The Future of Ship Design

DELTAMARIN™

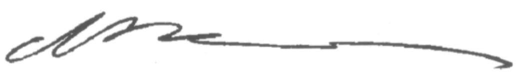
Dear Colleague,

The Future of Ship Design is intended for naval architects, marine engineers and for professionals working with the development of newbuilding ship projects from first feasibility studies through project and contract design into planning, basic and detail design. Working and design methods are also considered.
The authors are experienced naval architects, marine engineers, mechanical engineers and electrical engineers, all working at Deltamarin Ltd on the date of publishing. The background of the is in the Finnish shipbuilding as many of them have been working at the Finnish shipyards. All the authors have international experience mainly from European shipbuilding. This experience covers consulting, design and engineering tasks for various shipping companies, shipyards and suppliers as well s supervising and commissioning tasks of newbuildings and conversions.

The Future of Ship Design sets out to inform the practising Naval Architect and Engineer of the latest design techniques at his disposal, and then puts these in context by quoting case study material. Attention is paid to the important aspects of the hydrodynamics, machinery, structure and equipment design considerations, vital parts of the ship design mix.
The tremendous technical development we have faced in ship design during the last decade issues us as designers a big challenge for the 21st century. New technology has been introduced faster than ever before in our industry.

The issue is to maximise the efficient revenue generating space at minimised investment and running costs but taking into account availability and environmental impacts as well.
The tendency is clear and promising, new products and innovations are introduced and completely new ship configurations can he developed based on new system and machinery products on the market.
It is important to understand and consider the life cycle costs but taking into account possible additional revenue when considering new configurations.
Typical case studies are handled showing the importance of a techno-economic design approach.
Finally innovative design techniques available today and signifying a quantum leap in the ship design and project coordination in the early 21st century are discussed.
I hope this book will offer a platform for further exchange of valuable information in our common effort for more efficient products: ships for the 21st century.

Yours sincerely,



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1. Design Guidelines

1.0 General

This chapter is a collection of practical experience further developed into practical design guidelines. They concentrate on the typical naval architectural and marine engineering problematics in developing a new design. These guidelines are intended for helping the designer or project engineer to make the first approach and later on when the design is available to enable easy cross checking.

Historic trends are also explained when they can be used as general guidelines in developing future ship configurations. Items covered are the most typical problematic areas in ship design.

Examples shown are taken from typical ro-ro ships, ro-ro passenger ferries, passenger cruise ships and other similar twin-screw ships; product tankers, chemical carriers and other single screw ships are covered as well.

1.1 Hull Forms and Hydrodynamics

1.1.1 Basic parameters

The purpose of the hull form is to carry the defined load in accordance with the given transport task. As such it sounds simple but design of an efficient hull form consists of a great number of parameters and conditions, which have to be taken into account.

The tendency has been towards higher capacity, which has meant bulkier ships above waterline and under waterline, towards higher block coefficient. Another clear tendency has been towards higher speeds. Average contract speed for all ships above 1000 grt has increased by about one knot within the last fifteen years. For some special ship types, such as container feeder ships, ro-ro ships and ferries the increase has been several knots.

At the same time also the main dimension ratios have changed remarkably, the length-beam ratio has decreased, in some cases even below 5 and the beam-draught ratio has increased: for example for ro-ro passenger ferries from 3,0-3,5 up to 4,5-5,0.

Up to the end of the 70’s the hull forms were typically defined by using hull form series, such as Series 60, Taylor, BSRA and similar. Today these series are no more feasible and typically hull form design is based on reference vessel(s). A series of good recently tested and built ships is, of course, a perfect starting point. It is, however, difficult in today’s evolutionary world to find one single organisation having good references for different types of hull forms, main dimension ratios and other related parameters. It is always worthwhile to check the references against the state of the art on the market of any organisation to be worked with, even model basins, before starting any further cooperation.

General tendency has been to increase the earning capability of a vessel in comparison with the price. Length is still considered as one of the main parameters in the price definition, as well as restricted in many cases due to harbour and route limitations. Beam and block coefficient has been increased. Some limits, however, may have been met. Let us look at a few typical examples of different types of vessels recently built.

An extreme example is a product/chemical carrier with main dimensions: L_{PP} = 115 m, B = 24 m, T = 12,4 m and main dimension ratios L/B = 4,79, L/T = 9,27 and B/T = 1,94. A pram type stern hull form was applied with a slender centre skeg to accommodate propulsion machinery. Course stability was carefully studied with model testing and special attention had to be paid for the design of the centre skeg, slender enough, and the rudder size, somewhat larger than normal. Figure 1-1 shows the body plan.

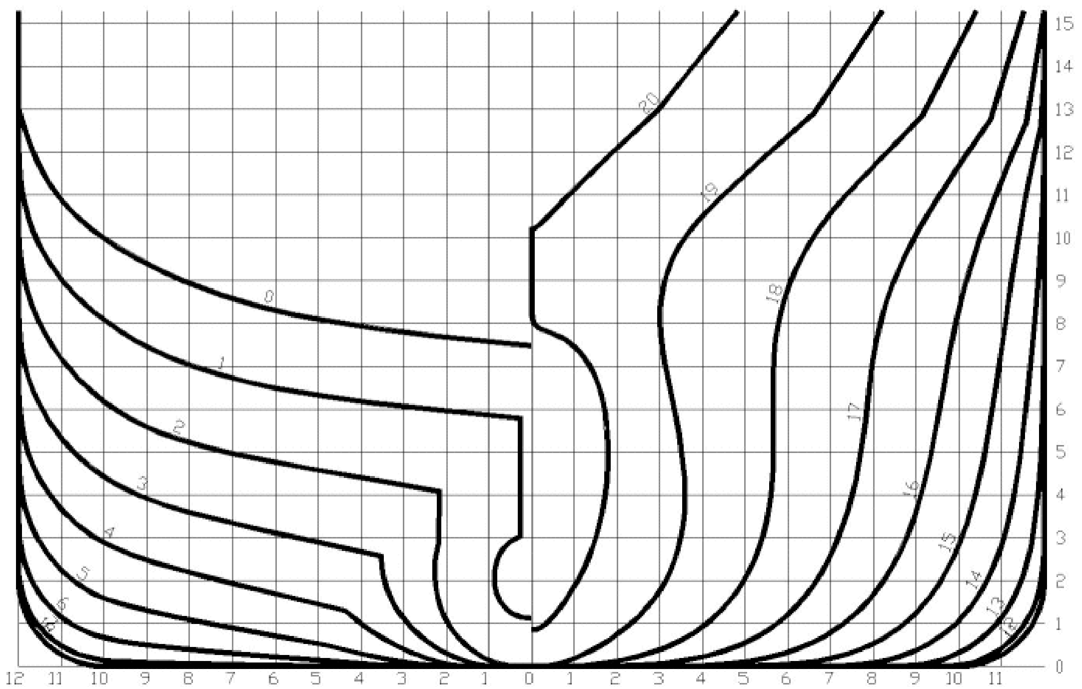


Figure 1-1 Body plan of a chemical/product carrier with main dimension ratios of L/B = 4,79, L/T = 9,27, B/T = 1,94.

The block coefficient (CB) has increased for chemical and product carriers below 40.000 dwt close to 0,80 or even slightly above. Applying pram type hull form has allowed a shift of longitudinal centre of buoyancy (LCB) afterwards enabling smoother forward shoulders and lower waterline entrance angle. Typical figures today vary between -1,8%…+1,0% of L_{PP} aft or forward of $L_{PP}/2$. The design waterline angle can be reduced at the same time with 1,5…2,5 degrees (half angle). Hull form developed with the above design philosophy gives a 5-12% reduction in required propulsion power when compared with good conventional hull shape with LCB more forward. Midship section coefficient (CM) is typically between 0,985 and 0,998 avoiding, however, a bilge radius below 1 meter, which leads to high vortex deformation.

Wide, shallow draft twin screw tanker with limited length may offer an interesting configuration for most of the different sizes and types of product, chemical and crude carriers. Twin skeg arrangement has been applied already since the 1930’s, but the electric machinery and pod propulsion (twin units) will offer a possibility for extremely simple barge type hull form, excellent manoeuvrability and high power availability and more efficient cargo volume. Simple hull form supports also simple, standardised hull structure.

Ro-ro’s and ro-ro passenger ferries have today two clearly different families: conventional high displacement ferries with Froude number

$$(Fn = \frac{v[m/s]}{\sqrt{g[m/s^2]L_{wl}[m]}}) \tag{1}$$

around 0,30 or below, and high displacement fast ferries with Froude number clearly above 0,35, reaching today already 0,40 or even above.

The typical conventional ferries have L/B ratio from 4,8 up to 6,5, draught between 5 and 6,8 meters and B/T ratio can be as high as 4,8…5. Block coefficient is between 0,64 and 0,72 and recommendable midship section coefficient is 0,985. A good LCB value for a ferry with full-length superstructure is between -3,7 and -4,7%, and for a ferry with a forward superstructure only between -2,5 and -3,2%.

The fast high displacement ferries are today built with L/B ratio not less than 6 and some even well above 7. Block coefficient is varying between 0,52 and 0,60, 0,57 being a typical value. Length is increased and width is restricted to get the block coefficient down and to reach a longer waterline, i.e. lower Froude number. Midship section coefficient is varying from 0,955 up to 0,988, the lower figure is from a rather short vessel (L_{PP} only 111,8 m), the longer vessels being between 0,98-0,988. LCB is varying from -2,6% with forward superstructure up to -3,6% with full length superstructure.

The big passenger cruise vessels are approaching quite standard main dimension ratios: L/B from 7,0 up to 8,4, and B/T between 4,0 and 4,5. Perhaps in the near future we will see standardised main dimensions, for example for a Panamax size cruise vessel and the next step would be standardised hull form with related cost and time savings. The block coefficient varies from 0,62 up to 0,71, LCB from -3% up to -5,1%, and midship section coefficient from 0,88 up to 0,985. Good experience has been gained with CB of 0,65, LCB -4% and CM of 0,98 for a typical Panamax size vessel with trial speed of 25 knots.

1.1.2 Hull form characteristics

Hull form of a new vessel is something with which you are married actually for the whole lifetime of the vessel, even though some small modifications could be carried out later on. It is therefore advisable to invest on the optimisation process of hull lines not only for resistance and propulsion but also for seakeeping and manoeuvring; it will pay off quickly.

Too many unfortunate references are still sailing unable to utilise the full sea margin or carrying unnecessary ballast water or consuming too much fuel per sailed ton mile.

Bulbous bow

In principle a bulbous bow fits for any kind of a ship, the only exceptions being, for the time being, icebreakers and very high speed ships. The effect of bulbous bow in the required propulsion power is typically from -8% up to -15% in ships with modern main characteristics and hull forms.

Typical bulbous bow today is a so-called ‘goose-neck’ bulb with an upside down drop form. Length of the bulb is 4-4,5% of waterline length but surprisingly good results have been reached with length of up to 5% and above, especially for Froude numbers above 0,30. Sectional area of the bulb is between 6 and 11%, 9% being a typical value.

Upper contour, profile, is rising forward and recent series of model tests have shown that it is preferable to place the contour clearly above the design waterline, 40-60 cm. Figure 1-2 shows a good example, profile of a bulbous bow designed for a car passenger ferry with service speed of 23 knots, Froude number 0,29. This kind of a profile gives good performance also at lower draughts (65-70% of design draughts) and speeds as well.

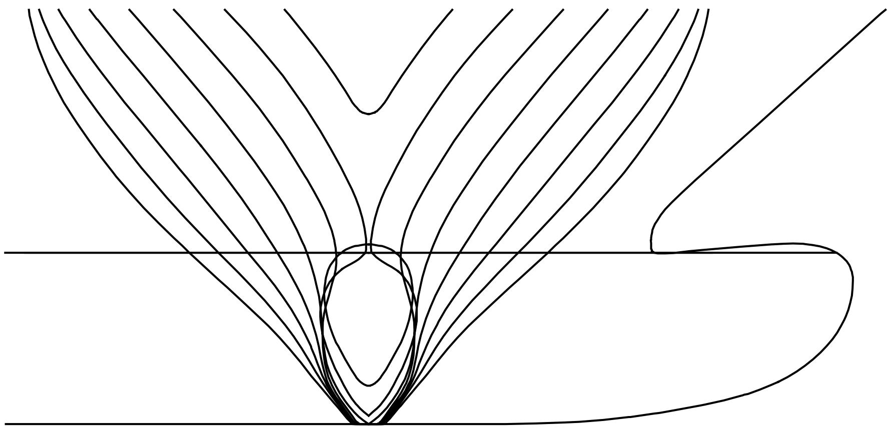


Figure 1-2 Profile for a modern bulbous bow with upper contour above design waterline.

It is extremely suitable for retrofits, the higher and larger upper part can be placed on top of the existing bulb, the gain being between 3-5% in propulsion power.

Fore ship

The design waterline is preferred to be straight and waterlines below convex. The higher the Froude number the more convex the waterlines below should be. In a chemical/product tanker with a block coefficient close to 0,80 waterlines remain nearly straight all the way but in a fast high displacement ferry the lower waterlines should have a more pronounced convex shape to allow smooth flow around the bilge underneath the flat bottom.

A forward shoulder should be avoided, it is always creating an additional wave system and increasing resistance. Curve of sectional areas is a good tool to check this, see figures 1-4 and 1-18.

The design waterline half angles vary a lot even in same type of vessels: chemical and product carriers between 21 and 35 degrees, ro-ro passenger ferries between 13 and 22 degrees, and passenger cruise vessels between 10 and 20 degrees. Lower entrance angle certainly gives lower resistance assuming the above two design criteria of straight design waterline and avoiding forward shoulder can also be met. With given main dimensions the best way to reduce the entrance angle is to remove longitudinal centre of buoyancy aftwards. Of course a good combination should be found, but an increase of sectional area of bulbous bow can at least partly compensate the high waterline entrance angle, if not otherwise possible. Some examples with good powering results are shown in figure 1-3, ship no. 4 is with high iceclass. The chemical/product carriers have an entrance angle of 21 and 24 degrees, the ro-ro passenger ferry 21 degrees, the ro-ro ship 20 degrees, the paper carrier 21 degrees and the passenger cruise ship 13,5 degrees. The ro-ro passenger ferry value is high due to rather forward location of the LCB, -2,5% aft of $L_{PP}/2$, but even so the powering results were unexpectedly good. Removing LCB more aftwards and reducing entrance angle would obviously make the performance even better.

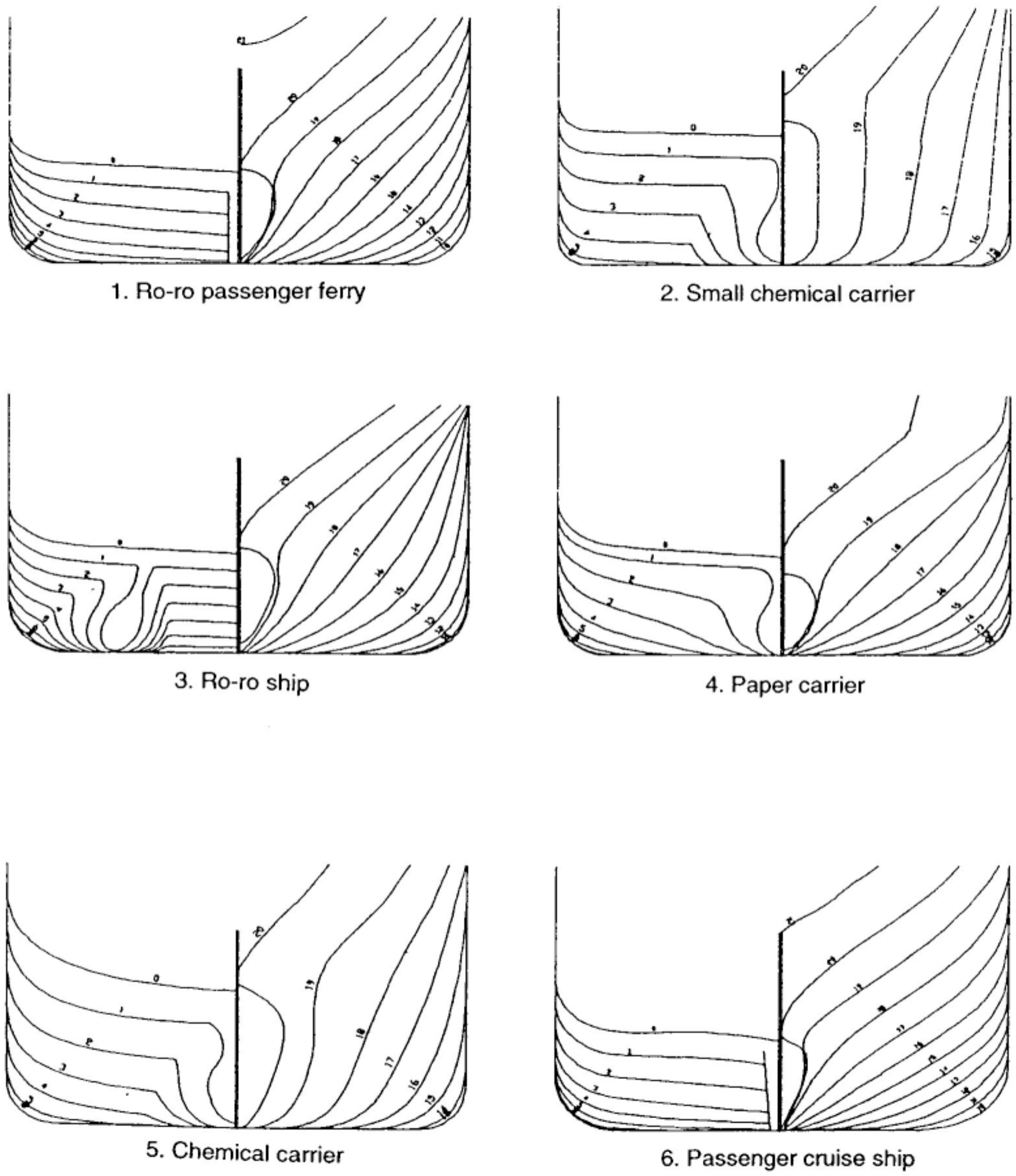


Figure 1-3 Typical hull form examples with good powering results, main dimensions L_{PP}/B/T: 1. 170/28,7/6,0; 2. 91,2/16,2/6,4; 3: 165/25,5/7,3; 4: 146/23,5/6,25; 5: 166/31/10,8; 6: 238/32,2/7,9

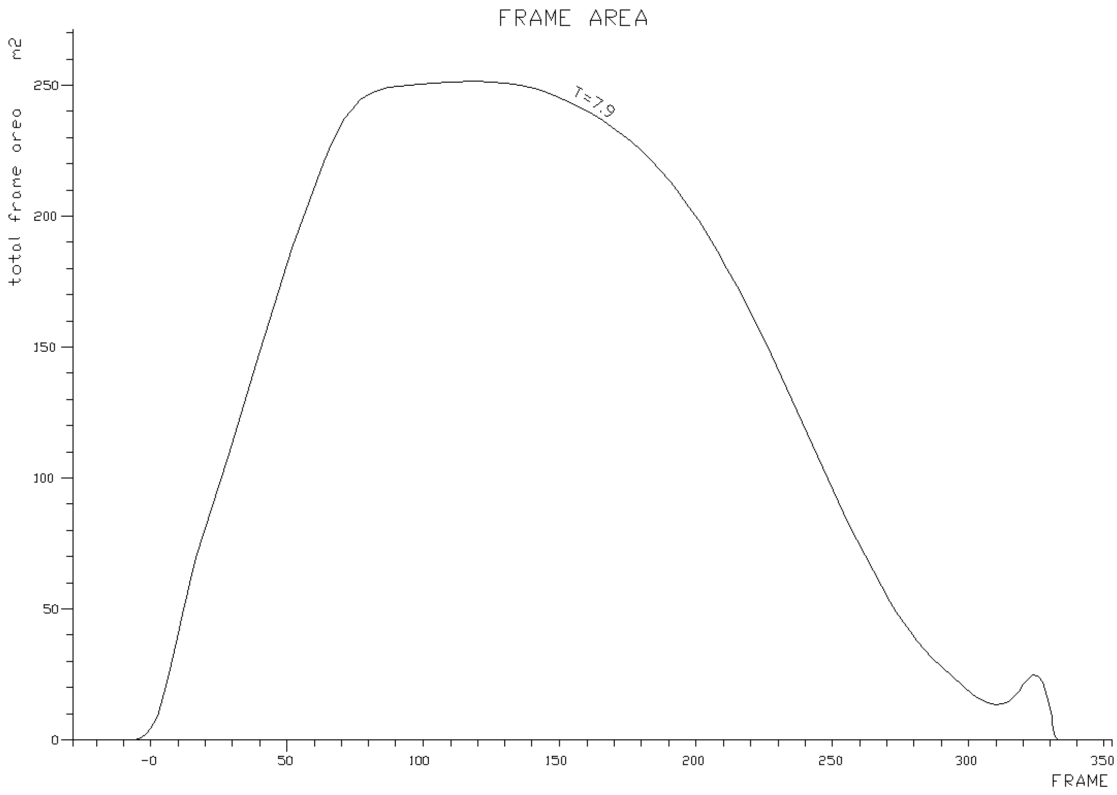
<u>Aft ship</u>

Pram type stern or buttock flow stern has become a typical hull form first for twin screw vessels and later on for single screw vessels as well. Figure 1-3 presents a good collection, one twin skeg (No. 3) and one moderate pram (No. 5), the rest being typical pram type hull forms.

Pram type aftship with semi tunnels is a generally applied hull form for passenger vessels and ferries. Some designers prefer to locate propellers and shaftlines as close to the centreline as possible to be able to minimise the shaftline length, however, that is restricting the propeller diameter heavily.

A big family of recently built ferries and cruise ships were analysed and two interesting parameters were found out having correlation with required propulsion power: radius of verticals (buttocks) in the transition area from the flat bottom into the rising verticals and angle of verticals towards baseline (at the shaft centreline). The radius varies between 0,42 and 1,16 times perpendicular length, average being 0,7. The vertical angle is between 9 and 15 degrees. Astonishingly there is no difference between ferries and cruise ships, which means that these parameters are dictated by the local restrictions not by the general parameters, such as block coefficient, longitudinal centre of gravity or Froude number. Ships with bigger radius and lower vertical angle had lower propulsion power requirement and typically better wakefield as well.

Model tests and CFD calculations have shown that a too heavy aft shoulder has a big impact on the viscous and wave making resistance as well as introduces heavy vortices into the flow towards the propeller and causing propeller induced vibrations. Figure 1-4 shows a reference with too blunt shoulder area which can be seen also in the enclosed curve of sectional areas. Waterlines 1 and 2 have too small bilge radius ‘preventing’ smooth, undisturbed water flow over the bilge and creating high pressure change and vortices. The vessel has suffered from heavy propeller induced vibrations.



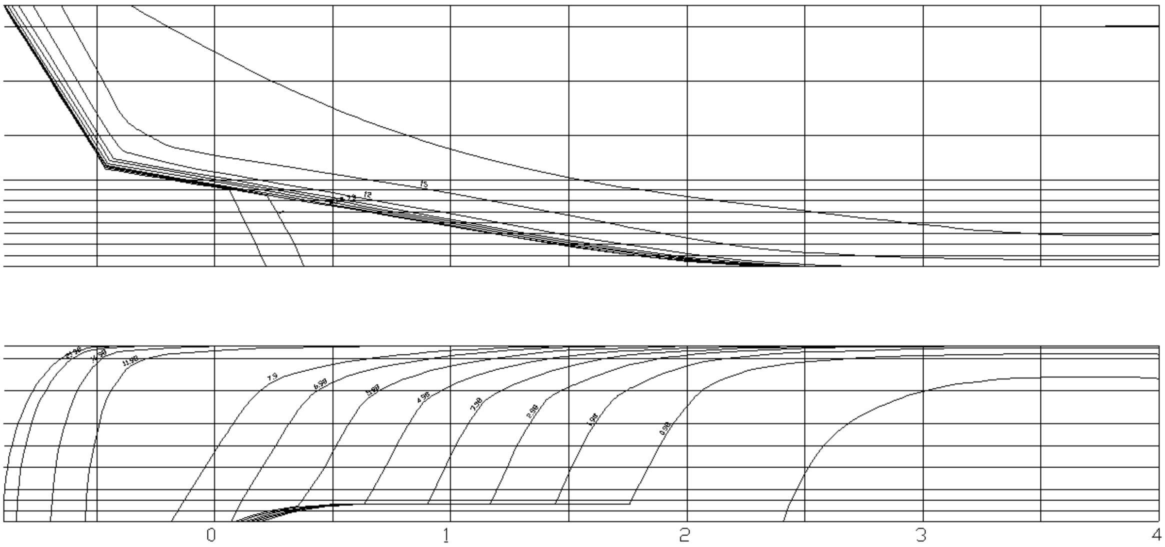


Figure 1-4 Example of too heavy aft shoulder

Optimised hull form for a passenger cruise vessel should have a transition radius not less than L_{PP} of the vessel, preferably closer to 1,5, and vertical angle not more than 9-10 degrees. These figures apply well for fast high displacement ferries as well. Good results for conventional ferries are reached with transition radius of 0,9-1 L_{PP} and vertical angle of 9,5-11,5 degrees depending on the LCB and block coefficient.

Most recent development for aft ship hull forms is the application of trim wedge combined with a ducktail. A trim wedge is located under the aft part of the verticals, typically starting just aft of the propeller plane/rudder plane and extended 3-5 meters aftwards and inclining the verticals downwards from horizontal between 2 and 9 degrees depending on the Froude number and hull geometry. The aft end of the trim wedge should be about 10-40 cm above or below the design waterline depending on the Froude number, less clearance/more submerged with higher number. Figure 1-5 shows a typical reference, a high displacement fast ferry, L_{PP} 111,8 m, design speed 24 knots. The trim wedge has proven to be successful already at Froude number 0,28 for ro-ro passenger ferries leading to a propulsion power reduction of 3%. At higher Froude numbers 0,34…0,38 power reductions up to 9% have been measured for already well optimised hull form. Successful design of trim wedge can be seen in model tests from the clearly better transom wave pattern and reduction of dynamic stern trim and sinkage into even keel.

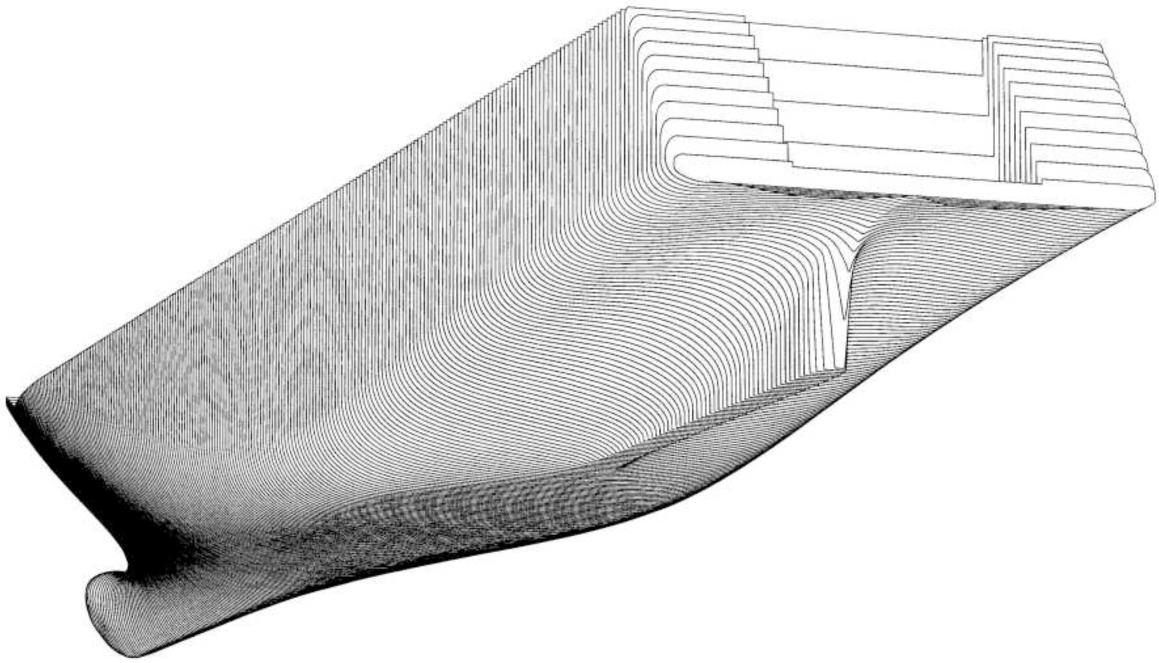


Figure 1-5 Fast handy size ferry with efficient trim wedge.

1.1.3 Appendages

Appendage resistance in a twin screw vessel has a big impact on the total powering performance and the biggest part of this is coming from shafts, brackets and hull bossings. Typical examples of conventional and fast high displacement ferries were analysed to see if differences would be found in the hydrodynamic design of shaftlines and their supports. Presented model test data belong to vessels having design speed range between 19.0-28.0 kn.

Four reference vessels were selected for this comparison with modern hull form having low total resistance values, main characteristics are presented in table 1-1. Bilge keels are not included in given resistance values and all reference vessels have fin stabiliser recesses and two bow thrusters except Ref. 2 which has only one bow thruster and no stabiliser recesses. Bossing size comparison is presented in figure 1-6.

Table 1-1 Vessel particulars and appendage resistance.

Vessel	Lpp (m)	Design speed (kn)	Shaftline inclination deg	Fn	Rudder Area % Lpp x T *	R_{app} %
REF 2	111.8	24.0	0.79	0.373	2.15	7.0
REF 3	165.0	23.5	0.82	0.300	2.75	13.7
REF 4	166.0	20.0	1.70	0.255	1.78	14.1
REF 5	159.7	28.0	0.57	0.364	2.98	8.1

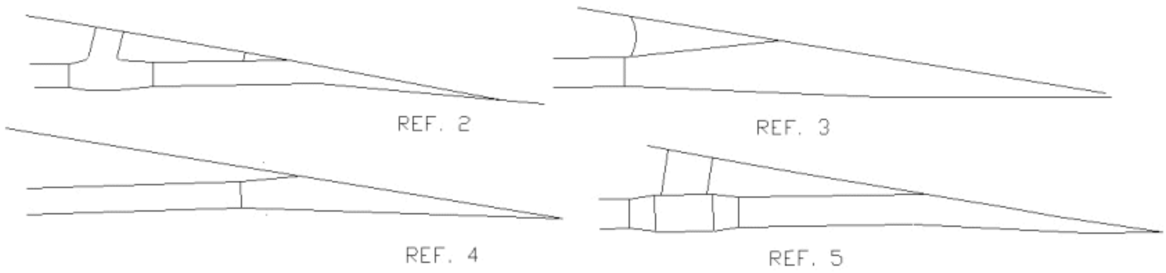


Figure 1-6 Hull bossings and intermediate brackets.

Ref. 3 has flap type rudder, Ref. 4 and 5 have spade rudders and Ref. 2 semi spade rudder with somewhat thinner profile than the flap and spade type rudders.

Ref. 2 and 5 are equipped with intermediate bracket in order to have as small hull bossing as possible. Ref. 3 and 4 have larger hull bossings and no intermediate bracket.

Appendage resistance

Appendage resistance R_{app} % is defined by formula

$$R_{app}\ \% = \frac{R_{bare}}{R_{appended}} \times 100 \qquad (2)$$

Where

R_{bare} = barehull resistance
$R_{appended}$ = resistance with appendages

$R_{appended}$ is model test measured value extrapolated to full scale without correction for different scale effects than applied to the hull itself. All the appendage resistance values have been analysed using ITTC 57 -method resulting in comparable resistance values.

Values of R_{app} % as function of speed are given on fig. 1-7. The low appendage resistance of Ref. 2 and 5 shows the impact of small bossing size which can be seen by comparing resistance values and bossing shapes in fig. 1-6. It is interesting to see that the optimised/minimised shaft supports and bossings are leading to similar appendage resistance figures as for example for a patrol boat (Ref. 1) with really small optimised appendages with no hull bossing.

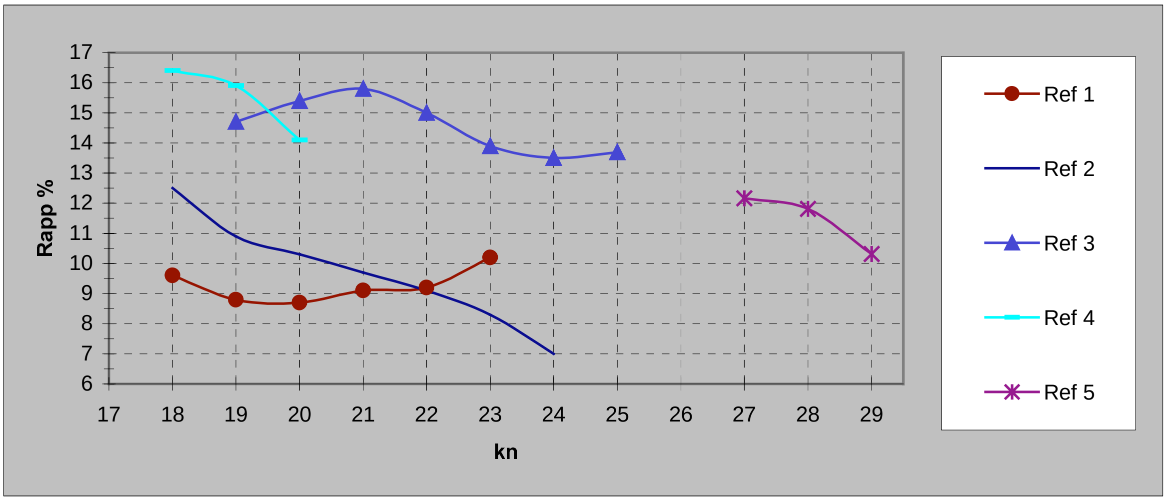


Figure 1-7 Appendage resistance as the function of speed.

<u>Wake</u>

Axial wake is given in fig. 1-8. Ref. 2 has the smallest size hull bossing and shaft slope of ≈ 0.8° resulting in low and narrow wake peak.
Ref. 4 has also small hull bossing but the wake quality is not as good as one could expect. This can be explained by high shaft slope of 1.7° which increases the “shadow” effect caused by hull bossing and shaft itself. There seems to be a correlation between the bossing size and shaft orientation.

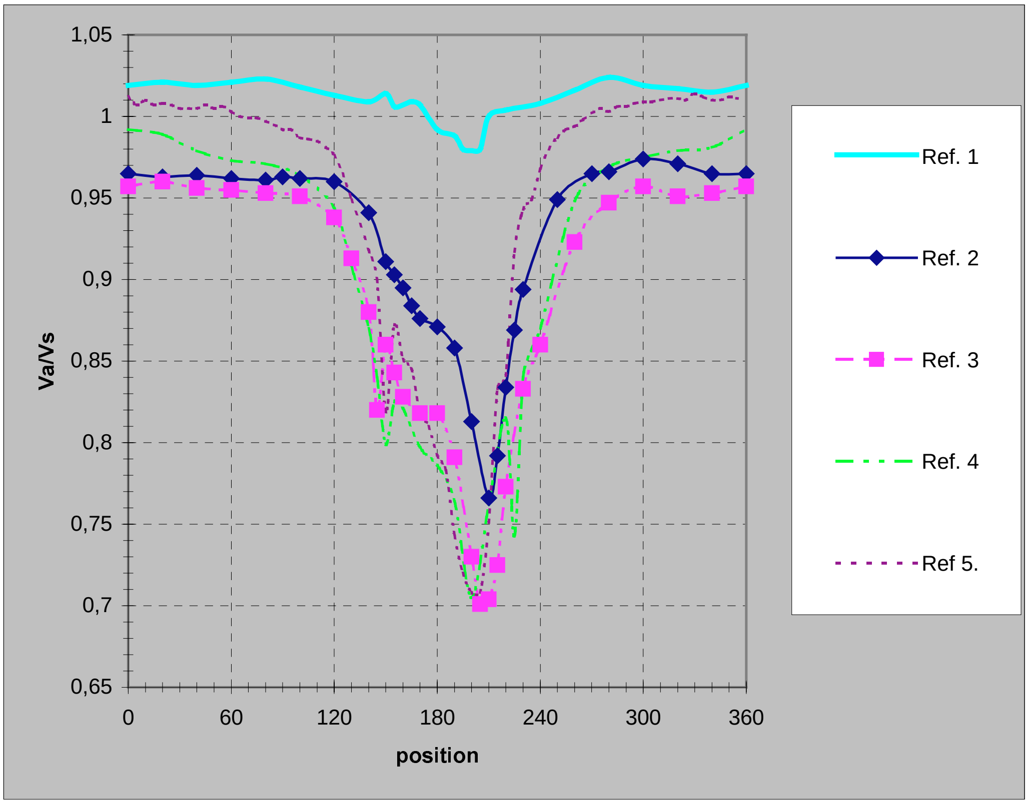


Figure 1-8 Wake axial velocity component at abt. 86-89% propeller radius.

Ref. 3. has the largest hull bossing resulting in deeper and wider wake peak.
It should be noted that wake characteristics of Ref. 3, 4 and 5 cannot be judged as poor. They are considered to be good among similar vessels as the typical criteria for wake peak are: depth not more than 30% and width not less than 120 degrees to avoid too rapid flow variations and risk for high propeller induced pressure pulses.

Ref. 3, 4 and 5 all have separate wake peaks caused by V -brackets. Ref. 1 and 2 totally lacks this effect also on tangential and radial wake plots.

All the reference vessels have abt. the same ratio for parameter *c* defined by

$$c = \frac{\text{bracket thickness}}{\text{distance from bracket CL to propeller plane}} \tag{3}$$

This value is between 0.085 and 0.095 which can be considered adequate. Ref. 1 and 2 have the lowest value of 0.085 which together with well aligned brackets possibly explains the absence of bracket peaks on the wake diagram.

Use of twisted brackets is suggested if large variations in flow angles are found on wake measurement, tangential wake variation more than ±20%. If these angles are found out to be small it is sufficient to use uniform alignment angle and for example NACA 64 -021 profile allowing higher variations in flow angles without increase in drag coefficient in comparison with for example NACA 00 -series profiles.

<u>Design guidelines</u>

Main parameters affecting bossing size are the tailshaft length and distance between tailshaft bearings. With long tailshaft one can move cylindrical shaft coupling as far inside the hull as possible and thus reduce bossing diameter.

In order to keep bearing distance in the range 22 - 28 x shaft diameter an intermediate bracket should be installed. This way the hull bossing size can be significantly reduced since now only forward seal, intermediate shaft bearing installation and cylindrical coupling diameter define the bossing size depending on particular vessel arrangement. In fig. 1-9 two possible appendage designs for the same vessel are presented. Despite the longer bearing distance larger hull bossing is required to accommodate tailshaft forward bearing. Both cases have equal total shaft lengths.

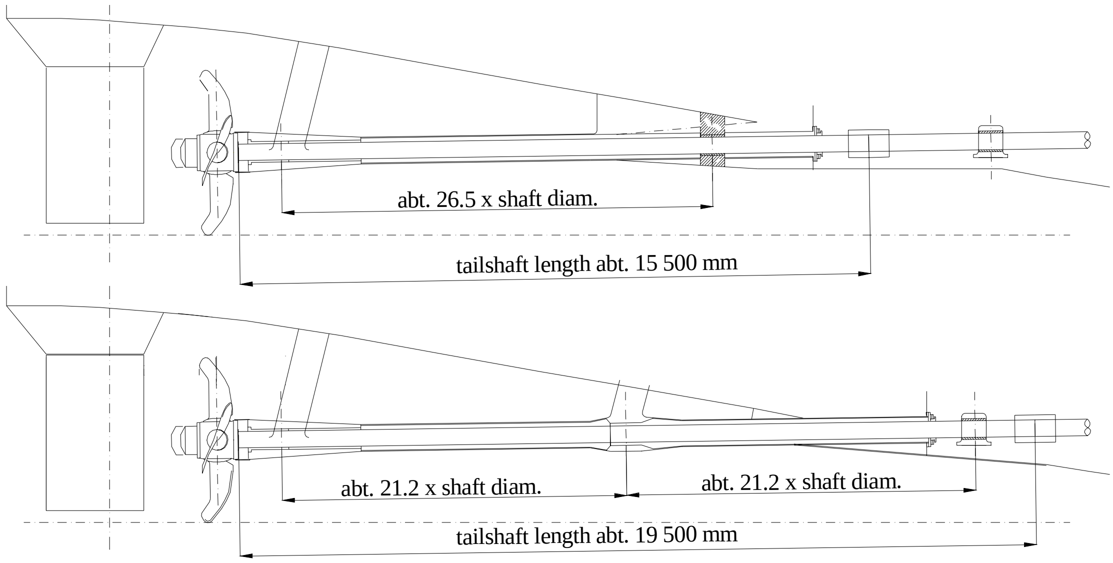


Figure 1-9 Two possible shaftline designs for the same vessel with and without intermediate bracket. Despite longer bearing distance the hull bossing size is significantly increased and the only advantage is shorter tailshaft length. Shaft diameter abt. 400 mm.

With intermediate bracket and somewhat longer tailshaft both bearing distances and hydrodynamics of the design can be improved.

Aperture between headbox and rudder should not be below propeller tip level in order to avoid erosion caused by propeller tip cavitation.

Sufficient clearance is needed to avoid erosion on rudder blade. On latest successful ferry designs the clearances have been 35 % to 50 % of D_p which is clearly more than suggested by classification society. Rudder cavitation erosion is particularly a problem with fast high displacement ferries.

Rudder headbox profile should be as long as possible to avoid flow separation on the aft part of the profile. Headbox alignment should also be determined in model tests.

Clearance between propeller plane and V -bracket trailing edge should be high enough for the turbulence caused by the brackets to dissipate. Parameter *c* (formula 3) should be less than 0.09.

Longer profile causes more disturbances in wake than shorter one when misaligned in flow. Therefore preferred bracket profile is NACA 64-021 over for example thinner 0018 series profiles. 64-021 also can sustain abt. 5 degrees change in incidence angle without increase in drag.

It is also suggested to measure alignment values for the brackets at several radii during model tests. With twisted brackets it is possible to improve wake quality by avoiding bracket “shadows”.

Conical aft bossing causes less disturbance on the wake than wider cylindrical one. Brackets are connected tangentially on the aft shaft bossing not radially, this is to open a better flow between the brackets.

Open shaftline with waterlubricated bearings is becoming also more popular. It gives an opportunity for minimised shaft disturbance.

1.1.4 Rudder

Spade rudders

To maximise rudder force at high rudder angles spade type rudder is usually selected. This kind of a rudder can act as “reaction blade” by deflecting propeller outflow using its total movable area. Especially when equipped with flap this type of rudder offers the best crabbing performance.

Disadvantage of spade type rudder is thick profile and often unfavourable profile shape from the resistance and propulsion point of view leading to higher appendage resistance and thrust deduction. Thick profile results from rudder stock that have to have large diameter in order to carry the rudder forces. Typical profile thickness of current flap rudder designs compared to conventional semi spade rudder profile is presented in table 1-2.

Table 1-2 Typical profile thickness of current flap rudder designs compared to conventional semi spade rudder profile.

	Thickness % of chord	Position of thickest part % of chord measured from trailing edge
Flap rudder (spade)	27 %	58 %
Semi spade rudder (NACA 63-021)	21 %	64 %

Semi spade rudder

Rudder supported by horn has smaller movable area than comparable spade rudder resulting in lower maximum lift generated at high steering angles. In normal operation propulsion power loss due to rudder is lower than with spade rudder because of the thinner profile. Use of thinner profile is possible because of rudder horn carrying the rudder forces instead of rudder stock.

When selecting profile a good alternative is NACA 63-021 and thinner profiles. These profiles allow quite high variation (3-5 deg) in angle of attack without change in drag coefficient. This variation can be caused by rotation component in propeller wake or constant steering caused by for example side wind.

End plates

End plates are horizontal plates fitted on bottom of the rudder blade to reduce flow around the blade tip and thus increase pressure difference over rudder blade. Since it has been found out in model tests that rudder operating in front of propeller producing astern pull does not product any significant side force end plates are useful only on inboard side of the rudder. Fig. 1-10 shows one arrangement of end plates on a 118 m ferry.

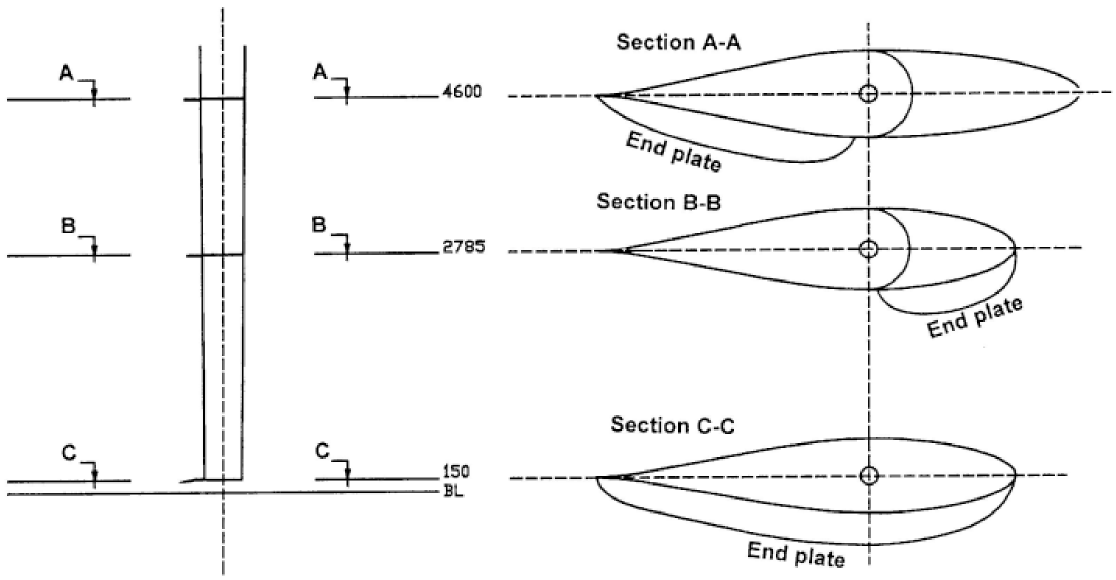


Figure 1-10 End plate installation on a 118 m high displacement fast ferry. End plates fitted only on inboard side.

Harbour manoeuvring, crabbing, example

Extensive harbour manoeuvring simulations and crabbing tests were carried out for a passenger train ferry (L_{PP} = 186,2 m, B = 29 m, T = 6,2 m) which had to operate in very confined waterways fully exposed to winds from all main directions /5/. Due to occasional ice conditions, owner preference was not to have flap type rudder, which would have otherwise been a natural selection for this kind of requirements. The speed of the vessel was also critical and thus a special fishtail profiled rudder with a wide trailing edge was also undesirable.

