - currently set at 30 years. The method is based on the fact that a future cash flow worth less than cash now because cash now available can be invested. For example: with an interest rate of 6% the net present value of a cash flow of $ 1,000 in thirty years time can be calculated as follows:

$$PV = \frac{\$\ 1{,}000}{(1.06)^{30}} = \$\ 174.1,$$

where:

PV = Present Value [$]

Hence the present value of the annual cash flows minus the initial investment gives the net present value and thus an idea of the economic viability of an investment. In general, the net present value of future cash flows can be calculated according to the following equation:

$$NPV = PV - I = \frac{CF}{(1 + ir/100)^{n}} - I,$$

where :

NPV = Net Present Value [$]
I = Investment [$]
CF = Cash Flow [$]
ir = interest rate per period, assumed 6% per year
n = term, assumed 30 years

The total present value of the annual cash flows (see Figure 72) is calculated as follows:

$$\sum_{1}^{30} PV_i = \frac{CF_i}{(1 + ir_i/100)^{i}}$$

The present and net present values for the 11 gas turbine configurations are displayed below.

	Extra investment cost (table 30)	Present value of cash flows	Net present value
SP1	$ 2.1m	$ 40.9m	$ 38.7m
SP2	$ 6.7m	$ 26.8m	$ 20.0m
SP3	$ 7.7m	$ 12.9m	$ 5.1 m
GP1	$ 3.4m	$ 75.7m	$ 72.3m
GP2	$ 4.3m	$ 62.9m	$ 58.6m
GP3	$ 8.8m	$ 50.2m	$ 41.4m
GP4	$ 13.5m	$ 25.6m	$ 12.0m
CA1	$ 7.0m	$ 41.8m	$ 34.8m
CA2	$ 8.0m	$ 41.3m	$ 33.4m
CA3	$ 8.9m	$ 36.6m	$ 27.6m
CA4	$ 13.5m	$13.0m	$ -0.5m

Table 33: Net present value of the annual cash flows

Except for configuration CA4, the net present values are positive. Significant profits can be made not only with 'WR-21 only' configurations, but also with the 'combined' configurations.

7.9.3 What if scenario

This section looks at the effect on net present value in the event of variations in first cost, operating costs and revenues. Realistic events are believed to be:

1. $ 30/tonne HFO bunker price increase for sulphur free fuel
2. Occupancy decrease from 100% to 75%
3. NO_X emission limit of 6 g/kWh
4. Interest rate increase from 6 to 10%
5. Only 75% of the extra available engine room space can effectively be used for additional cabins
6. Due to a more favourable layout, 125% of the extra available space can effectively be used for additional cabins
7. Diesel genset specific acquisition cost down to $ 300/kW (including controls and lub oil and cooling systems)

The difference between case two and five is the fact cabin construction has already taken place for case two. Case seven is based on the assumption that if necessary diesel manufacturers will reduce the genset acquisition cost to prevent gas turbines from entering the cruise market. The net present value for each change is shown in Table 34.

	Net present value ($m)							
	Base	1	2	3	4	5	6	7
SP1	38.7	53.6	14.2	43.4	25.1	14.6	60.8	34.9
SP2	20.0	35.0	-1.4	24.8	10.9	-1.1	39.3	16.3
SP3	5.1	20.0	-7.7	9.9	0.4	-7.4	23.6	1.3
GP1	72.3	93.1	35.5	80.2	47.4	35.8	105.8	66.6
GP2	58.6	79.4	23.0	66.5	37.8	23.6	90.6	52.9
GP3	41.4	62.1	8.4	49.3	24.5	8.9	70.9	35.7
GP4	12.0	32.8	-13.5	20.0	3.0	-13.0	40.1	6.3
CA1	34.8	51.4	11.2	41.6	20.8	11.6	55.5	29.7
CA2	33.4	50.0	10.5	40.2	19.5	10.9	53.4	28.5
CA3	27.6	44.2	4.4	34.5	15.3	4.8	48.0	22.8
CA4	-0.5	16.1	-13.5	6.4	-5.4	-13.2	17.0	-5.3

Table 34: Effect on net present value in case seven different events

Clearly, the additional cabin revenues is the most critical factor (case 2 and basically case 5 as well). An occupancy decrease from 100 to 75% drastically reduces the net present value of the investment and in some cases even make the investment non-viable. This is certainly the case for the '601-R only' configurations.

It becomes clear that reducing diesel engine first cost does not have a significant effect on the net present values. In general, a variation in initial cost results in a relatively small change in on net present value. The pay-back period however, depends largely on the initial cost.

7.10 Availability, Reliability and Maintainability (ARM)

The availability *A(T)* of a system is the average part of a period during which a system can be used. Unavailability *U(T)*, the complement of availability occurs in case of failure, testing/maintenance or repair. Clearly they are related as follows:

$$A(T) + U(T) = 1$$

The reliability *R(T)* is the probability (*P*) that a system during a certain period *t*, under the specified ambient and usage conditions remains functional:

$$R(t) = P(t_s > t),$$

where T_s = time to failure

The maintainability is the probability that the time required for maintenance of a system (θ), given certain maintenance conditions, requires at most a certain period t:

$$M(t) = P(0 < \theta \leq t)$$

For this report it is not possible and moreover not considered useful to quantify the above mentioned properties. Not possible because insufficient information is available and not useful because the prime movers are merely a part of the total power system. The overall availability of (electric) power is namely dependent on the prime movers and all its auxiliaries.

An aspect closely related to ARM is redundancy. Redundancy is the ability of a system to maintain or restore its function when component failure has occurred. The power availability and the influence on maximum speed will be investigated for every configuration in case of either prime mover failure or department flooding. The relevant data from Tables 1, 7 and 13 are summarised below. The values are valid for conditions in practice (summer).