In the test set up there was a captive model in towing tank, instrumented so that all horizontal forces and moments acting on it could be measured. First it was realised that the rudder angle at backing side propeller had insignificant influence on crabbing forces, so zero angle on that side was used during the rest of the program.

Test procedure was quite simple, full absorbable power for the astern running propeller and enough power for the ahead running propeller to balance the longitudinal advance. Applicable power to the ahead running propeller varied strongly with different rudder set-ups (size, type and especially angle). Eventually one could even apply full available power on the ahead running side and start to reduce power on astern side.

Rudder size and location

Already in the beginning it became quite evident that the size of the rudder blade as well as rudder angle played an important role. Testing also clearly pointed out that a lifting surface has another side force peak after the stalling angle. For a foil with a low aspect ratio this second peak appeared to be higher than the first one before the stalling. Thus, only the area of rudder blade exposed to propeller slipstream, with the combination of angle of course, had impact on the side force generated. Location of rudders off the propeller shaft centre line was disregarded due to higher required propulsion power for the trial speed. The final total rudder area was selected to be 4,5% of underwater lateral area (L_{PP} x T). Rudder area of recently built ferries is presented in figure 1-11.

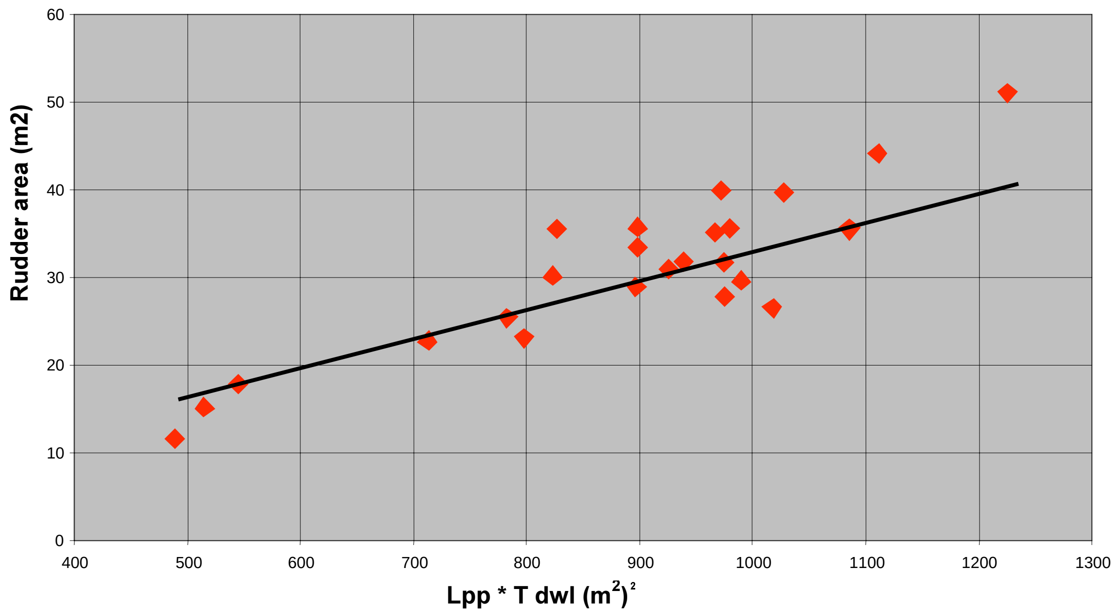


Figure 1-11 Ro-ro passenger ferries rudder area.

Rudder type

Having the rudder in line with propeller centre, the weakness of spade rudder became more obvious. Classification society allows only 23% of movable blade area to be balanced. Thus a great portion of propeller's jet stream would pass the rudder without an exaggerated large blade area. It would have also led to a very blunt profile and/or to additional profile and head box thickness.

In the semi-spade design the class allows the use of balanced section length of 35% of total chord length as long as 23% area limit is not exceeded. NACA 63-type profile was chosen with which the maximum thickness is exactly 35% from leading edge and it has low drag. It appeared that up 20% larger rudder blade could be used without any measurable power penalty compared to NACA 00 profile.

Good propeller slipstream blockage at high rudder angle gives also high drag. This is essential in reaching good crabbing performance, since balancing thrust is usually the limiting factor for conventional rudders. Due to pitch distribution and limited stroke in the hub bollard thrust astern is normally not more than 50-55% of ahead thrust for CP propellers.

End plates

The bigger the rudder angle becomes the more cross flow over the tops appears. Therefore small end plates were introduced to inner sides of the rudder blade only. Influence to the propulsion power appeared to be insignificant in model scale. Increment in side force as well as in drag was 10-12%, the peak appearing between 60-65° rudder angle.

A small wedge at trailing edge like in a fishtail profile was also tested, but like the end plates, at the inner sides only. An increased side force of 8-9% was achieved, the peak value shifted to approximately 50° rudder angle. Unfortunately powering performance was not measured in this case.

Manoeuvring performance

Table 1-3 presents manoeuvring performance comparison of recently built ferries showing impact of main dimension ratios and rudder size/type. Today most of the passenger ships and ferries have turning diameter below three times the length of the vessel, some even below two. Model test results are also fairly well confirmed by full scale trials. Results for residual rate of turn and overshoot angles, pull-out test and 20/20 or 10/10 Z-manoeuvre, are interesting. A few cases indicate course instability and/or problems with course checking ability. Low overshoot angles coincide well (model/ trial), but when high overshoot angles have been measured in model scale they have not been found in full scale trials and these ships are operating well. An other aspect is that designers tend to make vessels too stiff, quite many of the operators want to have a certain 'instability' to have a better and faster vessel to handle in difficult manoeuvres.
It always seems to be a compromise, better turning ability should lead to worse course stability and course checking ability, however, references exist where both turning ability and course stability have both been increased with rather simple modifications, e.g. large rudders, extended centre skeg and similar.

Table 1-3 Manoeuvring performance comparison of recently built ferries

Ref.					
Cb	0,62	0,607	0,62	0,689	0,59
L/B	5,50	6,06	5,20	6,42	5,90
B/T	4,35	4,23	3,97	4,68	4,00
Rudder area % Lpp * T	3,00	2,75	3,00	4,50	2,96
Type	Flap	Flap	Mariner	with horn	Mariner
Tact. Diam./L	1,46	2,475	2,26	2,186	2,50
20/20 oversht.	22,2	15	27	17,5	15,3

Thrusters

Bow thruster dimensioning has been very much based on references and simple design guidelines. Most of the recently built ferries can operate up to wind speeds 12-15 m/s. However, today especially with fast operating speeds it has become essential to be able to operate up to maximum wind speeds which means in most of the operating areas up to 20-22 m/s continuous wind speed.

Average bow thruster power in ferries is 0,54 kW/m^2 (total bow thruster power/projected windage area), varying from 0,28 up to 0,96 kW/m^2. The tendency today seems to be towards 0,6-0,8 kW/m^2 to be able to operate fast and safely under all prevailing wind conditions without any tug assistance. Stern thrusters seem to be dimensioned unanimously at 0,2-0,25 kW/m^2.

1.1.5 Seakeeping

Several factors are to be evaluated in a typical seakeeping analysis. This chapter presents first some practical design tools for evaluating factors relating the ship behaviour in waves to the geometry of a conventional displacement ship. Assessment of operability in waves requires the use of seakeeping criteria, which are given for motions and derived responses. A specific tool for bow flare estimation is presented and applied for an example case.

In practice, the ship’s main dimensions and hull form are largely determined by other design factors than ship motions in waves. When the main dimensions and hull form are fixed, there is not much to be done to reduce ship motions or the related derived responses. However, quite a good insight into the ship seakeeping characteristics can be obtained by examining the basic ship dimensions and coefficients, which calls for a seakeeping analysis at an early design stage.

Preliminary to computations, which are routinely used in design work, a good indication of seakeeping characteristics can be obtained by examining the natural periods of ship motion components. For the most important motion components heave, pitch, and roll, the uncoupled, undamped natural periods are

$$T_z = 2\pi\sqrt{\frac{\rho\nabla + A_{33}}{\rho g A_w}} \quad \text{for heave,} \tag{4}$$

$$T_\theta \approx 2\pi\sqrt{\frac{\rho\nabla k_{yy}^2 + A_{55}}{\rho g\nabla \overline{GM_L}}} \quad \text{for pitch, and} \tag{5}$$

$$T_\phi = 2\pi\sqrt{\frac{\rho\nabla k_{xx}^2 + A_{44}}{\rho g\nabla \overline{GM}}} \quad \text{for roll.} \tag{6}$$

ρ is the mass density of water, ∇ is the displacement volume, A_{33} is the heave added mass, g is the acceleration of gravity, and A_w is the waterplane area. k_{yy} is the pitch radius of gyration, which typically equals 0.25*L*, where *L* is the ship's length. A_{55} is the pitch added moment of inertia, and $\overline{GM_L}$ is the longitudinal metacentric height. k_{xx} is the roll radius of gyration, which is approximately equal to 0.35*B*, where *B* is the ship's beam. The roll added moment of inertia A_{44} can be estimated to be about 20% of the roll moment of inertia $\rho\nabla k_{xx}^2$. $\overline{GM}$ is the transverse metacentric height. If further approximations are desired, it may be roughly assumed that $A_{33} \approx \rho\nabla$ and $A_{55} \approx \rho\nabla k_{yy}^2$. Then the following simplified formulae are obtained:

$$T_z \approx 8.9\sqrt{\frac{\nabla}{A_w g}} \ , \tag{7}$$

$$T_\theta \approx 2.2\frac{L}{\sqrt{g\,\overline{GM_L}}} \ , \tag{8}$$

$$T_\phi \approx 2.4\frac{B}{\sqrt{g\,\overline{GM}}} \ . \tag{9}$$

The above periods ought to be compared with the wave encounter periods to be expected for a ship in the operational sea area. A scatter diagram gives the probability distribution of an apparent wave period *T*, such as the zero crossing period, which can be used to estimate a range of typical wave periods appearing with a high probability. Using the fundamental relationship for the encounter frequency, the following formula for wave encounter period T_e can be obtained

$$T_e = \frac{T}{1 - \frac{2\pi}{gT} V \cos\mu}, \tag{10}$$

where V is the speed of advance and μ is the heading angle (180° for head seas). Negative values for T_e indicate that the ship overtakes the waves in quartering or following waves. Comparing with a natural period T_n, the tuning factor $\Lambda = T_n / T_e$ can be evaluated for each motion component, and if $|\Lambda| \approx 1$, violent resonant ship motions may occur. Depending on the wave period and ship's speed and heading, different measures can be taken to avoid resonance motions. From a design point of view, it is noted that most ocean going ships operate in rough weather in the subcritical zone, where $|\Lambda| < 0.75$ for heave and pitch motions. If the natural periods for these motion components can be shortened, a somewhat higher speed may be used for subcritical operation. On the contrary, a fast ship in head waves may operate in sheltered waters in the supercritical zone, where $|\Lambda| > 1.20$, and ship motions are small. Reduction of natural periods would in this case require even higher forward speed to attain supercritical operation. In following waves, T_e is larger and resonant vertical plane motions may occur for fast ships as well.

One can analyse the problem of ship motions in waves also in the wave length regime. It is well known that wave induced ship motions in head waves are very small in the vertical plane if the wave length λ is shorter than about three-quarters of a ship's length. Data given as a function of wave length can be transformed to the wave period or circular wave frequency $\omega\,(= 2\pi / T)$ domain using the dispersion relation for deep water waves

$$\omega^2 = \frac{2\pi g}{\lambda}. \tag{11}$$

In case of an oblique wave encounter, the effective wave length encountered by the ship increases to $\lambda / \cos\mu$, and even shorter waves may excite ship motions. Also in this case, the encounter wave period can be calculated from equation (10).

It is a normal practice in ship design to perform seakeeping calculations with a strip theory computer program, which gives among others much more rational estimates for natural periods, and these computations are highly recommended. In addition to ship motions and loads, various associated dynamic effects, or derived responses, should be taken into consideration. These include motion induced accelerations, slamming, deck wetness, and added resistance. Many strip theory programs include evaluation of responses in random waves, which make it possible to obtain statistical seakeeping data. Extension to include long-term statistics is also possible.

The result of a seakeeping study should be presented in a simple form, which can be used in comparison of alternative designs. Several seakeeping indices have been proposed, and a widely used measure of merit is the operational effectiveness, which is given as the percentage of time the operation of a ship is possible. In order to evaluate the operational effectiveness, governing criteria have to be set for various responses of interest. Depending on the mission of a ship, the governing criteria vary, and a general set of criteria is given in Table 1-4.

Table 1-4 : General operability limiting criteria for ships (Karppinen et al., 1988).

	Merchant ships	Naval vessels	Fast small craft
Vertical acceleration at FP (rms)	0.275g ($L \leq 100$ m) 0.05g ($L \geq 330$ m)	0.275g	0.65g
Vertical acceleration at bridge (rms)	0.15g	0.2g	0.275g
Lateral acceleration at bridge (rms)	0.12g	0.1g	0.1g
Roll (rms)	6.0°	4.0°	4.0°
Slamming probability	0.03 ($L \leq 100$ m) 0.01 ($L \geq 300$ m)	0.03	0.03
Deck wetness probability	0.05	0.05	0.05

For vertical acceleration at FP, a nearly linear relationship exists for ships with length between 100 m and 330 m. For slamming the corresponding relationship is linear. When a special type of work or passenger comfort is considered, Table 1-5 gives criteria for accelerations and roll.

Table 1-5 Criteria with regard to accelerations and roll (Karppinen et al., 1988).

Vertical acceleration	Lateral acceleration	Roll	Description
0.20g	0.10g	6.0°	Light manual work
0.15g	0.07g	4.0°	Heavy manual work
0.10g	0.05g	3.0°	Intellectual work
0.05g	0.04g	2.5°	Transit passengers
0.02g	0.03g	2.0°	Cruise liner

In addition to the magnitude of acceleration, a human being is sensitive to the frequency at which the accelerations occur. A typical seasickness frequency range is from about 0.6 to about 3 radians/second. It has been found out that most severe seasickness occurs at a frequency equal about 1.07 radians/second, which can be encountered on many ships in rolling motion. A third important factor is the period of exposure. For short time periods, much larger accelerations can be allowed than for longer periods. The limit curves for vertical accelerations of the international standard ISO 2631/3 are often used in this context, and compared with predicted accelerations.

Bow flare

Extreme deck shape leads easily to extreme type of hull form and e.g. bow flare. Introduction of wide bow door and ramp is a typical and good example of an utmost difficult design task: When are we going too far in the deck and bow flare shape? Unfortunately several sad references exist. Greed for the last deck square metres or lane metres or ramp centimetres has led to poor or unacceptable performance in heavy or even in moderate head and bow quartering seas, heavy loss of speed, wave induced impact loads, noise and whipping vibrations occur. And to avoid this the master reduces speed or changes course and cannot keep the schedule.

It is not a straight forward design task to combine a slender design waterline (low resistance) with a wide trailer ro-ro deck and ramp or passenger cabin and public deck. The bow flare tends to become extreme. Excessive bow flare means high wave induced impact loads, high accelerations, noise, whipping vibrations, involuntary speed loss and at the end also voluntary speed loss and difficulties in keeping the schedule. The most extreme case, unfortunately not very rare, is when the applied dimensioning loads are exceeded and structural damages are met.

With bulky bow flare lines the applied sea margin becomes useless, it is not possible to use the installed power in heavy weather due to too high wave impact loads in the bow flare. A good rule of thumb is to avoid bow flare angle against waterline below 50 (45) degrees in unlimited service (unsheltered waters) and below 45 (40) degrees in limited service (in sheltered waters), figures in brackets showing absolute local minimum.

A simple guidance tool has been developed on basis of seakeeping model test results, the Bow Flare Estimator. Bow flare impacts have been measured for several ferries and passenger cruise ships in head and bow quartering seas with moderate and high seas, significant wave height varying from 1,5m up to 8m. The amount of impacts and pressures has been measured. Results have been compared with full scale behaviour of the ships.

This Bow Flare Estimator, denoted *BFE*, is defined for station located at distance x forward of midships as

$$BFE = \frac{x}{L\tan\alpha} \qquad (12)$$

where L is the perpendicular length of a ship and α is the smallest angle of flare against waterplane at the station. Ferries with a *BFE* value below 0.50 typically show good performance track record in full scale. Maximum measured full scale bow wave impacts in these cases were below 220 kN/m^2 in typical wave conditions, and below 300 kN/m^2 in extreme wave conditions. These impacts caused neither noise nor vibrations and no voluntary speed loss either. Examples of *BFE*s for ferries are shown in Figure 1-12, and for cruise ships in Figure 1-13. Frame number 20 is located at the forward perpendicular, i.e. x/L=0.5.

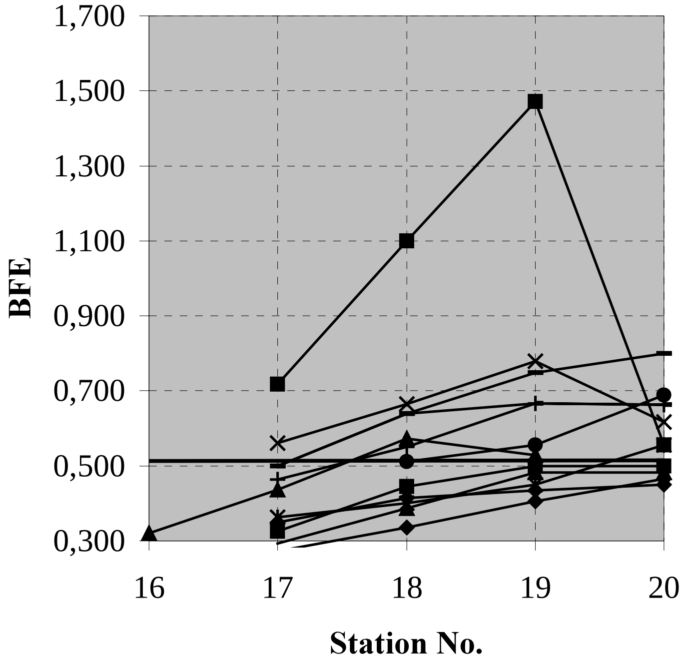


Figure 1-12 Bow Flare Estimators for ferries.

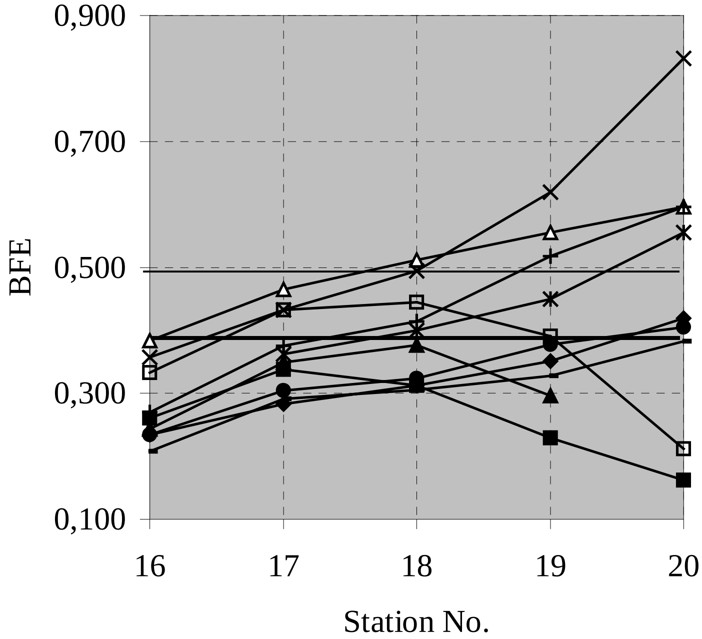


Figure 1-13 Bow Flare Estimators for cruise ships.

One representative example of bow form modifications is shown in Figure 1-14. The ex MS ‘Skandinavia’, today MS ‘Viking Serenade’ suffered from heavy bow flare impacts already at a significant wave height of 1.5 m in her service on the US West Coast. A bow form conversion was carried out and the effect of the modified body plan was checked with seakeeping calculations, which indicated a clear improvement in the behaviour of the ship. The modified bow lines are also shown in Figure 1-14, and service experience has proven the validity of calculations. The ship performs well on the route without any bow flare impact problems.

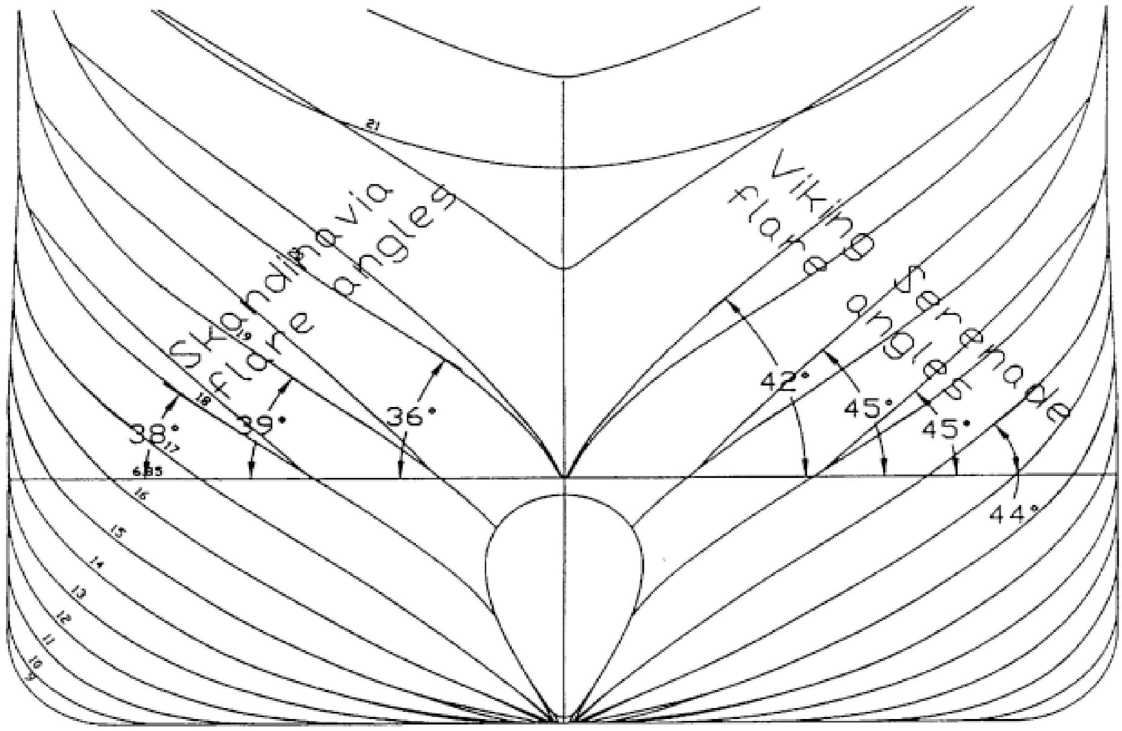


Figure 1-14 The original (Skandinavia) and modified (Viking Serenade) bow forms.

In the original 'Skandinavia' bow, the *BFE* was between 0.5 and 0.7, the modified bow resulted in an essential reduction in the *BFE* values. A comparison in Figure 1-15 shows these values, and only at one station the *BFE* value for the modified bow form lies above the level 0.50.

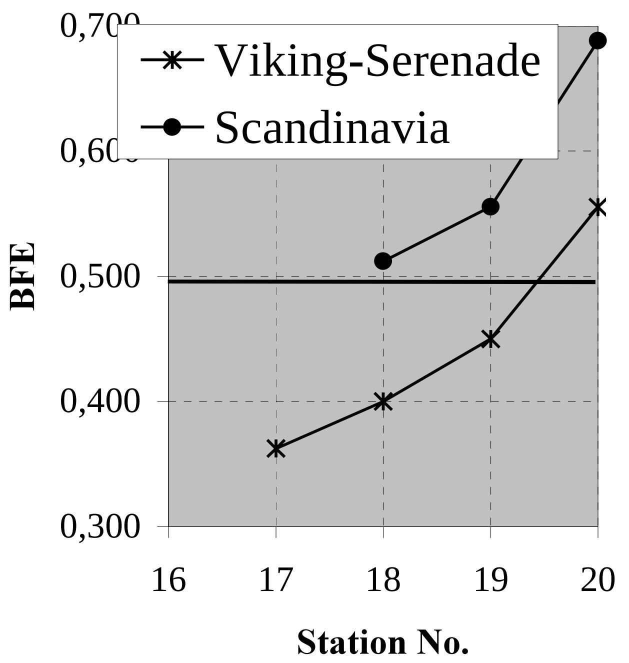


Figure 1-15 Bow Flare Estimator values for the original (Skandinavia) and modified (Viking Serenade) bow forms.

This hull form is a good example of a relatively small hull form modification, which leads to an essential improvement in seakeeping behaviour. In many cases it is erroneously assumed that such hull form modifications are of minor importance only.

Sustained speed

The old liners have very slender bow and low values whereas the recent Caribbean cruisers have rather high estimator values, close to 0,7 and even over. The Caribbean wave climate is not the most severe one and sailing schedule versus maximum service speed gives some relaxation. But operation in more harsh environment, e.g. cruising world wide or just from the UK to the Mediterranean and passing the Bay of Biscay may cause harmful bow flare impacts, noise, vibrations and speed drop.
Bow flare estimator of about 0,5 is a good limit for Caribbean service and 0,4 for world wide cruising.

Figure 1-16 shows body plans of two cruise ships for world wide operation. Bow flare estimator for both ships is presented in figure 1-17. The difference in minimum flare angle is 4-6 degrees, but the differences in seakeeping capabilities are obvious. Seakeeping calculations with downtime analysis for North East Atlantic service were carried out.

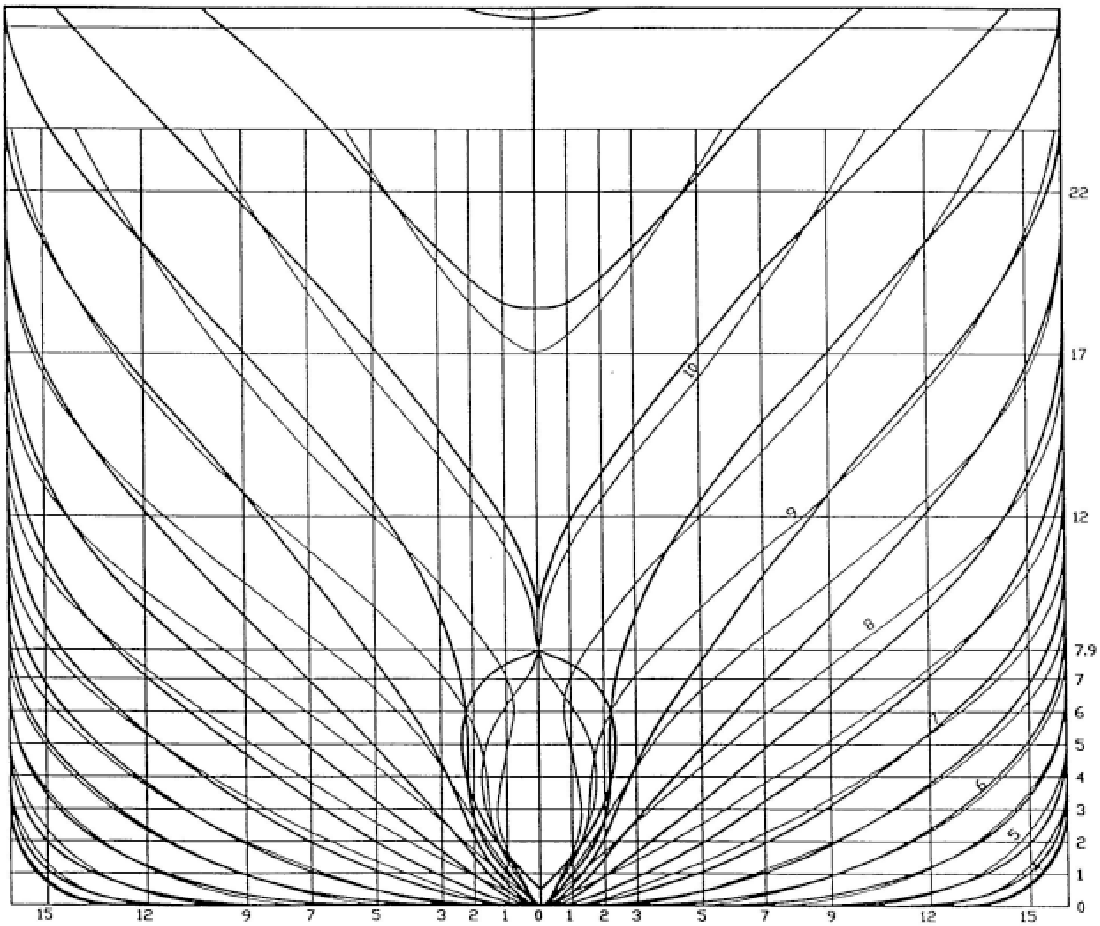


Figure 1-16 Body plan for the comparable cruise ships.

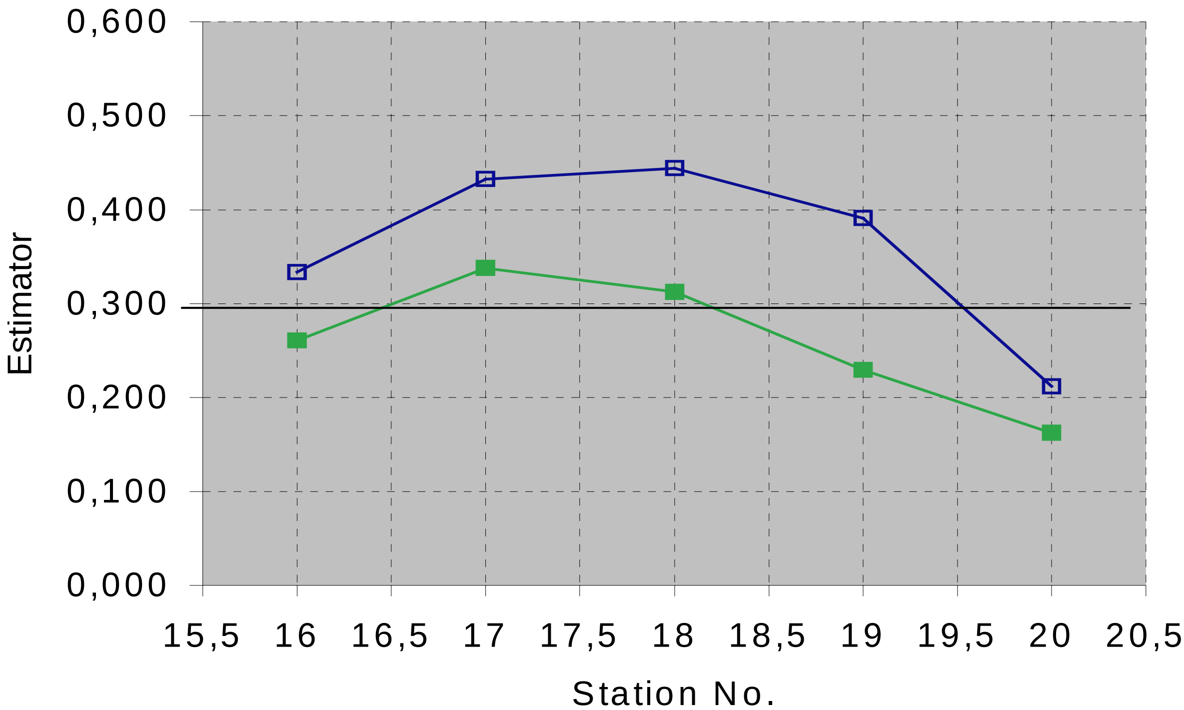


Figure 1-17 Bow Flare Estimator values for two cruise ships compared.

Bow flare impact criterion of 60 kN/m^2 was applied as descriptive criterion for the passenger comfort. In bow quartering seas at the North East Atlantic this criterion is already met at significant wave height of 4,5m with the worse performing vessel, heavier bow flare, and reducing the speed does not help. There is, however, power enough to sail 20+ knots speed at 4,5 m wave height sea state, but this power cannot be used due to loss of passenger comfort. The vessel with better bow flare, however, can handle waves up to 7,5m without exceeding the criterion. Resulting downtime is small, 2,2% of the time constantly sailing in bow quartering seas.

The difference in deck areas between the two bow flares is minimal and 2-3 additional passenger cabins hardly can justify the worse seakeeping performance.

1.1.6 Conclusions

It is worth while paying attention to the hull form design of a newbuilding. Look for references and compare with the state of the art on the market. There seem to be many ‘truths’ but at the end performance at the prevailing service conditions counts. You need to discuss with the model basins, designers, yards, shipping companies and the crew.

Figure 1-18 shows a comparison of two curves of sectional area (frame area) from the different hull forms prepared for the same project, i.e. they have the same main dimensions, displacement and LCB. In the aft ship they can both accommodate the same machinery arrangement. In the bow area version B lacks some deck area. Comparing the curve of sectional areas shows that the A version has a smaller bulbous bow, less displacement in the forward entrance area and clearly more pronounced forward shoulder, midship section coefficient is the same, A version has again a more pronounced aft shoulder and less displacement in the aft part.

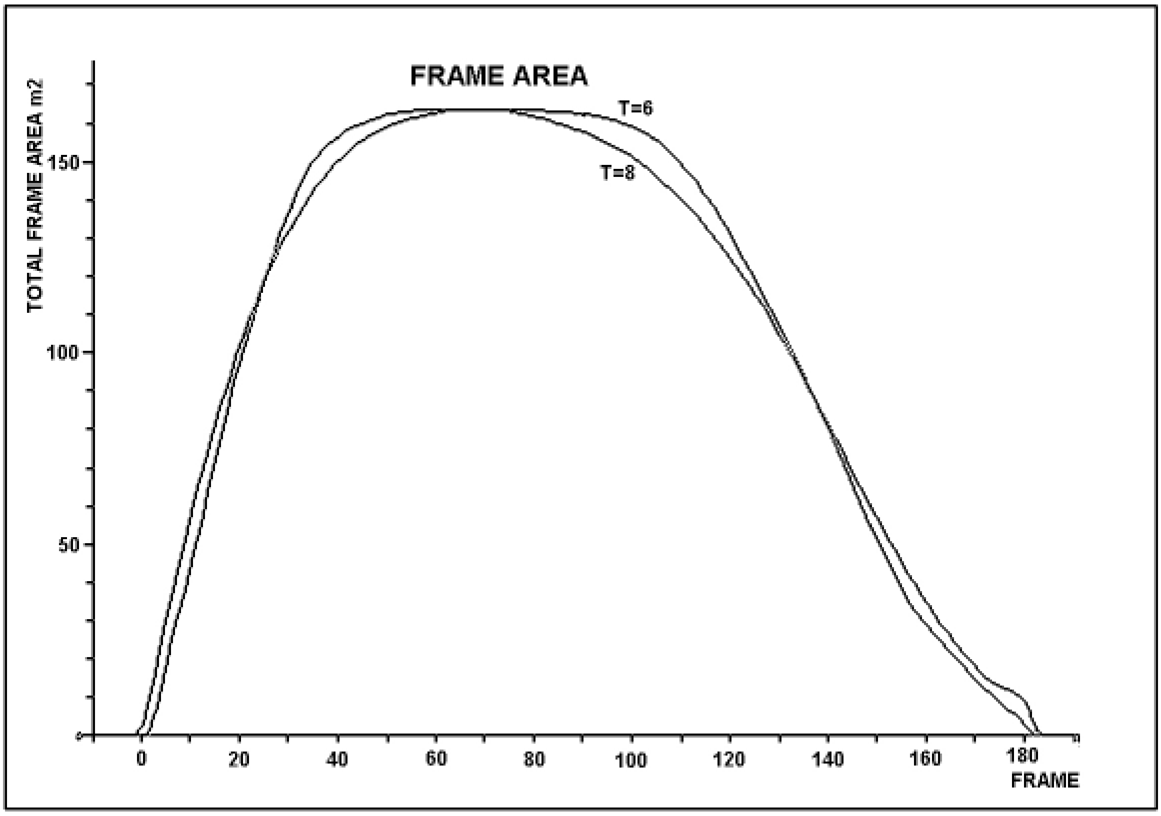


Figure 1-18 Comparison of curves of sectional area of two alternative hull form designs for the same project.

The expected difference in trial propulsion power is minimum 10% in favour of the version B. The operational downtime in moderate and heavy bow and bow quartering seas is about three times higher for version A than for version B, and a higher risk of disturbed flow in the aft ship and propeller induced vibrations exists for version A. The difference in bow flare angles was from about six degrees on an average, version A having minimum angles from 34 to 48 degrees and version B from 41,5 to 51 degrees respectively.

Stability lever KM was 0,50 m higher with version B, i.e. version B can take more weight on the superstructure or bigger superstructure and/or has better stability margin.

Generally the stability lever in recently built passenger cruise ships varies from 51,5% of the beam of the ship up to 56%. A good figure as starting point for a new project is 54-55%, but this typically requires a pram type stern or otherwise a pronounced forward shoulder is required.
In ro-ro passenger ferries the range is from 52% of the beam of the ship up to 58,5%, the highest figures are reached with a wide full transom close to design waterline, typically applied today in fast high displacement ferries.

1.2 Propulsion

1.2.1 Powering performance

Design for efficiency should be the target of every commercial ship design. Fuel consumption and speed-power performance are good indicators to show the adaptation of proper design criteria and state of the art know-how. Propulsion fuel efficiency can be verified with various measures and the final outcome is the daily consumption in actual service, which is a sum of hull efficiency, resistance, propulsion efficiency and machinery and transmission efficiency. Resistance and propulsion efficiency can further be subdivided into calm water and under sea state performance. The easiest to measure are the calm water figures and the most reliable to compare are the model test results. Concentrating on the hull and propulsion efficiency, one way to measure the degree of fuel efficiency is to use coefficients, e.g. Heickel coefficient:

$$K = \left(\frac{\sqrt{\nabla}}{\mathrm{P_B}} \right)^{1/3} \mathrm{x}\ \mathrm{V_S} \qquad (13)$$

∇ = displacement in m^3
P_B = engine power in kW
V_S = ship trial speed in m/s

at the same Froude number

$$Fn = \frac{V_S}{\sqrt{gL}} \qquad (14)$$

L = length of waterline in m

When comparing this Heickel coefficient of some recently built ferries and passenger ships at equivalent Froude number surprisingly wide range can be found: differences up to 30%. The same differences exists in the fuel bill as well. That is of course an unnecessarily big difference especially if and when these ships may compete on the same market. A good example is the Adriatic route Patras - Ancona where modern high displacement fast ferries are sailing with big differences in tonmile fuel consumption.

Lower required propulsion power means not only lower fuel consumption but also smaller main engines. The smaller required engine room space and also the weight are giving larger service and cargo (ro-ro/passenger) spaces and smaller investment cost.

Several examples exist where comparative model tests have been performed and differences ranging from 3% up to 30% have been found between the various designs, hull forms and propulsion arrangements, however, always with the same main dimensions and displacement. Most recently extremely good results have been reached with pod propulsion.

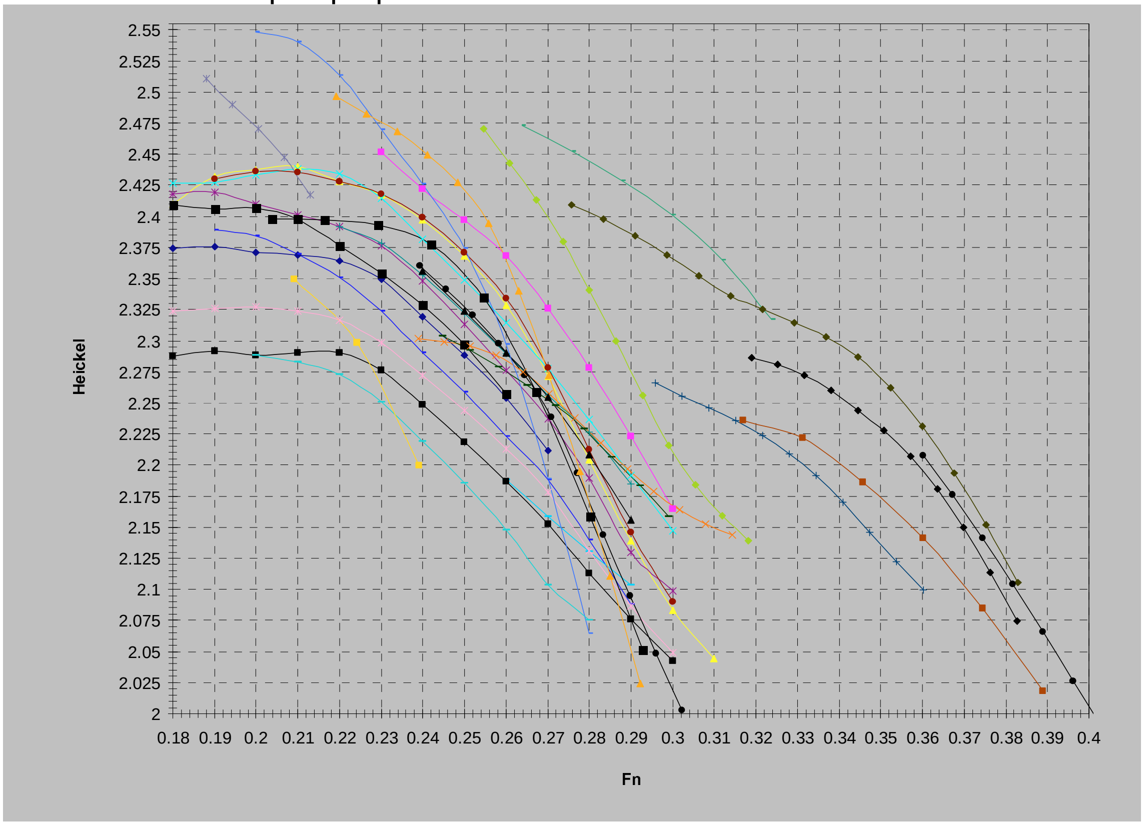


Figure 1-19 Heickel coefficients for recently built passenger cruise vessels from 4300 up to 110000 grt

Figure 1-19 presents typical Heickel coefficients for passenger cruise ships. The curves should be read as follows: the higher the Heickel coefficient the better the performance (less power required to reach the same speed), and the smaller the ship (smaller displacement) the higher the coefficient should be. The highest curve in the figure is for a small size cruise ship, main dimensions L_{PP} = 78,8 m, L_{WL} = 80,4 m, B = 15,3 m, T = 3,85 m, displacement 2875 m^3, propeller diameter 2,8 m, trial speed 16,3 knots and maximum speed 17,3 knots. Cruise ships of 70.000-75.000 grt are in the range from 2,22 up to 2,29 at Froude number 0,24. The recent Panamax size ships of 83.000-87.000 grt at the same Froude number are from 2,265 up to 2,31, all with pod propulsion. The improvement in propulsion performance with pod propulsion can be clearly seen, 16-19% bigger ships in volume are reaching the same speed at lower propulsion power. Of course, some development has been in the overall design of the hull as well but most of the benefit is coming from the pod arrangement. And these ships are the first generation of cruise ships originally designed for pod propulsion. It is obvious that we are just witnessing, April 1999, the first steps in the development of specific vessels based on pod propulsion and arrangement. The challenge and the potential is still for major improvements.

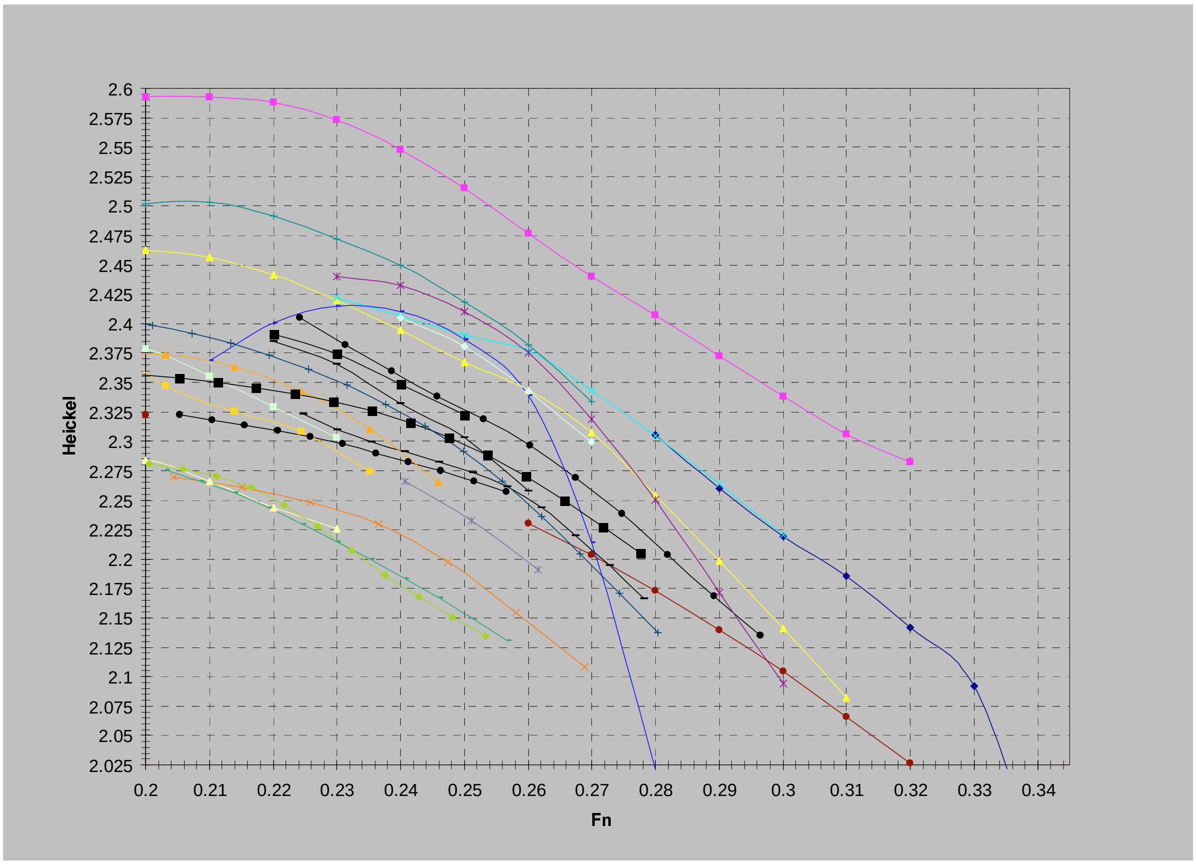


Figure 1-20 Heickel coefficients for recently built ro-ro passenger ferries

Figure 1-20 presents Heickel coefficients for recently built ro-ro passenger ferries. The curves on the right hand side are for fast high displacement ferries with trial/operational speed between Froude numbers 0,34-0,38. The curve in the middle of these six curves is for a ferry with L_{PP} = 158,5 m, L_{WL} = 160 m, B = 24,8 m, T = 6,45 m, displacement 15.600 m^3, propeller diameter 5,0 m, trial speed 28 knots and maximum speed about 29 knots. On the left hand side the vessels included are from big cruise ferries of 60.000 grt down to small ro-ro ferries of 9.000 grt, a few ro-ro ships are included as well. Hull forms vary from rather conventional V-type aft ship through twin skeg aft ship into modern pram type aft ships, and in some of the fast high displacement ferries even efficient trim wedges are included. Some of the ferries have also 1A and even 1A Super ice class.