	No. of prime movers (-)	*Available electric power (kW)*	*Maximum speed (knots)*
SPD	4	4x11,200=44,800	22.4
SP1	2	2x21,300=42,600	22.4
SP2	5	1x21,300+4x5,700=44,100	22.4
SP3	7	7x5,700=39,900	22.4
GPD	6	6x11,200=67,200	23.9
GP1	3	3x21,300=63,900	23.9
GP2	5	2x21,300+3x5,700=59,700	23.9
GP3	8	1x21,300+7x5,700=61,200	23.9
GP4	11	11x5,700=62,700	23.9
CAD	4	4x14,300=57,200	25.5
CA1	3	3x21,300=63,900	25.5
CA2	5	2x21,300+3x5,700=59,700	25.5
CA3	7	1x21,300+6x5,700=55,500	25.5
CA4	10	10x5,700=57,000	25.5

Table 35: Summary of available genset power and maximum ship speed

The effect on power availability and maximum ship speed will be investigated for the following cases:

- loss of one prime mover at sea; in case of configurations SP2, GP2, GP3, CA2 and CA3 this will be loss of either a WR-21 or a 601-R (two possibilities).
- flooding of (or fire in) one out of the two watertight compartment at sea comprising gensets; in case the prime movers are not equally distributed over the two watertight compartments this obviously yields two possibilities for each case (see Appendix 15, Figures 8 to 10).

If necessary the hotel load can be reduced by (temporarily) switching off non-vital users from galley, air conditioning, accommodation, etc. Obviously this can only be done temporarily. If this is not possible the maximum ship speed will be further reduced. The estimated hotel load under these circumstances are: 8 MW, 14 MW and 9 MW, respectively. In case of an emergency this load is further reduced. The available genset power and the maximum attainable speed (engines 100% load) are presented below. The power available for propulsion has been reduced by the adjusted hotel load.

The left-hand column under 'Loss of one prime mover' depicts the loss of one WR-21 whereas the right-hand column represents the genset power or maximum speed in case of a 601-R loss. The left-hand column under 'Flooding of one compartment' displays the consequences in case of flooding of the left-hand watertight compartment in Appendix 15, the right hand column ...., etc.

	Available genset power (kW)				*Maximum attainable speed (knots)*			
	Loss of one prime mover		*Flooding of one compartment*		*Loss of one prime mover*		*Flooding of one compartment*	
SPD	33,600		22,400		21.9		18.8	
SP1	21,300		21,300		18.5		18.5	
SP2	22,800	38,400	22,800	21,300	19.0	22.4	19.0	18.5
SP3	34,200		22,800	17,100	22.2		19.0	16.5
GPI	56,000		44,800	22,400	23.9		22.2	15
GP1	42,600		42,600	21,300	21.7		21.7	14.3
GP2	38,400	54,000	32,700	27,000	21.0	23.6	19.4	17.5
GP3	39,900	55,500	44,100	17,100	21.3	23.8	22.1	<10
GP4	57,000		45,600	17,100	23.9		22.3	<10
CAD	42,900		24,150		24.3		19.4	
CA1	42,600		42,600	21,300	24.2		24.2	18.2
CA2	38,400	54,000	42,600	17,100	23.4	25.5	24.2	16.0
CA3	34,200	49,800	21,300	34,200	22.5	25.3	18.2	22.4
CA4	51,300		22,800	34,200	25.5		18.8	22.4

Table 36: Maximum attainable speed in case of prime mover loss

As can be derived from in Table 36, the maximum attainable speed in case of loss of one prime mover is more than 83% for each ship. In most cases the ship will be able to proceed the cruise without substantial delays.

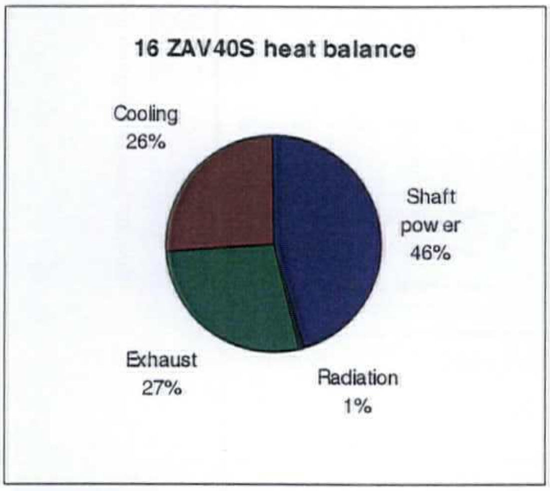
In case of flooding of one compartment the effects are clearly more severe, especially in case of unequal distribution of the gensets (GP3 and GP4). If necessary emergency gensets, required by the 'rules' will be switched on.

Another issue is the effect of either intercooler or recuperator failure. When subjected to a failure of the intercooler system, both the intercooler and recuperator are run in bypass mode. Under these conditions, the WR-21 will remain operational, but the maximum power output is limited to about 13.5 MW at ISO no loss conditions (25.2 MW under normal cicumstances). In case of failure, the recuperator will be bypassed. This has no restriction on power capability. The effect on specific fuel consumption is discussed in section 7.1.3: **Meeting the steam demand**.

8. Alternatives for optimum plant usage

8.1 Waste heat

As mentioned previously, current practice with diesel engines on cruise ships is to use the exhaust gas to generate process steam (section 7.1.3: **Meeting the steam demand**) and the high temperature cooling water as heating mediumfor evaporators. As shown in Figure 73, the heat balances for diesels and gas turbines differ notably. The amount of energy rejected with the exhaust gases is significantly more for gas turbines. This is reflected by the maximum steam production: 4,500 kg/h and 18,000 kg/h respectively, whereas the rated power for the gas turbine is about twice as high. For simple cycle engines this amount is relatively larger, due to the lack of an intercooler.