With proper references it is easy to check whether the proposed design is of high quality or should it be reconsidered. It is advisable not to stick to predetermined configurations, open-minded approach gives typically better results, of course considering at the same time any possible risks.

Propulsion system and propulsion design criteria are typically demanding and some times also counteracting: high efficiency through all operational modes, high thrust at manoeuvring modes forward and astern and short reaction times, no harmful propeller induced vibrations and no harmful cavitation on the propeller blades. These design criteria always lead to a compromise solution and rather big differences can be found in the performance figures when comparing designs from different makers. It is advisable to request calculated propeller/propulsion system performance characteristics as early as possible in the development of a new project. It should be possible to receive both calculated figures and guarantee values for efficiencies, thrusts, and forces induced against the ship hull. For the designer of the vessel it is important to know what is possible to be reached and which are the consequences for his selection of design criteria.

1.2.2 Propeller diameter and location

Propeller diameter is typically selected as big as fits within the hull and this is to get the rpm and propeller loading down and efficiency up. But this should not increase the hull resistance or induce poor hull-propeller interaction. The first thing to start with is to locate the rudder-propeller combination as aft as possible. This will allow the use of maximum diameter as well as give some freedom for the hull design to have adequate propeller hull clearance without going into extreme vertical (buttock) angles and tunnels above the propeller. A good rule is to take the rudder up to the transom. For passenger vessels and especially for ro-ro passenger ferries a clearance of about 20% of the propeller diameter should be left between transom and rudder trailing edge to avoid air ventilation into the rudder. This can be avoided well by applying ducktail and the rudder can be taken even partially under the ducktail. Typically the ducktail length is 3-5 m or even more depending on the stern configuration and speed, the higher the speed the longer the ducktail should be.

Rudder-propeller clearance should be between 15 to 50% of propeller diameter depending on the propeller loading. 15% is adequate for 400-500 kW/m^2 loading (power per propeller disc area) and should be increased up to 50% when loading is increased up to 1500 kW/ m^2 or even above. In extreme cases twisted rudder profiles are recommended to adjust the rudder in accordance with the flow and to avoid cavitation on the rudder blade itself.

Propeller loading (kW's per disc area) for passenger cruise vessels varies from 300 up to 950 kW/m^2, the average being 650 kW/m^2. Only QE2, originally being a Transatlantic liner, has a clearly different propeller loading 1670 kW/m^2.
The average propeller loading in ferries is slightly higher than in cruise ships, 700 kW/m^2, but the tendency is upwards due to high displacement fast ferries in which the propeller loading is already approaching 1500 kW/m^2 with successful results, i.e. high propeller efficiency is reached together with acceptable propeller induced vibrations. Of course GTS Finnjet had already at the end of the seventies propeller loading close to 1800 kW/m^2, and navy frigates have loadings above 2100 kW/m^2. This, of course, requires a very good flow into the propeller with minimum disturbance from the hull, shaft and shaft supports.
Pod propulsion offers a good opportunity for even higher propeller loadings as the flow towards the pod propeller is not disturbed by any shaftline or shaft supports.

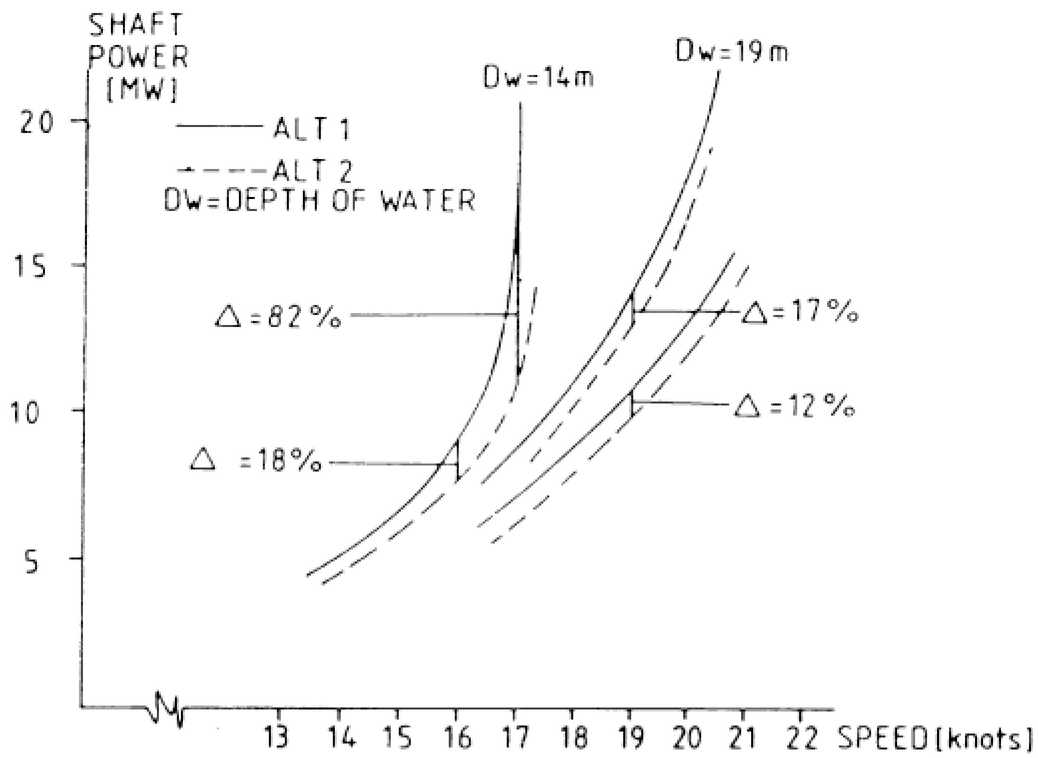


Figure 1-21

Transverse location of propellers in twin screw ships varies between 14% of beam of the ship up to 25%. The closer the propellers area located the shorter the propeller shaft outside the hull becomes, i.e. reducing the appendage resistance. But locating propellers close to the centreline reduces the propeller diameter, i.e. it is difficult to introduce enough space for large propeller diameter and adequate clearances. One should take into account propeller loading, propeller hull clearances (not only as a function of propeller diameter but loading as well), propeller shaftline and its supports and the hull shape when selecting the final compromise. A good starting point is 20% of beam for passenger cruise ships and 22% for ro-ro passenger ferries. This gives freedom in the hull form design in the aft shoulder transition area aft of the flat bottom. To ensure best flow for propellers the hull should allow good mixed flow both from the sides across the bilge and underneath. The risk of creating hard bilge vortices ending up into propeller inflow should be minimised.

Propeller diameter for typical twin screw ships with speed up to Froude number 0,27 is about 75% of the design draft. References exist with propeller diameter up to 82% of design draft. Baseline clearance is 100 mm and propeller-hull clearance between 25-30% of propeller diameter, 26-27% being typical value. In high displacement fast ferries propeller diameters up to 85% of the design draft are being used. The baseline clearance is taken down to zero, but in any case not more than 50 mm. Propeller-hull clearance is the maximum possible, typical values being between 22-27% of propeller diameter, 25% being a good design guidance. A risk is clearly taken with the high diameter for air ventilation into the propeller at the manoeuvring modes, especially when going astern but also when accelerating. Five bladed propellers are being used to allow some freedom for the propeller design.

1.2.3 Shallow water and wash effect

An efficient combination of the different design criteria is needed. Operation in limited water depths in harbours and routes calls for a careful study between maximum possible draft/propeller diameter and dynamic sinkage and trim in shallow waters. Pram type hull form combined with a long ducktail and trim wedge minimises the dynamic trim into zero. Bigger propeller is also additionally more efficient in shallow water than a propeller with smaller diameter compared at same power and ship speed. Figure 1-21 shows a comparison of two hull forms and propeller diameters in shallow water. Model tests in shallow water were carried out for two ferries with same displacement of 15.300 m^3 and draft of 6,0 m, one with more conventional aft ship hull form and with propeller diameter of 4,50 m and the other with pram type aft ship and semi tunnels and propeller diameter of 5,0 m. An astonishingly big difference was measured in the shallow water performance. The pram type stern was able to reach almost one knot higher speed. A big difference was measured in the wash effect as well. The pram type hull shape with bigger propellers was creating only half of the wave height at the same speed.

1.2.4 Air ventilation

Air ventilation into the propeller in general appears in manoeuvring and acceleration modes if the propeller shaft immersion becomes less than 80% of the propeller diameter. This has been studied in both cavitation tunnel and towing tank and seem to correspond quite well with full scale performance. Continuous air ventilation will cause thrust breakdown of the propeller, i.e. propeller is consuming power but thrust is dropped down to 10-15% of the original thrust. Aft ship hull form can help to push the limit of incidence abt. 3-5%, e.g. pram type hull form with trim wedge.

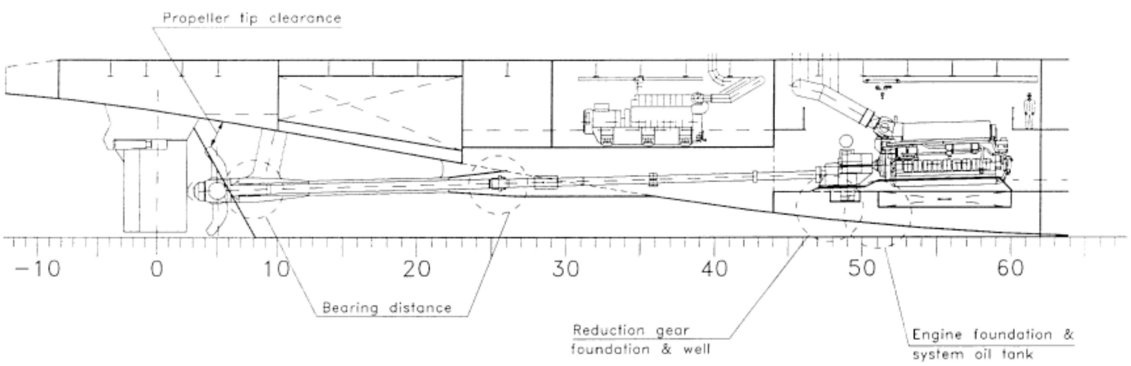


Figure 1-22

1.2.5 Pod propulsion

The freedom of optimised hull form design is offered by the pod propulsion arrangement as shown in figures 1-22 and 1-23 pointing out the limitations of unconventional shaft arrangement for a twin screw vessel and on the other hand possibility for optimised hull design with pods.
The propulsion power saving is at least between 10-15%, even differences up to 20-25% have been measured in comparable model tests between conventional twin screw shaft arrangement and twin pod arrangement. The difference in propulsion power is summed up from the optimised hull form, absence of shaftlines and rudders and optimised position and orientation of pods. The pod propeller shaft should be inclined towards the baseline at about 50% of the respective vertical angle of the hull form, i.e. with a vertical angle of 9 degrees the pod orientation should be about 4,5 degrees. The orientation in the horizontal should be as well towards the flow, a typical figure is 2-3 degrees leading edge outwards referred to the centreline, i.e. propellers are oriented properly against the actual water flow, which hardly is possible with conventional shaft arrangement. This also gives the propeller designer some additional freedom as the inflow angle is optimum. The pods are developing a quick change in the flow velocity around themselves as well as around the hull, they are working as stern bulbs and when properly located they will reduce the transom and aft ship wave system. This is a big benefit especially for higher speeds, Froude number above 0,27, when aft ship wave making becomes an important part of the total resistance. This offers an interesting opportunity especially for high displacement fast ferries. The most efficient pod location, both transverse and longitudinal can be found out in model tests.

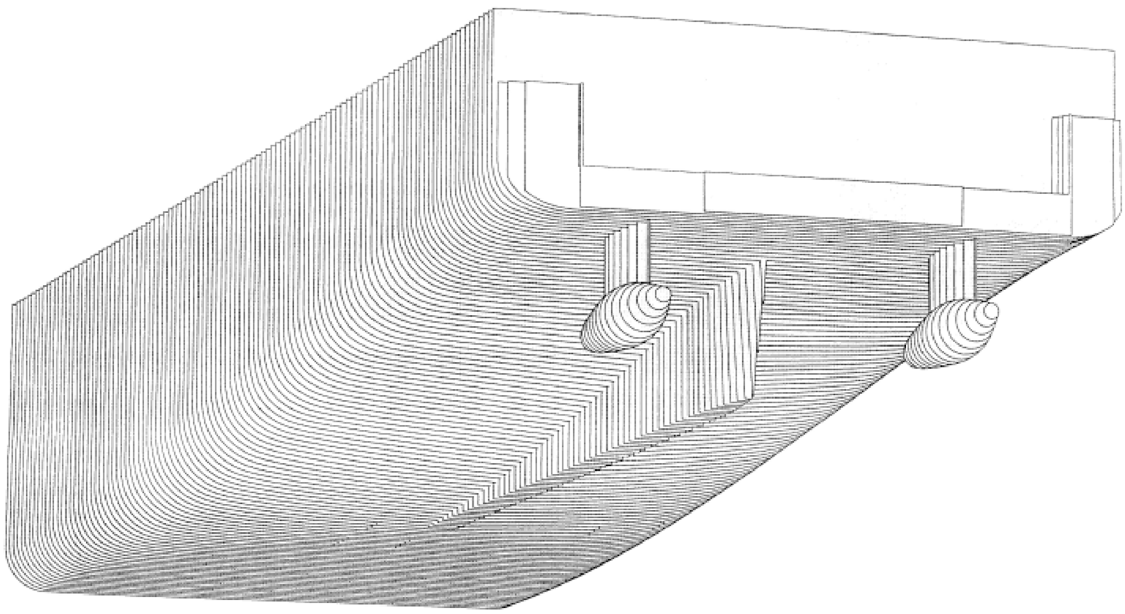


Figure 1-23

The total unit, pod and propeller together, open water efficiency varies between 0,67-0,72 for the typical twin pod configurations measured mainly for passenger cruise vessel applications.

Propeller tip - hull clearance can be minimised due to the homogenous flow into pulling pod, i.e. due to no disturbances of shaftline, bossing and brackets and possibility for flow optimised hull form. The total aft ship displacement can be the same for pod configuration and the longitudinal centre of buoyancy can be moved even more aftwards without disturbing the good flow properties. Propeller induced hull forces can be handled efficiently without sacrificing the efficiency of the unit. At the best only the non-cavitating pressure fluctuations are measured at the hull, which is 50-70% less than measured with conventional shaft and propeller arrangements.

Other hydrodynamic benefits are improved performance in shallow water, improved stopping capabilities and extraordinary manoeuvrability especially in harbour crabbing mode.

The optimised aft ship hull induces higher dynamic lift which is reducing the additional resistance in shallow water.

Stopping can be carried out by just turning the pod units into an angle of 30-45 degrees without changing the propeller turning direction. This is especially important at lower manoeuvring speed below 12 knots. This is absolutely the fastest and most controlled way of stopping a vessel. It applies also to crash stops, on the other hand several stopping practices are available and most suitable can be selected for each specific vessel and case through simulations.

Crabbing becomes efficient as propeller thrust can be steered exactly to the intended direction of motion at minimum power; no more dredging of harbour basins and minimum amount of exhaust gas emissions.
With proper setting of pods for manoeuvring mode reaction times can be dramatically reduced from those of conventional arrangement: 30 to 60%.

Pod propulsion system is also making its entrance into single screw vessels either with single or even with twin pod arrangement. Some further ideas and results of studies are described in chapter 3.

1.3 Machinery Systems

1.3.1 Machinery selection

The driving force in machinery development is and has always been the search for higher power. This is followed by other targets, such as better economy and more compact design.

Power plant as a whole defines the feasibility of machinery and ship configuration. An attractive power plant configuration may need electric drive or electric drive is an inherent part of a new, more efficient propulsor as applied on pod propulsion. Electric propulsion, and widely the complete machinery, is no more a separate item: it combines modern ship design with modern construction principles and latest development in component sector.

Typically the evaluation of a new concept is carried out by considering the traditional design as a base option and the new solution as an improvement of that. This is not the correct way because the principal idea of the general arrangement and functions are then tailored to feature the traditional machinery and not the new one. The starting point should be different: the new machinery concept and its advantages should be utilised in the general arrangement development already from the beginning.

Guidelines

Some general rules have to be considered when searching for the optimum machinery. The fuel selection must be made because the fuel type has a major impact on the feasibility of any option. The machinery shall support ship concept development and vice versa; each machinery must be applied by utilising its features, such as low weight or small size, in novel ship design. You should concentrate on the power plant configuration. The most important features for the owner should be considered: cargo space, low emissions, good manoeuvrability, high speed, low fuel cost, low first cost, or a combination of the above. The operational profile of the ship / cargo transport mission must be defined. Power demand, propulsion, electric and heat must be estimated. Rough estimate of differences in earning potential with alternative machinery concepts must be made. The rough first, fuel and other running costs must be calculated. The differences in earnings must be reflected to differences in costs. The machinery must be in line with the standard and mission of the ship; do not design Rolls Royce if the target is Ford.

Today most of the new machinery concepts are based on high utilisation of electric power transmission. However, you should bear in mind that the electric propulsion is actually a variety of options as shown in figure 1-24.

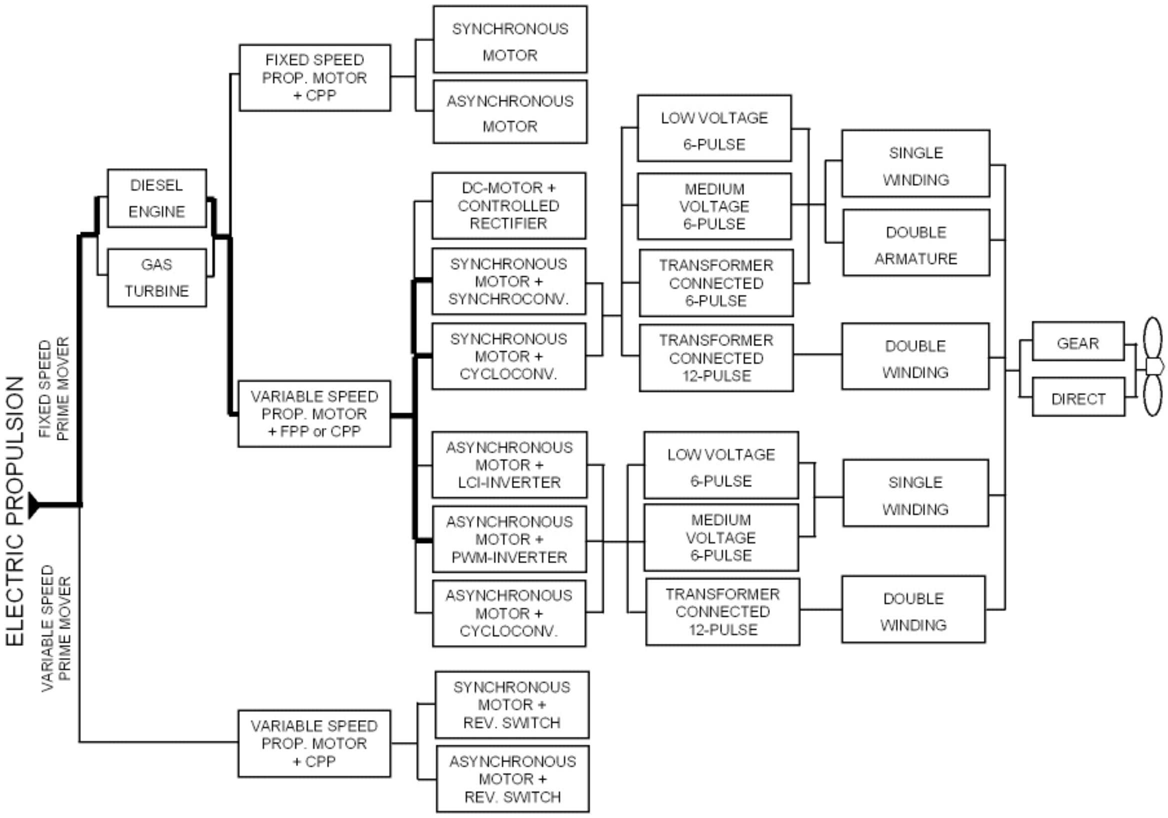


Figure 1-24 Possible options in selecting electric machinery

The selection of electric propulsion system can be divided into several steps. The power plant is most important because it affects strongly the ship arrangement and thus the earning capability of the whole vessels. Operation pattern and modes affect the choice of motor control whereas motor and converter details must be in line with the requested degree of redundancy. The total cost of the plant can be adjusted by selecting a correct combination of the converter and motor details whereof motor speed is most important.

It seems that the most feasible solution on cargo vessels, such as tankers, with slow running, 70-80 rpm, propeller would be high speed propulsion motor connected to reduction gear. This concept is applied on all diesel-electric tankers lately built for Stolt Parcel Tankers.

Passenger and ro-ro ships have faster running propellers than cargo ships, typically 150-180 rpm, and thus benefit of the reduction gear is small. On these vessels the choice is a directly connected motor.

Consideration of consequences

After manning, maintenance and repair (M&R), is normally the largest single item among operating costs, and in some cases it can even exceed manning costs. Like manning costs, M&R is also one of the major areas where owners/operators are capable of saving during periods of depressed revenues. The cost implications of damage or unscheduled repairs are largely unquantifiable, but this is not the case with scheduled repairs. Part of the scheduled work is undertaken by the crew and covered therefore by manning and store/spares costs. This activity has, however, declined considerably due to the reduction of crew number on the ship. Unlike manning, M&R costs do not vary greatly with flag. Fuel and machinery configuration have significant impact on M&R costs especially when diesel engines are considered. Huge differences can be found when evaluating scheduled M&R costs of machinery on a ship with electric propulsion to same kind of a vessel with mechanical propulsion. This difference is further pronounced when different fuel grades and different cylinder sizes are considered. Relative figures on scheduled maintenance, based on one detailed study, are:

- Diesel-Mechanical with HFO 100%
- Diesel-Mechanical with MDO 77%
- Diesel-Electric with HFO 75%
- Diesel-Electric with MDO 63%

Gas turbine based power plants

One of the decisive features in electric propulsion is the freedom to select power plant configuration and propulsion unit to suit optimally for each project. In most cases power plant allocation (location) and configuration is the dominating factor when determining the economical feasibility of the machinery configuration, and thus the main interest must be paid on the choice of power plant and simultaneous evaluation of the possibilities each option gives for the ship designer. Reduction of 6000 meters in machinery ancillary piping or increase of passenger cabin number by 50 cabins are attractive targets which have been calculated and shown when comparing alternative options on same ship project.

Together with the economy, safety and environmental issues are getting increasing importance. Today's vessel calls for a simple, safe and low emission machinery at reasonable cost.

The gas turbine is a real option especially for passenger cruise vessels but also for other ship types. The combined cycle is needed when applying gas turbine on other vessels than high speed craft in order to gain lower fuel consumption and thus economically feasible installation. On fast ships the space is limited and the simple cycle is the only choice, and higher fuel consumption must be then accepted if gas turbine will remain as an option. In most combined cycle applications the turbines are driving generators and thus also electric propulsion is a request. The pressure towards gas turbine propulsion rises actually from various reasons:

- The shipowners are interested in low weight and compact size.
- The authorities are looking for low emissions.
- The shipyards want simple design, low number of ancillaries and clear turn-key possibilities.

As with electric drive, there are also several options available when selecting gas turbines as shown in figure 1-25.

With this choice one can heavily affect the performance properties as well as the first and fuel cost of the machinery. In every respect, the choice of gas turbine plant must be closely tied to actual power demand (propulsion, hotel, heat) and ship operation data.

Following choices have to be made when selecting the machinery:
- Definition of operation patterns, check of all possible operation modes.
- Choice of engine type: aeroderivative or industrial.
- Simple cycle, combined cycle or even recuperative.
- Cogeneration or with steam turbine.
- Base load plant or booster plant.
- With single or several heat recovery boilers.
- Light or intermediate fuel.

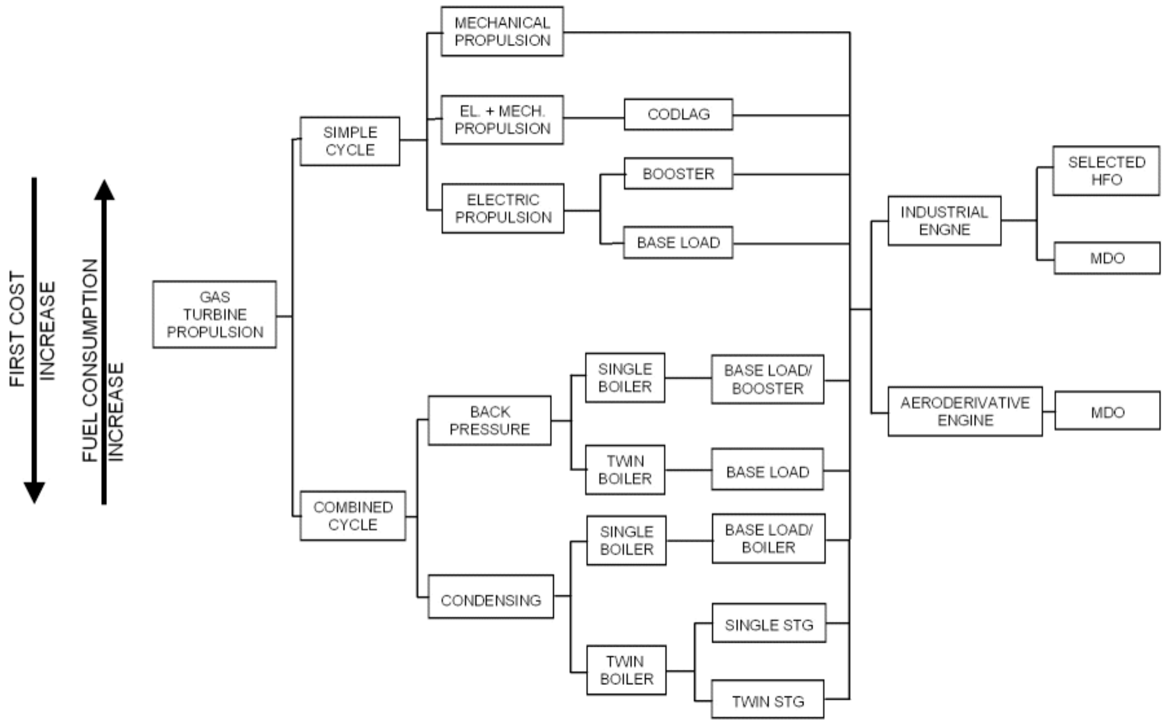


Figure 1-25 Possible options when considering gas turbine propulsion. This graph is the content of the first box “gas turbine” on the electric propulsion graph presented in figure 1-24.

The first gas turbine based cruise ships will feature electric propulsion, twin independent base load plants with aeroderivative engines and common back pressures turbine.

The solution on gas turbine based high speed ferries and other ships with strictly limited space demand would be simple cycle machinery connected to reduction gear and mechanical power transmission. However, the right solution must always be selected case by case.

Modern shipyards are using extensively labour saving methods, such as prefabricated units and pipes and even modular design. Ancillary modules made in workshop including several units and all related pipes in the area are becoming more and more typical. Machinery configuration supporting this development is preferred by the yards.

The electric propulsion and power plant concept with a variety of prime mover options supports this development in the most efficient way by allowing standard designs for ancillary systems. This can be one of the major issues in the future why electric propulsion will be selected.

A viable machinery option is compact in size, produces high power at low weight, releases minimum pollution, is fuel efficient and reliable, gives a good support and flexibility for the ship arrangement development and is easy to install and maintain.

The electric propulsion has these features and thus it is justified to believe that the concept will remain. Major questions in the near future concern the configuration of power plant and the time needed for the podded propulsor to gain remarkable market share also in the traditional merchant fleet.

1.3.2 Fuel selection

Could MDO be a fuel of the future? Availability is not a problem when burning MDO but the fuel price is. MDO would be perfect fuel for a green ship: less CO2 emissions due to lower fuel consumption, SCR units are functioning better due to no risk of blocking by ammoniumsulfite, low SOx emissions due to low sulphur content, no risk for too low exhaust temperature.

The fuel type has several impacts. Maintenance cost equals easily to some 15-20% of the fuel cost. Maintenance of the engine itself is decisive when evaluating the maintenance cost of the total machinery; some 80% of the cost is accumulated from the diesel engine where fuel type and quality plays a major role. It is calculated that some 30-40% can be reduced from scheduled maintenance cost by burning MDO instead of HFO. This can generate almost $ 100 000/year on a 14 MW machinery. The diesel-electric machinery is not as sensitive to fuel quality as the diesel-mechanical; there the reduction potential is only 20%, being still a remarkable amount.

More interesting is the relation between spare part consumption and cylinder size. Some engine makers use such definition as ‘wear rate’ when comparing spare part consumption on different engine types. The empirical equation of wear rate for HFO engine:
number of cylinders x *cylinder diameter* x *mean effective pressure* x *piston speed*
gives a guideline when comparing different engine types. This equation indicates about 20-30% higher maintenance cost for a 320 mm bore engine than for about 400-500 mm bore engine. Therefore, on heavy fuel diesel, it is feasible to select bigger bore engines and low cylinder number which is also normally done. However, the equation does not apply as such for MDO use; the cylinder number and diameter has less impact on maintenance cost and this changes the economical importance from M&R towards engine purchase cost. Price of a 320

mm engine is only 70% of the equal powered 500 mm engine giving thus one totally new cost aspect to the machinery choice.

With the diesel-electric propulsion this difference is even bigger because the smaller bore engines have also higher revolutions giving additional reduction in generator price. The conclusion is clear: the MDO ship should have smaller cylinder diameter engines than the HFO ship, and benefit from the lower first cost of the engine. This means that already at about 15 MW power level the difference in engine-generator price can vary up to 1 M$ even without making ultimate choices or changing make of the engine. The engine cylinder size is clearly dominating the cost factor to be considered in the fuel choice due to extra cost from ancillaries, piping, tanks, heating, etc.

Figure 1-26 shows the cost difference in machinery systems when selecting HFO instead of MDO for a 15 MW machinery. The graph does not include the possibility to use smaller cylinder engines for MDO ship. 29% of the cost comes from the fuel purifying system. The total difference in ancillary systems in this case was $400 000.

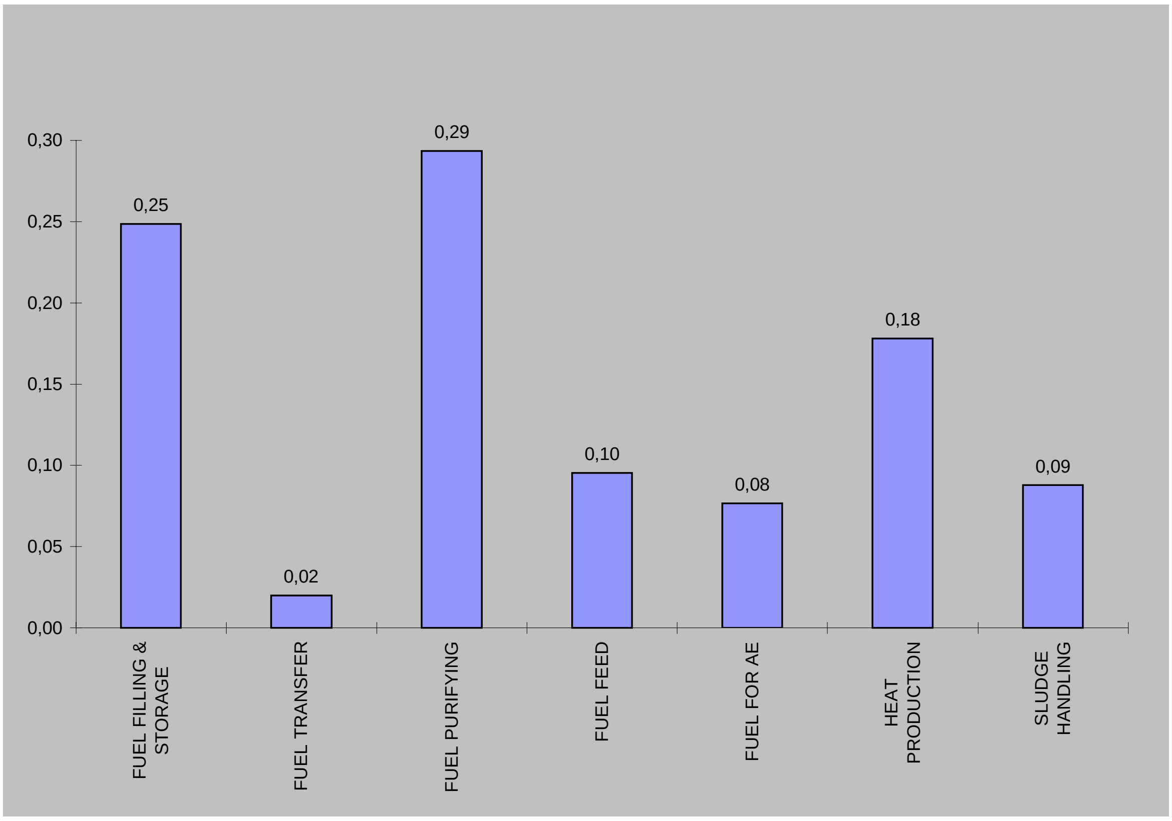


Figure 1-26 Distribution of cost differences when providing 15 MW for heavy fuel capability.

It is possible to carry out a calculation for determining the most feasible fuel quality for certain ship and operation pattern. However, this calls for an effective simulation model which can take into account all the different variables which are depending on fuel type and must be included.
Table 1-6 shows the most important variables in such an evaluation.

Table 1-6 List of the main variables and calculation results to be considered when selecting fuel for a certain operation pattern.

Investment data		**Operation data**	
	Engine type Machinery configuration Fuel filling system Fuel storage system Fuel transfer system Fuel purification Fuel feed Heating system Sludge system		Operation profile Engine efficiency Power demand Fuel analysis & price Lubrication oil analysis & price Emissions Emission fees Heat demand Maintenance and spares
Economy		**Output of the calculation**	
	Interest rate Investment method Calculation period Inflation Taxation		Purchase cost Fuel cost Lubrication cost Emission cost Heating cost Maintenance cost Total economy

By running the above variables in a simulation program it can be estimated, when it is more feasible for a shipowner to choose MDO instead of HFO machinery.

A case study

Following case example gives an idea about what kind of results can be expected. A ro-ro ferry with following power demand is considered:

Mode	**propulsion**	**auxiliary**
at sea	12.500 kW	800 kW
manoeuvring	4.000 kW	1.500 kW
in port	0	1.000 kW

Mode	**at sea**	**manoeuvring**	**in port**
1	2.760 h/a	700 h/a	5.300 h/a
2	5.300 h/a	700 h/a	2.760 h/a

Result of the simulation is shown in figure 1-27. The graph shows the price difference between HFO and MDO at break-even economy condition. Profile 1 = 60% in port, Profile 2 = 60% at sea. Engine type included means that MDO ship has 320 mm bore engines and HFO ship has 480 mm bore engines. In the base case both options have similar engines.

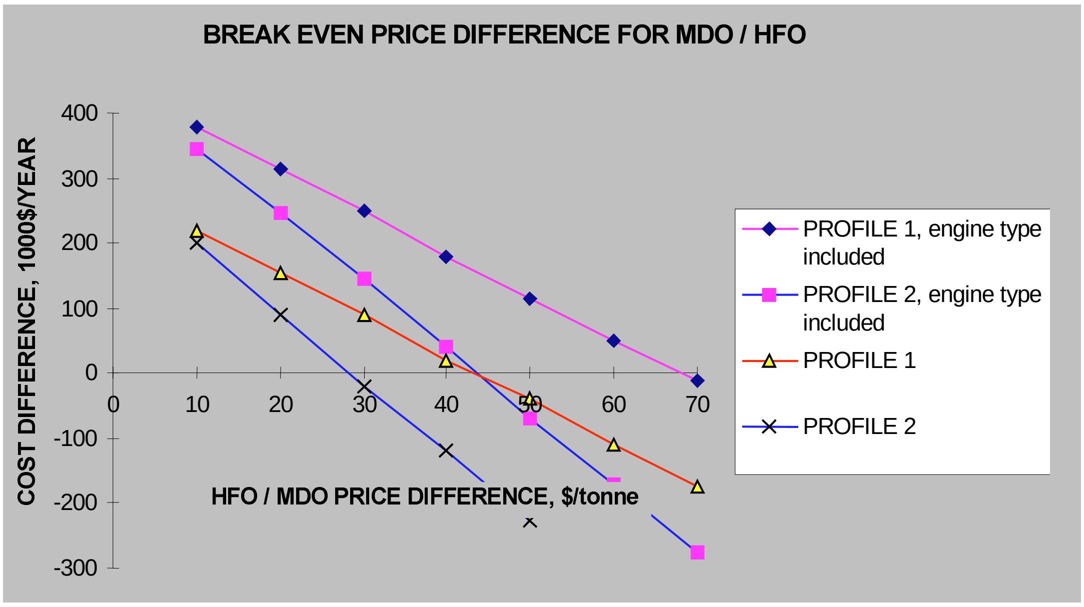


Figure 1-27 Results of fuel quality simulation for a ro-ro passenger ferry.

Running on MDO can be justified by considering the long term price difference between HFO and MDO which has been between $60-80/tonne. However, local specialities in fuel availability and cost structure as well as political and environmental trends must always be checked and valued to higher degree than just the world-wide average figures. In some areas the price difference of Low-S HFO and MDO can be only $ 20-30/tonne.

The result of the case study is quite clear:

- The less the ship operates at sea, the more feasible it is to design the ship for MDO.
- MDO ship should be based on smaller bore engines than HFO ship.

1.3.3 Electric balance

Electric power consumption onboard most of the ship types is continuously increasing. This is mainly due to the general trend towards higher vessel speed, bigger ships, increased amount of electronics and electrically driven systems onboard, as well as improved standards of living. Better heating, ventilation and air conditioning systems and larger illumination installations with special show properties, increased number of reefer receptacles, etc. are typical features of modern ships when compared with just a few years older solutions.

Electric power consumption on a modern 70-80 000 GRT cruise ship is typically from 8 to 9 MW at sea in summer condition, maximum being about 10 MW. Ventilation and air conditioning takes some 60 % and lighting systems 15% of the whole consumption. The rest 2000 kW is shared between machinery, galley, deck, navigation and audio-visual consumers.

Due to the fact that the HVAC group is clearly dominating, the highest attention must be paid to evaluation of this group, especially when calculating electric balance for a cruise ship. When cooling power demand is determined, the corresponding electric power demand can be estimated. Electric power for compressors is about 20% of cooling power and the compressors correspond to 50-60% of the total electric power demand of the air conditioning plant. A year-around loading of cooling compressors on a Caribbean cruise ship is 60-70% of plant rated efficiency. In more detailed calculations attention must be paid on selected system, especially concerning:

- heat recovery configuration
- possible use of fan coil units
- actual coefficient of performance (COP) and number of the compressors
- actual operation point of the compressors

Decisive load in respect of generator station rating is typically:

- HVAC and side thrusters on cruise ships
- reefer sockets on container ships
- side thrusters on ro-ro ferries
- cargo handling on tankers
- main engine ancillaries on other ship types.

Figures 1-28 and 1-29 show typical electric consumption share for three different types of ships at sea and at manoeuvring.

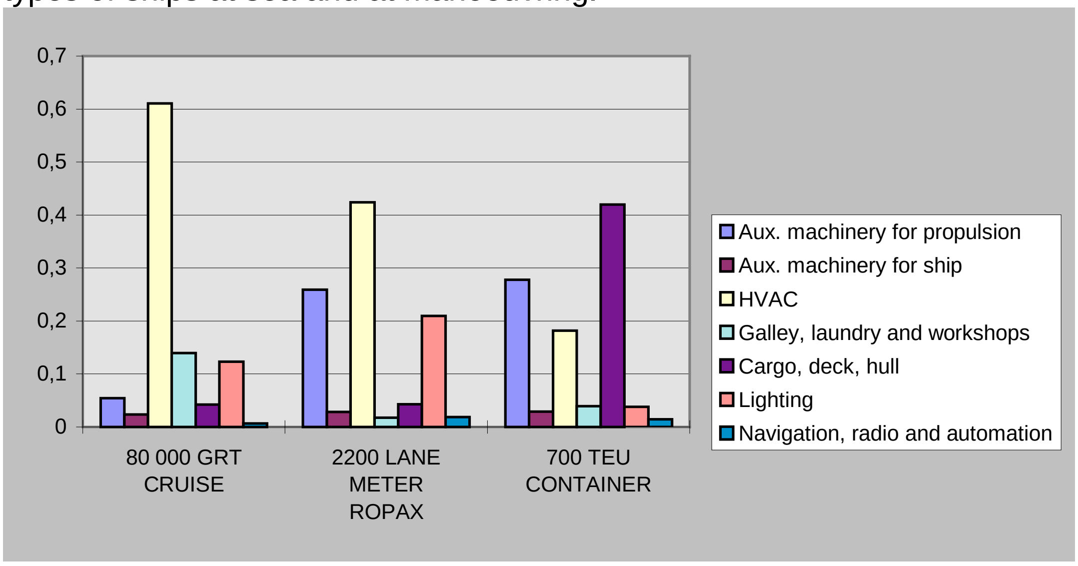


Figure 1-28 Share of electric consumption in different groups on three vessel types, at sea condition. Typical total consumption at sea is 8000 kW for cruise ship, 1400 kW for ro-pax and 800 kW for container vessel.

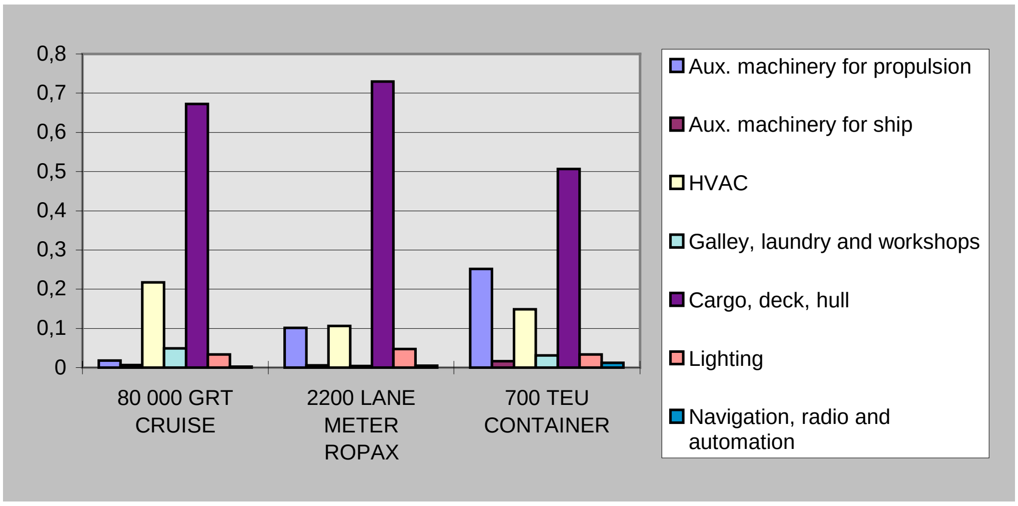


Figure 1-29 Share of electric consumption in different groups on three vessel types, manoeuvring condition. Typical total consumption is 22 000 kW for cruise vessel, 6400 kW for ro-pax and 1000 kW for container vessel.

Thrusters

Special attention has to be paid to starting of large consumers. Modern ships have improved manoeuvring properties and thus also powerful side thrusters. There the starting may become a problem, especially in upgrading cases. Stardelta and autotransformer starting is commonly used but the softest method would be to apply inverter starting. This method has several benefits over other methods due to high, constant power factor:

- Small cables
- Lower short circuit level (especially with stern thruster installations)
- Lower installed generator power
- Lower installed engine power

When all this is included, a FPP thruster at 1-2 MW level with inverter control can be a less expensive solution than the conventional CPP thruster and Stardelta starter.

Emergency generator

Adaptation of the highly efficient water mist based extinguishing systems have increased the installed emergency generator power. Fire fighting is today clearly the decisive consumer in generator dimensioning. Automatic fire extinguishing systems can alone require 150 kW of electric motor power on a modern ro-pax ferry with high pressure sprinkler arrangement.

Adaptation of twin emergency generators would increase the overall availability of emergency power. One unit could be nominated as emergency generator and the other as emergency/auxiliary generator. This would allow the use of this latter generator more frequently and assure that the engine is really working when needed. This is also an arrangement found in most recent cruise vessels.

1.3.4 Power availability

Reliability technology is commonly used in space and nuclear technology. Reliability, safety and control of operating costs have become important also in shipping industry. Reliability technology offers a wide range of tools for analysing, and developing ships to be more safe and economical.

Methods

The most commonly used methods are:

- Reliability block diagram analysis
- Failure mode and effect analysis (FMEA)
- Operability study
- Availability analysis
- Fault tree analysis (FTA)

The nature of these first three methods is qualitative. Availability analyses and FTA can also be carried out as quantitative.

Reliability block diagram analysis

Reliability block diagram of system shows clearly critical system components' relation to each other. The principle idea of the diagram is that all consecutive components must function and one of the parallel components or component lines must function, so that the system can carry out its mission.

Redundancy rate of different components and component groups are quickly seen from the diagram. In few simple systems the diagram is identical with system diagrams, but in more complex systems the same components will appear in many places in the same diagram. Figure 1-30 shows an example of sea water cooling system diagrams for pump station and corresponding reliability block diagram, describing reliability connections of the pump station.

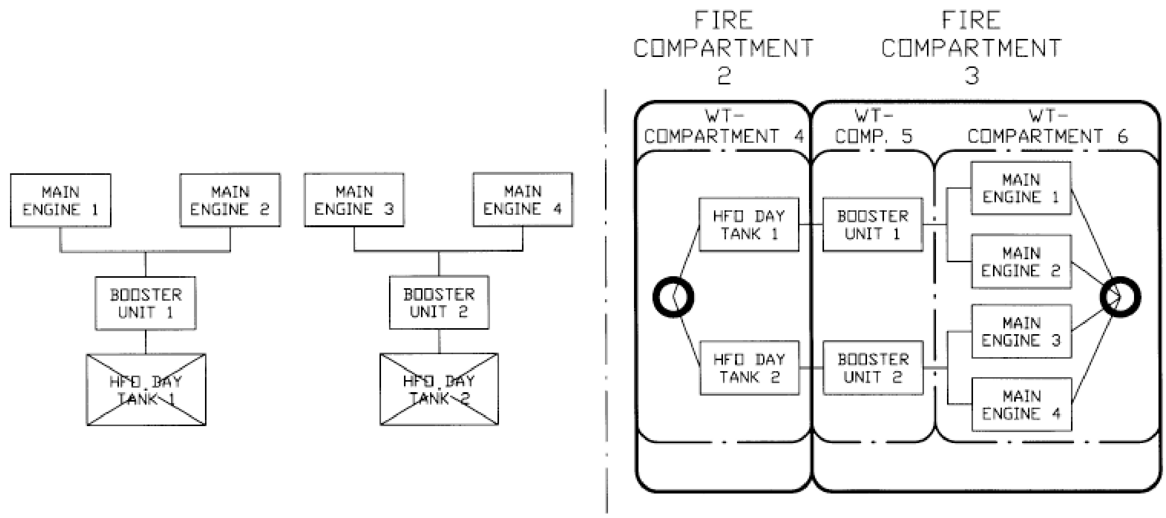


Figure 1-30 Block diagram of pump station. Two of three pump lines are needed for sufficient cooling water flow. In case of leakage of the non-return valve in the stand-by pump line the flow will be insufficient.

Failure mode and effect analysis (FMEA)

FMEA is a systematic approach for finding out system components' failure modes, theirs effects on system level and corrective actions needed for preventing component failure from causing system failing.

There are many different types of FMEA tables standardised. For example IEC-812 (International Electrotechnical Commission) or SFS-5438. On basis of these a specific table form has been developed suitable for ship configurations.

Analysis can be used for:

- finding out failure modes and mechanism for components and systems
- defining systems requiring improvement
- defining failure-finding instructions.

Operability study

Operability study is originally developed for evaluating safety in chemical process plants. The basic idea of operability study is that it assumes problems in systems not occurring until operating parameter changes (temperature, pressure, concentration etc.).

The study will be made in table form. Table 1-7 shows part of the investigation made from the system in figure 1-30.

Table 1-7 Example of operability study for seawater cooling system as presented in figure 1-30.

Vessel:	Delta Ship	Made by:	JL
Auxiliary system:	Sea water cooling system	Checked by:	
Pipe line:	Cross over tk.-line 4		

Operability study

NO:	KEY WORD	DEVIATION	PROBABLE REASON	CONSEQUENCES	NEEDED ACTIONS ONBOARD	PROPOSALS
1	Less	Less pressure	Loss of pressure of pump 1 and non-return valve in line 1 will leak to back direction	Insufficient flow to central cooler, LT water temperature will increase, automatic slow down of engine	Closing of shut-off valve in line 2	Pumps and non-return valves in pump station must be tested regularly
2			Loss of pressure of pump 2 and non-return valve in line 2 will leak to back direction	Insufficient flow to central cooler, LT water temperature will increase, automatic slow down of engine	Closing of shut-off valve in line 2	as 1

Advantage of operability study compared to FMEA analysis is that it makes it possible to investigate consequences of fault combinations.

The operability study is a good tool for qualitative reliability evaluation of engine auxiliary systems. If the reliability analysis includes also a quantitative part the FMEA is more convenient for this purpose, because in that case RAM (reliability, availability) data for single component failures must be determined. The quantitative analysis will be made by building a logical fault tree of system, in other words fault tree.

Fault tree analysis (FTA)

Fault tree is generally used as a reliability modelling and observation method. FTA means analytical investigation of system, process, or subsystem, so that such failures or failure combinations will be find out which are causing a remarkable system or process failure directly or indirectly. The target of the analysis is to find out component failure effect to the appearing of system or process failure.

The FTA will be carried out by drafting a logical diagram (fault tree) from the system which shows how the component failures are causing the system failure. FTA analysis shows reliability connections of a system and is thus a reliability model of the system.

FTA is also suitable for determining fault combinations, unlike FMEA. It also makes it possible to investigate human errors that could cause the appearing of system failure.

FTA is a top-down analysis, where the building procedure begins from the top event and continues by the help of logical thinking to lower levels. Fault tree is a logical diagram, where the weakest parts of the system can easily be found.

FTA can be used for:
- Recognising weak or remarkable parts of a system.
- Describing the reliability model of the system.
- Evaluation of control and safety systems sufficiency.
- Making the failure localising easier and by this way the maintenance actions quicker.
- Evaluation of the effect of human and software errors on the reliability of system.
- Charting the actions that can be used for system repairing.
- Describing system's reliability model.
- Quantitative determining of a system reliability.

Figure 1-31 shows a fault tree describing sea water cooling pump's reliability model. The top event of the tree is chosen to be "Pump will not start to produce output, which will cause stopping of engine".

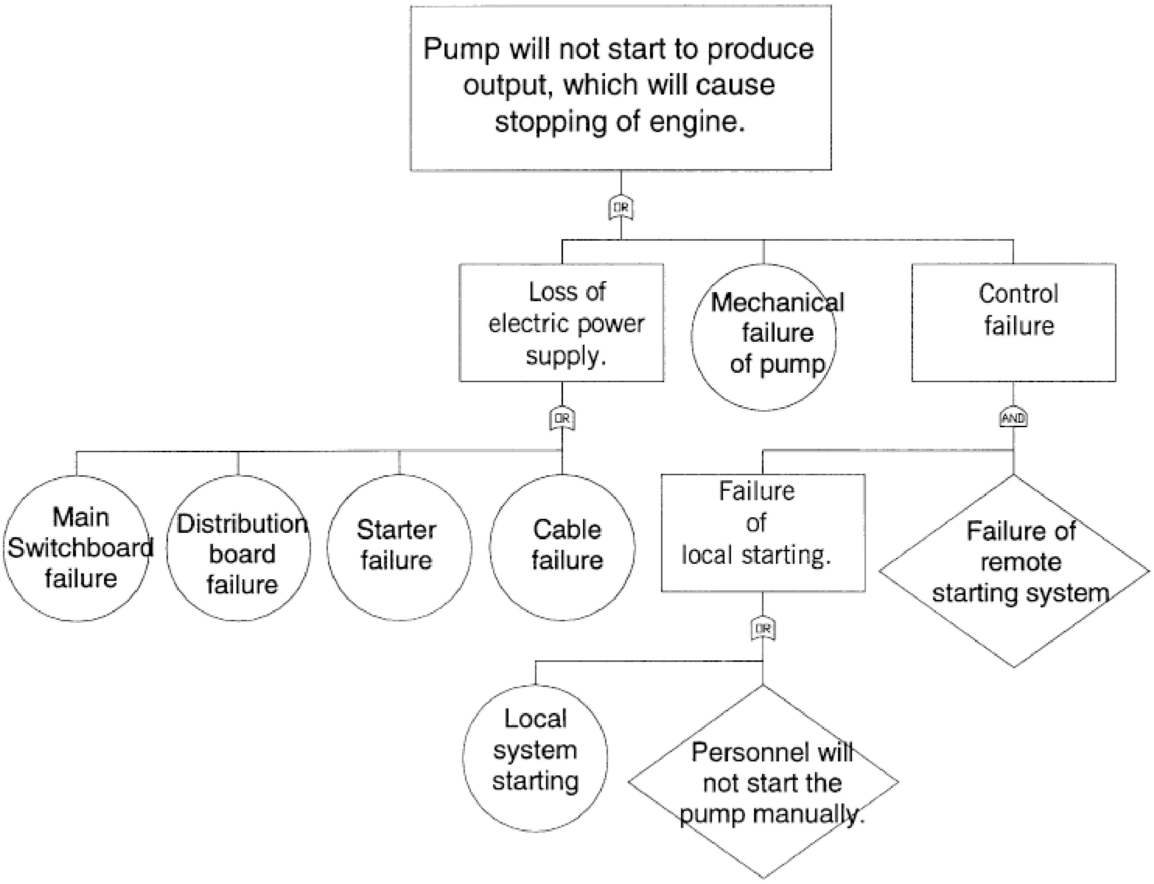


Figure 1-31 Fault tree of a pump.

Fault trees include different kinds of logic symbols. Rectangles are fault events caused by basic events. Circles are basic events, i.e. RAM data must be determined in case of quantitative analysis. Diamonds are events not developed to its cause (not significant or outside of boundary). Output through "AND" gate exists if all input events exist and output through "OR" gate exists if any of input events exist.

FTA can be completed by quantitative analysis if it is necessary and RAM data for basic events exist.

Examples

Reliability and availability analysis can be utilised already from the first studies for a new vessel configuration, to compare different machinery concepts, power availability versus costs.

The next step is to compare average spare part costs for different machinery concepts as well as to find out the number of annual unexpected failures and related costs.

The reliability level of the machinery concept can be found out and decision can be made if action has to be taken for increasing the reliability and availability of machinery, and this way the safety onboard and total economy of the vessel can be increased.

The weak parts of a system can be recognised and decision can be made if more reliable components or even higher redundancy level of machinery are needed when considering safety and economical aspects.

Figure 1-32 shows meantime between any failure of propulsion machinery requesting repair for a LNG tanker study. Three different machinery concepts are included: diesel-electric propulsion machinery with two pods, diesel-electric machinery with reduction gear and single shaft line, and slow speed engine machinery.

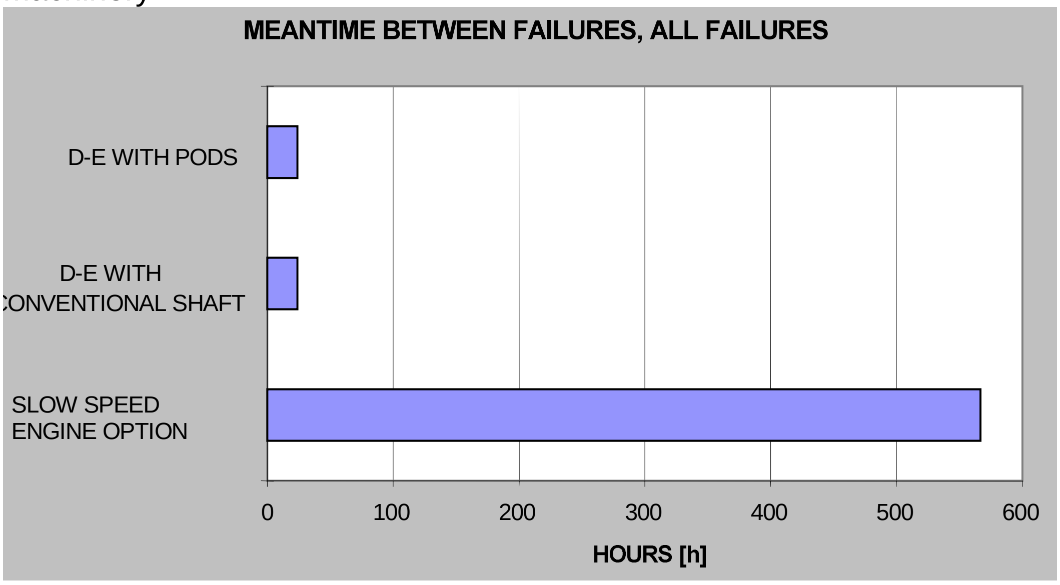


Figure 1-32 Meantime between failures in different options, all failures requesting some kind of repair.

The magnitude difference between slow speed engine option and diesel-electric options is caused by multiple number of components in diesel-electric machinery compared to slow speed engine machinery.

Figure 1-33 shows results when only critical failures are included, i.e. when the vessel has no propulsion left.

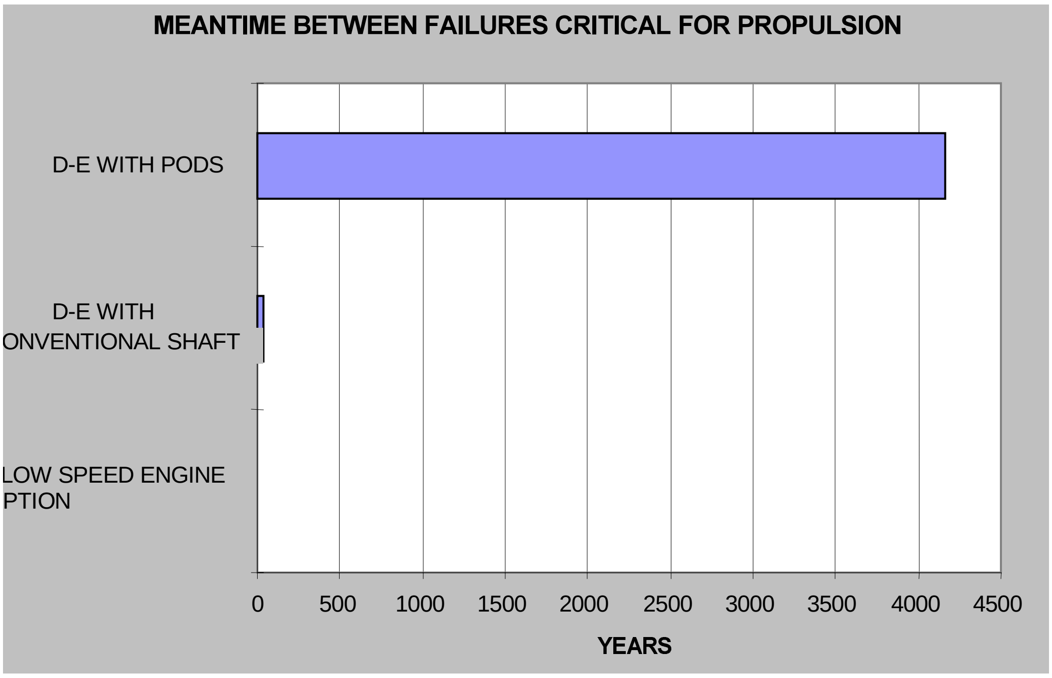


Figure 1-33 Meantime between failures in different options, only critical failures included.

The effect of redundancy can be seen from the results. All components in diesel-electric option with two pods are redundant. Diesel-electric option with single shaftline includes many non-redundant components and therefore the meantime between failures is calculated to be about 40 years, where most of the failures are caused by the reduction gear. The slow speed engine machinery is almost completely non-redundant and therefore the meantime between critical failures is calculated to be only about half a year.

The RAM analysis also gives a tool for finding out components which are causing most of the propulsion system failures. Figure 1-34 shows an analysis carried out in a vessel equipped with a slow speed engine. The diagram shows percentage share of component failures leading to situation where the vessel has no propulsion.

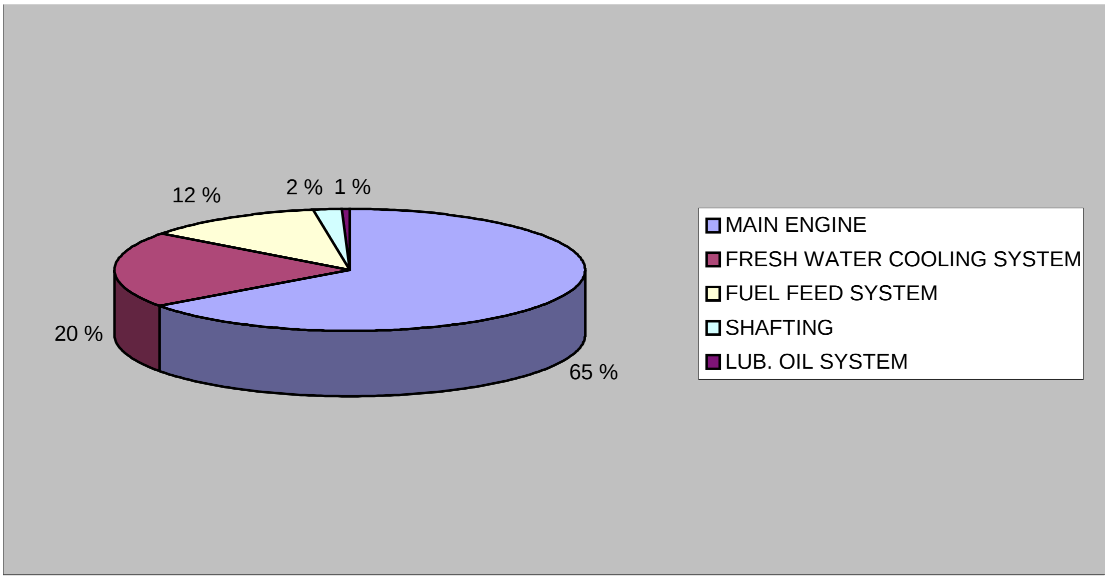


Figure 1-34 Percentage share of reasons causing a standstill situation in a slow speed engine vessel.

RAM analysis gives a tool for finding out the components or systems with which it is worth increasing reliability in comparison with costs required. The analysis works also the other way: ‘unnecessary’ additional equipment and system can be detected.

1.3.5 Ancillary services

This chapter consists of a list of guidelines for designing ship and machinery ancillary services. The values and rules of thumb are well applicable not only at project design stage but also during basic design when actual components have not yet been chosen.

Auxiliary oil fired boiler

- Steam production ability
  Saturated steam, 7bar g, 170°C.
  1 kW corresponds to about 1,6kg/h steam or
  1 MW corresponds to about 0,42kg/s steam

- Fuel oil (FO) consumption
  1,0kW corresponds to 0,105kg of HFO/h
  1,0kg/h steam corresponds to 0,066kg of HFO/h

Fuel oil systems

- Main engine (ME) and auxiliary engine (AE) fuel oil consumption
  Normally engine suppliers give the specific fuel oil consumption (SFOC) based on ISO 3046/1 standard, table 1-8.

Table 1-8 Typical values of SFOC.

CYLINDER DIAMETER (mm)	SFOC (kg/kWh) according to ISO 3046/1	SFOC (kg/Kwh) With tolerances (typically +/- 3-5 %)	SFOC (kg/kWh) With engine driven pumps (typically 1-3g/kWh)	SFOC (kg/kWh) with HFO 40200 kJ/kg, with Tolerances, with engine dr. pumps
300 – 380	186	193	195	207
400 – 480	182	189	191	203
500 -	178	185	187	199

- Emergency diesel generator
  SFOC: typical value 0,25kg/kWh
  100 kW of power means about 28-litre fuel oil consumption per hour.
  FO tank to be dimensioned at least for 36 h constant running according to SOLAS.

- Heavy Fuel Oil (HFO) Tanks

  Storage tanks

  Minimum temperature in storage tanks depends on the pour point of the HFO. The temperature of HFO should always be kept higher than pour point to avoid filter blocking, and other similar problems.

  If HFO 380 cSt/50°C is used; the temperature in storage tanks should be about 40-45°C to reach 600cSt good pumping viscosity. Pour point for HFO 380cSt/50°C is 30°C.

  Settling tanks

  To allow reasonable separation the tank should be sized for 24h consumption.
  The settling temperature to be calculated to be about 70^{O}C.
  According to the latest SOLAS rules double settling tanks are needed.

  Service tanks

  Service tanks should be sized for 10h - 12h consumption.
  The temperature in service tanks to be calculated to be 75°C.
  According to the latest SOLAS rules two separate service tanks are needed.

FO heaters
To avoid thermal cracking of FO a thermostatic control is to be provided.
Temperature in steam heaters should be below 170°C.
The electric heater power loading should be limited to 1,0W/cm^2.

Lubrication oil systems

- Lubrication oil consumption
  LO consumption for medium speed engines in average is about 1,0g/kWh.

- System oil tanks
  In case ME's are so-called dry sump engines, there should be a system oil tank on the double bottom.

Heat balance

To define the required heating capacity heat balance should be made.
Typical summary for heat consumers on a ro-ro passenger ferry is presented in table 1-9, total output of main engines 4 x 4140kW.

Table 1-9 Summary of all heat consumers on a ro-ro passenger ferry.

	pcs.	Total heat KW	F	H – U KW	f	N. O. KW
HFO tanks	5	307,9		206,0		101,9
BILGE	1	15,93		3,30		12,63
ME LO SUMP	4	36,03	1	14,51	0	0,00
OVERFLOW	1	20,35	0,75	15,26	0,75	27,02
SLUDGE	1	13,31	1	13,31	0,75	9,98
DIRTY OIL	1	12,00	0,8	9,60	0,7	8,40
FO DRAIN	1	6,31	0,75	4,73	0,75	4,73
HFO separator heater	2	120,0	0,25	30,0	0,5	60,0
ME LO separator heater	4	112,0	0,25	28,0	0,5	56,0
ME HFO heater	2	130,0	0	0,0	0,85	110,5
ME FW preheater	2	0,0		0,0		0,0
AE HFO heater	1	27,0	0,85	23,0	0,85	23,0
AE FW preheater	1	0,0		0,0		0,0
AC heater	2	1744,0	0,375	654,0	0,45	784,8
Domestic water heater	2	170,0	0,75	127,5	0,75	127,5
AE LO separator heater	1	9,0	0,85	7,7	0,85	7,7
Pipe tracing		25,0	0,9	22,5	0,9	22,5
Boiler HFO heater	1	7,00	1	7,0	0	0,0
TOTAL		2755,8		1166,4		1356,5

Notes: 1. f loading factor
2. H - U Heating Up Energy
3. N. O. Normal Operation i.e. heat energy consumption during sailing

Cooling system heat flow calculation

To define the needed cooling capacity heat flow calculation should be made. Typical summary of cooling system heat flow calculations for the same ro-ro passenger ferry as in the previous example is presented in table 1-10.

Table 1-10 Summary of cooling system heat flow calculation

MAIN ENGINE COOLING SYSTEM	Heat(kW)
HT-circuit	1382
LT-circuit	592
LO cooler	479
MAIN ENGINE TOTAL	2450

PROPULSION AUXILIARIES COOLING SYSTEM	Heat(kW)
Reduction gear	1x108
Hydraulic unit for CP- propeller	1x20
Shaft bearing	3x10
PROP. AUXIL. TOTAL	158

TOTAL HEAT FLOW	Heat(kW)
Main engine total	4900
Propulsion auxiliaries total	158
TOTAL	5058

PROPULSION MACHINERY CENTRAL COOLER	Heat(kW)
Demand	5058
SELECTED CAPACITY	6120

One central cooler for 100% heat rejection
Reserve capacity to be 10 %.
Central coolers fouling factor to be 10 %.

Sewage systems

It is highly recommendable to specify a biological sewage treatment plant for all types of ships because of the environmental reasons. The plant is typically dimensioned to treat full black water load.

Galley waste water is normally not led to sewage treatment plant, because it slows down the biological process.

Grey waters have been discharged directly overboard or collected to grey water storage tanks. Some times grey waters have been chlorinated before discharging overboard but not really biologically or chemically treated onboard.

The amount of the produced sewage onboard depends on several parameters, such as type and route of the vessel, but some rough guidelines can be given to estimation purposes.

Grey water system

It can be estimated that in ro-ro passenger ferries grey water production is approximately from 150 to 200 l/person/day.

In cruise ships grey water production can be estimated to be over 200 l/person/day, the more accurate grey water load to be calculated taking into account particular project demands.

Table 1-11 Grey water production

	Water l/day/person
Showers / wash basins	100 – 150
Laundry	50 – 60
Galley	40 - 50
Other consumers	10 - 50
Total	200 - 300

Table 1-12 Black water production

	Water l/day/person
Vacuum toilets 1,5 l / flush	7,5 – 10,5
Gravity toilets 5-8 l / flush	25 – 56
Hospital drains 200 l / day	-

1.4 STRUCTURES AND WEIGHT

Basis for steel structures

When a new ship project is started the first thing should be the idea of the capacities, functions and general arrangement of the vessel. The development of a new project follows principles below:

- utilising reference vessel
- utilising literature and studies
- utilising own similar designs and studies
- combination of the above

When the basis is settled rough space reservations can be done (1st stage project general arrangement).

In a very early stage of the project the spacing of the frames in longitudinal direction and in transversal direction should be selected (based on experience or reference vessel) and the general arrangement drawing should be adjusted accordingly and vice versa.

Following basic steel construction systems have to be created:

- Framing system (longitudinal, transversal, mixed, i.e. longitudinal and transversal framing system is mixed according to structures; e.g. decks and bottom are framed longitudinally and shell and bulkhead transversally or vertically).
- Web frame spacing system, normally n times frame spacing
- Supporting structure system, i.e. pillars and supporting bulkheads, normally n times web frame spacing.
- Longitudinal girder system, normally n times longitudinal spacing.
- Watertight bulkhead system, n times web frame spacing; distance between watertight bulkhead to be more than extent of damage in passenger vessels and less than floodable length in cargo vessels, anyhow n times web frame spacing.
- Longitudinal bulkheads according to deck longitudinal system.
- Fire divisioning bulkheads according to watertight bulkhead system.

When the above mentioned items are defined then the first General Arrangement drawing can be issued which means that arrangement and structure should always be coordinated right from the first project idea.

Specialties of passenger and ro-ro passenger vessels

In passenger vessels the frame spacing (actually the web frame spacing) should be selected according to cabin modules, i.e. cabin width being multiple of frame spacing.

The next step should be the definition of watertight bulkheads. The distance between watertight bulkheads is to be between minimum extent of damage and nearest multiple of web frame spacing.

The selection of spacing of longitudinal should be according to cabin length. The beam of the vessel should be divided into fractions and supporting pillars should be located between cabins and/or close to cabin corridors. The recommended distance between pillars is about maximum 7 metres.

The length (i.e. web frame spacing) and spacing of longitudinals should be equal because of the local vibration behaviour. The natural frequency of the structures is highly depending on the span of the structures, and if the span and spacing varies a lot it is almost impossible to tune the natural frequency of the deck panels between allowable range of 1st and 2nd blade passing frequency of the propeller.

In cases where supporting pillars cannot be fitted, the supporting arrangement can be for example hanging structure. The typical examples are large public spaces, theatres, show lounges and ro-ro decks where pillars are not accepted. The best way to support this kind of areas is to have them hanging from above, to have one-two decks high supporting structure above with continuous longitudinal and transverse bulkheads.

When the above mentioned is fulfilled, the steel general arrangement drawing can finally be created. This drawing is necessary for steel designers, and a management tool for architect and other design disciplines.

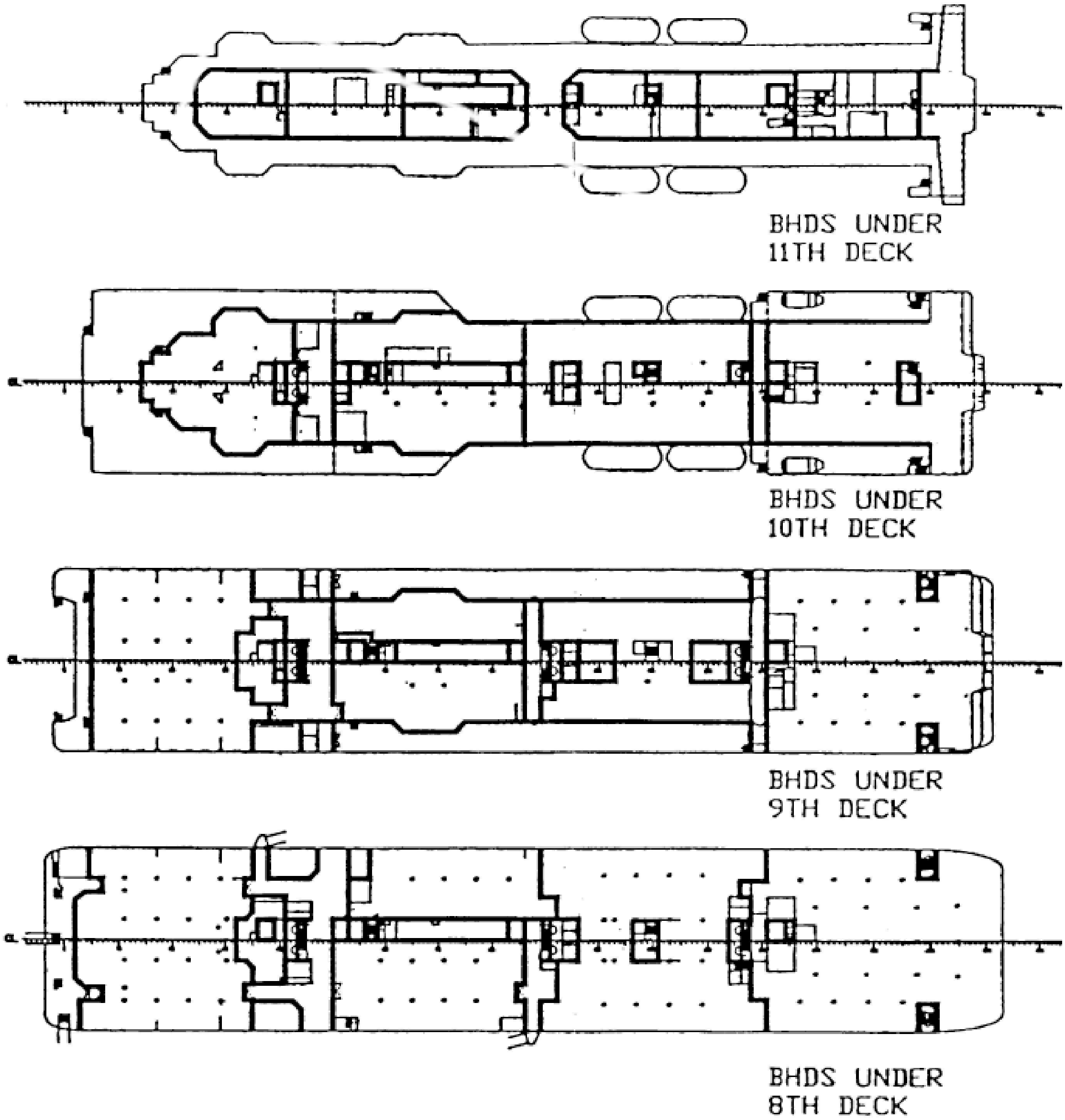


Figure 1-35 Steel general arrangement

Cost optimisation, weight and number of pieces

It is generally accepted that by minimising the weight, the costs are minimised. In general this is true concerning material costs, but not building costs.
The building costs are highly depending on the type of structures, degree of automation and number of pieces.

The type of structures should always be as simple, standard and with as good continuity as possible. Good general arrangement allows to make simple vessels, and simple vessels are light in weight.

Structures should be designed to support the shipyard's production facilities, e.g. if the shipyard has welding automates, design should fit to them.

Number of pieces is a very important subject. The steel weight of the vessels is mainly formed of steel plates, girders and stiffeners. The fraction of steel weight is formed of small pieces, such as brackets, lugs and other small pieces. However, these small pieces can be up to 45 % of all pieces per gross section but only a few percentages of the total steel weight. The chart in figure 1-36 shows one example of the distribution of pieces. As can be seen the number of small pieces (i.e. brackets, small fat bars, lugs, etc.) is remarkable. It should also be remembered that fitting of small pieces is manual work. To reduce the number of small pieces gives potential for building cost reduction.

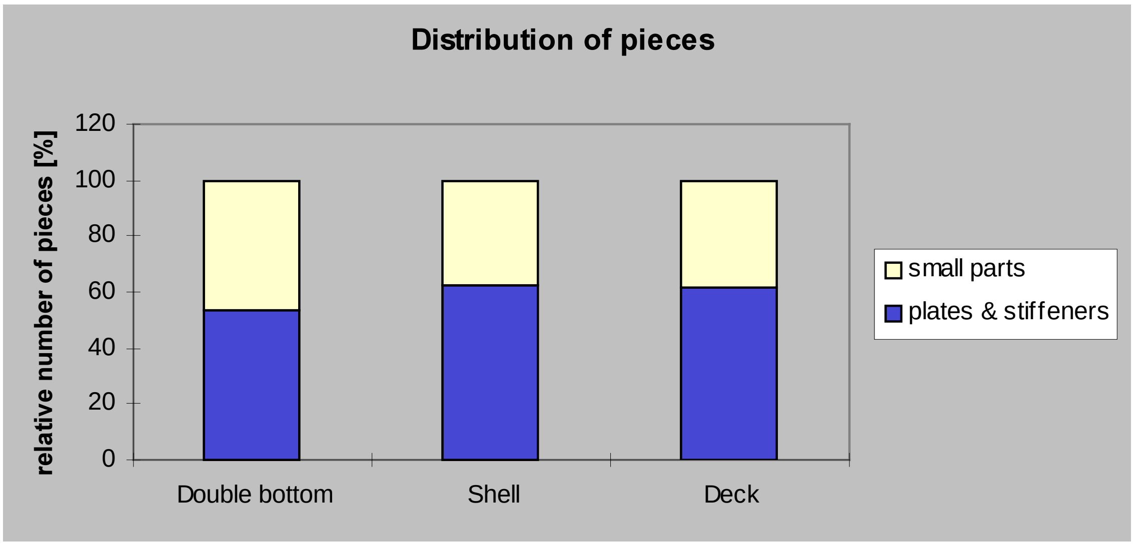


Figure 1-36 Distribution of pieces

The importance of small structural details can be recognised, see figure 1-37. This figure shows a comparison between midship sections of recently built about 70.000 grt cruise vessels, number of pieces and weight/gross section. In principle all vessels in the chart are equal in size, they have similar framing system and similar web frame system. However, the differences in weight and number of pieces are significant between each other. A noticeable thing is that number of pieces and weight do not correlate as normally expected: less pieces and high weight, many pieces and low weight. This study does not answer the question which structure is most advantageous, but the results are relative ones. The weight is formed of plating and girder and stiffening system while the number of pieces is formed of small structural details. It shows, however, clearly that the challenge exists to reduce both weight and number of pieces, and possibilities for that exist.

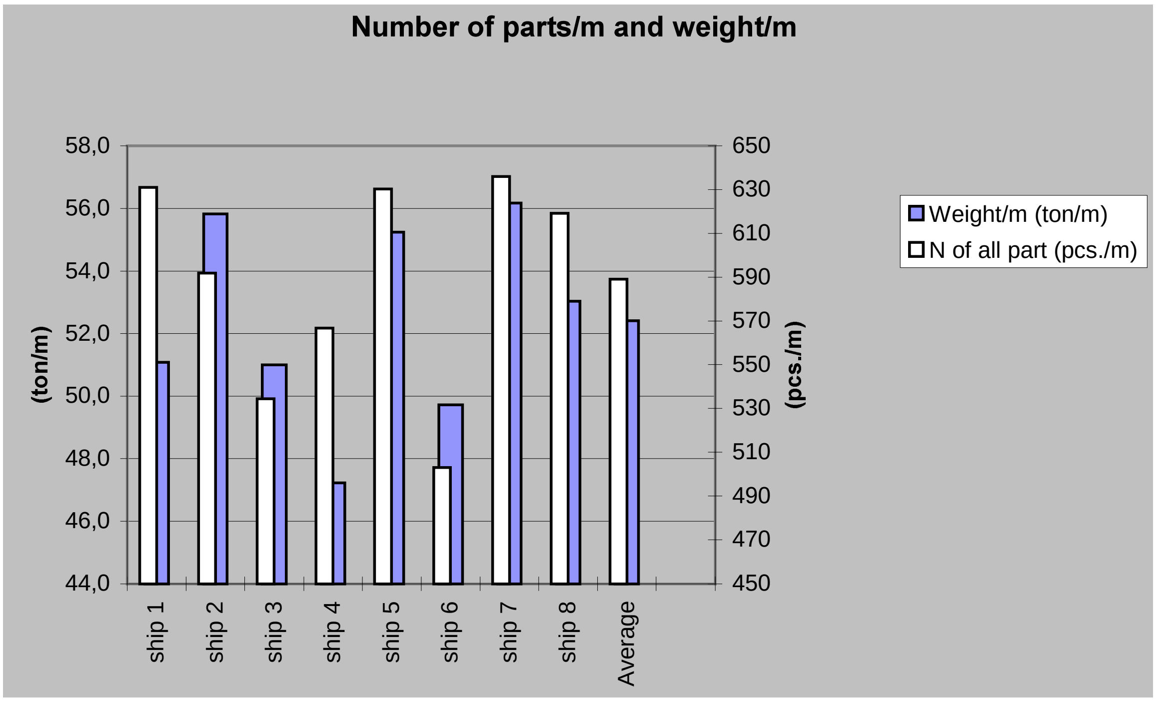


Figure 1-37 Number of parts/m and weight/m in cruise vessels of about 70.000 grt.

Weight control

The weight calculation and weight control system should be based on specification numbering system, e.g. as follows: steel, hull outfitting, interior, HVAC, machinery, auxiliary machinery and electrical.

The first weight estimates are based on the general arrangement drawing, midship section drawing and ship's specification. Weight estimation is based on direct calculation of areas, volumes and statistical figures of previous projects.

The second step in weight calculation is basic design. In this phase the weights are based on more accurate direct calculations (steel weight based on classification drawing, etc.), purchased equipment and statistics where more detailed data is not available.

The third step in direct weight calculation is based on workshop drawings. The steel weight is based on block drawings, etc.

The fourth step is actual weight control during the building period based on weighing.

Schematical weight control procedure is shown in figure 1-38.

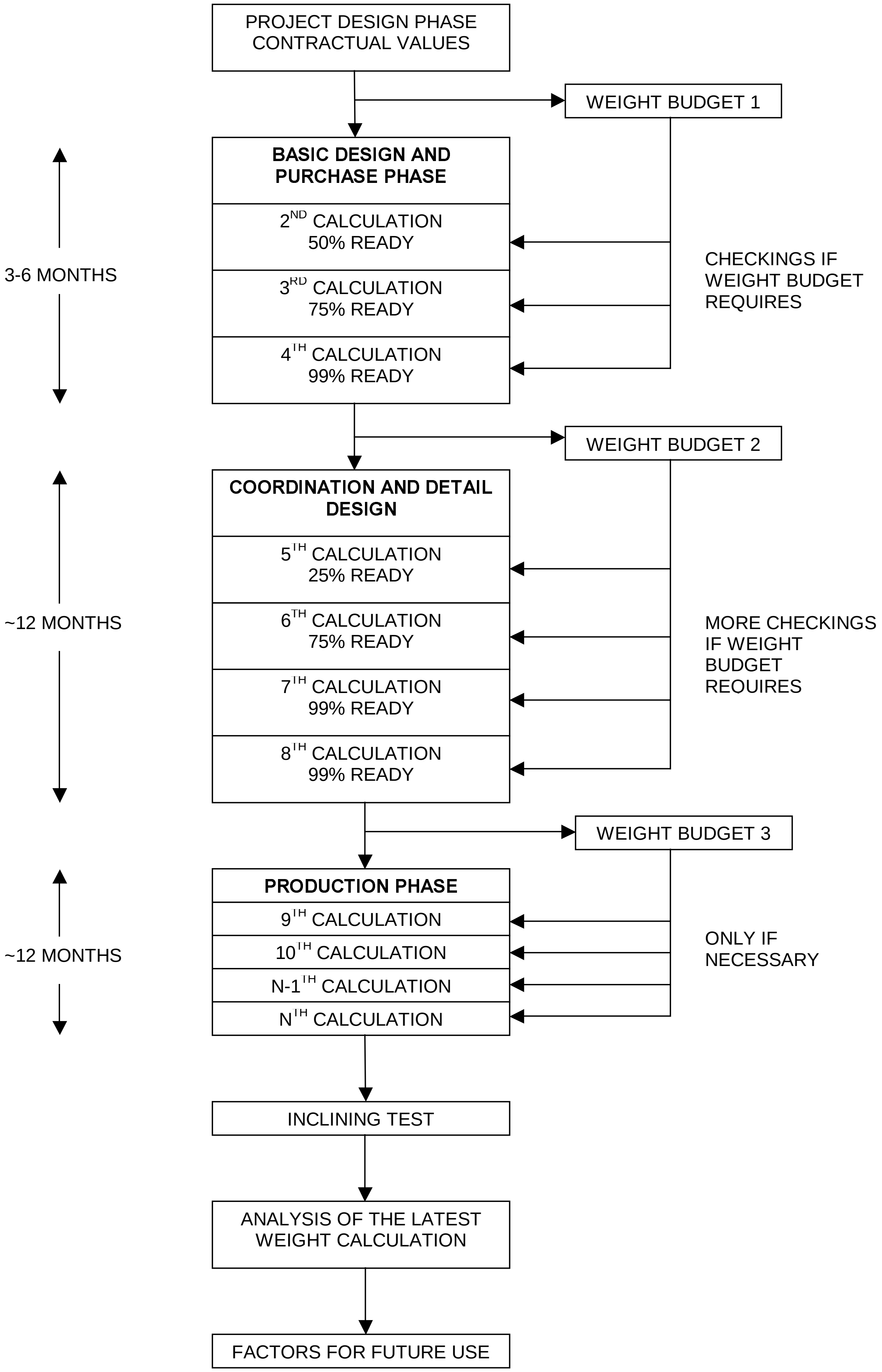


Figure 1-38 Description of the Weight Control Procedure

Weight statistics

The project stage weight calculations should be based on weight statistics. It is impossible to estimate the weight of a project ship accurately enough without good statistics in the project stage. It is also a good tool to check more detailed project calculations.

The most important statistical weight figures are:
- Lightweight versus total volume
- Centre of gravity/centre of volume versus total volume

The statistical weight figures are, of course, highly depending on ship type. Examples of weight statistics are shown in figures 1-39 and 1-40.

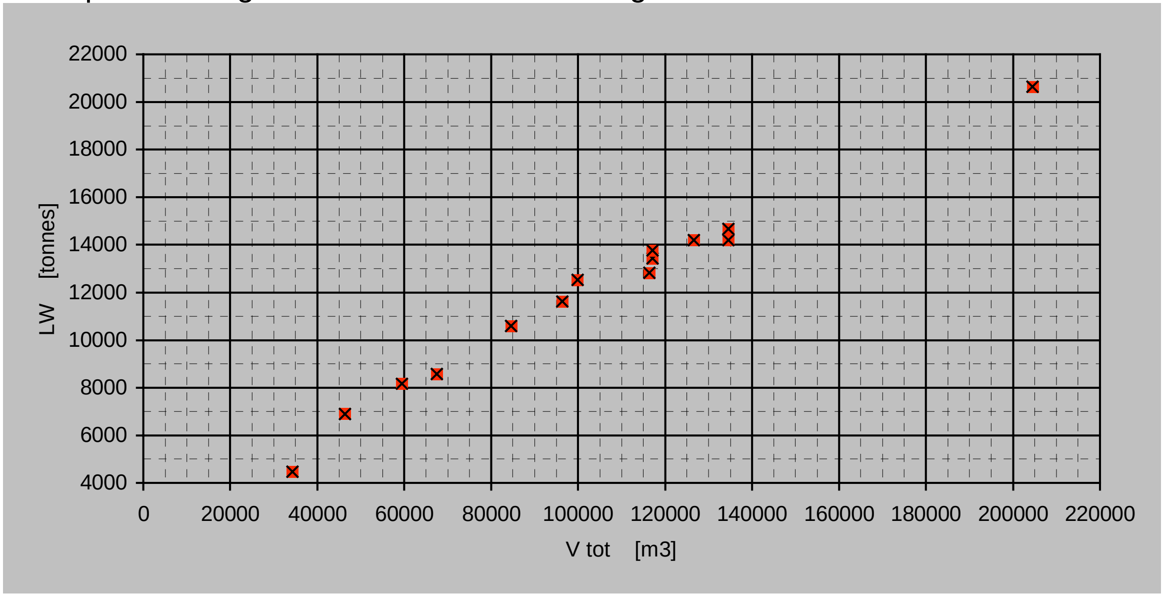


Figure 1-39 Lightweight versus total volume

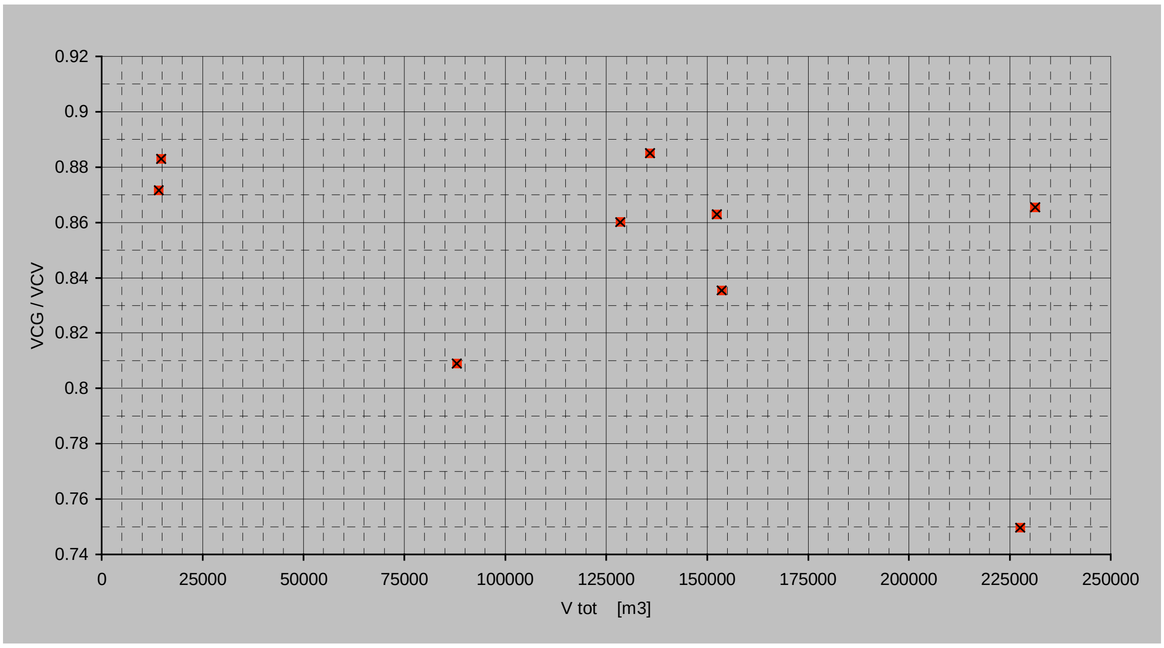


Figure 1-40 Centre of gravity/centre of volume versus total volume

1.5 DAMAGE SAFETY

Damage safety, especially for ro-ro passenger ferries, is discussed in this chapter.

The design approach in the ro-ro passenger ship market is rather contradictory. The use of lower cargo hold under freeboard deck has several different interpretations, not only for damage stability but for other safety features as well. Side casings versus centre casing is another subject very much debated and affecting the overall design of the vessel.