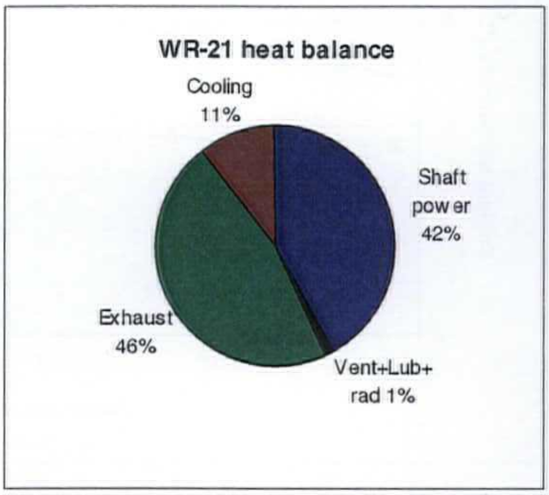


Figure 73: Diesel and gas turbine heat balance (conditions in practice, 100% MCR)

The high temperature diesel jacket cooling water is available at 90 °C with the existing heat exchangers. This temperature is high enough for heating evaporators, because flash or vacuum evaporators work with pressures below 1 bar. As a result the boiling point of water is lower. The boiling point at 0.07 bar for example is only 40°C. Obviously, maintaining a low pressure requires a certain amount of energy. The associated energy flow for a 16 ZAV40S is about 5 MW.

The WR-21 dissipates 11%, or 6.5 MW of the total fuel energy through its intercooler. Consequently, the possibilities for effective use of this waste heat will be investigated. It has to be emphasised that it is not possible with the 601-R, because this engine does not incorporate an intercooler. Potential users for this low temperature heat are evaporators, swimming pool heaters and sanitary fresh water heaters. If the coolant heat can effectively be used for either of these, the steam load can be reduced and this should result in fuel savings and/or a reduction of waste heat and oil fired boiler capacity.

The WR-21 intercooler coolant inlet/outlet temperatures and associated heat dissipation is shown in Figure 74. The latter represents the heat that has to be rejected for optimum intercooler use. No more and no less.

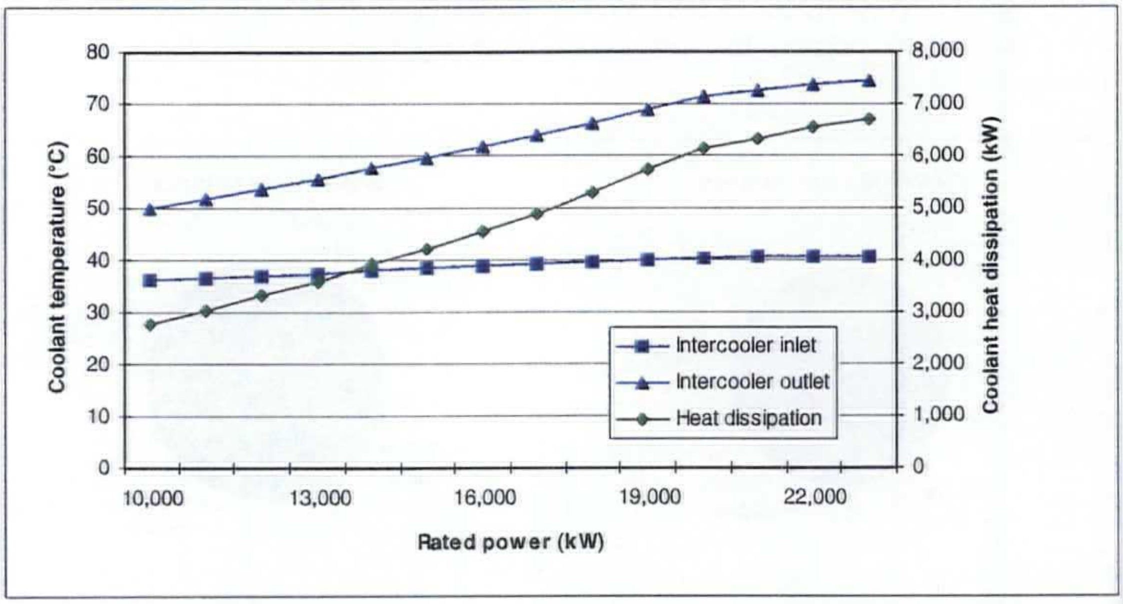


Figure 74: WR-21 coolant inlet/outlet temperature and heat dissipation (25°C)

Regarding the amount of heat that is available, the obvious choice is to use the heat for evaporators. A simplified flow diagram of a multi-stage flash (MSF) sea water desalination plant is illustrated in Figure 75[40].

The sea water (feed) flows under positive pressure through the tubes of a number of condensers from the last stage to the first stage whereby it is heated gradually by the vapour condensing in the various stages. After leaving the first stage condenser, the sea water flows through the brine heater where the heat input to the plant causes a further temperature increase. This heat input can either be delivered by steam, cooling water or a combination of both.

[40] This illustration is copied from a data sheet from Serck Como

The sea water leaves the brine heater at the brine top temperature of approximately 80°C. Up to this point, the pressure of the sea water is above atmospheric pressure and therefore below boiling pressure.

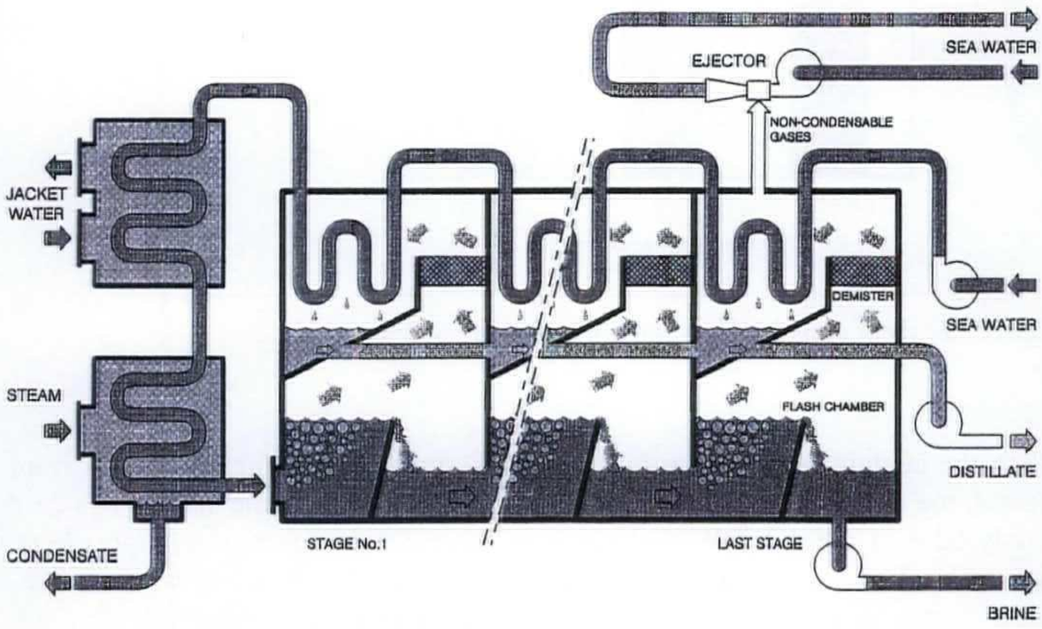


Figure 75: Simplified MSF evaporation plant flow diagram

The sea water is then directed into the first stage of the plant which is at a pressure below boiling pressure. In order to return to a state of equilibrium, part of the sea water flashes off such that the saturation temperature corresponds to the pressure in the stage. This process is repeated from stage to stage whereby the pressure and the temperature in each stage is less than that of the preceding stage. The brine is discharged from the last stage by the brine pump. The distillate is drawn through from the first to the last stage condenser where it is discharged by the distillate pump. The non-condensable gases released in the various stages are discharged by the ejectors.

As discussed with Serck Como, a manufacturer of heat exchangers and evaporators, when the coolant temperature falls below 75°C, steam assistance is required. For example: the heat required for a single-stage-flash (SSF) evaporator, producing 180 m^3/day is 5.8 MW. Dependent on the coolant temperature, this heat is distributed as shown in Figure 76. As the return temperature is 46 °C (Figure 74), the surplus heat has to be discharged through the sea water cooler. The calculations are based on a 32 °C sea water temperature. Clearly, when this temperature is lower, the heat consumption will be higher.

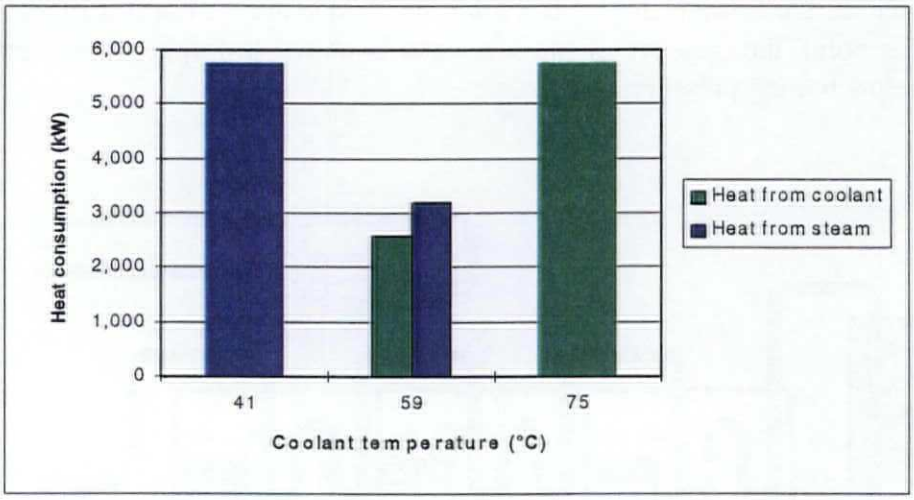


Figure 76: Heat consumption for a 180 m^3/day SSF evaporator.

When the coolant temperature falls to 59 °C, only 2.6 MW can be used from the coolant, the remaining 3.2 MW will have to be delivered by the steam. The 2.6 MW is only 62% of the 4.2 MW that needs to be rejected. Again, the surplus heat will have to be discharged through the sea water intercooler. In conclusion can be said that a certain amount of the heat rejected by the coolant can effectively be used. For preliminary steam load calculations (see Appendix 13) it is assumed that per engine half the amount of heat as shown in Figure 76 can effectively be used to reduce the steam load.

8.2 Intakes and uptakes

A wide variety of alternative intake and uptake arrangements can be applied to cruise ships. In section 7.4.2: **Power plant dimensions**, an arrangement conform the existing diesel engine casing was chosen for the comparison. Both intakes and uptakes were mounted in a central casing in the ship. This resulted in similar casing dimensions for the diesel plant and the gas turbine plant.

As mentioned before, another position for the air intakes in on each side of the ship. This arrangement for a 80,000 tonnes COGES ship[41] (see also next chapter) is shown in Figure 77. This results in a compact engine casing, above deck 6. On the other hand, the filters and plenum chamber take up a significant amount of space on deck 6. Typically located on this deck are restaurants and bars.

[41] [Nurmi, 1996]

Moreover the filter requirements will be more strict, since there will be more spray and probably green water. This will require a three stage filter possibly in combination with a primary louvre. Also shown in this drawing is an alternative ventilation air intake duct, mounted downstream on the combustion air intake duct. Clearly this short ducting results in some space saving.