The philosophy for attaining improved safety is based on three basic assumptions. First of all the ship should be easy to operate, and loading should be done rationally and with the best stability in mind. Secondly, the configuration, arrangement, itself should provide a better damage safety. Finally, the design and shape of the hull should eliminate the possibility of unsafe operation in harsh weather, thus increasing safety onboard.

Shortly after the disaster of the ‘Herald of Free Enterprise’ it became obvious for the ferry industry not to locate passenger cabins below freeboard deck anymore. The volume below the main ro-ro deck is large and leaving it as void would be waste of valuable space. It could be used for cargo and/or for storage. The only efficient way to utilise this space is to use it for cargo, to apply the so-called lower hold configuration. The hold is typically limited by longitudinal bulkheads inside B/5 line and with deck above B/10 line, B being the beam of the ferry. The damage stability for the first modern lower hold ro-pax vessels was simply calculated disregarding the lower hold from any damage cases and the floodable lengths were calculated applying the principle of equivalent bulkheads within the hold area applying the ‘B/5 rule’. Another option was to use the A265 method specifically intended for ferries with longitudinal subdivision.

Later investigations proved that the B/5 limit was not an adequate limit for collision damages. The B/5 limit still exists in the SOLAS but most of the authorities have difficulties in accepting it as physical limit for damage cases.

The A265 method was tested in a few ro-ro passenger ferries but it was soon understood that it was clearly limiting the maximum number of passengers and transverse bulkheads may be required in the lower hold destroying the cargo flow.

It was necessary to have a wider approach to the lower hold configuration. SOLAS 90 gave a starting ground. The lower cargo hold in ro-ro passenger ferry ‘Normandie’ of Brittany Ferries, delivered in 1992, was still limited with bulkheads inside B/5 and deck above B/10, but both damage cases and floodable lengths were considered with damages extended into the lower hold. Two compartment damage cases (without lower hold) were calculated in accordance with SOLAS 90 criteria. And two compartment damage cases together with the lower hold were calculated in accordance with the SOLAS 90 intermediate stage criteria (the intermediate stage criteria were applied as criteria for the final flooding stage also). The floodable lengths were compensated with the direct damage stability calculations including lower hold. This approach has been used since ‘Normandie’ for several newbuildings.

The obvious further step was to check if it would be possible to fulfil even full SOLAS 90 criteria with all two compartment damages together with the lower hold. This principle is now being applied in some of the most recent ro-ro passenger ferry newbuildings.

It is possible to construct a modern, efficient ro-ro passenger ship without using doubtful limitations for the damage definitions. There seem to be, however, at least six different interpretations on the market for lower hold vessels and thus the probabilistic method is really welcome for the industry to clarify the situation.

Probabilistic method, impact on designs

We participated in the Joint Northwest European project “Safety of Passenger/RoRo Vessels”. The project focused on the development of a new safety standard for new passenger/ro-ro designs with particular focus on stability and survivability in the damaged and flooded (water on deck) condition, the new probabilistic stability framework.

In order to test and demonstrate the consequences and feasibility of the new probabilistic stability framework three example designs were made:

- 2500 Passenger Cruise Ferry
- 1000 Passenger Ro-Ro Ferry
- 800 Passenger Handy Size Ferry

Common to all the three example designs was a moderate sized lower hold for ro-ro cargo. In order to increase survivability they were designed with reserve buoyancy above the trailer deck, either in form of side casings or, as is the case for the first example, utilisation of space high up as reserve buoyancy. Furthermore, in order to cope with the more stringent requirements to cross flooding, the double bottom height is increased, allowing ample cross flooding ducts.

The main findings from the calculations of the example designs are summarised.

The rule framework provides a logical way of evaluating survivability, giving the designer, in principle, free hands to make his design. The philosophy follows the cargo ship convention, but the amount of analysis work has increased substantially.

A high level of survivability is achievable as a technical exercise. Practical considerations, e.g. to emergency escape, ventilation, piping systems etc. will limit the achievable level. The example designs show a practical attainable index ranging from 0.73 to 0.85. It may be noted that it does not seem to be the usual correlation between ship size and attainable index, in fact the smallest ship obtained the highest index. (but also the highest capsize index). The sample is of course too small to draw any firm conclusion on where the level should be, but there are indications that an attained index of abt. 0.80 and capsize index of abt. 0.10 is achievable using the proposed framework.

Sensitivity with respect to wave height was less than anticipated. This may be explained by the fact that the survivability factor "sa" is not a function of the wave height. Most of the index was achieved through one and two-compartment damages, which were all showing very high critical wave height. The variation with wave height was then only visible for the three-compartment damages, which accounted for only a small portion of the total index.

Sensitivity with respect to GM (metacentric height) is high. This is expected because GM is an important parameter both to the GZ (stability arm) curve and critical amount of water on deck. The GM used in the example designs is close to 3.0 m, about 3,5 m GM is typically considered the maximum tolerable for comfort and sea-keeping.

Capsize index approach appears effective against rapid capsize due to loss of stability and/or excessive asymmetry. It seems difficult, however, to bring the capsize index lower than approx. 0.10.

Handling of openings in the new framework becomes very complicated. This may be explained with lack of adequate computational tools for the moment. The approach itself should not be regarded as more complicated than the traditional method.

There may be a conflict of interest between evacuation and watertight integrity. A high level of survivability may be restricted by a need for evacuation routes. This is one of the main factors limiting the achievable subdivision index, and has to be born in mind when setting the required level.

Openings present very different risks, all from spilling over half height car deck gates, to grey water scuppers. They should be treated differently with regard to seriousness, e.g. size.

Guidelines for treating ship internal systems are needed. In a probabilistic framework endless numbers of critical penetration depths may be generated, one for each position of pipe connecting two tanks or rooms.

Proposed level of requirement for instantaneous flooding may prove to be impractical. It is correct and necessary to address transient flooding, but the 10 seconds interval should be discussed.

Cost effects are dependent on how it is possible to utilise volumes and deck areas. Increased beam may be utilised for cargo or accommodation, or it may be a waste of space. In any case, volumes below the main deck will increase substantially, and it will be difficult to utilise economically. A cost increase in the range of 5 % is found in some of these examples. The total cost increase from the rule proposal can be determined only after the required level has been set.

The new proposed probabilistic stability method opens possibilities for new, more efficient ship configurations and it remains to be seen if they can be developed less expensive as well.

Figure 1-41 presents the arrangement of the 800 passenger handy size ferry.

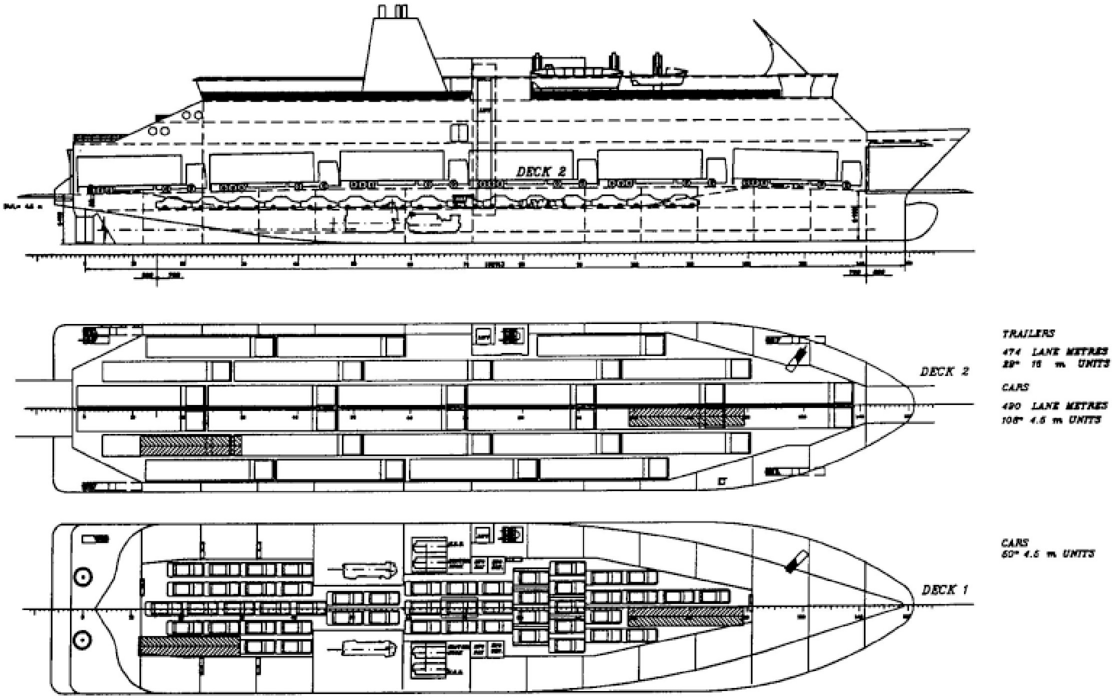


Figure 1-41 Principal arrangement of the Joint Nordic Project Handy Size Ferry

Main findings from the example designs are presented in the Summary Report DNV Doc. No: REP-T00-001.

Centre or side casings

Typical arrangement for ro-ro passenger ferry has been a centre casing accommodating funnels, lifts, stair cases, garbage room, fire stations and similar. Some stores and cargo offices have been located in the corners of the main cargo deck (freeboard deck). Side casing arrangement was used only in exceptional cases. One central casing was typically considered to be cheaper and less complex to build.

SOLAS 90 damage stability requirements, however, changed the situation. Stability characteristics required after flooding and during intermediate flooding stages urged for additional buoyancy volume compared to previous requirements. This could be arranged by increasing the height of the freeboard deck and increasing the beam of the vessel. Raising freeboard deck is not a very effective measure, vertical centre of gravity (KG) is raised at the same time. Increasing beam increases also damage volume. SOLAS 90 offered a possibility to take advantage of compartmentation above freeboard (margin line). The side casing configuration became much more attractive. This is even more evident when lower hold is arranged and considered in damage cases and the more recent water on deck requirements make the side casing arrangement even more attractive.

Typical arrangement with side casings and large lower hold is presented in figure 1-42.

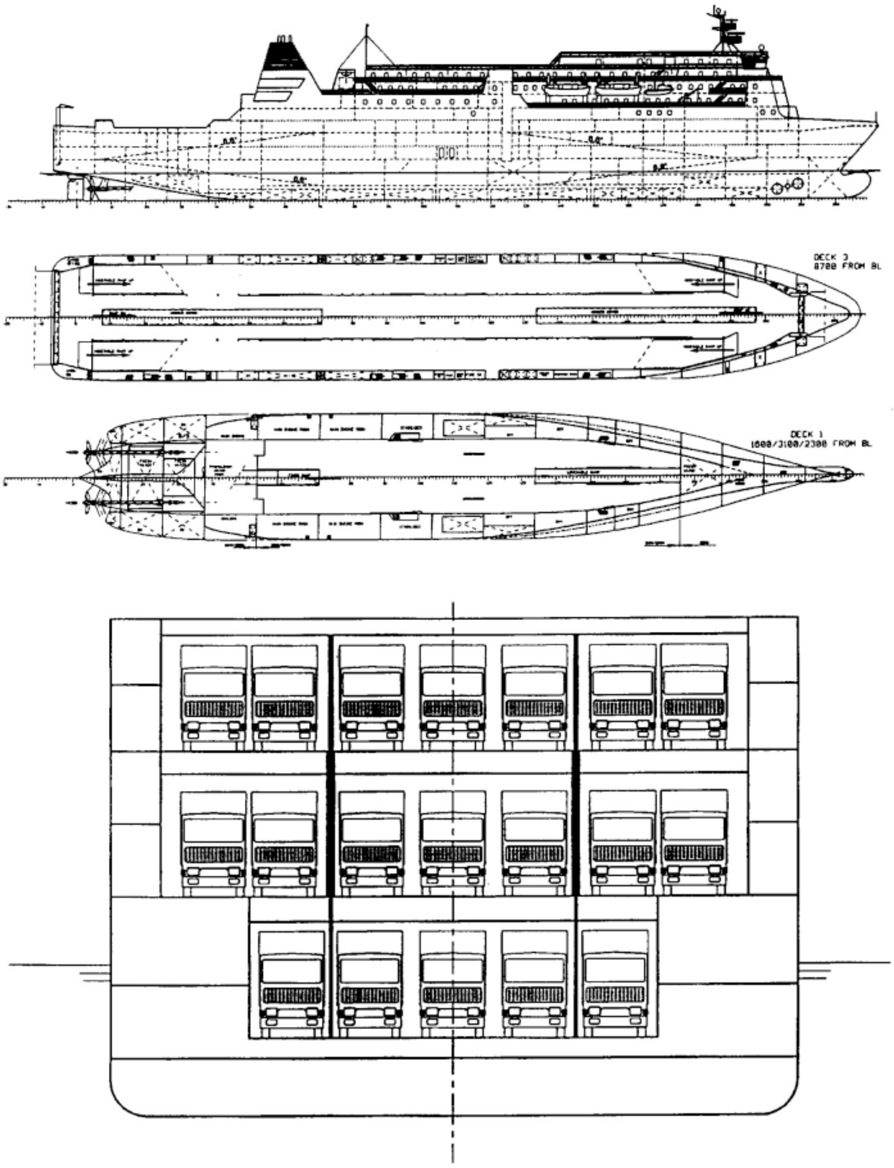


Figure 1-42 General arrangement and mid section of the TT-Line Ferries 'Robin Hood' and 'Nils Dacke' showing the safety barriers, side casings and longitudinal bulkheads.

These TT-Line ferries 'Robin Hood' and 'Nils Dacke' are the first modern ro-ro ferries with diesel-electric machinery and due to the location of diesel generators beside the lower hold, outside the B/5 bulkheads the hold volume and length was maximised. A more detailed description of the vessel is given in chapter 2.

The side casing arrangement has been studied and model tested to find out the optimum casing width and arrangement, impact on water on deck and damage behaviour and compared as well with the centre casing arrangement.

It is possible to reach a lower freeboard deck height with side casing arrangement, especially when water on deck requirements (Stockholm agreement) are to be fulfilled. According to model tests a ferry with side casings can typically fulfil the Stockholm agreement without any flood preventing gates (doors) on the freeboard deck (main ro-ro deck). Side casings are also favourable when a large lower hold is considered and damages are extended within the lower hold. Reduction in freeboard deck height can be as much as 300-500 mm depending on the size of the vessel, lower hold size and criteria applied for the lower hold damages.

Figure 1-43 presents results of a study in which the side casing width was systematically varied for a 170 m long and 28,7 m beam ro-ro passenger vessel with design draught of 6 m. The impact of the side casing width is studied for the GM requirements. Side casing seems to be the most efficient for the higher draughts (GZ range) and at higher draught at least a 5% of the vessel's beam is required in side casing width before they become really efficient.

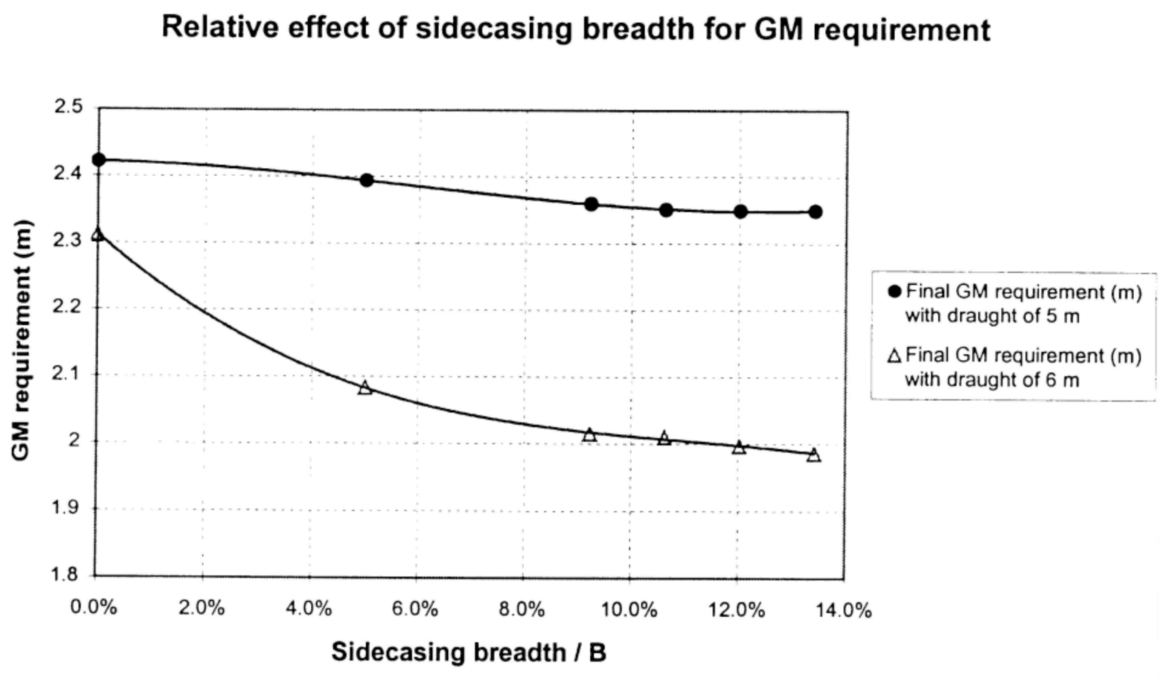


Figure 1-43 GM limiting curves with two draughts

An extensive model test program was carried out at MARIN, Holland, to check the survivability of a ro-ro passenger ferry as presented in figure 1-44, side casing width 2,650 m. Figure 1-44 presents the arrangement of the vessel with large lower hold, side casings, two longitudinal bulkheads on the main deck, i.e. two lanes between the side casing and longitudinal bulkhead and three lanes between the two longitudinal bulkheads. Flood preventing doors installed at the both ends of the longitudinal bulkheads closing towards the side casings, i.e. main ro-ro deck divided into these separate compartments concerning water on deck.

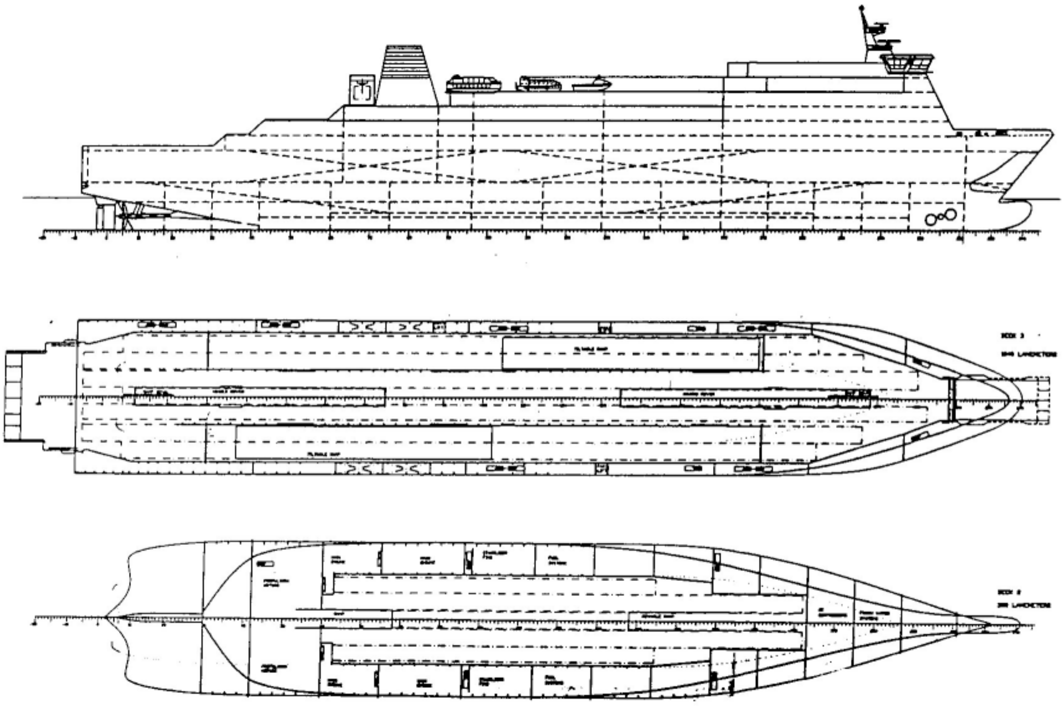


Figure 1-44 Example of effective arrangement to satisfy the new Nordic/IMO requirements for water on deck.

Model tests according to the Stockholm agreement procedure were carried out at design draught of 6,0 m with 4 m wave height (length overall 185 m and beam 28,7 m) starting with the worst SOLAS damage case and continuing with combined worst SOLAS and lower hold damage and gradually taking out all the doors and bulkheads on the freeboard deck. Results were amazingly good.

No water ingress on main deck in the worst SOLAS damage. The worst SOLAS damage combined with lower hold, 100% permeability gave the following results.

- The ship survives with initial GM = 2,20 m and all bulkheads and doors present on main deck.
- The ship survives with GM = 2,20 m and transverse doors removed, but with longitudinal bulkheads present.
- The ship capsizes with GM = 2,20 m and all bulkheads removed (in this case we had permeable cars on the main deck), but only after more than 1 hour (full scale) of continuous testing, i.e. very slow process.
- The ship survives with GM = 3,80 m and all bulkheads removed.

Actual loading cases show GM varying from 2,7 m up to 4 m. Observation for lower hold damages: the slight trim forward causes water to accumulate at the forward end of the main deck, the ship does not roll, and water on deck cannot really slosh back and forth; water never comes any further aft than 0,75 L from FPP. The main difference compared to many existing designs is that the maximum heeling was only 11 degrees as initial heeling and final equilibrium heeling remained below 5 degrees. Ship was very stable and did not roll at all.

For the same ferry detail damage calculations were carried out. The damage stability calculations for the maximum lower hold configuration with all two compartment damages (i.e. adjoining compartments aft and fore and/or side compartments) show good GZ-curve capabilities meeting easily all SOLAS 90 requirements.
Figure 1-45 presents the most severe case lower hold together with the motor room aft and the adjoining side compartment.

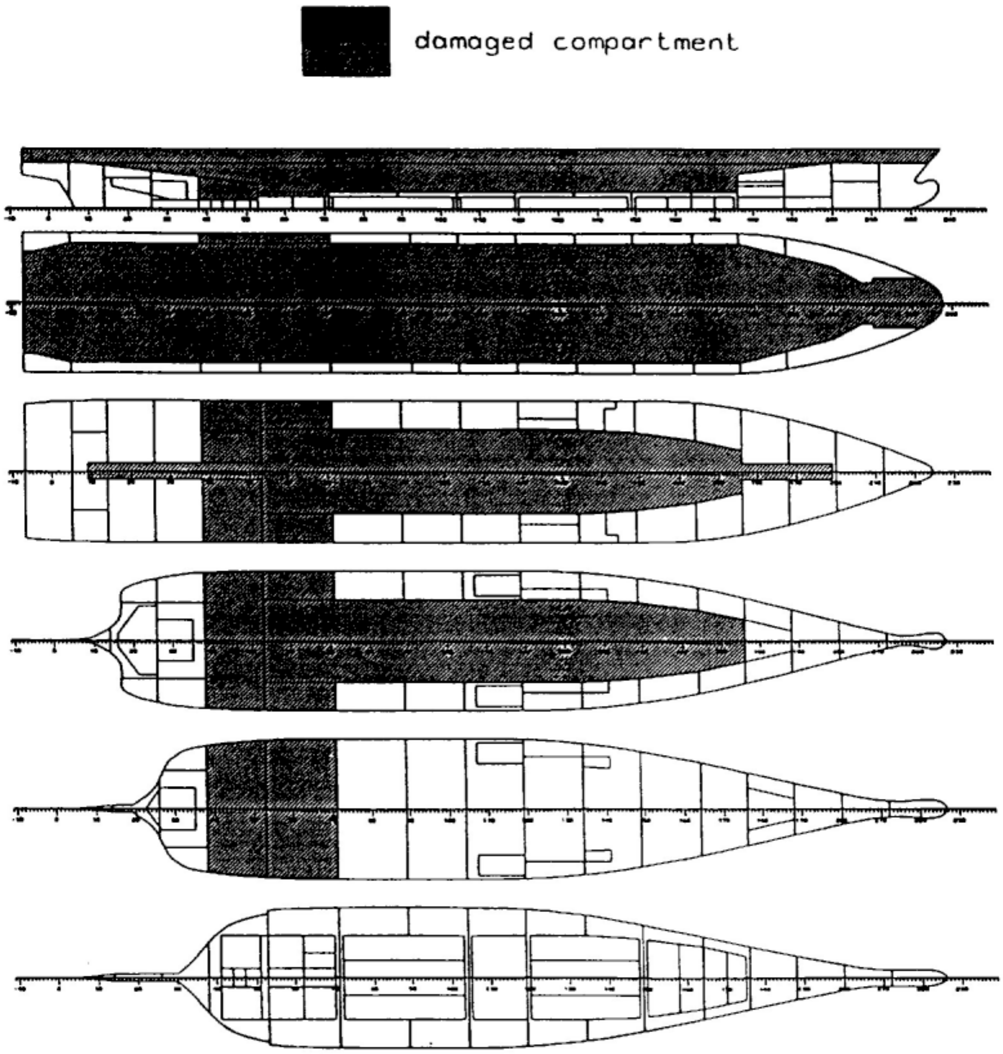


Figure 1-45 The worst damage case with lower hold damaged

In addition to the above ‘normal’ damage cases the following typical and most probable damage situations were studied:

- Three side compartments plus lower hold damage, SOLAS 90 without margin line
- Complete double bottom damage, SOLAS 90 without margin line
- Collision damage extending over 9-11 compartments from bow including lower hold and bulkhead deck, SOLAS 90 without margin line
- Maximum amount of water on deck over three meters corresponding to over 6000 tonnes, simulating an open bow door situation, SOLAS 90.
- Combined lower hold and two side compartments damage plus simultaneously water on deck, survival.

All the above damage cases could be met fulfilling Solas 90 final stage criteria, except in some of them the margin line criteria. The lower hold damages with the longest possible hold actually show the best survivability as there is no trim included.

The side casings above bulkhead deck are an essential part of the survivability and according to model tests give a possibility to leave out flood preventing doors on the main deck. The longitudinal bulkheads, see figure 1-42, within the main deck give also an option to limit the amount of water on the deck if seen necessary but they also give a good possibility to limit cargo movements on the deck.

1.6 AUTOMATION, ELECTRIC AND NAVIGATION

1.6.1 Machinery Automation

System characteristics

Machinery Automation Control Systems in different vessel types are all very alike in system architecture. They are microprocessor based integrated and distributed systems with open architecture.

Typical system consists of Operator Stations communicating with Main Computer Units by local area communication network (LAN). The Local Process Units are connected to each other and Main Computer Units via redundant field bus, as presented in figure 1-46.

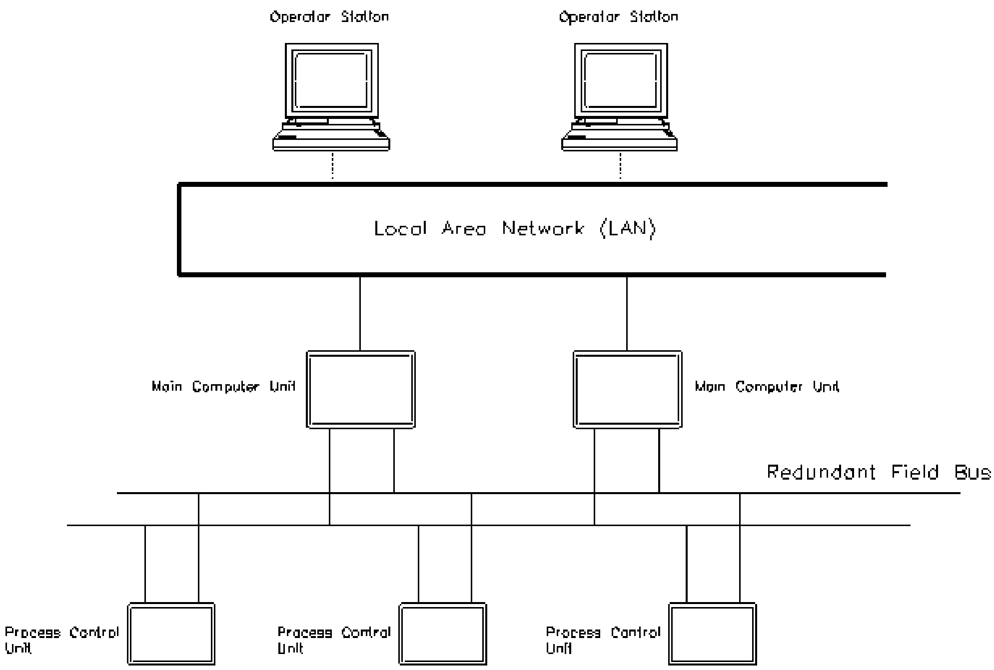


Figure 1-46 System Architecture

Automation systems can be divided into three groups according to system capability; small, medium and large systems.

Small size system

Small systems are typically used on modern tankers, bulkers and container ships. These vessels have some 1000 I/O channels for the alarm handling and control functions. Average system includes 2 Operator Stations connected to 5 Process Control Units.

System operations are mainly alarm and monitoring of main engine and diesel generators and also cargo and ballast control. One important function for the system is power management, with less than 100 input/outputs.

Medium size system

This system is suitable for ro-ro passenger ferries which are technically between ordinary cargo vessels and cruise ships. These vessels have high installed power and machinery is based on multi-engine installation with medium speed engines.

Typical modern ro-ro passenger ferry has 1000-1500 I/O channels. Medium size system typically consists of 4 Operator Stations and 7 Process Units. The number of Operator Stations is a result of auxiliary systems.

Large system

Large automation systems are typically for passenger cruise ships. Modern big cruise vessels are based on diesel-electric machinery with four to six main engines. In the future gas turbine machinery of combined cycle (COGES) is also one possible option as well.

Due to high demand for safety and complicated machinery for hotel services (double auxiliary systems for machinery, air conditioning, different water systems and extensive piping systems) the automation system meets special demands regarding capacity on operability, especially concerning large number of graphic display pages.

A typical modern cruise ship has 4000-6000 I/O channels. A large system typically consists of 6 Operation Stations and some 12 Process Units.

Automation System Trends

Future vessels will have small but qualified crew due to new requirements by STCW-95 (The international convention of Standards for Training, Certification and Watch keeping for seafarers). This is a part of ISM (International Ship Management code) overruling all major maritime functions in the future.

Extensive use of a comprehensive integrated machinery monitoring and control system of the complete ship (ship operation centre) would be the most effective method to meet major part of the STCW demands by giving totally new possibilities for crew training and familiarisation to ship and her systems.

The systems would then be based on extensive integration with all ship systems including fire detection, fire door control, fire fighting, air conditioning control and indication. The system could be a common data and operational centre including all information of the ship.

Based on failure statistics a human error is the major risk for safety. Thus the automation system should be able to prevent from doing such mistakes. Alarm functions should be self-diagnostic evaluating how serious each alarm is and that only selected alarms will be indicated. Unnecessary automatic shutdowns must be avoided.

Trend in maintenance already today is towards on-condition maintenance rather than scheduled maintenance.

1.6.2 Electric

Power and voltage levels

The installed electrical power and voltage in vessels have steadily been increasing due to growing vessel size and application of electric propulsion and other electric consumers.

The installed electric generation power level on vessels today is described in the following:

small passenger ferry	up to 5 MVA
cargo ships (mech. propulsion)	up to 8 MVA
big ro-ro passenger ferry/small passenger cruise ship	5 … 15 MVA
passenger cruise ship (mechanical propulsion)	10 … 25 MVA
passenger cruise ship (electric propulsion)	30 … 75 MVA
large passenger cruise ship	abt. 100 MVA

The voltage at which power is generated is generally determined by the total power demand of the system, the current and power levels of heavy consumers and the short circuit capacity of the breakers and the switchgear. The common frequency is 60 Hz or 50 Hz.

A recommended maximum for distribution of current is 2000 … 4000 A due to required cable/busbar size, magnetic fields and losses.

Generation and distribution of power including emergency power on low voltage (LV) has been the conventional technique until the fault level in the LV distribution networks increased with the power level to extremely high figures. Practically the upper power limits were reached by the figures below.

Low voltage levels with maximum installed (parallel running generator) power:

- 400 V … 9 MVA
- 450 V … 10 MVA
- 690 V … 11 MVA

As the medium voltage technique has been developed and has become more cost attractive the recommendation for highest LV generation power installed is about 8 MVA.

Recently the 690 V has become a common LV power distribution system normally supplied by generators or step down transformers. The 690 V system, reducing distribution costs and weight compared with 400 … 450 V systems, has typically following characteristics:

- standard 380 … 440 V (D-connection) motors can be used as 660 V (Y-connection) motors
- lower nominal currents
- cables with smaller cross-sections
- reduced weight
- lower fault currents
- equipment maximum fault level 50 … 75 kA

Medium voltage is applied on ships to generate and distribute high power with lower current. As the power demand has increased on modern ships, mainly due to electric propulsion, the use of medium voltage is ordinary.

Medium voltage levels with recommended and maximum installed power levels are:

- 3,3 kV … 25 MVA
- 6,6 kV 10 … 60 MVA
- 11 kV 40 … MVA

Medium voltage distribution is recommended for motors from about 750 kW upwards.

<u>Busbar architecture</u>

The electrical power distribution system based on the busbar architecture provides the secure electrical supply required for maintaining the ship in normal operational and habitable conditions.

Today the common busbar architecture is a subdivided main switchboard, each side supplied by 1 - 3 sources. This basic architecture provides an ordinary redundancy and prevention of blackout and avoiding loss of propulsion and steering.

The operational practice to have at least two generator sets running and supplying the network, including harbour and sea mode, should reflect the definition of the number and size of generators where possible. Furthermore, allowance for one ‘spare’ generator set should be considered in case of unforeseen need of service in any part of the generation system.

Utilising shaft generators to supply large electric motors, e.g. side thrusters, cargo pumps, etc. is often found cost effective. The architecture can be a common ship distribution network or a separate shaft generator supplied net with 'tolerant' characteristics.

A typical LV busbar architecture is illustrated in figure 1-47.

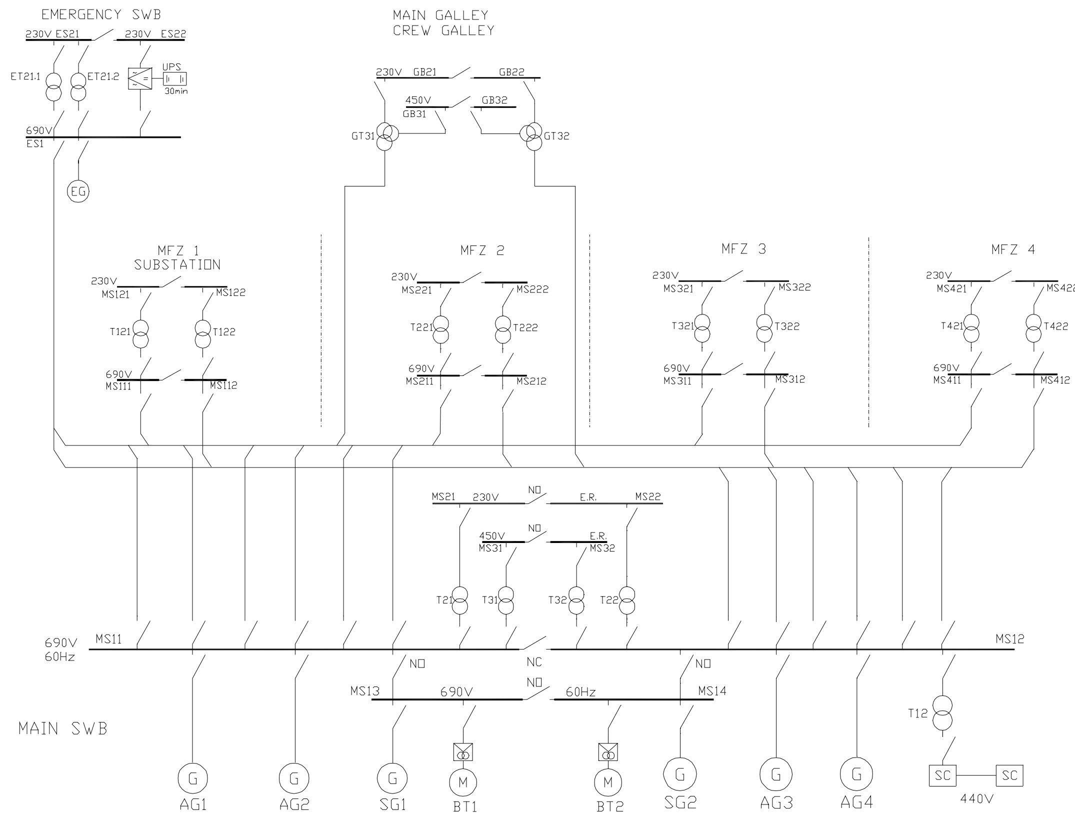


Figure 1-47 Typical busbar architecture on a ship with both diesel-electric and shaft generators.

Extended redundancy is achieved by embracing a structural subdivision of the busbar to two independent main switchboard rooms including division of supply sources into two compartments. The electrical separation is thus supported by separation in respect to fire and flooding.

With growing power demand the voltage is increased and the amount of voltage levels to distribute, as well as the network complexity, grows.

The emergency load grows accordingly as well as the emergency generator services to achieve an extended safety specified by the owner. An option to manage these extensions is to provide two emergency generator sets, each with capacity suitable for either emergency services required by the pure SOLAS or emergency services specified by the owner.

Network characteristics

Some noteworthy characteristics of the electric network are highlighted in the following.

Motor starting capacity of the network

The network capability to allow starting of a large motor is evaluated in new projects but especially in ship conversions regarding additional side thruster or new sprinkler/drencher pumps to emergency network.

In ship conversions it is not enough only to consider available spare power for the required conversion, first of all the starting capacity should be evaluated.

The motor starting capacity, actually the allowed transient voltage drop in the network, depends mainly on the generator reactances, voltage regulator and motor starting characteristics.

Water cooling of electrical equipment

The heat losses derived from electrical components are considerably high in a large ship, so high that alternative cooling methods to air have been introduced. Water cooling is today common for generators, large motors and converters and also large transformers have recently been equipped with water cooling. Thus the capacity of required fan coils and air ducts can be reduced.

Separate insulated networks

In order to limit frequent earth faults a separate insulated network has been provided for e.g. following common groups:
- galley area consumers in passenger vessels
- reefer socket outlets on ro-ro vessels

The networks are insulated from the ship distribution net with transformers.

Selectivity

A selective protection is disconnecting only the faulty part of the network. This means that all the series connected over-correct relays, direct acting circuit and time delayed breakers and fuses shall be coordinated to achieve correct selectivity (discrimination) during fault conditions. Correct selectivity shall be maintained for the minimum and maximum prospective fault currents.

A total selectivity can generally be ensured by combining different types of discrimination techniques:

- current discrimination by different fuse values or magnetic trip values
- time discrimination with intentional trip delay devices
- zone (accelerated) discrimination with microprocessor interface (pilot wire)
- energy based discrimination, a recent improvement to selectivity

The selectivity is only partial when it is ensured to a certain level of the prospective current and above that simultaneous tripping of more than one protection may occur.

An analysis of the selectivity can be performed using suppliers discrimination tables or available computer programs.

Impacts of electric propulsion

The feasibility of electric propulsion originates from the power plant. A power plant concept where all the engines are connected to generators and all power consumers are supplied from the main source can be feasible if at least one of the following criteria is fulfilled:

- Considerable amount, say 30-50%, of the normal power consumption is somewhere else than on propulsion. Typical case is a cruise ship.
- Podded propulsion would give a fundamental impact on arrangement, power demand or manoeuvrability. This can be the case with high vessel speeds where pod propulsion can reduce power demand by up to 20% from the case with traditional shaftline.
- The central power plant can be allocated so that ship design can be improved toward increased income potential. Good examples are the diesel-electric tankers.
- The dynamic positioning mode is a major operational mode.

Correspondingly, especially in the last case a comprehensive evaluation of the whole ship concept including also initial transportation mission must be carried out prior to making any decision about the feasibility of electric propulsion.

Electric propulsion with large converter drives brings some totally new aspects to be considered in machinery design:

- System efficiency is about 8% lower than in mechanical propulsion. The total value depends on the selected solution of the complete machinery plant
- The power factor is not constant over the complete range of operation.
- The current waveform is distorted leading to distorted voltage waveform when the current meets inductances.
- Special attention must be paid to how the reverse power is handled and controlled during stopping and crash-stop. In this respect the advanced manoeuvring philosophy allowed by pod propulsion should be utilised.

The distorted waveform initiates typically most of the discussion. It should be remembered that all semiconductors widely used on ships create this harmonic distortion, not only electric propulsion. Typical sources are converter controlled air conditioning and ventilation fans, passenger lifts, sea water pumps, compressor motors and side thruster motors.

Distorted waveform does not create later problems if it is considered in design phase. Main effects of the harmonic distortion are:

- Additional heat losses in machines, transformers and coils of switchgear and control gear
- Additional losses in compensated lighting
- Distortion of the accuracy of some measuring devices
- Interference of all kind of electric equipment, such as regulators, communication and control systems, position finding systems and navigation systems
- Disturbance in different onboard computer systems.

In distribution systems mainly consisting of conventional consumers such as lighting, motors, etc., and powered by synchronous generators, the total harmonic distortion in voltage waveform shall normally not exceed 5% (rules by DNV). Special care is to be taken by harmonic distortion level in normal case but also the worst case (low hotel load, high propulsion load) must be checked. This can be done effectively by applying computer simulation already at an early design stage. It is also worthwhile to carefully consider the acceptable distortion level on main propulsion busbar to avoid too expensive or bulky solutions. This can be higher than 5% when special care is taken that all components connected on this network can sustain the actual distortion level.

The clean network including segregated clean cables is kept apart from the ‘unclean’ converter networks. The clean network supplies conventional low voltage services, such as power, lighting, telecommunication, navigation, automation, etc.

Separation of this clean network was earlier done mainly by applying rotating converters. This is a well known and secure method but has several disadvantages such as efficiency loss, noise, heat load and space demand. Today there are also other, more advanced, methods available and applied, such as:

- Use of transformer supply and twelve or higher pulse power converters for large powers
- Installation of filters on transformer secondary for the suppression of dominant harmonics
- Over-sizing of generators and transformers
- Applying low sub-trancient reactance on generators
- Compensation by duplex reactors

Later modifications with additional thyristor controlled load must be considered by leaving some margin for the future growth.

Distribution principles

Following distribution principles are commonly adapted particularly in passenger ships with large electrical loads on the hotel side. The principles are used in both power and lighting distribution levels.

Power distribution in a ship with a centralised generation plant and load concentration in the area of generation will obviously be compact and the radial distribution principle is typically adopted.

The power transformers and main switchboard are typically located close to the main generation plant. In the radial system power is distributed by dedicated cables directly to large consumers, motor control centres and sub-power boards throughout the ship from the main power switchboards on different voltage levels (MS1 and MS2 in the example). This results in a large amount of feeder cables running through most of the ship areas and penetrating watertight and main fire bulkheads.

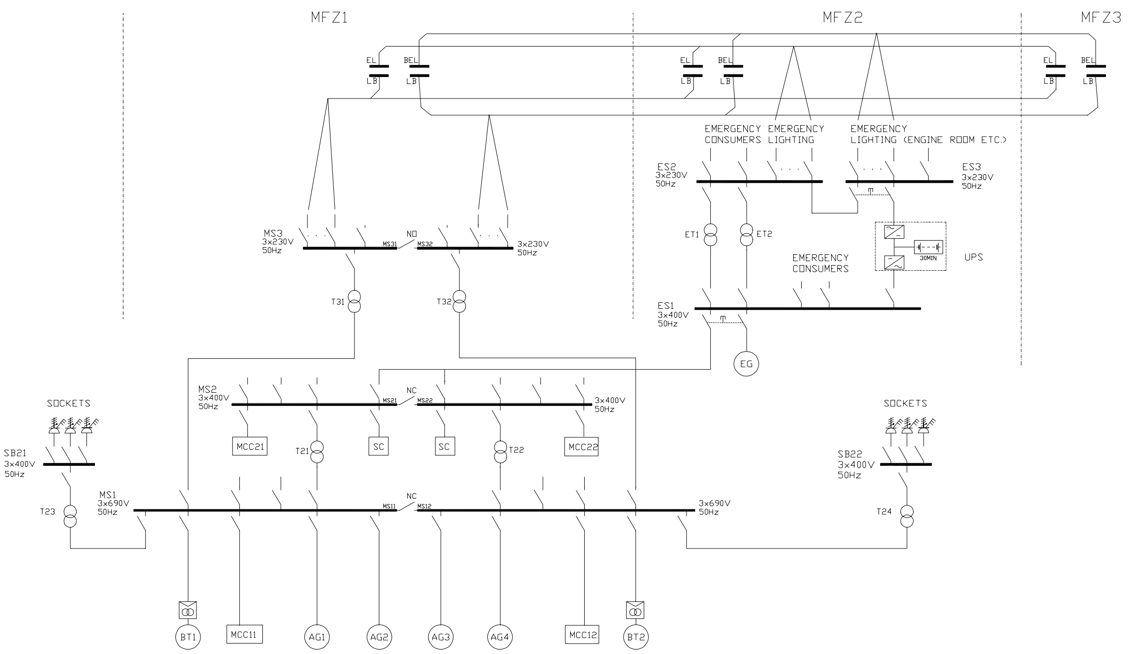


Figure 1-48 Example of radial distribution principle

Radial distribution is direct distribution and compact in size, it is uncomplicated and
cost effective in ships with limited distribution currents as well as cost effective in ships with limited distances and areas.
The disadvantages are increased installation costs with large distribution currents and distances, and less redundancy due to concentration.

Main fire zone (MFZ) related distribution

The MFZ related distribution has distributed power sub switchboards located in electric sub-stations, one for each main fire zone, see figure 1-47. This MFZ, except for the main supply, is principally self contained. Sub-distribution cables are installed vertically and located only in that MFZ. This requires vertical cable trunks to facilitate straight and short cable installation which reduces costs.

Due to good possibilities to reach high redundancy, e.g. 100 % spare capacity of spare supply, this principle has also been utilised in smaller ships.

MFZ related distribution advantages are reduced installation costs (mainly lighting distribution) compared to radial distribution with growing ship size, and redundancy improved by the distributed system.
On the other hand the sub-station area demand means space loss and increased amount of equipment.

Circular distribution

The circular or ring-main distribution principle is a distributed system, a variant to the MFZ related distribution, with a back-up supply arranged in a ring-main, see figure 1-49. The MFZ power sub-station, containing typically a high voltage (HV) transformer and the sub-switchboard, is connected into a supply ring. This ring has two or more supplies from the main power switchboards. The ring-main shall have sufficient capacity for any possible load and supply configuration.