An estimation of the total intake arrangement volume is 350 m^3 for the dual engine configuration. The gas turbines in the figure below are similar in size and power to a WR-21, so it is allowed to compare the COGES intakes with the intake arrangement in section 7.4.2: **Power plant dimensions**. Since an 80,000 tonnes ship is typically one or two decks lower, the estimated intake arrangement volume for two gas turbines is 430 m^3. Consequently a space saving of 80 m^3 can be achieved by mounting the intake filters on the sides of the ship.

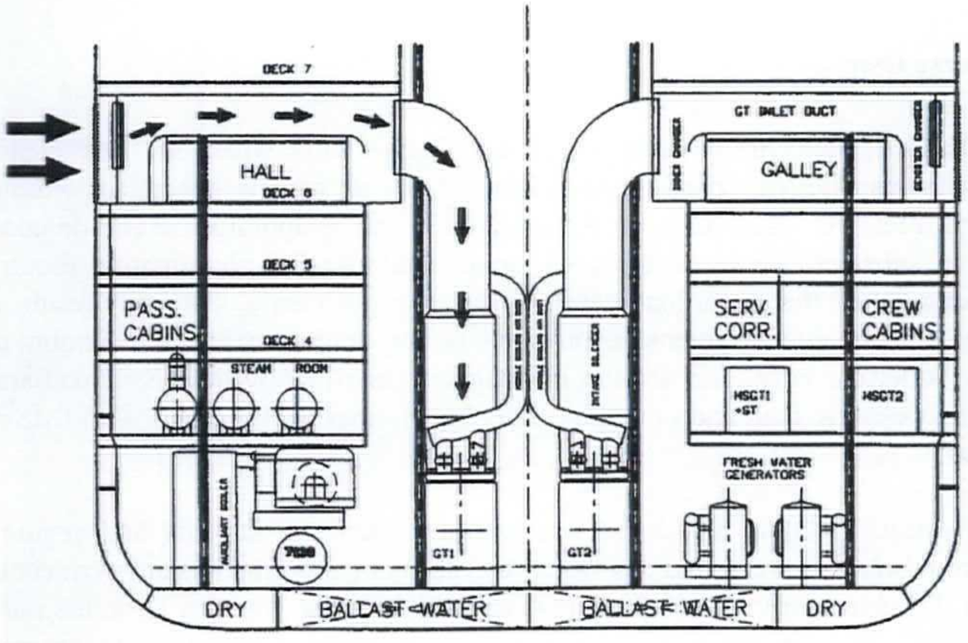


Figure 77: COGES air intake arrangement

Complete elimination of the central casing can be achieved with an alternative uptake ducting arrangement at the aft end, on both sides of the ship. Although this will result in longer uptake ducts, this option produces valuable space savings in the middle of the ship. Virtually, the ship is no longer split in two by the engine casing. This allows for a higher degree of freedom when designing large spaces on the decks in question.

It should be emphasised here there is no reason not to apply these alternative arrangements for diesel engines.

8.3 Alternative desalinator plants

Reverse Osmosis (RO) takes an increasing share of the naval market while membrane technology continues to improve. On cruise ships however, flash evaporators remain favourite for making fresh water because of the maximum utilisation of the waste heat available from prime movers.

Another desalination process which does not require steam is the so called Vacuum Vapour Compression (VVC). This distillation process is obtained by the application of heat delivered by compressed vapour. The VVC process is one of the most efficient distillation processes in terms of energy consumption and feed water recovery ratio. As the system is electrically driven, it is considered simple to operate and maintain.

8.3.1 Reverse Osmosis

Reverse Osmosis plants are nowadays only installed on ships when the 'free' steam production is insufficient. 'Free' steam as in steam that can be generated without using extra fuel. As illustrated in Appendix 13, the evaporator steam demand represents a substantial part of the total steam load. As RO desalinators require electric energy only, the steam load can be reduced significantly. This may result in superfluity of maintenance intensive oil fired boilers and possibly a reduction in waste heat boiler capacity. This section investigates the effect on the steam load and the impact on electric load and operating costs by comparing RO plants with (MSF) evaporators.

Reverse Osmosis (RO) is a process which allows removal of salts and organic material from brackish water and sea water, by means of a synthetic semi-permeable membrane. Pressure (typically 40-60 bar) is applied to saline water to force the pure water molecules through the membrane. The majority of the dissolved salts, organic material, bacteria and suspended solids are unable to physically pass through the membrane and are discharged from the system in the rejected brine. The pure water is then ready for use without further treatment.

Operating costs

An RO plant consumes typically between 10 and 13 kWh per tonne of produced fresh water. In other words, the electric power demand from a 430 tonnes/day unit (Table 38) is typically 210 kW. An additional $ 0.5 per tonne must be added for consumables such as chemicals, filters, etc. The effect on the electric load demand (at sea) is shown in Table 37.

The RO electric load is based on installed FW generating capacities as displayed in Table 7. The electric load when manoeuvring and in port remain the same because as evaporators, RO plants can only generate fresh water at sea. Also there is a risk of damaging the membrane with contaminated water.

	Maximum electric load at sea[42] (kW)	*RO electric load (kW)*	*Total (kW)*	*Increase (%)*
Sun Princess	38,800	780	39,580	2.0
Grand Princess	59,000	1,080	60,100	1.8
Capricorn	54,200	630	54,800	1.2

Table 37: Effect of RO on maximum electric load

Except for option GP2, the currently selected genset power is still sufficient to meet the increased electric load. Clearly the power plant service factor will increase slightly.

The cost of generating one kWh of electric energy with gas turbines can be estimated as follows. Each operating profile yields a certain engine load profile. Every period from the speed and operating profiles (see chapter 4) multiplied by the total electric load for that specific period yields an number of kWh's. Thus the amount of kWh's per year can be calculated. This results in $1.48*10^8$ kWh for *Sun Princess*, $2.44*10^8$ kWh for *Grand Princess* and $2.09*10^8$ kWh for *Capricorn*. The operating costs for options SP1, GP1 and CA1, including fuel, maintenance, lub oil and manning are shown in Figure 71. This amounts to \$ 8.5m, \$ 11.4m and \$ 8.9m, respectively. As a result, the costs per kWh varies from \$ 0.043 to \$ 0.057.

A prediction of the total operating costs of an RO plant therefore amounts to \$ 1.1 per tonne of produced fresh water (\$ 0.5 + 12.5x\$ 0.05). The daily fresh water consumption per vessel is shown in Table 8. Consequently, the estimated annual operating costs for a RO plant are \$ 390k, \$ 430k and \$ 240k, respectively.

The only moving part is the high pressure pump, consequently the maintenance required is negligible. The life time of a membrane is normally over five years.

Acquisition cost

The acquisition cost of two suitable one stage models is displayed below[43].

Model	*Estimated effective capacity (m^3/day)*	*Acquisition cost (440V, 60Hz manual)*	*Acquisition cost (440V, 60Hz,automatic)*
30MS840	360	\$ 480k	\$ 510k
36MS840	430	\$ 540k	\$ 570k

Table 38: Acquisition cost of two suitable RO plants

[42] See table 12.

[43] Manufactures quotation: Marinco, October 1998

Unfortunately no information on MSF acquisition cost is available.
Weight and dimensions

The dimensions of the 36MS840-model are approximately 7x2.5x2.5 metres ($44m^3$). The volume including maintenance and overhaul clear space is about 158 m^3.

The dimensions of 740 tonnes/day MSF evaporator unit is 9.2x3.3x3.8 (115 m^3). The overall volume, again including maintenance and overhaul clear space is 292 m^3. Possibly, the space savings can be used for the construction of extra cabins.

The comparisons as discussed above are summarised below.

	Sun Princess		*Grand Princess*		*Capricorn*	
	RO	*MSF*	*RO*	*MSF*	*RO*	*MSF*
Energy consumption (kWh/tonne FW)	10-13	180-200	10-13	180-200	10-13	180-200
Number of units (RO model: 36MS840)	4	3	5	3	3	2
First cost ($m)	2.28	?	2.85	?	1.71	?
Operating costs ($k per year)	390	≈	430	≈	240	≈
Equipment volume (m^3)	620	700	790	880	470	540
Steam load reduction (tonnes per year)	102,000	-	114,000	-	64,000	
Extra cabin construction cost ($k)	48	-	48	-	48	-
Additional cabin revenues ($k)	210	-	210	-	210	-

Table 39: Multi Stage Flash / Reverse Osmosis comparison

8.3.2 Vacuum Vapour Compression

Vapour compression is a distillation process, where evaporation of sea or brackish water is obtained by the application of heat delivered by compressed vapour.
The effect of compressing water vapour is obtained by means of an electrically driven, mechanical centrifugal compressor. The saline water is evaporated at sub-atmospheric pressure on one side of the heat transfer surface, and on the opposite side it is condensed into fresh water which is collected and extracted as product water.

The remaining, concentrated saline water (brine) is also extracted as blowdown, which has an average concentration factor about two times that of the initial value. Due to its high efficiency it becomes the obvious choice for a single purpose installation.

Typically, the system includes a high efficiency centrifugal blower with a low compression ratio. Because the thermal differential in the evaporator/condenser is lower than 5°C, less compression work is required, which results in a low energy consumption of typically 12 kWh per m^3 of produced fresh water.

9. COGES versus advanced cycle (WR-21)

Royal Caribbean Cruises recently ordered up to six gas turbine/steam turbine ship sets from General Electric Marine Engines for its Millennium and Voyager class newbuildings. By doing this Royal Caribbean becomes the first cruise line to build cruise vessels powered by gas turbines. Each newbuilding will be outfitted with a pair of LM2500+ aeroderivative simple cycle gas turbines and a single steam turbine in what is referred to as a COGES configuration. The orders have been placed at Chantiers de l'Atlantique and Meyer Werft, respectively. The COGES configuration is shown schematically in Figure 78.