The circular distribution is applied on big passenger cruise ships, and the ring consists of a HV-cable. For an average sized ship the ring can as well utilise low voltage (LV) cables.

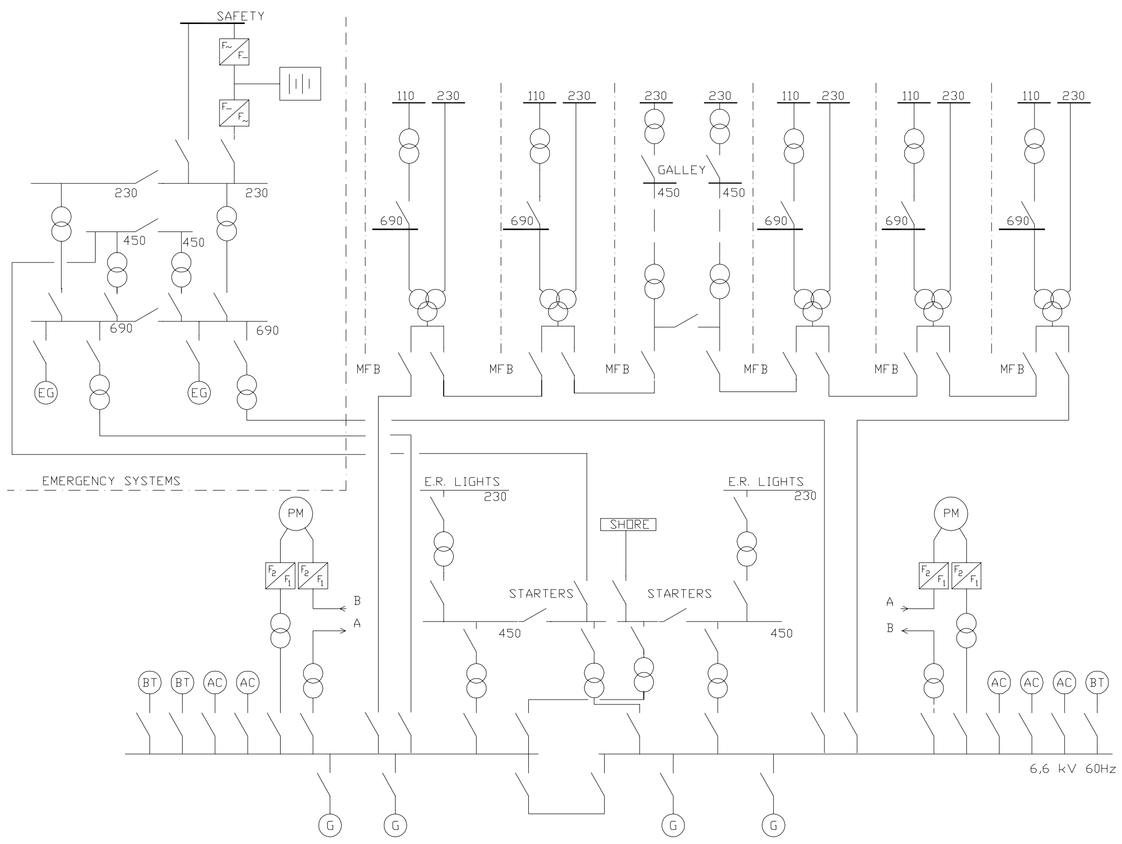


Figure 1-49 Principle of circular distribution

Distribution of power on primary voltage level is cost effective with circular distribution. Installation costs are reduced compared to radial distribution with growing ship size, and distributed system improves redundancy.
The sub-station area demand means loss of space in each MFZ, and the complexity and amount of equipment increases.

1.6.3 Navigation Bridge

All the facts affecting the bridge design have to be discussed in detail with the owner and related authorities. This chapter gives guidelines for navigation bridge design, items to be considered and methods for efficient ship operation centre development.

Typical design criteria can be listed as follows:

- Design criteria and references required specially by the Client
- Ship type
- Panama Canal Commission: vessel requirements
- Operation area of the ship
- Class notations DNV W1, W1-OC, Lloyd's NAV-1, ABS OMBO, e.g.

- Authorities requirements
- Standards, IEC 1023, 936, Document 18, 534, ISO 8468
- IMO SOLAS Ch IV,V, Resolutions A 708, MSC circ. 566
- others.

Contract material typically defines the scope of the navigation, communication, ship safety and other bridge equipment and maker(s). Wheelhouse geometry and geometry above water line is defined by the general arrangement of the vessel.

Equipment data is defined in the specification or alternatively by the selected maker(s), including consoles, equipment 3D drawings, etc.

Bridge and wheelhouse arrangement

A good base to start the design is to prepare a catalogue of the equipment, pre-locate them to the workstations and evaluate the required space for each equipment, and to define the number and size of video display units and conventional analogue instruments.
Possibilities for using multipurpose displays in the systems (ECDIS, ARPA, Conning, IMAS, etc.) should be checked as well, and electroluminescence displays instead of analogue instruments.
A list of the equipment located to the equipment room close to the wheelhouse should be made as well as a list of power source, power consumption and heat dissipation.

Navigation workstations

The planning should be started from the most important, main navigation workstation, command centre, its form, size and arrangement.
The modern workstation should be planned for two navigators (pilot - co-pilot system) with E-formed cockpit or 45 degrees open cockpit, where both navigators have own radar display. Conning, steering and manoeuvre equipment in the middle, common use for both navigators.
The cockpit should be placed in centreline or starboard from the CL if e.g. deck cranes are located in centreline and will disturb the optimum field of vision.

Station for manual steering should be located behind or in front of the cockpit in the centreline.

Stations for route planning, communications and for safety operation are located to the aft part of the wheelhouse together with the place to study drawings and books, and to carry out office works, etc.

Checklist for the planned wheelhouse area:

- Wheelhouse shape, clear height
- Window arrangement, height of lower and upper edge above deck and division between the various windows, inclination
- Location of different workstations
- Panama conning positions and instrumentation
- Passageways between workstations

- Dimension of consoles
- Location of main stand-alone equipment (radar, ECDIS)
- Ceiling panels
- Access to the bridge
- Toilet facilities.

Field of Vision

Installations outside the bridge, masts, deck area in bow and below e.g. shall not reduce the field of vision. Total arc of blind sectors must be in accordance with the class notation .

Check the field of Vision from following workstations:

- Traffic surveillance / manoeuvring / navigation workstation
- Navigation workstation
- Workstation for safety operation / communication
- Workstation for docking manoeuvres
- Workstation for manual steering

Deck area in front of the bridge superstructure shall be visible from inside the wheelhouse.
Side of the ship shall be visible from the bridge wing.

Check list for good ergonomic design of the wheelhouse

- Working environment.
- Colours and materials.
- Working in seated position with optimum visibility and integrated presentation of information and operating equipment.
- Safety operation and fast action.
- Navigator’s safety with hand rails, no sharp edges, height of instrument and panels above workstations.

A typical example for wheelhouse principal lay-out is presented in figure 1-50.

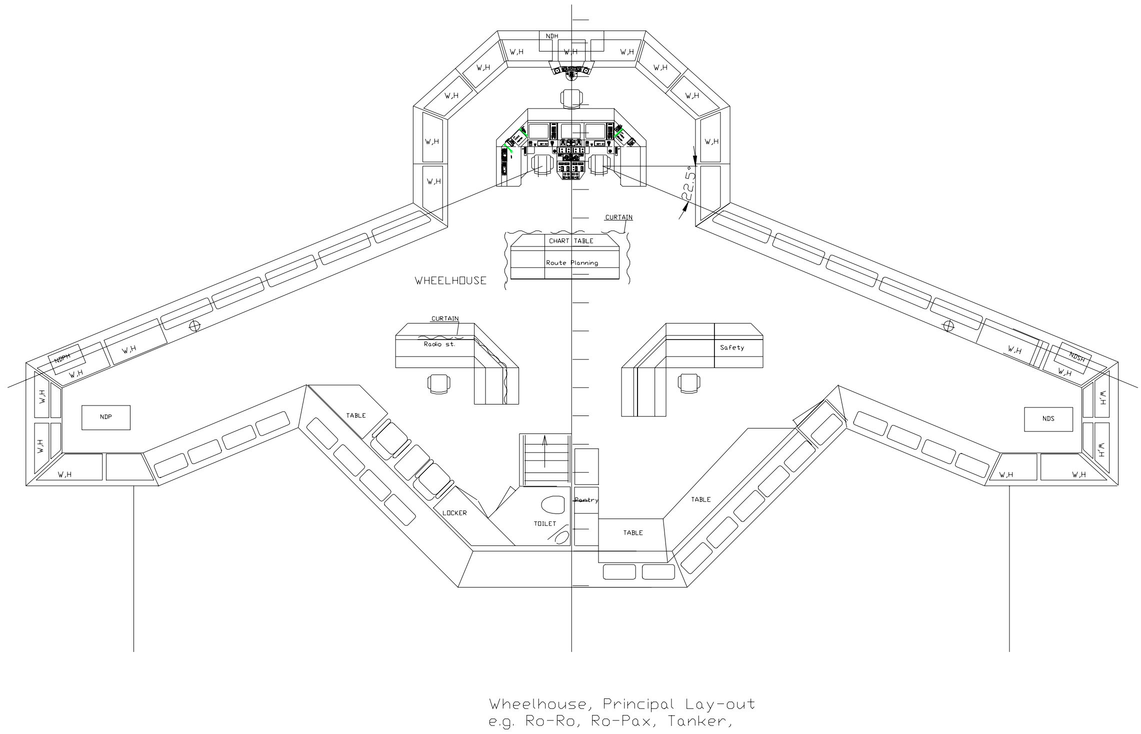


Figure 1-50 Wheelhouse principal layout

In this example of the main navigation workstation, the command centre is located off the front bulkhead for the following reasons:

- Displays are off from direct sunshine
- Deckhead consoles above windows are in proper distance from working place
- Helmsman is located in front of the cockpit for better watching, alternative location is behind the navigators in the centreline.
- Pilot conning positions are directly behind and close to the windows in CL and both side (Panama).
- Consoles and equipment service possibility on both side of the consoles.

Consoles are mechanically integrated together and there is a clear passage to every workstation.
Navigator seats are located on the same line to minimise blind sectors.
Bridge wing is as narrow as possible for optimum vision downward along the ship side.
Afterpart of the wheelhouse has route planning station, station for external communication, safety station and table for drawings and plans, and office, pantry and place for navigation and safety books as well as toilet.

4D Bridge Design

4D modelling is the most recent design tool to generate the virtual reality model, in which you can move, make modifications, create alternatives, check visibility and ergonomics.
User inputs can readily be examined and optimised in the model. It is easy to analyse the design in every stage of the project work. The layout optimised in a 4D model saves efforts, time and especially expensive mistakes afterwards. Coloured views and prints from different positions are easy to understand. It is easy to check and ensure desired field of vision in all directions. Ergonometrical and operational requirements can be optimised. The same model can be used for training and as a database for training simulator (STCW and ISM code). It is also easy to modify the reference bridge layout for any conversion or generating a new ship configuration.

Bridge standardisation

The bridge layout on every ship has been different, the position of equipment and workstations has varied to provide field of vision and passageways. In the future, as we see it, the purpose is to standardise the layout and design of the navigation bridge.
A good starting point for standardisation would be design for standard main navigation workstation, command centre and berthing workstations. Other workstations to be standardised according to the type and purpose of the vessel.
The manning in different types of vessels will vary, but the main navigation tasks are quite similar.
Standardisation of main navigation equipment functions, module size, models, interfaces, colours, etc., can be carried out quite easily; the next step being standardisation within the shipping company, within the ship type, etc.

Key benefits for standardisation are obvious. Simplified training of officers, a company standard can be created. Less training for a newbuilding project is required, the bridge is known already well before delivery. Possibilities to use standard bridge module in training simulators becomes reality. Risks in pilotage communication become less. Ready made virtual mock-up enables good and fluent design and building coordination, shorter design and building time, up to several months, and efficient test facilities before installation as well as better understanding of building costs.

Integrated Bridge System

Well designed, manufactured and tested integrated bridge system can save a lot of human operator work load. Computers have very high capacity and can calculate different tasks fast and more accurate than the human operator can do.
The total system for the bridge performance of bridge function, comprising bridge Personnel, integrated system, man/machine interface, and procedures.

Mechanical integration comprises a number of stand-alone equipment incorporated into ergonomically designed consoles. All functions are electronically integrated, where various equipment are linked to each other with duplicated high redundancy navigation network. All necessary information is displayed on multi-functional display screens. The basic integration system should consist of ARPA radar(s) with multifunction display, a very accurate gyro-compass with dynamic (ballistic) error correction, a very accurate dual-axis speed log, differential corrected GPS receiver (Loran-C for back-up), ECDIS (Electronic Chart Display and Information System), route planning station, chart digitiser, printer, centralised navigation alarm system,
ANTS (Automatic Navigation and Track-keeping System) which can produce navigation lines, marks, curved headline and needed symbols displayed in Radar screen.

The integration shall also include such additional functions as engine monitoring, machinery status, pumps control, fire alarms, cargo condition/control, hull condition monitoring, and others as required.

The future is certainly for integrated and standardised bridges supplied by turnkey suppliers. There seems to be a lot to be learnt from aviation and car industry.

1.7 STATISTICS, EFFICIENT TOOL FOR PROJECT DEVELOPMENT

There has been tremendous development in all kinds of ships since the early eighties. The size of ships has progressively increased as well as speed. Increased safety together with higher required capacity and efficiency has led to application of new design configurations and technology, at a rate never seen before in the shipbuilding world.

Increased competition, however, will always put the focus on the investment and running costs of any investment. It is evident that lifecycle cost analysis is required to back-up introduction of any new configuration or solution.

The issue is to maximise the efficient revenue generating space at minimised investment and running costs but taking into account system availability and environmental impacts as well. These items are to be clarified and their impact on the lifecycle economy of the vessel calculated before a decision can be made for example between different machinery configurations.

The tendency is clear and promising, new products and innovations are introduced and completely new machinery and ship configurations are developed for efficiency and economical reasons. The shipbuilders are no more just preventing this development, but they have also realised the potential for cost savings in new conceptual thinking.

Considering cost efficiency we end up with four items: space, weight, power and equipment (materials). The efficient area and volume of the vessel compared with the total area and volume is a good indication of revenue generating capability and costs. Weight is directly related to building and fuel costs. Installed power onboard relates to the efficiency of hull, propulsion system and power generation. Equipment and materials are directly and indirectly, through required man-hours, cost related.

Space efficiency

We have gathered and analysed more than 50 recently built passenger cruise ships of all sizes. Rather big and sometimes amazing differences can be found in space utilisation. Figures 1-51 and 1-52 present good examples. Figure 1-51 shows machinery space volume compared with the total volume of the ship of 32 passenger cruise ships. Smallest value is 8% and highest 22%, the average being 11,3%. The highest figure is from QE2 and is taken into this comparison only to show the general tendency. A further study of engine casing volume of selected vessels, figure 1-52, shows the same kind of variation. Some of the differences can be explained by different vessel speeds, but for example both diesel electric and diesel mechanical types are included, and that gives no explanation, nor the year of built.

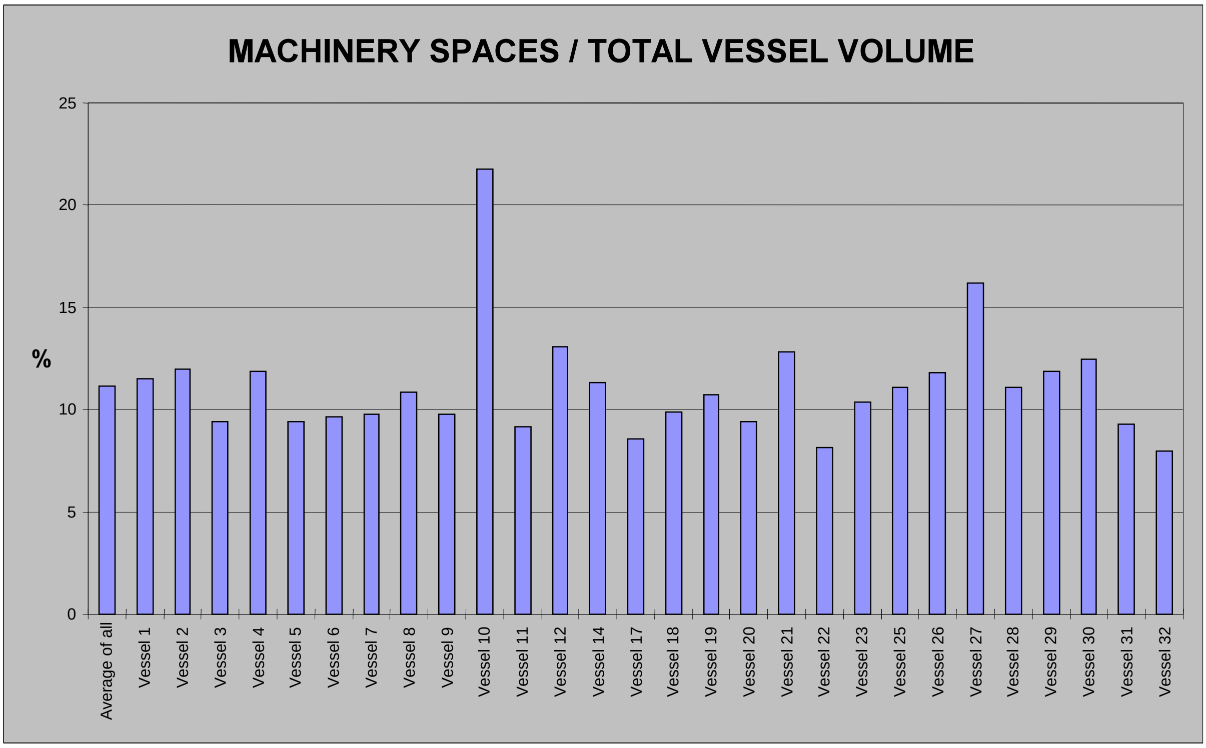


Figure 1-51 Machinery space volume compared with the total volume of the ship.

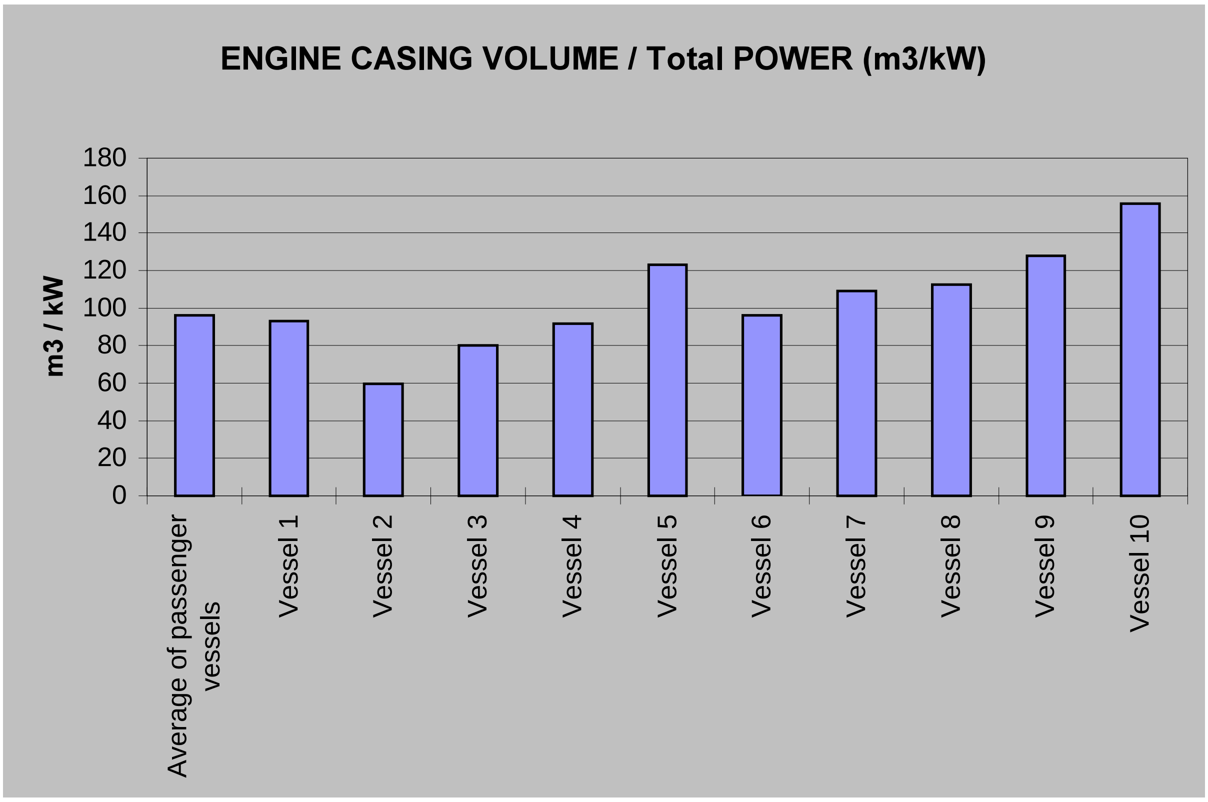


Figure 1-52 Engine casing volume comparison

Bearing in mind that the machinery spaces and especially engine casing are located in the most centralised area of a vessel it is certainly worth while considering carefully the efficiency and location of machinery spaces and casing.

Space efficiency can be considerably increased through clever machinery configuration selection and design. There are four different steps and selections to be considered:

- power generation
- propulsion system
- machinery type
- machinery location.

Electric propulsion is today an industry standard for quite many types of ships, such as passenger cruise vessels, offshore vessels of several types, research vessels, icebreakers, and it is becoming more and more typical in ferries, tankers (chemical and product). It seems to be only a question of time when the first container, ro-ro and car carriers are built with electric machinery.

Electric power generation gives the freedom to select optimum power source and optimum number of power generating units. This leads to direct and indirect space savings. Big slow- speed diesels can be replaced by much smaller units and due to the power plant principle the total installed power can be reduced, the same power generation can be used for propulsion and for other consumers, such as cargo handling and hotel load.

The selection of propulsion system has traditionally been between fixed pitch and controllable pitch propeller. Electric power generation already changed the stable market situation at the early nineties. Today the big challenge is the pod-propulsion. It is an industry standard today in passenger cruise ships and has been applied also for product tankers (two refits), icebreakers and offshore vessels. Space saving is obvious, big propulsion motors are moved from the tank top outside the ship.

The selection of machinery type has typically been between different types of diesels, between two stroke and four stroke, but today gas turbines have made their entrance into fast ferries and passenger cruise vessels, again with high space savings. In the new Panamax size RCI newbuildings, ‘Vantage’ and ‘Millennium’ class ships, it was possible to gain 50 additional passenger cabins by changing into combined gas and steam turbine electric machinery. Gas turbine machineries are now studied for all coming cruise ship projects but interestingly also for other types of commercial vessels. Space saving and environmental issue are playing a major role.

Optimising the location of machinery was one of the main criteria in the selection of diesel-electric machinery for chemical and product carriers.

Tanks

Analysing the total number of hull tanks in seven recently built cruise vessels, of abt. 70.000-80.000 grt, an astonishing variation was found, maximum being 91 and minimum 45 tanks, a difference of 50%, figure 1-53. Some explanation is given by the different service profiles but certainly not all. It is a matter of design efficiency and related with compartmentation and damage stability. Some of the ships can fulfil the same SOLAS 90 damage stability requirements with less tank subdivision.

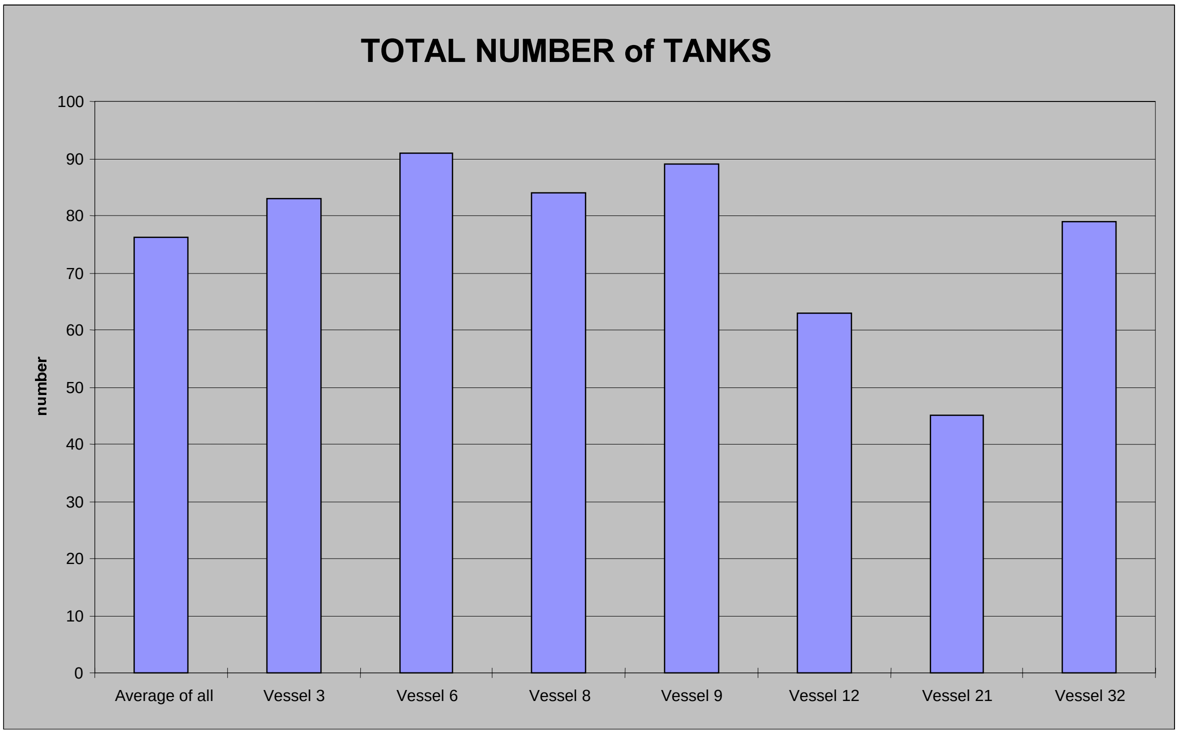


Figure 1-53 Total number of hull tanks in recently built cruise vessels.

Air conditioning

Area of air conditioning fan rooms was analysed and compared. Again quite large variations were found from 3,6% of the total interior area up to 10,3%, average being 5,8%, figure 1-54. Today the so-called fan coil system is becoming more popular and with this system the fan rooms can be even further reduced as the cooling media is brought into the cabins instead of the cooled air. It is worth while paying attention to the location and space allocated for air conditioning spaces as well as air intakes and outlets, i.e. fan rooms located far from intakes may lead to big loss of space in inlet and outlet ducting.

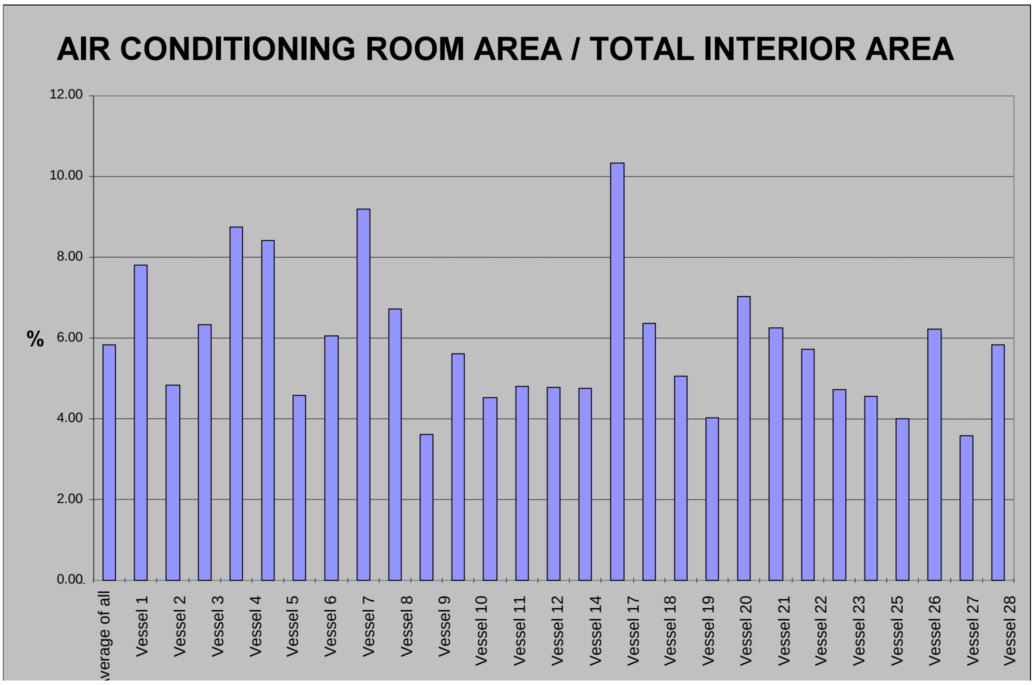


Figure 1-54 Comparison of the area of air conditioning fan rooms

<u>Space - Weight</u>

Analysing the space-weight relation of recently built cruise ships, figure 1-55, shows that the most efficient ship has 40% more effective area per lightweight ton compared with the least efficient. A typical difference is about 20% which can be considered to be high. Standard of the vessel may have an impact on these numbers, but otherwise it is difficult to find other explanations except efficiency in the design.

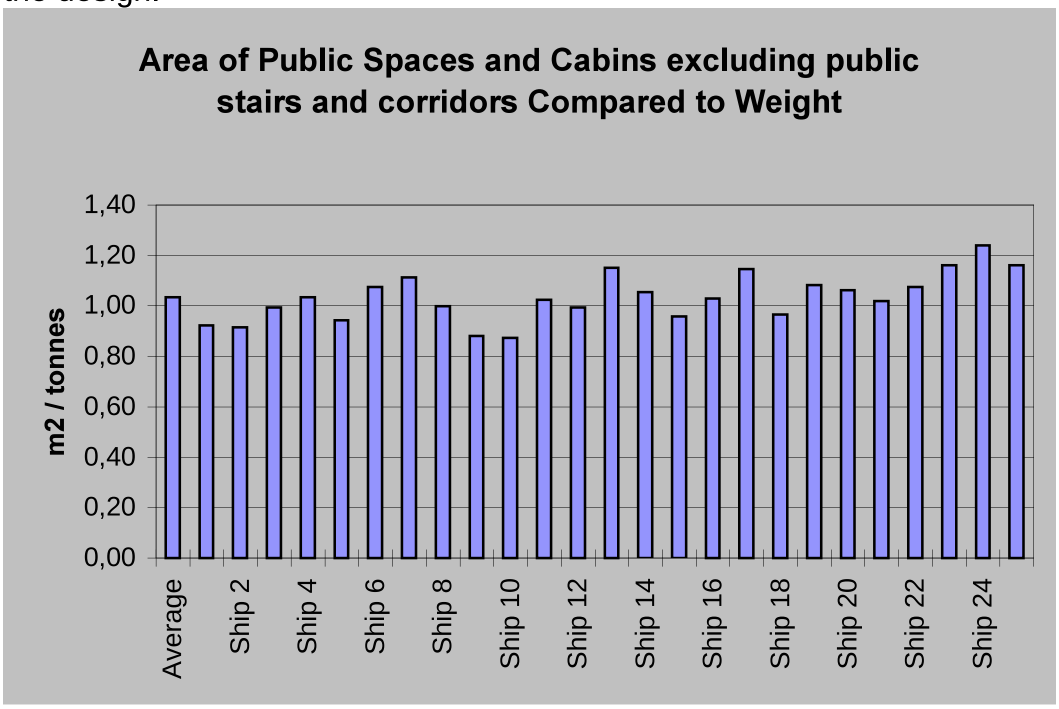


Figure 1-55 Space - weight relation analysis of recently built cruise ships

It is certainly worth while analysing in detail a new design and compare the space efficiency of the intended project with similar built vessels. A quick general study can be made already on basis of first arrangement drawing but more detailed conclusions require also more work with the arrangement. On the other hand the approach can be turned upside down, space reservations for different systems, equipment and functions can be made on basis of existing statistics. Programs based on statistics offer a good, basic tool for both of approaches and for quick checks of different options.

Alternative technical solutions may also offer valuable space savings and impact on the complete configuration of the ship should be carefully considered.

Passenger cruise vessels

Cruise vessels have developed rapidly during the last twenty years. The increase in vessel size is impressive. 85.000 grt has for a long time been the upper limit of cruise vessel size. The first mega size cruise vessel was the 101353 grt Carnival Destiny delivered in 1996.
The total number of such mega cruise vessels, built or on order is now 9.

Figure 1-56 shows vessel price counted in US $ per gross tonnage. The statistics for vessels to be delivered between 1996 and 2002 is assuming an inflation rate of 2%.
Relative costs decrease clearly with size although the curve flattens out at above 100,000 grt.

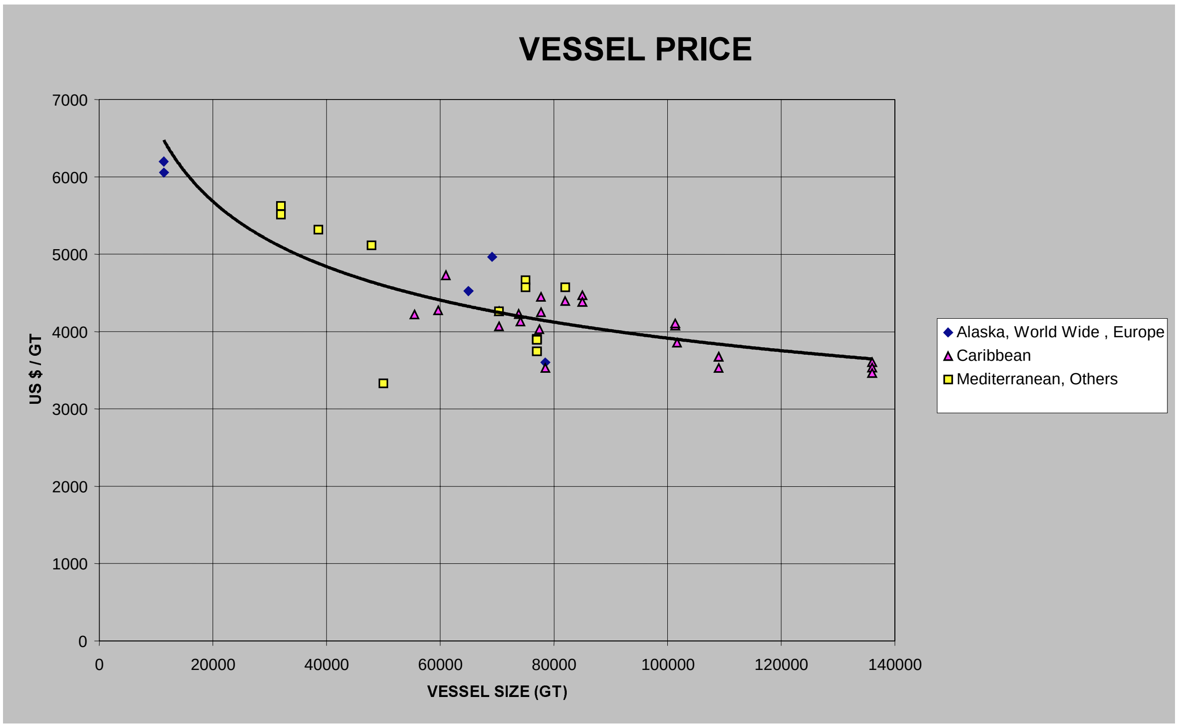


Figure 1-56 Cruise vessel price

An interesting point is that the operational area does not affect the price level. The difference between vessels built for different market segments is perhaps not in more expensive solutions as such, but in space per passenger. Figure 1-57 presents tonnage per passenger as a function of year of delivery of passenger cruise ships.

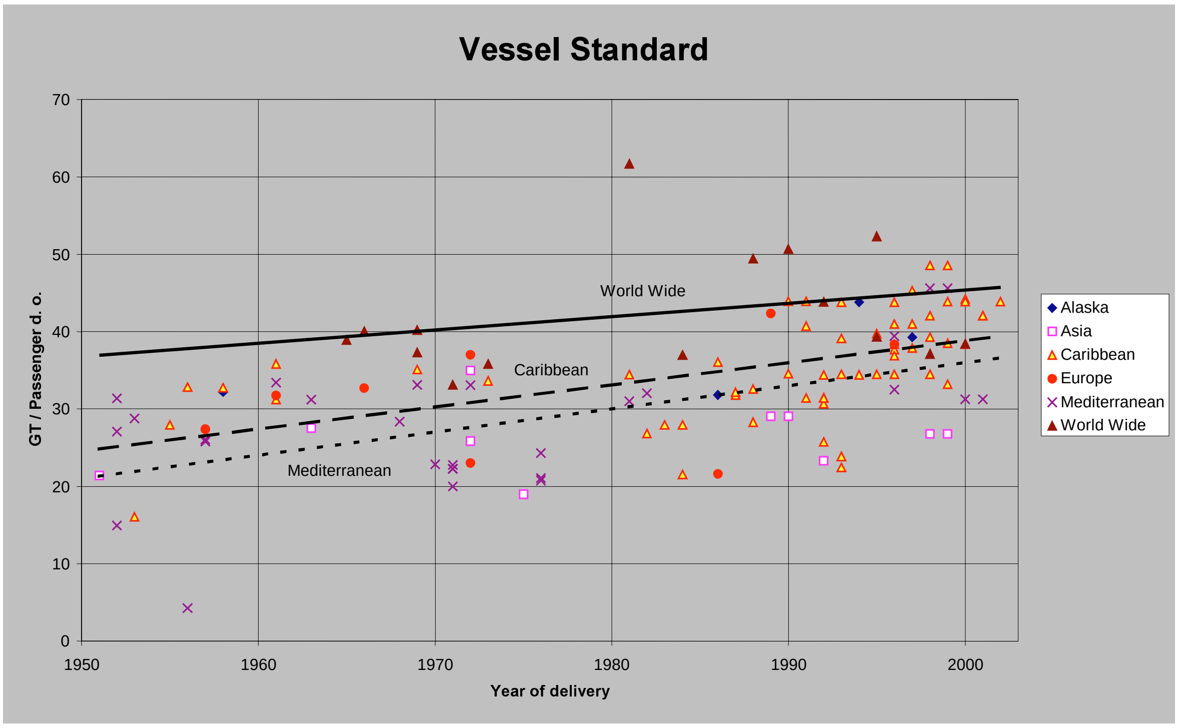


Figure 1-57 Cruise vessel standard

Historically the trend towards more space per passenger is clear. It can also be noted that space per passenger is about 10 grt more on vessels intended for world wide operations compared to vessels intended for the Caribbean or the Mediterranean area. The difference between Caribbean and new Mediterranean cruise vessels is small. The practice of using the same vessel in the Mediterranean area in summer and in the Caribbean area in winter is thus very feasible.

Gross tonnage clearly presents the volume of a ship, and is perhaps not what the average cruise passenger thinks about when he walks along the ship. But he will most certainly know if his cabin is spacious and if there are enough public spaces. Figure 1-58 shows cabin size versus public area of some recently built cruise vessels. The horizontal axis shows the public space area divided by passenger number (double occupancy), and the vertical axis shows the average cabin size. The higher standard of world wide cruise vessels can be seen.
This is very logical as passengers on longer cruises appreciate more space, and obviously can afford it.
The Disney vessels are surprisingly spacious, considering that they are intended for short cruises, whereas Oriana for example is within the average figures only.

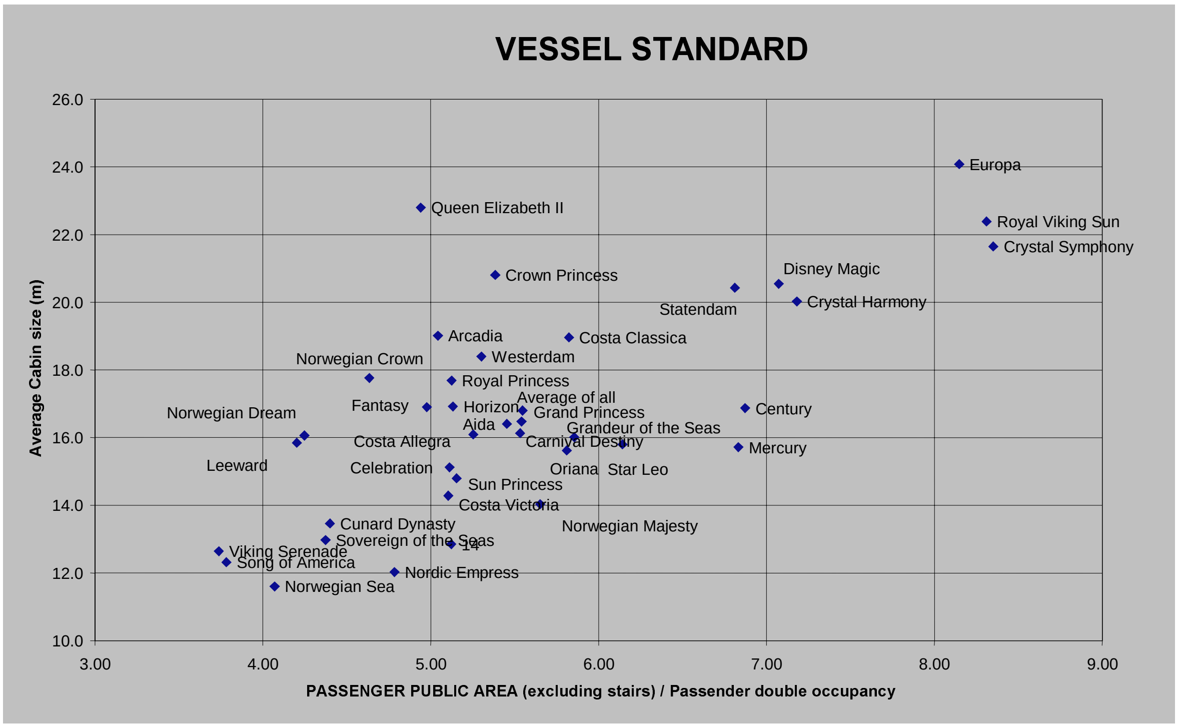


Figure 1-58 Cabin size versus public area in some recently built passenger cruise ships.

Statistical methods can be of help in the project development work, and especially in judgement of the result. For example when designing a Caribbean cruise vessel with 2000 passengers, the database can tell you that a typical total air conditioning room area for the vessel could be 3200 square meters but a more compact solution can be made with a 2500 square meter area.
This 700 square meter difference could be used for 30 more cabins!

Combining the database approach with modern expert systems and virtual reality 3D computer models is most probably the future tool for ship designers.

Figure 1-59 shows the relative area used by passengers (cabin area and public area divided by vessel size (grt)). This is a rough measure of the vessel efficiency, revenue making area divided with total area. Extremely big differences can be found, some of the recently built vessels are located down on the scale and the general tendency seems to be towards less efficient space utilisation when the size grows, a challenge for the designers.

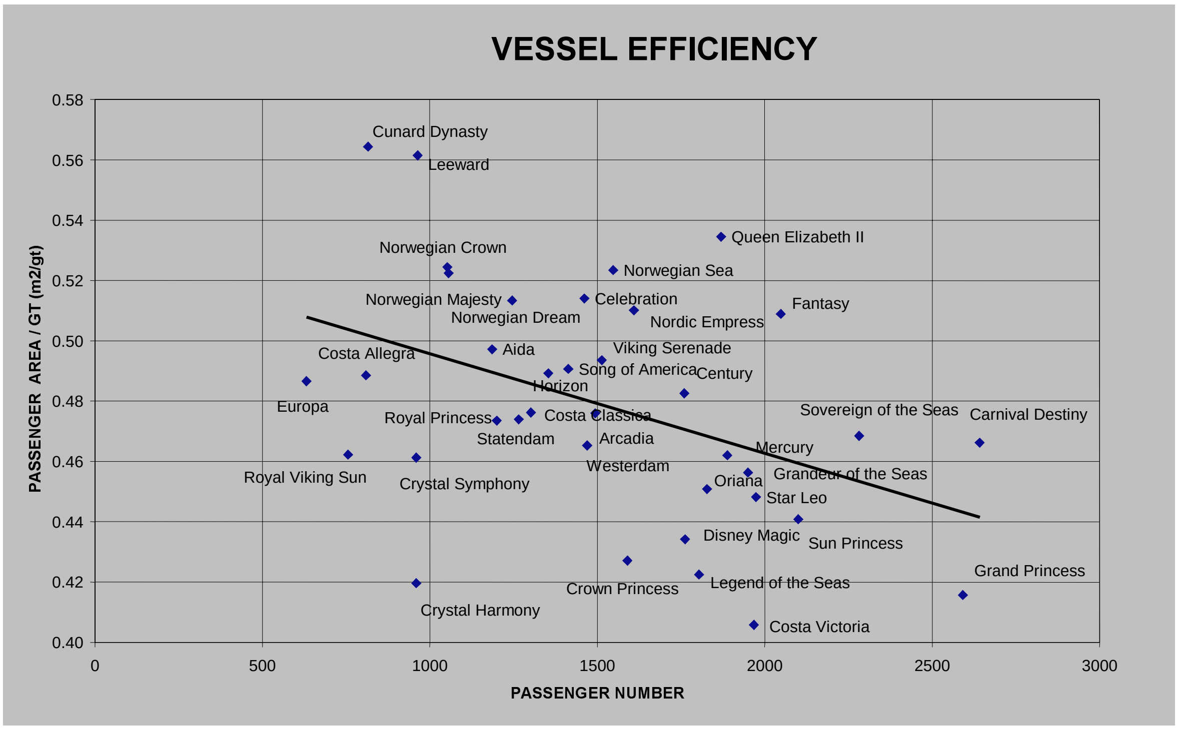


Figure 1-59 Efficiency of general arrangement for some recently built cruise vessels

2. Design Experience

2.0 General

This chapter describes some practical experience to verify the guidelines presented in chapter 1. It also gives detail insight into project management tools and into technical project development. Selected references are all prototypes concerning the vessel itself, its technical solution or the way the project has been managed.

2.1 Project Management

<u>General</u>

For completing successfully a project it should be managed properly, considering costs and schedule, based on agreements and technical specifications.

Project Management should have an active role. It is not enough to know afterwards where and why mistakes were made, but risks and possible problems must be considered beforehand and be prepared accordingly to take care of corrective actions.

Project management is discussed in this chapter considering typical ship engineering and design projects.

<u>Characteristics</u>

A proper starting point for any kind of project management task is to have adequate management hours reserved in order to take care of the complete project successfully.