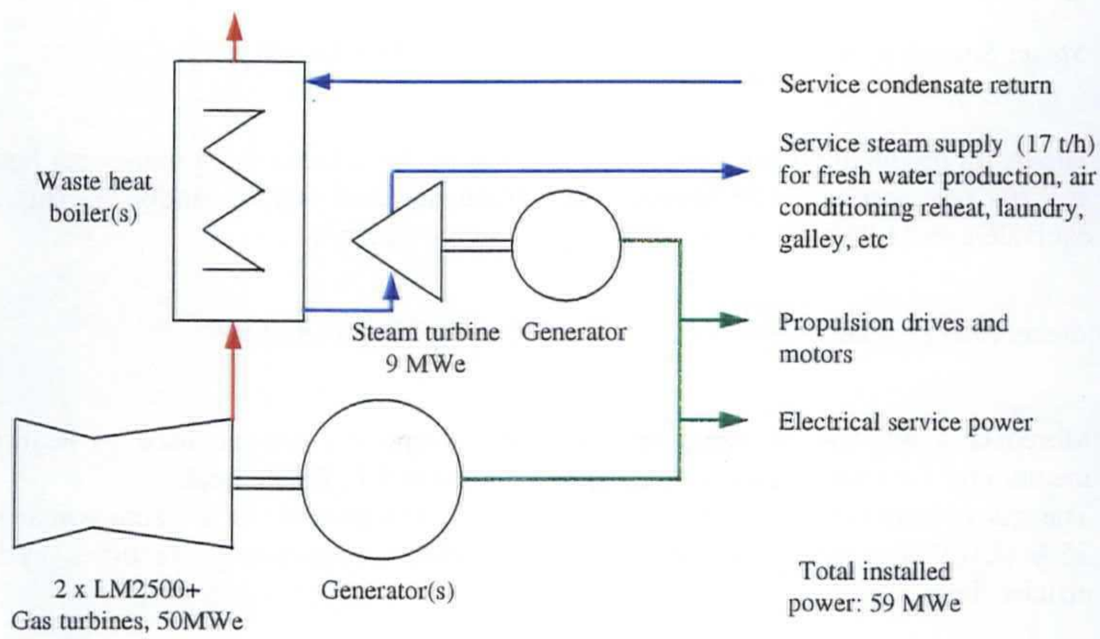


Figure 78: GE Combined Gas and Steam Turbine Integrated Electric Drive System

The exhaust gas form the LM2500+ gas turbines is utilised for steam generation with waste heat boilers. These boilers produce superheated steam at 32.7 bar, which is supplied to a steam generator (also supplied by GE). Process steam (17t/h, 3 bar) will be tapped directly from the steam turbine to feed evaporators, HVAC systems, laundry and galley. The LM2500+ engines are rated at 25,000 kW and the steam turbine delivers another maximum of 9,000 kW. GE claims an overall plant efficiency of 45 to 50%, depending on the amount of steam required for shipboard services (maximum of 17 tonnes per hour).

The main differences between the proposed WR-21 (or 601-R) gas turbine-electric/steam system and the COGES system are discussed below.

- Performance

As mentioned above, the LM2500+ is rated at 25,000 kW at the generator terminals over an ambient temperature range from 0 to 30°C, at the installed intake and exhaust conditions. The maximum power output from a WR-21 genset for the same conditions is 21,800 kW. The maximum power output at ISO, no loss conditions are 28,500 and 25,200 kW, respectively.

The waste heat from two LM2500+ gas turbines can generate an additional 9,000 kW electric power and a maximum of 76 tonnes saturated steam at three bar (a) per hour. The latter represents another 48,600 kW of energy. The conversion, as illustrated in section 3.3.3: **Evaporators and oil fired boilers**, for three bar steam is:

$$Steam\ Energy = 76 * \frac{1{,}000}{3{,}600} * (4.2 * (403 - 373) + 2{,}172) = 48{,}600 \text{ kW}$$

The steam production from a suitable WR-21 waste heat boiler is 18 tonnes per hour at 9 bar (see section 7.1.3: **Meeting the steam demand** and Appendix 8). This is equivalent to 11,800 kW according to the following equation:

$$Steam\ Energy = 18 * \frac{1{,}000}{3{,}600} * (4.2 * (453 - 373) + 2{,}015) = 11{,}800 \text{ kW}$$

Moreover 6,500 kW of waste heat from the intercooler can be used as heating medium for the evaporators as investigated in section 8.1: **Waste heat**.
The gas turbine efficiencies for the above mentioned conditions are approximately 35% (LM2500+) and 41% (WR-21). This yields the following combined cycle efficiencies:

LM2500+: $\dfrac{50 + 9 + 48.6}{50 / 0.35} = 75.3\ \%$ (dual gas turbine plant)

WR-21: $\dfrac{21.8 + 11.8 + 6.5}{21.8 / 0.41} = 75.4\ \%$ (single gas turbine plant)

It should be emphasised here that at a large amount of heat from surplus steam will be discharged through dump condensers. The COGES ship in question for example, utilises on average only 17 tonnes per hour out of the 76 tonnes that are available. As a result the plant efficiency falls to 49%. This approximates the 50% plant efficiency as mentioned previously.

An equivalent plant would comprise three WR-21 gas turbines, delivering at least 65.4 MW. A total amount of 54 tonnes of steam per hour would then be available. A surplus of 37 tonnes, or 12.3 tonnes per engine for this specific case. The plant efficiency would then become:

$$\text{WR-21: } \frac{21.8 + \frac{18 - 12.3}{17} * 11.6 + 6.5}{21.8 / 0.41} = 60.5\ \%\ \text{(single gas turbine plant)}$$

With regard to high quality energy production (steam is not high quality energy) the efficiencies are similar: both more than 41%. However, to achieve this efficiency with the COGES plant, both LM2500+ engines have to be running at full power whereas only one WR-21 at typically 70% load can achieve the same efficiency.

Regarding efficiency, clearly both systems are almost identical.

- First cost

The LM2500+ budget price is \$ 9.90m[44], or \$ 350/kW (ISO, no losses). For a WR-21 this number is around \$ 400/kW. Bear in mind that the LM2500+ simple cycle efficiency is 38%, compared to 44% advanced cycle WR-21 efficiency (ISO, no losses).

To achieve the same electrical efficiency a high pressure steam system with steam turbine is required. An indicative price for a 9 MW steam turbine genset is \$ 3m according to a quotation from a steam turbine manufacturer. This excludes inlet steam pipework and valves, a cooling water system, all cabling and electrical equipment and installation. The estimated boiler plant acquisition cost is also \$ 3m. This results in a COGES main equipment acquisition cost of nearly \$ 26m or \$ 440/kW (conditions in practice). As shown previously, the main equipment acquisition cost of a triple WR-21 plant is \$ 30m or \$ 460/kW for conditions in practice.

- Emissions

The claimed LM2500+ specific NOx emission is 2.5-4 g/kWh[45]. As mentioned before, the 5 g/kWh prediction for the WR-21 is rather pessimistic.
As illustrated in section 7.5.1: **Origin from and reduction methods for exhaust gas emissions**, the SO_X emissions are proportional to fuel consumption and the sulphur content in the bunkered fuel. Both aeroderivative engines burn the same type of fuel and as a result the SO_X emissions will be similar.

[44] [Gas Turbine World, 1997]
[45] [Nurmi, 1996]

- Maintenance costs

A 10 year maintenance contract for a COGES plant on a 80,000 tonnes vessel has resulted in a cost per fired hour of $ 90 (5,000 accumulated fired hours annually). The extent of the maintenance cover however is not fully defined. WR-12 maintenance cost amounts to $ 112 per fired hour (see section 7.7: **Maintenance and spares**). Based on a ten year contract though, this can possibly be reduced.

- Intake and uptake dimensions

The intake and uptake ducting designs for an LM2500+ have resulted in a cross sectional flow area of about 4.7 and 2.8 m^2, respectively. Preliminary design calculations for the WR-21 yielded 4.0 and 3.1 m^2, respectively.

The LM2500+ waste heat recovery boilers maximum steam production is 38 tonnes per hour at 32.7 bar. The overall dimensions are 13x6x6, including ducting transitions. Waste heat boilers for WR-21 applications are shown in Appendix 8.

- COGES-diesel study

A comparison study[46] of diesel-electric and COGES machinery for a 80,000 tonnes luxury cruise liner resulted in the following statements:

- similar acquisition and maintenance cost for both machinery systems
- weekly higher fuel costs of $ 31,000 ($ 1.6 m annually), based HFO/MGO price gap of $ 70/tonne
- 23 additional passenger cabins (2.5% increase) plus 15 additional crew cabins
- lightweight: 800 tonnes less, dead-weight 80 tonnes less (lub oil bunker) ⇒ 1% propulsion power saving due to smaller displacement
- 1 MW lower heat load (fuel heating)
- 500 kW lower electric load (diesel auxiliaries)
- reduction of: 90 machinery main components, 5,600 metres piping and ducting, 150 tonnes machinery piping weight, 350 valves on machinery piping, 800 automation points

The conclusion of this study was that there is actually no technical nor commercial disadvantage for use of COGES machinery on large cruise vessels.

A significant disadvantage over diesel-electric and advanced cycle gas turbine-electric though is the complicated and expensive high pressure steam system. A system which is generally vulnerable and maintenance intensive. Moreover the shipyards in question have no previous experience with installing a high pressure steam system.

[46] [Deltamarin, 1995]

A COGES plus point is the extensive experience gained by the LM2500, an engine which first entered service in 1970.
In a way, this can also be said for the WR-21, because the rotating elements are based on the RB211 family. The compressors and turbines are derivatives of the Aero RB211-535 and 524 whereas the power turbine is based on the latest aero-Trent 700 and 800. The RB211 has gained a total fleet experience of 7 million running hours since 1973.

10. Conclusions and recommendations

Despite of the higher initial cost and increased operating costs, considerable economic benefits are achievable with an advanced cycle gas turbine plant in large cruise ship applications. This is mainly due to a significant increase in cabin revenues which is allowed by a high power density of gas turbines. The switch from a diesel plant to a gas turbine plant offers an increase in cabins varying from 5% on *Capricorn* up to 7% on *Grand Princess*. Additional savings originate from reduced lub oil and engine room manning costs.

Basically, the only two advantages of involving the relatively small 601-R gas turbine for this size of cruise ships in the comparison are an increase in redundancy and a slightly reduced NO_X emissions. Compared to the WR-21, the overall operating costs are higher: the specific fuel consumption is higher, the maintenance costs per engine are higher and the total lub oil consumption is higher. Moreover, the total machinery space required for a similar sized plant is higher, resulting in less cabin revenues.

Therefore it is hardly surprising that the profitability of the 'WR-21 only' configurations are highest. Further, the net present value the '601-R only' configurations is marginal or even negative. Clearly the configurations comprising both WR-21 and 601-R engines hold the middle. Especially on *Sun Princess* this might be a good alternative because the 'WR-21 only' configurations comprises only two prime movers. Regarding redundancy possibly not a good option.

It has to be emphasised that the outcome of the life cycle cost comparison depends highly on the amount of extra cabin revenues involved. The effect is demonstrated by reducing the cabin occupancy from 100 to 75%; for most configurations this results in a notable reduction of profits and in some cases even a negative net present value.

Other advantages, not directly resulting in a positive cash flow should of course not be forgotten. Improved shipboard habitability due to lower noise and vibration levels and cleaner exhaust gases will enhance passenger satisfaction. Additionally, a gas turbine ship has a significantly reduced impact on the environment. By fitting a gas turbine plant in a cruise ship, the NO_X emission reduction is about 70% whereas the annual SO_X emissions can be reduced by as much as 90%. These are issues that can not be left out of the equation.

It was recognised above that the cabin revenues represent a crucial factor in the equation. As this report is written from a marine engineering point of view, the effect of reduced machinery spacing and associated ship superstructure changes has not been thoroughly investigated. Significant changes to the machinery spaces and intake/exhaust casings will require a re-optimisation of the entire ship layout. This should result in a more favourable layout of either passenger spaces and ship service spaces resulting in additional cabin space or increase passenger comfort. This clearly this forms the basis of additional study incorporating naval architecture aspects.
If the actual amount of extra cabins is determined an investigation in the actual ticket sales should be conducted. This should give a more accurate estimation of the potential cabin revenues.

As mentioned above, it was also recognised that the '601-R only' configurations and to some extent the 'WR-21/601-R' configurations are less profitable than the configurations which consist of WR-21 gas turbines only. This however, has only been confirmed for vessels ranging from 75,000 to 109,000 tonnes. For larger vessels the conclusion will be the same. For 'small' cruise ships (<50,000 tonnes) the 601-R gas turbine could be a good alternative. Especially vessels that comprise high speed diesels. In that case the fuel cost difference will decrease significantly as high speed diesels also require distillate fuel.

Moreover a 'minimum size' vessel of 50,000 tonnes (as derived in the report) should be investigated. For this vessel size, a combination of a WR-21 and a 601-R may turn out to be a more economical solution than a power plant comprising two WR-21 engines.

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