Management includes work of project manager, sub-managers, secretary and of course meetings, on top of the management required for each discipline and task.

This is a big part of the complete management task and should not be forgotten, in which savings may become costly later on. Depending on the scope of work and type and size of the vessel the number of required management hours vary a lot. Table 2-1 presents some typical numbers as percentage of the complete required engineering hours. Typical numbers of required documents are shown as well.

Table 2-1 The number of required management hours

SHIP TYPE	Hours	Project Design	Basic Design	Detail Design
TANKER 50 000 dwt	Design Hours Management	500 10 %	15000 20 %	80000 12 %
CONTAINER 700 TEU	Design Hours Management	500 10 %	10000 20 %	50000 12 %
RO-RO 1 200 m	Design Hours Management	500 10 %	10000 20 %	40000 12 %
FERRY 500 pax 2 500 m	Design Hours Management	1000 20 %	25000 25 %	150000 15 %
CRUISER 2 000 pax	Design Hours Management	1000 25 %	100000 20 %	500000 15 %

The presented hours are average and typical ones and may vary depending on the complexity of the design and required modifications during the process.

Quality Assurance (QA)

The basis for good project management, as for the whole company as well, is a quality system built-up as a continuously developing process. It forms the steady foundation on which it is easy to build the procedures and regulations for the project management. Project management based on quality management starts with the commitment of the top management of the company and their setting the example, shows the quality thinking as their tool of management.

Figure 2-1 presents a typical quality system of a consulting and engineering company.

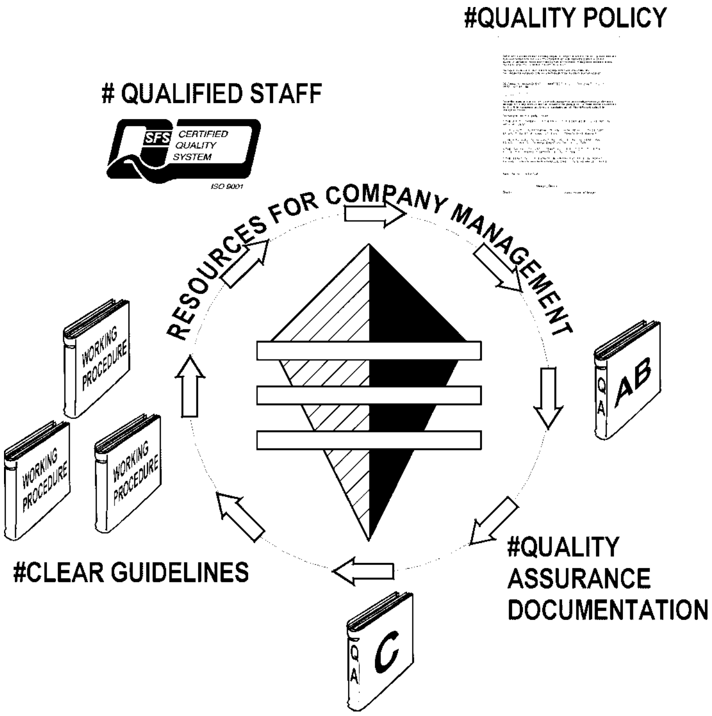


Figure 2-1 Typical quality system of a consulting and engineering company

The quality system consists of the following parts: quality policy, quality thesis of the company, QA-AB manual, documentation of the quality system, QA-C manual, quality plan for each specific project and work procedures, procedure descriptions for each specific discipline and task.

The quality system should be approved and continuously audited and followed up by an external quality auditor.

The QA-C manual plays an important role for the project manager, describing work procedures and instructions how he can build-up his project management system.

One of the most important tasks - if not even the most important - of the project manager at the start-up stage is preparing the project plan/quality plan.

Table 2-2 gives a list of contents for a typical quality plan.

Table 2-2 List of contents for a typical quality plan

Contents	
1.	Scope of work
2.	Organization and communication
3.	Schedule and drawing list
4.	Work breakdowm structure and hour report
5.	Project reviews - contract review - design review
6.	Project meetings
7.	Checking of drawings
8.	Filing
9.	Reports, source and progress
10.	Document info, mailing and copying
11.	Modification procedure
12.	Quality control
13.	Cad and data transfer
14.	Confidentiality

The first step is to agree upon the project plan with the customer. After that it is the project manager's tool to supervise his project ant to ensure that the customer's requirements are fulfilled according to the contract.

<u>Planning</u>

Contract review and project evaluation is the first thing to start with the project team.

Basic characteristics of the project are defined including main information of the vessel, scope of the work and main items of the contract. All related documents are listed and copied as necessary.

Project manager is responsible for the project supervisor or for the management group of the company.

Project manager with his project group is taking care of accomplishing the project. Discipline managers and project secretary are further key people.

Figure 2-2 presents an example of a project organisation with key-people and main responsibilities. It is a project based organisation not a line based.

Customer contact persons as well as other important partners are to be shown in the organisation chart as well as contact levels.

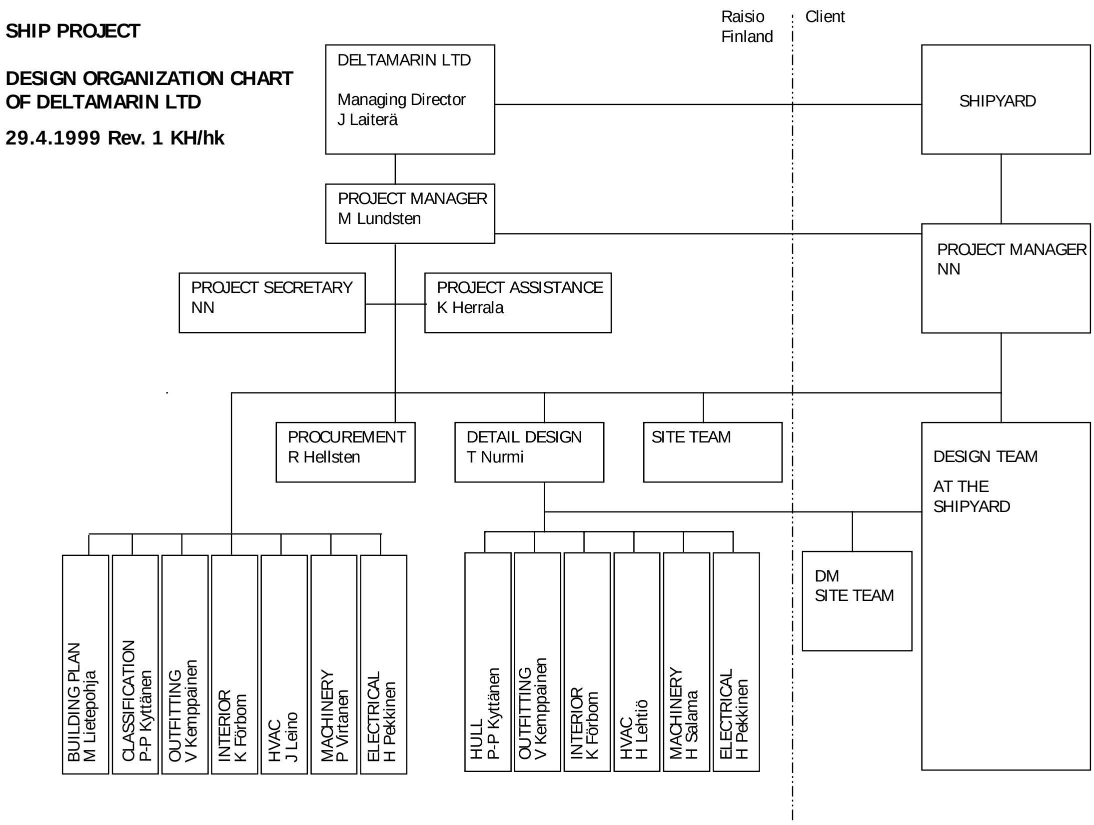


Figure 2-2 Typical project organisation

Project schedule is presented as bar charts, with information of the total design time, start and end dates, of the time for each discipline and each document or group of documents, and responsible designer for each document, dates for main events (milestones) as delivery date, feed-back and scheduled meetings.
Figure 2-3 presents an example of main project schedule.

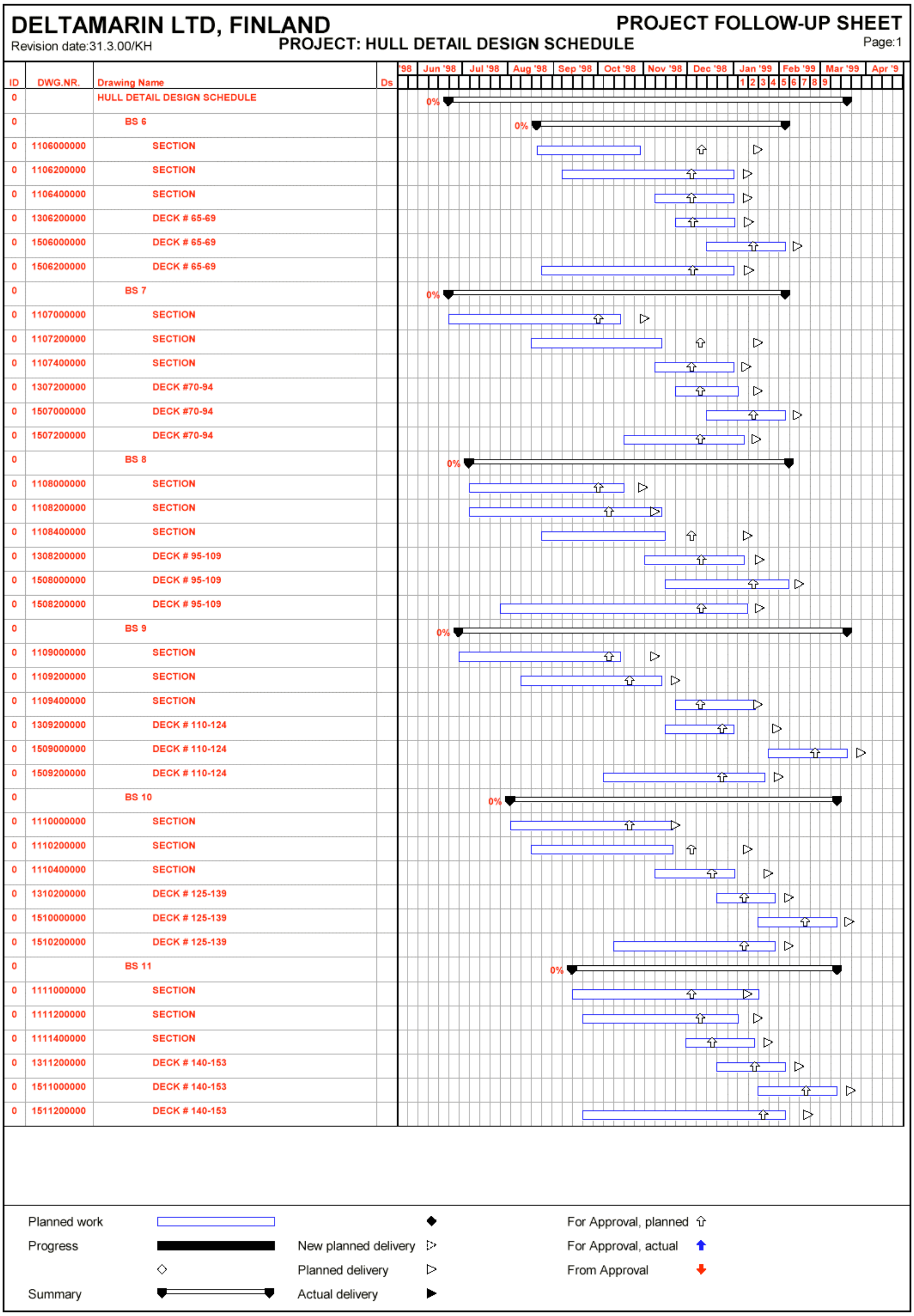


Figure 2-3 Typical project main schedule

S-curve with planned progress and man-hours in a very important tool for the project manager to follow up the general progress of the work.
Figure 2-4 presents an example of S-curve prepared at the planning phase.

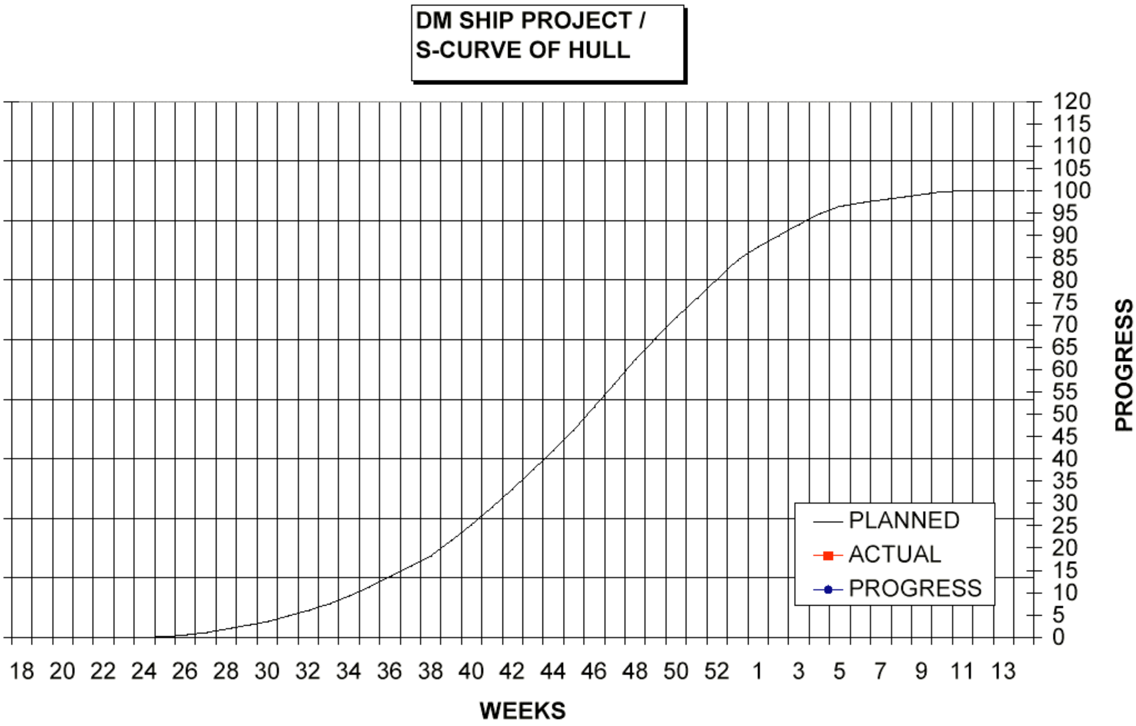


Figure 2-4 Typical S-curve of planned progress and man-hours

Manning plan is made to show the required capacity for each discipline as a function of time, a typical example is presented in figure 2-5.

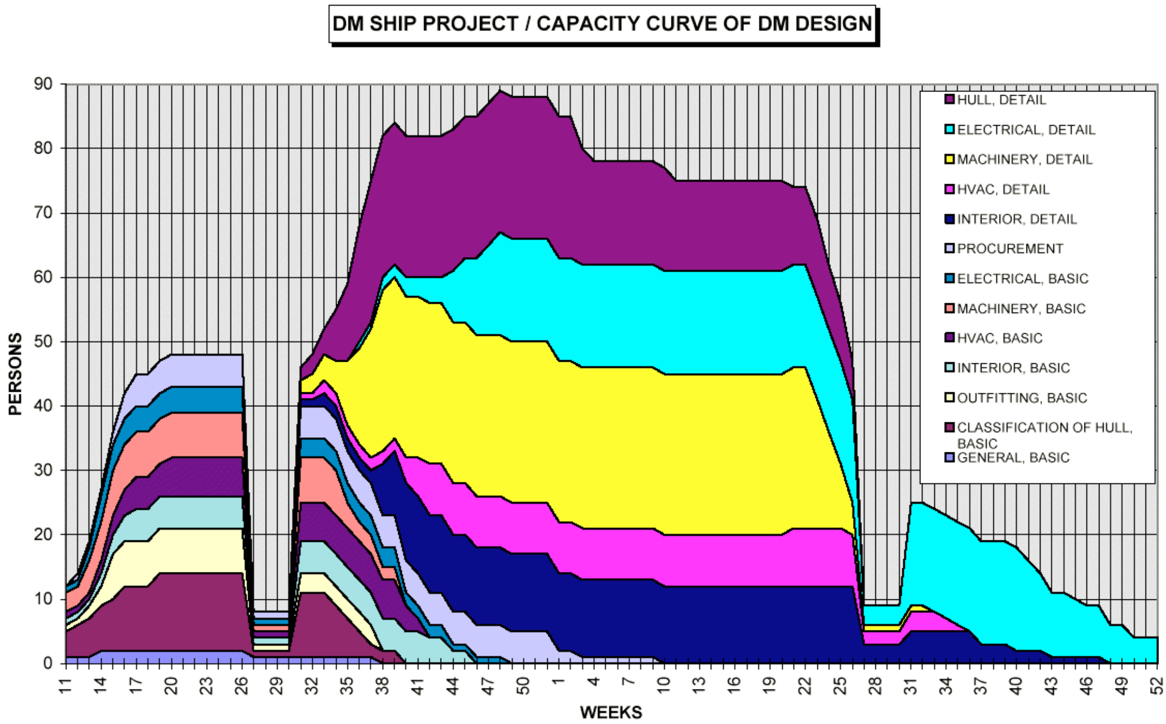
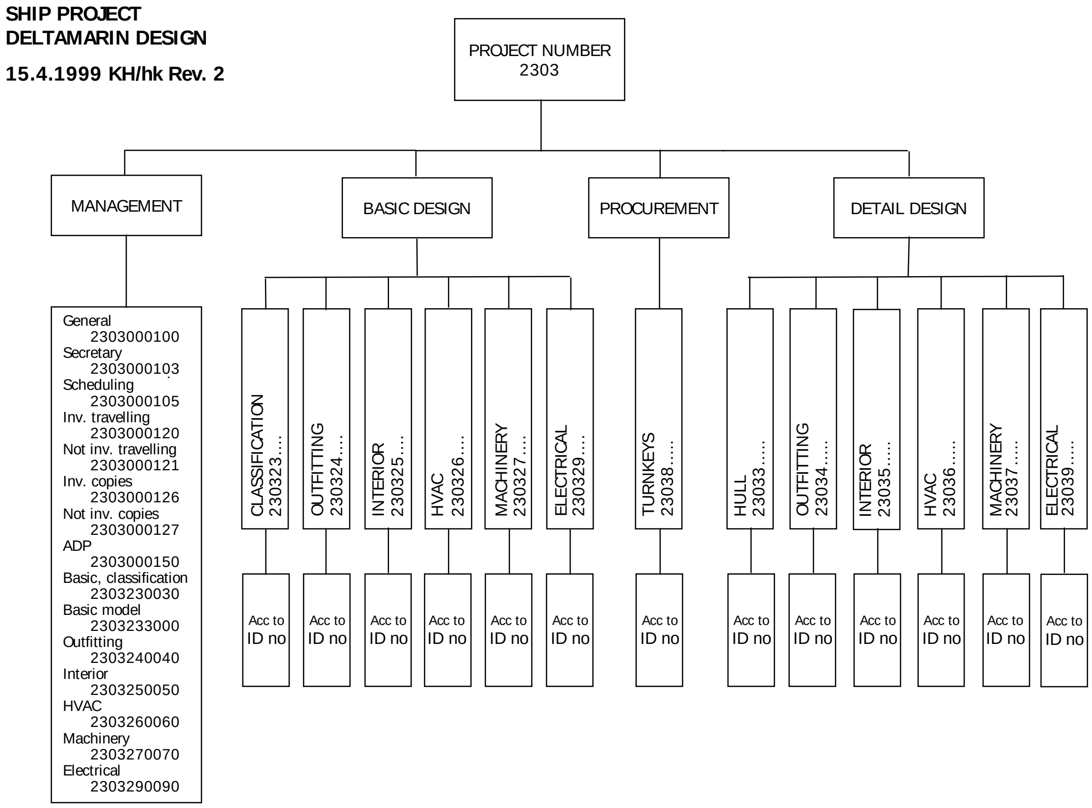


Figure 2-5 Example of project manning plan

Work breakdown structure means dividing the project into parts according to discipline responsibilities, main groups of documents, document numbering system, and any specific project related requirements. Necessary codes for follow-up of design hours and other costs are considered as well. A typical work breakdown structure is shown in figure 2-6.

Figure 2-6 Work breakdown structure for an engineering project



Source data or client information is one of the important project documents to be prepared at the start-up of a new project. The designer needs information to be able to start the design process as well as when proceeding with the design.

It is essential to have all specifications and other contract documents, vendors' equipment documentation, yard standards and it is preferable to have reference documentation as possible.

Information is needed for the systematical follow-up; including system number, document name, description of necessary information, date when requested, needed and received and any deviation remarks.

For managing this information a suitable system is required in order to collect the necessary information, e.g. input for certain design area or missing information. Figure 2-7 presents an example of missing source data list.

Source Data Listing — Date: 20.5.1999

DM Project No: 2303 — Contact: M Lundsten

Client's Project: — Phone: 09 47884420

Scope: 00000 Project — E-mail: mikael.lundsten@deltamarin.com

ID	Rel.	Name	Dwg. No.	Rev.	Appr.	DMR	DMP	DMH	K-P	Exp.	Rec.	Dev.	Remarks
10001	-	Yard standards								9909		-12	
11001	-	Safety Signs, Decks 9, 10 & 11	D.337.4940.4	C		4.3.99				9909	9909	0	sheet 1/6
11002	-	Safety Signs, Decks 7 & 8	D.337.4940.4	C		4.3.99				9909	9909	0	sheet 2/6
11003	-	Safety Signs, Decks 5 & 6	D.337.4940.4	C		4.3.99				9909	9909	0	sheet 3/6
11004	-	Safety Signs, Decks 3 & 4	D.337.4940.4	C		4.3.99				9909	9909	0	sheet 4/6
11005	-	Safety Signs, Decks 1 & 2	D.337.4940.4	C		4.3.99				9909	9909	0	sheet 5/6
11006	-	Safety Signs, Details and LLL in Crew Cabin	D.337.4940.4	C		4.3.99				9909	9909	0	sheet 6/6
11007	-	Emergency Exit and Main Fire Zones Deck DB, 1 -	KE-500-01/98			4.3.99	23.4.99			9909	9909	0	sheet 1/5
11008	-	Emergency Exit and Main Fire Zones Deck 3, 4, 5	KE-500-01/98			4.3.99	23.4.99			9909	9909	0	sheet 2/5
11009	-	Emergency Exit and Main Fire Zones Deck 6, 7, 8	KE-500-01/98			4.3.99	23.4.99			9909	9909	0	sheet 3/5
11010	-	Emergency Exit and Main Fire Zones, Deck 9, 10,	KE-500-01/98			4.3.99	23.4.99			9909	9909	0	sheet 4/5
11011	-	Emergency Exit and Main Fire Zones, Profile	KE-500-01/98			4.3.99	23.4.99			9909	9909	0	sheet 5/5
11012	B	General Arrangement: Deck 0-2	41-00-004	Post		12.4.99				9909	9914	-5	
11013	B	General Arrangement: Deck 3-5	41-00-003	Post		12.4.99				9909	9914	-5	
11014	B	General Arrangement: Deck 6-8	41-00-002	Post		12.4.99				9909	9914	-5	
11015	B	General Arrangement: Deck 9-11, Seite				12.4.99				9909	9914	-5	B
11016	-	Tank Plan	KE-PV1200-0	Prel.		18.4.99	19.4.99	20.4.99			9915		preliminary
11017	-	Fluchtwege Berechnung	amtw			19.4.99					9915		
16001	-	SFI-Baugruppenverzeichnis				18.4.99					9915		23.10.98
30001	-	Measure Drawing	D.337.3300.3	C		4.3.99	23.3.99	9.3.99		9909	9909	0	

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Page 1 of 20

Figure 2-7 Example of missing source/client data list

Filing system

A major engineering and design project includes thousands of produced documents, requiring a lot of source data and other managing information. For managing this vast amount of information a comprehensive filing system is required. Some documents are produced only as CAD-files others in paper copies. Correspondence, memorandums, etc. need to be filed as well.

Typical files for a major design project include basic information, (yard's standards and information), source data / client information, inspection copies (own documents with markings for corrections etc.), delivered documents, approved / commented documents and correspondence (covering letters, faxes, etc.).

Each document in the file must have a dedicated number, different colours in different files and each document file should include a list of contents. When material from the files is removed a "borrower's card" must be used showing the location of the document.

CAD-files are easy to be arranged in a main computer system according to disciplines, where at least subdirectories should be arranged for ready, delivered and update status of each document indicating also inspections and revisions.

<u>Potential Problems</u>

It is advisable to evaluate at the very early stage what kind of risks and potential problems a new project may include, such as new rules and their interpretations, missing or delayed source data, excessive time for feed-back, underestimated required capacity, new technical solutions and configurations, prototype equipment etc.

Possible corrective actions for each item should be carefully thought in beforehand.

And if a problem arise a short and quick response system should be planned.

<u>Follow-up & Reporting</u>

A regular follow-up is arranged on weekly or fortnightly basis at least for progress of work, in percentage, used design hours, percentage of hours, milestones, source data, separate open questions and possible problems and modifications.

Results of follow-up are reported, usually monthly or fortnightly. S-curves and other graphical form of showing information is used. Figures 2-8 and 2-9 present typical examples.

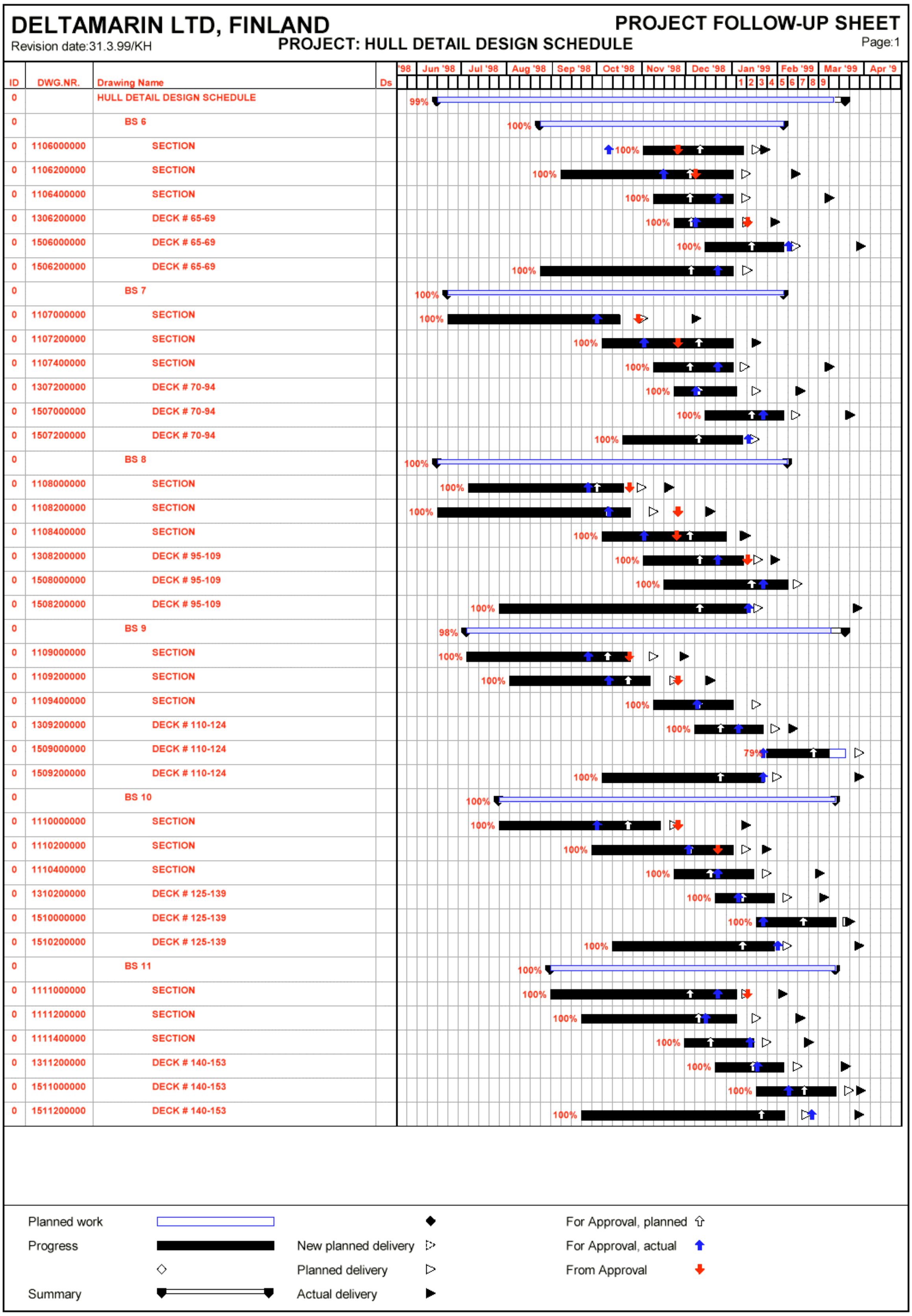

Figure 2-8 Example of document follow-up system

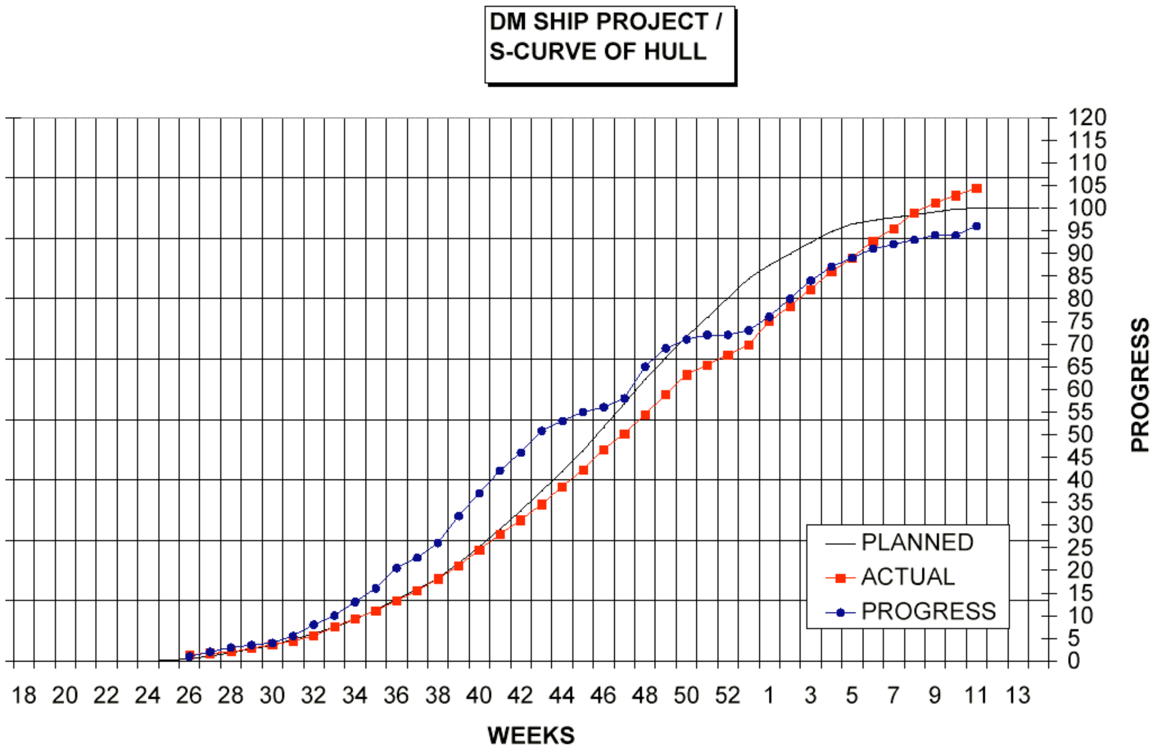

Figure 2-9 Example of S-curve for progress and man-hour follow-up

An example of resource follow-up is presented in figure 2-10.

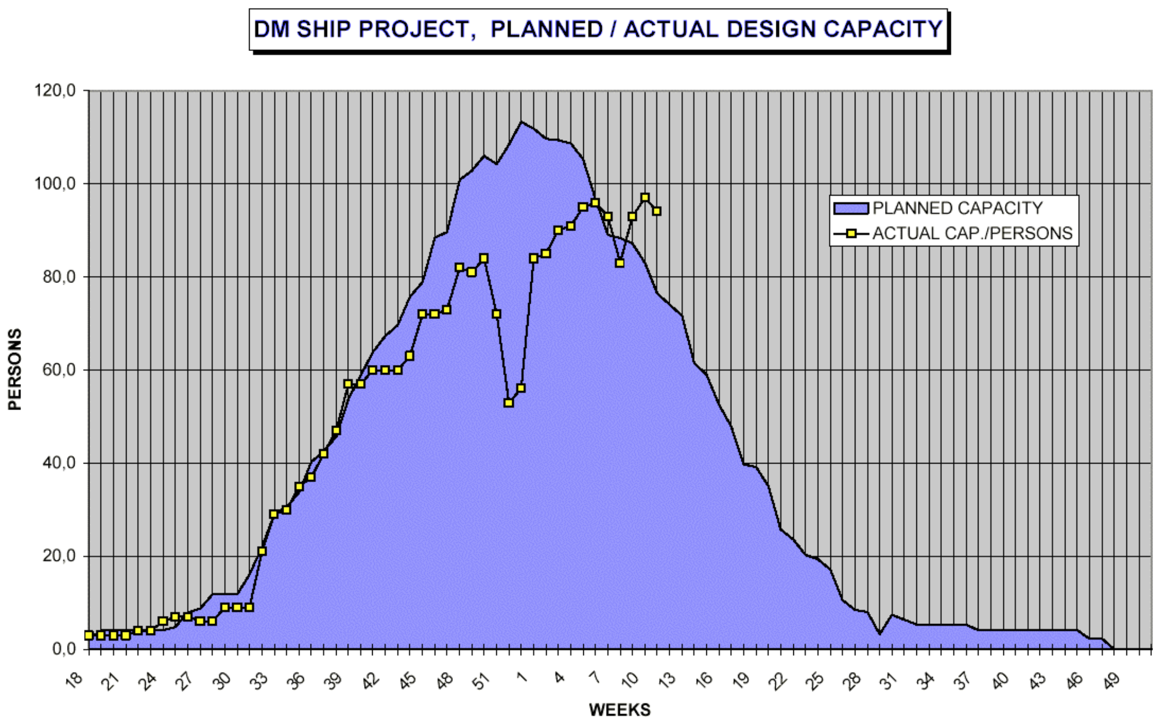

Figure 2-10 Example of resource follow-up

Modification Management

Usual updatings due to normal iterative process and modifications due to changes should be handled separately.

Updatings should be done in a reasonable way. It is not advisable to correct immediately all the typing errors and minor mistakes, which are not significant from the performance point of view, especially if all inspectors have not yet given their feed-back. Otherwise it may be, that 7-8 updatings are made instead of normal 1-3.

Each modification is reported for the client, including at least: reason for modification, effect on schedule, costs, weight, stability and any other specific requirement.

Modifications should generally be handled centralised via project manager, not between individual designers and inspectors.

Modifications should be agreed without delays, minor ones in a week, major ones in two weeks time.

Design should not be modified without an agreement in beforehand.

Summary

Closing a project should be made with a proper evaluation and preferably in a report form and at least partly with the client. Project feedback and experience is valuable statistics.

Check list for the project manager, including all the essential management tasks is presented in figure 2-11.

Figure 2-11 Check list for project manager

The project manager has to know the theories and also the tools of project management as well as how to use them. Yet this is not enough, as the most important quality of the project manager is to know how to lead his team. The “chemistry” of the project manager has to work in two directions, not only with the customer but also towards the project team.
It is easy to get people to work 7,5 hours a day but to get the team to fulfil customer’s requirements in time and with top quality requires top management skills.

2.2 Project Control System for a Turnkey Supply

General

An order for two Ro-Pax ferries with delivery at the beginning of 1998 was placed by Superfast Ferries at Kvaerner Masa Yards in Turku. The building schedule was very tight with both vessels to be delivered almost simultaneously (only 7 weeks difference). The yard split-up the ships into a number of Turn-Key areas of which Turun Prosessiasennus Oy (TPA), a company specialised on turn-key contracts, got the contract for all the ro-ro deck areas, main portion of the complete ship. Everything in the ro-ro deck areas was included except the ro-ro equipment. In this project Deltamarin was a sub-contractor to TPA, responsible for the design and the production and project control system. Figure 2-12 describes the area covered by the turn-key contract.

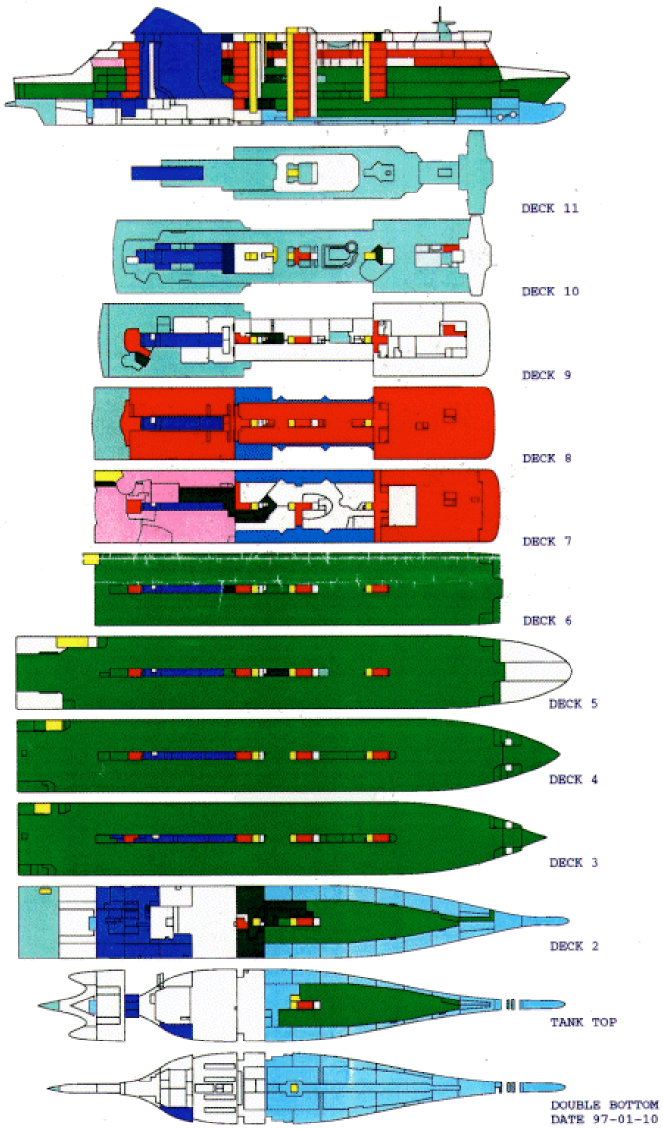

Figure 2-12 Turn-key contract area for Turun Prosessiasennus in Superfast ferries III and IV

The idea to introduce design control system also for production control and follow-up was coming from the experience in earlier similar projects with the same yard. TPA company management had a clear need to create a system with which it was possible to follow the project in "real time" and to have a reporting tool at the same time towards the yard. It was also from the beginning clear that some major activities will be sub-contracted to different companies and the progress of these companies had to be measured with the same tools. An other main concern was the great amount of steel blocks becoming available for outfitting work almost at the same time (two vessels almost simultaneously). The work and amount of labor had to be controlled carefully and continuously. The system had to enable also to identify possible peak loads so that necessary steps could be taken by the project management to reserve personnel and material.
Main features for successful project follow-up and control must include control of the project to follow the master schedule of the yard, checking of the progress proceeding weekly, making sure that the hours reserved for the job are not exceeded and reporting of work progress to the yard every second week.

Method

As basis for the project control and follow-up the system as described in chapter 2.1 was utilized. The project work break down structure, main areas onboard, was made similar to the yard work break down structure in order to make the reporting to the yard more easy and understandable. Further the project was split into logical groups and sub-groups so that each work content could be identified and checked separately.
When defining the work break down structure following items where notified:
yard area division, sub-contractors involved by TPA, works that could be identified and "measured" and works for which TPA wanted to collect statistics for later use where notified.

Input Information

The TPA internal time schedule and follow-up was prepared based on yard master schedules like steel production, block outfitting sequences and testing schedules. The aim was to plan the works to be done in the most favorable positions in order to save money and personnel efforts but anyhow to guarantee that the jobs are done in right sequences. Also critical work sequences, like installation of big items, were identified and incorporated to the schedule.
The main items can be highlighted including master schedule and requirements of the yard, time required to handle specified works, possible critical milestones,
input and requirements of sub-contractors and information and connections available concerning neighbouring areas

Documents

The main document is the project control follow-up sheet where all activities are listed either as work bars or as milestones. This is the basic document where all information is registered. The progress is indicated by a black bar which shows the percentage of work carried out. This percentage is then related to the status line which is date orientated. By this combination it can easily be checked which works are in schedule and which not.
An other important document is the S-curve in which following information is collected: planned work, cumulative progress and cumulative hour consumption. Knowing the background of the curves and how they normally behave, it is even possible for the management to make 1-3 months forecast how the project will continue. If the work break down is made correctly it is even possible to identify possible deviations before they reflect on the total project. This enables the project management to make corrective actions in time when needed. Figure 2-13 presents a typical S-curve of the project.

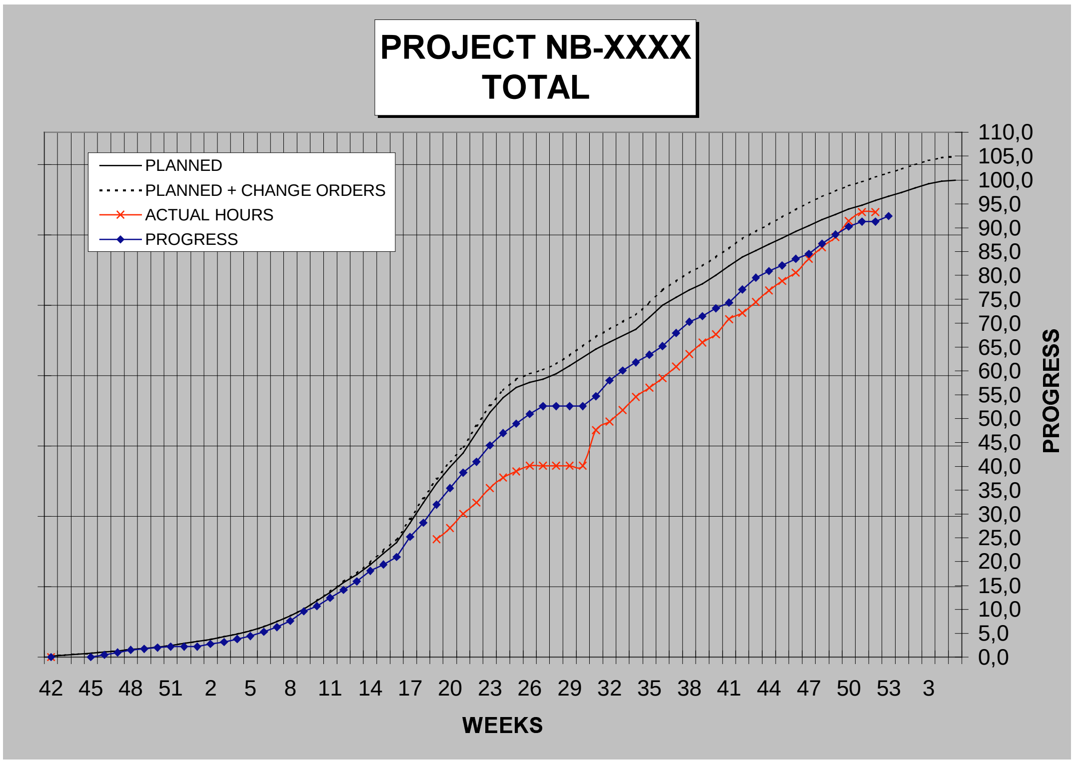

Figure 2-13 Typical S-curve for project control and follow-up

Summary and Results

To arrange a good project control and follow-up system some key information, sometimes classified, of the project is required. The need to arrange a project control and follow-up system was anyhow notified by the subcontractor and therefore the persons involved from different parties where motivated to cooperate and give necessary information.

The strategy of the turn-key company was to use sub-contractors for all the jobs which are not directly related to own know-how. When including these subcontractors into the follow-up system it was noted that some of them never had prepared any kind own schedules. A centralised follow-up system turned out to be the only way to keep all these companies informed when to start the jobs.
In the beginning the foremen where somewhat restrictive against the system due to the fear of additional bureaucracy. During the project it came obvious that there was in practice no additional works required, only a different way of reporting. The foremen must anyhow have a knowledge of what is happening during the week and now it was only to make this information uniform and possible to use by an outside person.
The turn-key sub-contractor was able to follow-up the progress exactly at their own workshops, at subcontractors workshops and at the yard. This was also reflected in the reporting and it was easy to detect reason for any changes, modifications, delays etc. and possible claims both sides were easy to control.

The work proceeded almost as planned, deviations became from non-finished basic design documents, from non reported changes and modifications and from missing material and equipment. This was reasonably easy to control and to prepare corrective actions.
For the first time all the outfitting work including painting and insulation were carried out already at the panel line. None of the outfitting works were left for the ro-ro decks, and works could be carried with proper working methods on the block and not upwards from the scaffolds. A lot of working hours and time were saved. Both vessels were delivered in time.

2.3 Project and Basic Design of 5400 m^3 Chemical Parcel Tanker

General

Prior to the design agreement with INMA SpA, La Spezia, Italy, there were various studies prepared for Stolt Parcel Tankers Inc., USA for the complete configuration of the newbuilding project and especially concerning machinery and propulsion system. Conventional diesel geared machinery was compared with diesel electric with single shift and even with twin thrusters. More detailed descriptions are presented in chapter 3.

When Stolt Parcel Tankers Inc ordered six 5400 m^3 Chemical Parcel Tankers from INMA, Deltamarin was selected to prepare the final project and basic design.

The contract was made based on general arrangement with conventional diesel-geared drive but the technical specification was defining diesel-electric machinery and propulsion. Therefore the work was started by definition of the final main dimensions, speed power estimation and lightweight calculation. The development on engine room arrangement was also in high priority since the compact engine room with diesel generators on the main deck located next to the transom allows maximising the cargo capacity within the limited length of the vessel. Table 2-3 describes the contract requirements and the final selected main dimensions.

Table 2-3 Main Dimensions for a 5400 m3 Chemical Parcel Tanker for Stolt

	Contract Specification	Final selected
Length overall	94...96 m	96 m
B moulded		16.2 m
B extreme	16,0...16,40 m	
Draught moulded, design	6.00m, max	
Draught scantling	Max load line + 1.0 m	6.4 m
Dead-weight, design draught	4300 tons	
Dead-weight/minimum freeboard	5200 tons	5300 tons
Cargo tank capacity (16 tanks)	5400 m^3	5400 m^3
Speed ,85% MCR , 15% sea margin	12.5 knots	12,5 knots
Number of diesel engines	3 - 4	4

Basic Design

When preparing the design agreement there were some key items which were included in the contract and proved to be very important to successful performance of the commitment.

Outside designers' project manager was participating in the different negotiations especially between the owner and the yard and had therefore clear understanding of the yard's requirements.

Typical reference drawings were enclosed in the design agreement, yard had clear understanding what will be the standard of delivered documents. Less consultation was required when delivering design documents.

Drawing list with preliminary schedule was enclosed in the design agreement. Each document was provided with "weight value". Weight value was the percentage of the total contract price and it was estimated for each individual document.

Penalty for delayed delivery was agreed upon. Penalty was related to the agreed "weight value".

In addition to the typical project design tasks such as definition of main dimensions, general arrangement, midship section, tank plan, electric load balance, cargo deck arrangement etc. a complete list of basic design documents were prepared. Final hull form was developed and model tests were coordinated.

List of deliverables included all safety documents, all loading plans and calculations, outfitting plans, machinery systems, structural documents and electric systems.

The designer had the responsibility to take care of all the approvals from the owner and classification societies (double class).

It was agreed upon on general level that direct meetings between the designer and the owner, classification societies, and suppliers, were to be avoided unless specifically agreed upon with the yard. Correspondence between the designer and other related parties was going via the yard.

However, to perform successfully basic design task in limited time frame (less than four months for the major part) it is necessary to have certain amount of discussion with possible suppliers in order to receive necessary technical data quickly enough. Therefore it was agreed upon that the designer could contact the suppliers directly in order to be able to proceed with the design work. The yard was informed in advance and all correspondence was submitted to the yard and the issues considered with the suppliers were limited to the technical items only.

<u>Follow-up</u>

The basic design was carried out according to Deltamarin "Basic Design Work Procedure" and QA-system. Quality assurance C-manual was prepared for the project.

All the participants were provided with the C-manual and the designers were provided with the target design hours for each individual task. The hour reports were divided into individual drawing level.

The progress was followed up weekly summing up the progress of individual drawings and the used hours. The sum curve and progress of individual drawings were provided to the yard as well.

Changes

There will be always unforeseen changes in project and basic design task like this.

Changes are necessary and positive phenomena, they are needed to improve the design and can not be avoided when designing prototype i.e. diesel electric powered small chemical parcel tanker with a completely new arrangement configuration.

A well defined procedure is needed to describe how to handle these changes so that the positive impact of the changes to the whole configuration will overrule the negative attitude of possible additional cost of changes.

In the project C-manual it was described the procedure for reporting the changes with a formula called **"**Additional work / Project Modification**"**. This procedure proved to be very valuable as it forced the designer to carefully check the impacts of the change and document them properly. It was also much easier to receive owner's and yard's decisions with a well prepared modification report. Figure 2-14 presents a typical report. This report describes however a non-typical change: lengthening of the vessel with two meters. This was a result of speed/power estimations. Relatively short vessel with high block coefficient and high Froude number typically introduces high resistance and propulsion power required. The length restriction was checked in detail by the owner and additional two meters could be added. Propulsion power requirement dropped with 4-5%. A simple pram type stern was selected for the aft ship hull form, and model tests confirmed exactly the speed/power estimations, contract specification could be met.

Additional work / Project Modification

REPORT NR:3785/4,

PAGE 1/1

26 April, 1996

TITLE: **LENGTHENING , PROJECT 1684B**

LOCATION: VARIES

DRAW. NR:		
	00.00.10	**LINES AND BODY PLAN , CHECKING PURPOSES ONLY**
	00.00.18	**PRELIMINARY LIGHTWEIGHT CALCULATION & DISTRIBUTION**
	00.00.19	**PRELIMINARY SHEAR AND BENDING MOMENT CALCULATION**
	00.00.02	**TANK PLAN**
	01.11.03	**MIDSHIP SECTION**

DESIGNER: **EP, KK ,MOV,JHA**
DISCIPLINE MGR:**EP,KK,MOV**

WEEK OF MODIFICATION WORK:
WEEK 19 AND 20, DEPENDING ON THE RECEIVED INPUT DATA FROM MARIN

DESCRIPTION OF THE MODIFICATION:
FORE SHIP LINES ARE MODIFIED AND L_{OA} IS EXTENDED WITH 2m. THE AFT SHIP LINES, CARGO TANK BOUNDARY, WEIGHT, SHEAR AND BENDING FORCES AND

MIDSHIP SCANTLINGS ARE MODIFIED RESPECTIVELY.

REASON FOR THE MODIFICATION:
OWNER CANCELLED THE RESTRICTION OF MAX. L_{OA} = 96 AND THE LENGTHENING OF THE HULL IS STUDIED IN ORDER TO IMPROVE THE POWER/SPEED PERFORMANCE.

EFFECT OF THE MODIFICATION:
ADDITIONAL WORK TO BE DONE BUT LIMITED ONLY TO THE FOLLOWING DOCUMENTS:

00.00.10	LINES AND BODY PLAN, CHECKING PURPOSE ONLY
00.00.18	PRELIMINARY LIGHTWEIGHT CALCULATION & DISTRIBUTION
00.00.19	PRELIMINARY SHEAR AND BENDING MOMENT CALCULATION
00.00.02	TANK PLAN
01.11.03	MIDSHIP SECTION

DESIGN HOURS: **QUOTED FIXED PRICE : xxx FIM**

REASON APPROVED AND MODIFICATION AGREED TO BE MADE ACCORDING QUOTED PRICE

____
I.N.M.A. SPA

____
DELTAMARIN LTD

Notes:____________________________.

Figure 2-14 Typical change report

Novel arrangement

An essential benefit with the diesel-electric power plant is that the machinery can be located freely to suit best for the complete arrangement of the vessel. For a chemical tanker it is essential to maximise the cargo tank volume and thus it was extremely important to cut down the engine room space. As a result the cargo volume with diesel-electric powered vessel is always higher than with diesel-geared propulsion design.

The comprehensive machinery studies showed the optimum number of diesels to be four equal size units.
The engine room layout of this design includes features which are unusual, the diesel generators (main engines) are above the main deck.

To reduce the probability that two different incidents may destroy the power production totally one needs to consider the location of the main switchboard (MSB). Typically the generator and the MSB are required to be located in the same wt-compartment.

IBC-code damage stability requirements requires buoyancy above the main deck in aft the ship to be taken into account. Therefore the progressive flooding of water to the undamaged space separated with aft peak bulkhead must be prevented with WT-bulkhead extending above main deck. As a result the diesel generators and the MSB are in different water tight compartments.

To reduce the probability that two different incidents may destroy the power production totally a longitudinal bulkhead was provided to separate the portside and starboard side diesel-generator rooms. Figure 2-15 presents the final arrangement.

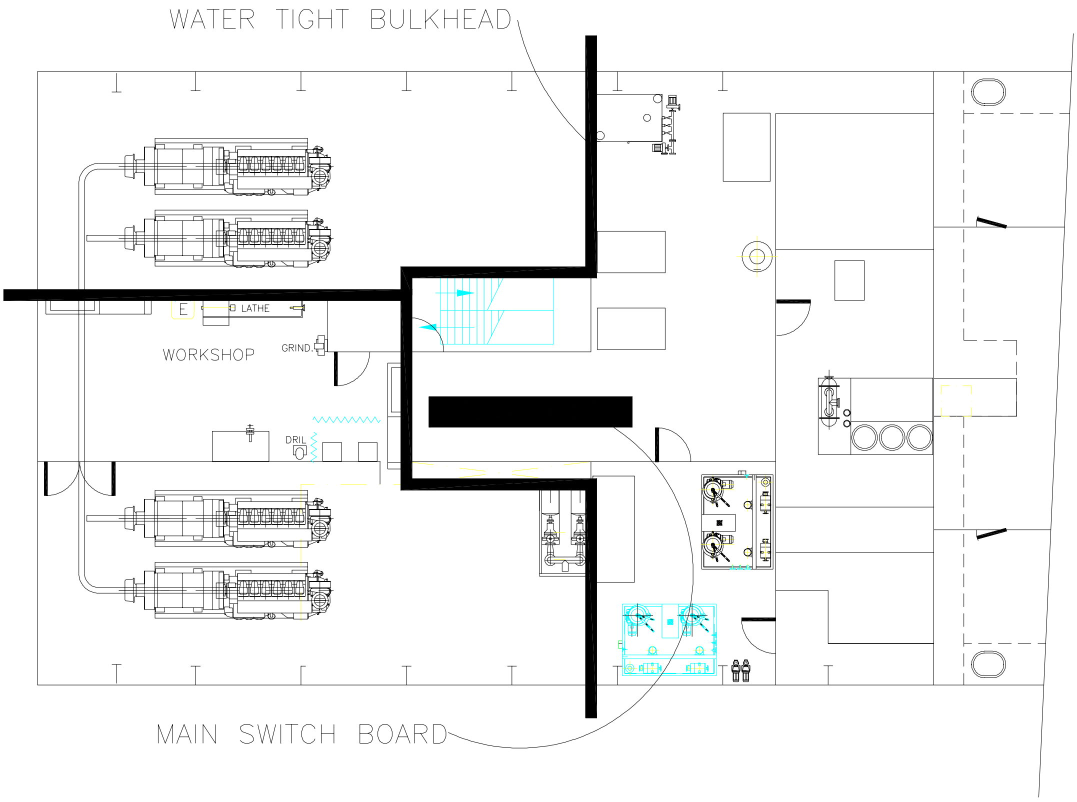

Figure 2-15 Separated engine rooms on the main deck. Diesel generators are in different WT-compartments

The possible damages in machinery spaces were analysed taking into account both flooding and fire.

Flooding studies showed the following results: damage in the wt-compartment with MSB causes a total black out, damage in portside diesel generator compartment leaves MSB undamaged, and starboard side generators running, damage in starboard side diesel generator compartment leaves MSB undamaged and portside generators running. Fire studies gave very much the same results.

With designed wt-bulkhead arrangement only one damage or fire in the main switchboard compartment will cause a total loss of power supply.

Conclusion

The discussions with the owner, yard and classification societies (DNV & RINA) were proceeding smoothly. The documents were delivered according to the agreed schedule. The problematic issue was the delivery of structural drawings of aft ship area. The development of machinery arrangement for diesel-electric powered tanker and the related approval was dictating the delivery of the structural drawings in that specific area. With open discussion and reporting of progress the issue was clearly understood by all parties and necessary adjustments could be agreed upon.

With diesel-electric propulsion the cargo tank volume was maximised.

2.4 Diesel-Electric Powered Ro-Ro Passenger Ferry with Large Lower Hold

New ro-ro ferry configuration based on diesel-electric machinery was developed already in 1990. This concept was picked up by TT-Line as an alternative project for their new Travemünde-Trelleborg vessel in early 1993. The design criteria for the projected vessel were very straight forward but after a more detailed analysis also very demanding: load capacity to be at least 2400 lane meters with minimised main dimensions, length times beam not to exceed 4884, all lane meters to be fully usable, not theoretical, within the intended harbour times, 1-2 hours, fully operable vessel, especially in Trelleborg harbour, on year-round service without any assistance and minimised maintenance with adequate, minimum crew.
It was obvious that a lower hold for trailers was needed otherwise the required capacity was not possible.

Trailers on three decks with such minimised harbour time immediately leads to a configuration with drive through lower hold. Single ramp or lift operation for lower hold were not considered feasible, too much time consuming. This meant that the ramps at aft and forward end of the lower cargo hold should not exceed 7 degrees inclination and to reach fully operable lower hold the length should be maximised.

Alternative locations were considered for the diesel generators and the spaces outside the lower cargo hold, outside the B/5 bulkheads, were found most feasible, each diesel generator in its own compartment. Aft ramp was lead between the electrical propulsion motors starting already at frame 10, 10,80 m forward of transom. Side casings were applied to have the cargo flow down to the lower hold in the middle and to the upper trailer deck on both sides with hoistable ramps.

Before going further with the design the economics of the proposed design were evaluated.

<u>Capital Costs</u>

The capital costs involved in the machinery plant itself were carefully studied for two machinery options: diesel-electric and diesel-mechanical. Basic machinery configuration for both options is shown in figure 2-16, the diesel-electric machinery equipped with five typical generator sets and mechanical option with four geared main engines with shaft generators and three auxiliary generator sets.

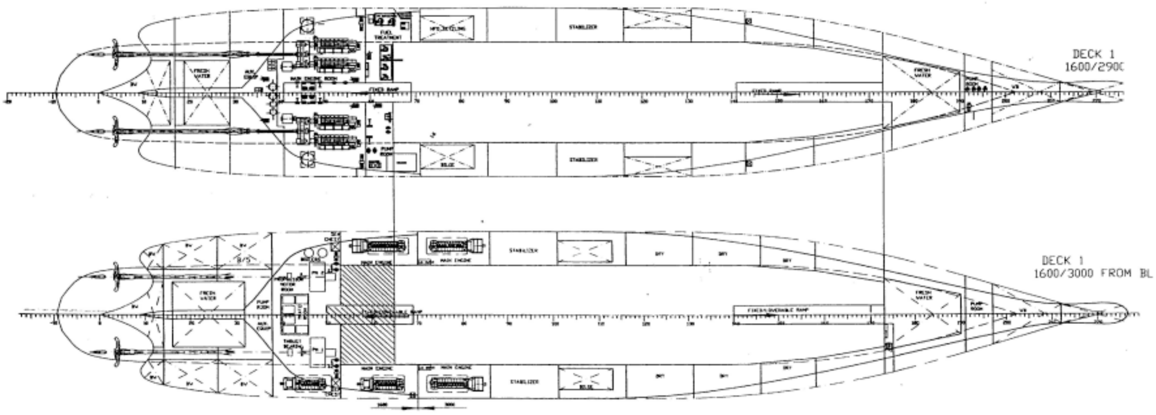

Figure 2-16 Machinery and lower hold arrangement with diesel-mechanical and diesel-electric machinery

Difference in investment cost concerning the plant itself is shown in figure 2-17. Both heavy fuel and marine diesel oil were considered. The first cost difference of the machinery plant, only, was 4,137 MDEM diesel-electric being more expensive, marine diesel options both.

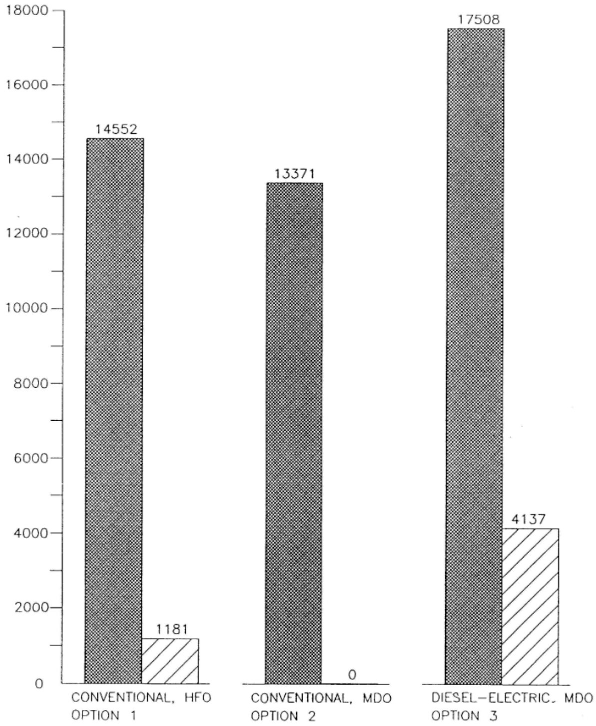

Figure 2-17 Ro-ro ferry, summary: first cost and differences (kDEM/year)

A more detailed study of the project configuration was required enable to create a novel and efficient general arrangement with large cargo hold and give attractive first cost for the complete ship.

It was estimated that items which were not considered in the cost comparison, would minimise or even level out the difference. Installation costs are higher for diesel-mechanical machinery due to bigger amount of machinery and equipment to be installed. Piping system costs are also higher for diesel-mechanical due to the same reason, and interesting enough no major difference in the cabling cost was found. On the other hand the diesel-electric arrangement gave 55 additional lane meters in the lower hold, which means 3,6 additional trailers, see figure 2-16.

The suppliers were able to meet the challenge and further progress was based on 12-pulse transformer connected system with double winding for the motors doubling also the degree of power availability.

The operational design criteria set already at the very beginning were encouraging towards diesel-electric machinery choice: low maintenance cost, low number of diesels, only one type of engine, good overall simplicity, easy control, slow propeller speed when manoeuvring to avoid 'dredging' and FP-propellers.

Annual Costs

Annual costs were compared between the two machinery options, including first costs, fuel cost together with an additional revenue from the additional cargo space. First cost included prime movers, power transmission, ancillary systems and propeller plant, no installation costs were included. Fuel cost included engine operation according to actual service profile, fuel oil heat value difference, difference in the amount of sludge, difference in electric power demand and difference in auxiliary boiler fuel. Figure 2-18 shows the calculated results, calculation is based on eight years life time and 10 % cost of capital, residual value is considered zero.

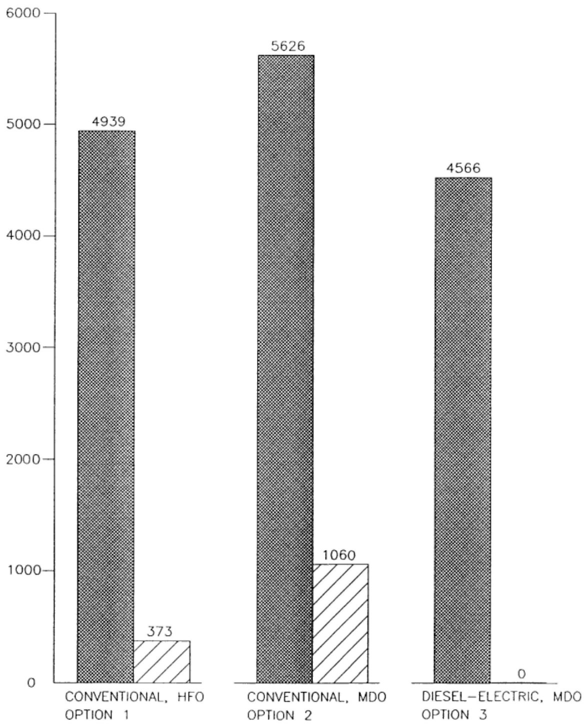

Figure 2-18 Ro-ro ferry, economy summary: total annual cost and differences (kDEM/year)

The diesel-electric option becomes most favourable, simply due to the additional revenue available through increased trailer lanes. Difference in fuel costs is negligible. This comparison clearly shows that one should not make a decision between diesel-electric and mechanical machinery only based on direct machinery related investment costs.

Maintenance

One of the most difficult operating costs to evaluate and quantify is the level of annual expenses attributed to maintenance and repairs, (M&R), diesel mechanical and diesel-electric machineries were compared as well as heavy fuel oil (HFO) and marine diesel oil (MDO) as fuel.

Engines by the same manufacturer were selected in order to avoid differences due to different suppliers. Total installed main engine power was about 18 MW, which is not typically the case: diesel-electric ship has lower total installed power due to the power plant principle.

All data was based on supplier's manuals and information for scheduled maintenance and spares for a machinery operating according to the specified profile.

Results were interesting when comparing diesel-electric and diesel mechanical, both with MDO.
The diesel engines dominate in both M&R hours and costs, 81 % of hours and 90 % of money was spent on engine service. Spare part cost is clearly more determining than service hours.
About 17 % more time is needed for engine service in a mechanical ship. The diesel-electric ship has only one type of engine and lower number of engines and cylinders as well as larger bore engines with more constant engine loading.
M&R work for diesel ancillaries accounts for 7-9 % of the total M&R hours and 5-7 % of the total M&R costs. There are big differences between the different systems typically in favour for diesel-electric machinery, but this has only a marginal effect on the total.
The electrical devices generate 5-11 % of the total service hours and 2-5 % of total costs, mainly due to low spare part consumption.

As a conclusion it can be stated that the difference in service hours, spare part costs and total costs is in favour of the diesel-electric machinery, in total costs abt. 18 %.

The second stage of the study was intended to show the fuel choice related consequences.

The increased complexity due to HFO calls for numerous additional maintenance tasks, especially in fuel systems, but the engine sector is still dominating at an equal portion as in the first evaluation; 80 % of hours and 90 % of costs are due to the engines. HFO brings, however, a significant, 29 % increase in the engine spare part costs.

HFO has also a major impact on fuel systems. A 250 % increase in service hours and 100 % in spare part costs are due to fuel quality. However, this is still a marginal cost, presenting only 1-3 % of total figure.

7 % increase in service hour demand and 19 % higher M&R costs can be expected when specifying HFO instead of MDO.

Damage Stability

The lower cargo hold concept has been widely applied in recent ro-ro passenger ferry newbuildings. The basic idea is to locate the cargo hold within B/5 bulkheads and thus, in principle, to have excluded, undamaged, from all damage cases.

The damage safety characteristics were carefully and thoroughly considered for this new design for TT-Line. The following design criteria were set:

- All two side compartment damage cases to fulfil SOLAS 90.
- Adjacent side compartment together with propulsion motor room to fulfil SOLAS 90 as well as all other two compartment damages aft and forward of lower cargo hold.
- All one side compartment damages together with the lower cargo hold damage also to fulfil SOLAS 90.
- Even in the case of two adjacent side compartments together with the lower cargo hold damaged the vessel to fulfil SOLAS 90 requirements as applied for intermediate flooding stages with applicable permeabilities.
- The above requirement also for the damage case of propulsion motor room, adjacent side compartments and the lower cargo hold!

These criteria were more strict than generally applied for similar vessels. But they were considered to be more in line with the general safety policy of the owner and they also give more margin for further extensions and conversions. See also chapter 1.5 with more detailed information.

The selected design concept includes together with the lower hold watertight side casings on the freeboard deck adequately subdivided to give stability and range after damage.

Fire Safety

The location of diesel generating sets into four different separated engine rooms and propulsion motors in their own compartment is clearly improving the internal fire safety. All these spaces are isolated with fire bulkheads and two of them also have the cargo hold in between.

Design Features

The vessel is operated through aft and forward ramps for ro-ro traffic. There is a simultaneous access to the main deck, to the lower hold and to the upper deck as shown in the principal arrangement drawing in figure 1-42. The length of the lower cargo hold is 56 % of the perpendicular length of the vessel, a world record, accessible with drive through ramps at both ends.

On the main deck seven lanes of 3,1 m each are arranged with the beam of 27,20 m, i.e. 3,89 m of beam required for each lane. Side casings are applied and two rows of pillars/narrow bulkheads to cut the span of the deck, to cut deck into three separate safety areas and to facilitate ventilation into the lower hold. The pillar lines also enables to arrange necessary passage areas as well as important safety barrier in case of cargo shift. Heeling risk due to cargo shift is minimised and has no practical importance.

The structural arrangement leads to a clear benefit concerning steel weight and structural heights. Transverse racking strength is easily supported by the selected structure and no special structures or reinforcements are required.

A lot of attention was paid to the hull form and propeller-rudder arrangement. Pram type hull form was applied with slender shaftlines and bossings. Fixed pitch propellers with outwards turning direction gave the best efficiency. Flap rudders were used to reach high manoeuvrability and controlled stopping in Trelleborg harbour by using rudders. Ribs were installed in the rudders to avoid cavitation erosion.

Model tests showed a power requirement of 10 690 kW for 19,5 knots at draft of 6,0 m (displacement 18 000 m^3, B 27,2 m, L_{PP} 166 m). Full scale trials of both vessels were showing even better results, 10 000 kW only.

The next generation is based on 19 500 m^3 displacement and the power requirement is 10 450 kW at 19,5 knots and 18 300 kW at 22 knots (L_{PP} 170 m, B 28,7 m).

<u>Summary</u>

The pieces for new ro-ro passenger ferry designs are available to meet the increased safety and environmental requirements without unnecessary increase of costs.
The design configuration applied for the new TT-Line ferries is based on diesel-electric machinery to be operable with marine diesel oil. The future requirements for low exhaust gas emissions are met without any extra investment or space required for cleaning devices.

The studies made to compare diesel-mechanical and diesel-electric machinery configuration show that the differences in machinery related investment costs can not be neglected. But taking into account all secondary costs for piping, cabling, installation etc., fuel and maintenance costs and the possibility for additional cargo space the diesel-electric machinery configuration becomes feasible.

2.5 Handy Size Ferries

Handy size ferries are typically used as day passenger ferries and as ro-ro ferries, but even as night ferries with some cabin capacity. The Joint North West European project on Safety of Passenger Ro-Ro Vessels was studying in detail the damage safety, water on deck and possibility of cargo shift. The theoretical studies were verified through example designs and we carried out the design for the handy size ferry. The ferry is intended for short international voyages with a significant wave height of 2,0 m. The main particulars and capacities of the ferry are presented in table 2-4.The vessel is only 102 metres in perpendicular length and 21 metres in beam. Small side casings, about 1,4 m, are designed to give additional buoyancy for damage cases and to help to fulfil the water on deck requirements. There is a lower hold for private cars through the complete feasible length. The main deck is raised in the middle, within the area of machinery spaces, to give additional height for machinery, but also to help in damage and water on deck stability. There are six trailer lanes on the main deck together with stern and bow doors. The lower hold is accessible through ramps at both ends of the hold. The height of the main deck does not allow a full trailer height for the lower hold and on the other hand the drive-through principle for trailers requires a vessel length of about 140-150 metres to keep reasonable ramp angles. Figure 2-19 shows the principal arrangement of this handy size ferry.

Table 2-4 Main particulars of a Handy Size Ferry, Joint Nordic Project

Length overall	113.90 m
Length perpendicular	102.00 m
Breadth, moulded	21.00 m
Draught dwl	4.60 m
Draught scantling	4.80 m
Depth to bulkhead deck	7.00 m
Deadweight	1300 t
Trial speed	18.5 knots
Trailer lanes, main deck	474 m
Car lanes, lower hold	230 m
Cabins	102 pcs
Passengers	800
Passenger public spaces	1400 m^2

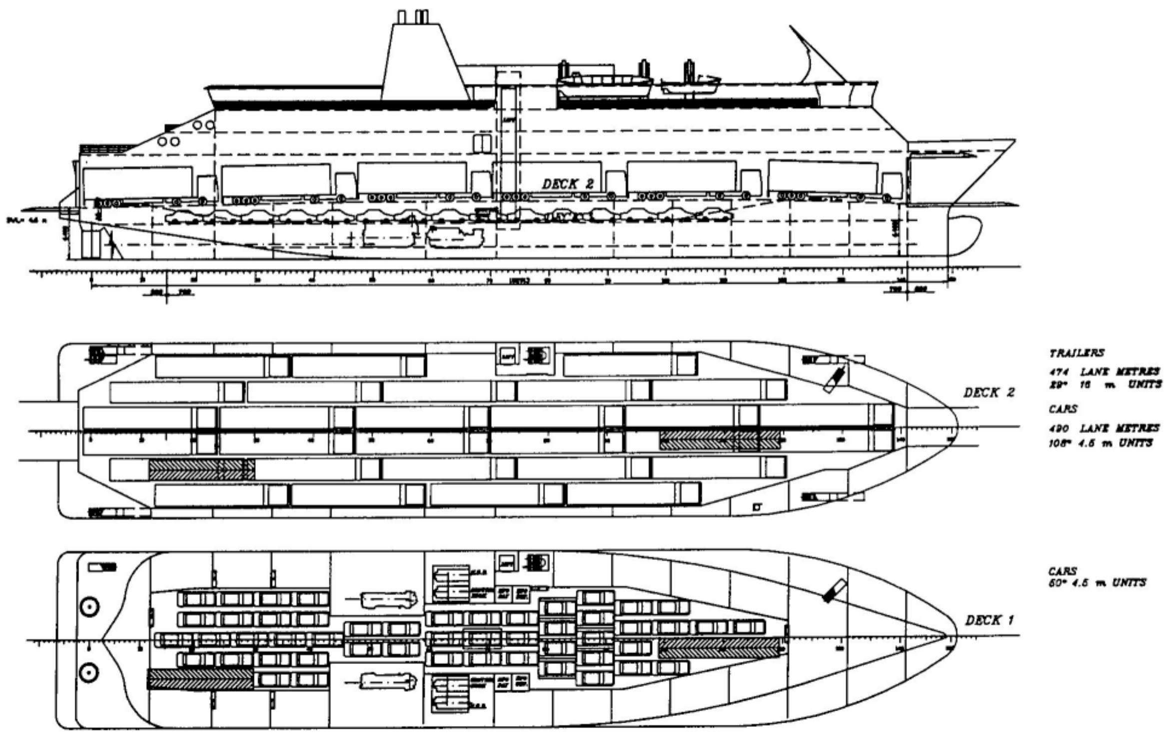

Figure 2-19 Principal arrangement of the Joint Nordic Project Handy Size Ferry

Damage stability was calculated in accordance with the SOLAS 90 including also the lower hold in all two compartment damages. Water on deck has also been calculated for significant wave height of two metres in accordance with the Stockholm agreement, no special arrangements are needed on the main deck (flood preventing doors or similar). Damage stability has also been analysed in accordance with the Joint North West European proposal for damage stability of ro-ro passenger ferries based on probabilistic method.

The biggest difference to the existing fleet of similar size is the relatively large lower hold for private cars. Only a few of the existing ferries have a lower cargo hold for ro-ro traffic operated through a ramp, and the operation in this case is even with drive-through principle, i.e. a ramp at both ends of the lower hold enabling an efficient cargo flow. Cabin capacity is relatively high, which area can on the other hand be converted into public spaces for a day ferry version with high passenger capacity.

<u>Fast full displacement handy size ferry</u>

There is a big interest on the market for fast handy size ferries with relatively high deadweight and within limited main dimensions to be able to operate economically into small harbours.

Strintzis Line of Greece was interested in the handy size ferry developed within the Joint Nordic Project, however, they pointed out immediately that higher speed, above 23 knots, is an obvious requirement, especially for the high season.

The main dimensions were modified to reach a more favourable length/beam ratio, perpendicular length was increased up to 111,8 m and beam was decreased to 18,90 m, thus the length/beam ratio became 5,92 in comparison with 4,86 of the original design. Length overall was restricted to 120 m due to harbour restrictions. The original block coefficient of 0,64 was reduced down to 0,60. Table 2-5 presents the main particulars of the design. The main deck has five trailer lanes and a central casing, side casings are provided in the aft and forward ends of the main deck. The upper deck is for private cars. Due to reduced beam full length side casings and lower hold became complicated and difficult to apply. However, in order to fulfil the SOLAS 90 damage stability requirements the bulkhead deck (main deck) had to be raised and in order to fulfil the water on deck requirements a further lift of 60 cm was required. Figure 2-20 shows the principal arrangement of the vessel.

Table 2-5 Main particulars of the fast handy size ferry for Strintzis Line

Length overall	113.90 m
Length perpendicular	102.00 m
Breadth, moulded	21.00 m
Draught dwl	4.60 m
Draught scantling	4.80 m
Depth to bulkhead deck	7.00 m
Deadweight	1300 t
Trial speed	18.5 knots
Trailer lanes, main deck	474 m
Car lanes, lower hold	230 m
Cabins	102 pcs
Passengers	800
Passenger public spaces	1400 m^2

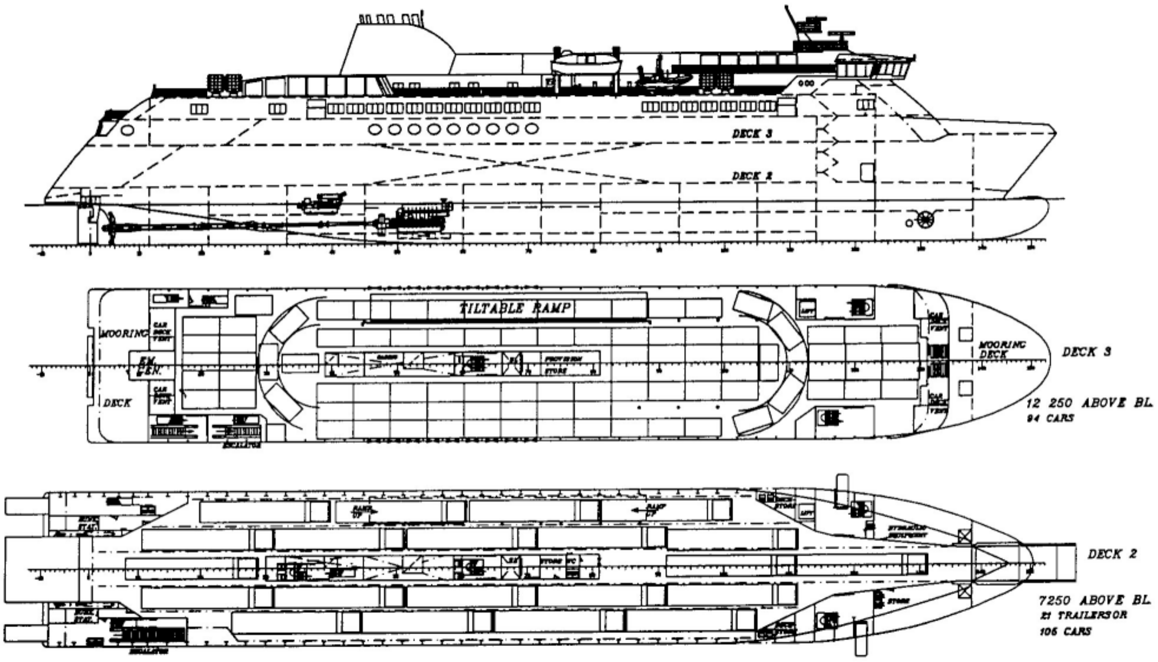

Figure 2-20 Principal arrangement of the Fast Handy Size Ferry for Strintzis Line

A lot of attention was paid to the development of the of the hull form as the request was to reach minimum 23 knots plus speed at 85% of MCR (at 14076 kW). The overall length was limited and the block coefficient was also on the high side for a typical high speed ferry at a Froude number[1] of 0,355-0,37. CFD (Computer Fluid Dynamics) calculations were carried out to optimise the hull form before starting model testing. Resistance, propulsion and wake measurement tests were carried out and the results were much better than expected: 24 knots plus was reached at 85% of MCR at maximum draught. The propeller diameter was 3,90 m and outward turning propellers were found favourable for powering. The wakefield was considered to be good with low propeller induced forces against the hull, see figure 1-5 in chapter 1.1.2.

An other interesting example is presented in figure 2-21 and table 2-6, a newbuilding project for Strintzis Line ordered at the Hellenic Shipyards, Greece, a medium size fast full displacement ferry with a relatively high speed of 26 knots, maximum 27 knots. Length overall is 140 m and length of design waterline 128 m (without the bulbous bow) leading to a rather high Froude number of 0,38. The main characteristics are presented in table 2-6, length beam ratio being 6,1, length draft ratio 24,6 and beam draft ratio 4,0. The maximum number of passengers being 2000 means that the ferry is an efficient day ferry with some cabin capacity. The basic idea is to increase the number of daily sailings as a day ferry especially during the high season, and to be able to sail economically at a lower operational speed during the off-season period, if needed.

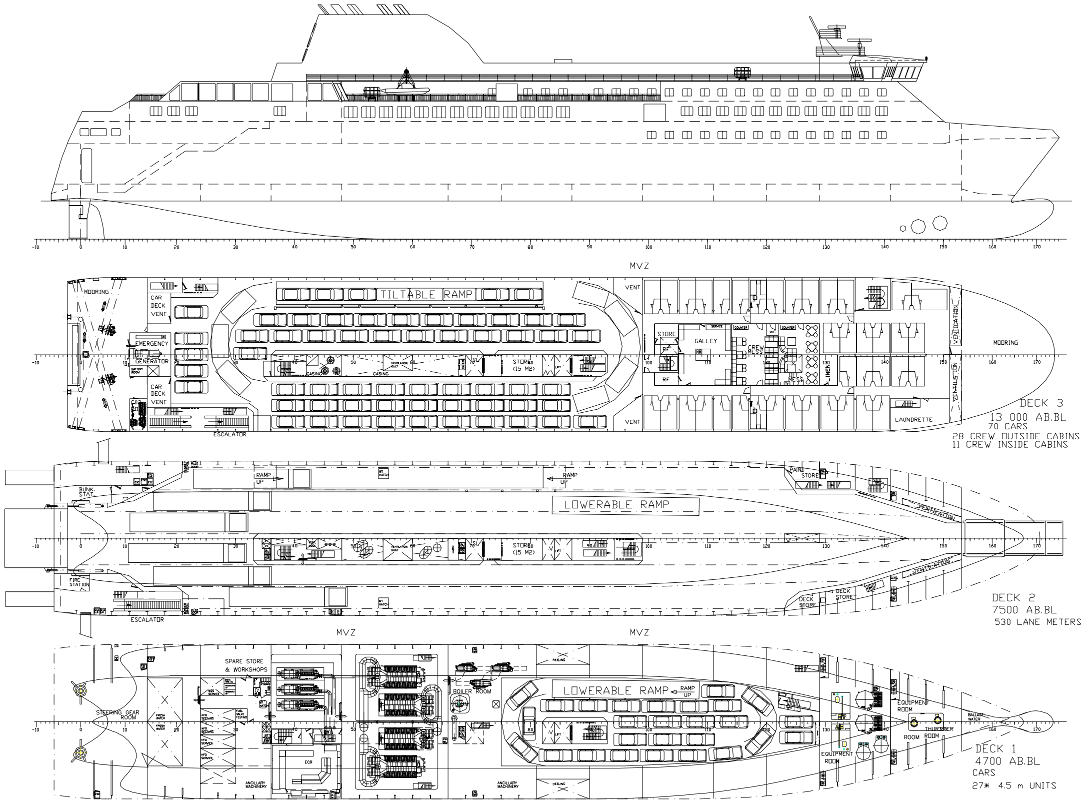

Figure 2-21 Arrangement of Strintzis Line Fast Full Displacement Ferry for 2000 passengers

Table 2-6 Main characteristics of a Fast Full Displacement Day Ferry for Strintzis Line to be built by Hellenic Shipyards

L_{OA}	136,7 m
L_{PP}	126,2 m
B	21,0 m
T	5,2 m
Depth to bulkhead deck	7,25 m
Deadweight	1960 t
Trailer lanes	530 m
Number of cars, upper car deck and lower hold	128
Number of cars on platform above trailer deck	104
Passenger cabins	146
Persons onboard	2100
Passenger public spaces (inside)	1800 m^2
Main engines	4 x 7920 kW
Speed	26 knots at 75% MCR

3. Machinery Considerations

3.1 General

This chapter describes and presents references of new machinery configurations studied and developed for different types of vessels on behalf of shipping companies. Diesel-electric machinery configuration is included, as well as combined gas and steam turbine machinery and pod-propulsion. The common feature for all the references is that consideration of the machinery concept itself has not been adequate, it is not possible to develop a new machinery configuration without developing and considering the complete configuration of the vessel at the same time. It is just an excuse for not carrying out the work properly.
Tankers, passenger cruise ships and ro-ro passenger ferries are described. Some future developments are also discussed.

3.2 Diesel-Electric Tankers

At the beginning of the 1990's the first modern electric powered tankers were established. Speed controlled AC-AC emergency propulsion was applied on tankers already at the beginning of the 1980's but the next decade introduced a shuttle tanker design with full electric propulsion machinery: the Statoil/Navion shuttle tankers. The reason for selecting electric propulsion was operational; a shuttle tanker has extensive periods of DP-operation calling for very high power availability for extended DP-operation under severe sea conditions.

The real breakthrough came in 1993 with the chemical carriers developed for Stolt Parcel Tankers. The extensive studies indicated that about 30 000-40 000 DWT would be suitable tanker size for modern electric propulsion. One of the main issues was the possibility to arrange some 1200 extra cubic meters for cargo tank volume due to the novel machinery arrangement and extremely short engine room as shown in figure 3-1. It was possible to show the electric ship to be also economically superior. This development was then soon followed by other diesel-electric tanker designs, such as the 5400 m^3 chemical tankers at INMA for Stolt Parcel Tankers, as well as the 12.000 to 24.000 m^3 chemical and product carriers for different European owners.

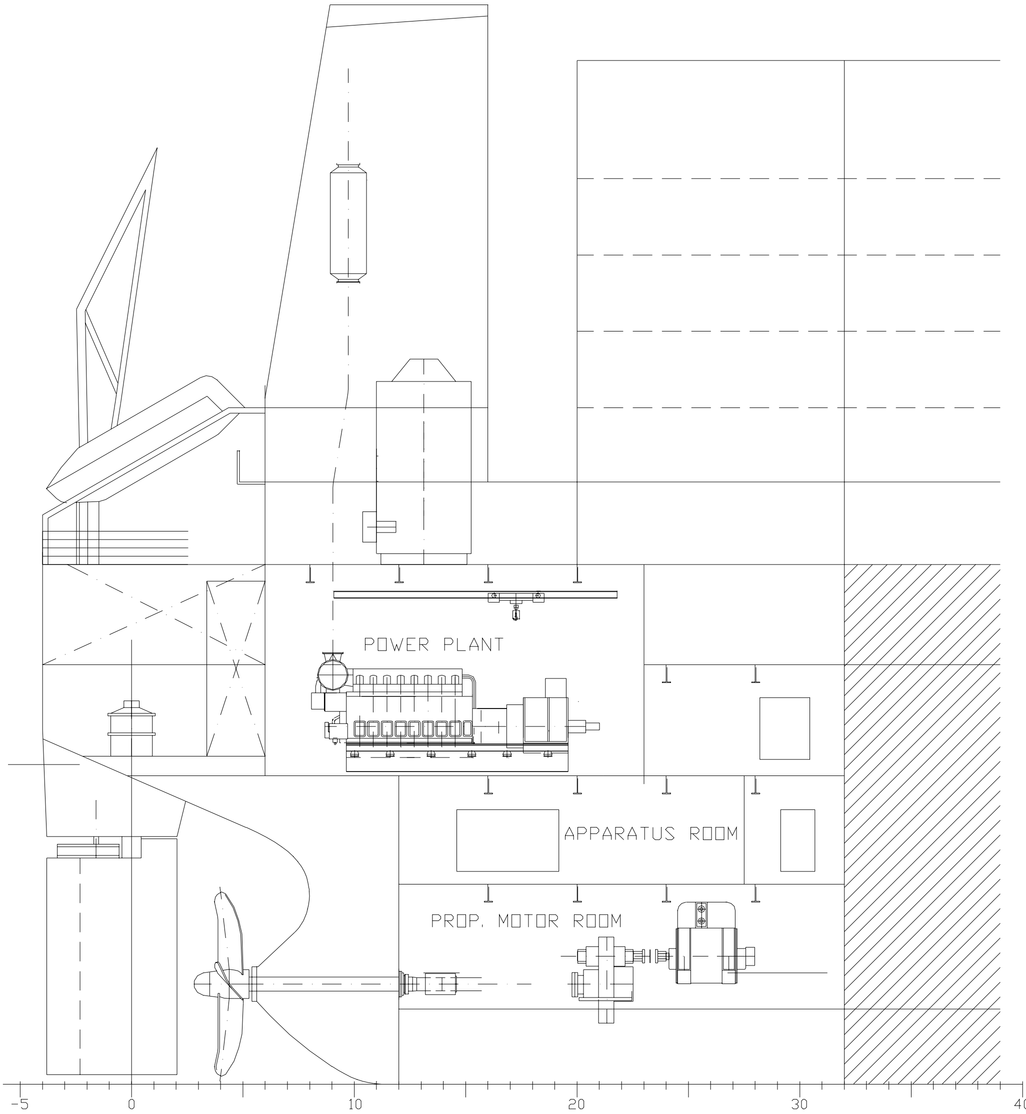

Figure 3-1 The vertical machinery allocation gives short compact engine room and increase in cargo tank volume

The recent success of the diesel-electric machinery, especially in tankers, is based on conceptual thinking.
The benefits can be listed: machinery location, cargo tank location, location of longitudinal centre of buoyancy, cargo capacity, loading capabilities, hull form, power requirement, structural principles, machinery and operational redundancy, maintenance and building and installation procedure. This list means that the total configuration of the vessel is to be considered before replacing a conventional machinery with a new concept.

Arrangement and Cargo Volume

The power generating machinery can be freely located to an optimum location to serve the cargo carrying capabilities in the best possible way. Several studies and newbuilding designs have shown that the optimum arrangement is to locate the machinery vertically, with propulsion motor on the double bottom, systems and electrical equipment in the intermediate level and generating sets, boilers and other big units on the upper (main) deck level. A straight forward and short routing is created for cabling and piping. The most important feature, however, is that the engine room can be made more compact and shorter releasing valuable volume for cargo tanks, 3-6,5 % additional cargo volume within the same main dimensions and displacement; in actual cubic meters: 1 300 m^3 for a 40 000 m^3 chemical carrier, 800 m^3 for a 13 000 m^3 chemical carrier and 200-350 m^3 for a 5 200 m^3 chemical carrier.

It is possible to utilise this extra volume released from engine room into cargo tanks by adding an extra tank or a pair of tanks or by adding the volume into existing tanks; cost-wise the third option is the cheapest and easiest.

The increase of cargo volume in the aft part automatically shifts the longitudinal centre of cargo aftwards when fully loaded, giving more freedom for hull form design and thus improving also loading capabilities.

Weight

The machinery weight including main engines, auxiliary engines, emergency generator, foundations, power transmission ancillary systems and propeller plant has been compared for different sizes of tankers and engine powers. The slow speed main engine option gives the highest total weight whereas the diesel-electric gives the lowest. Following relative differences have been found: slow speed main engine being 100, medium speed geared main engine 67, diesel-electric direct 72-73 and diesel-electric geared 54-55.

For a 37 000 dwt / 40 000 m^3 carrier the actual weight figures were: slow speed 656 ton, medium speed 438, diesel-electric direct 465-478 ton and diesel-electric geared 356-363 ton. This weight saving is giving same further flexibility for the hull form design as well as for the deadweight.

Hull Form

The additional cargo volume and the reduced machinery weight with the diesel-electric machinery configuration require careful attention to the hull form design.

It is possible to reconsider the principal type of hull form and related parameters, especially longitudinal centre of buoyancy in a non-typical way. The shift of machinery and cargo aftwards makes it possible to have a position of longitudinal buoyancy aft of midships and thus to avoid pronounced forward shoulder. This will reduce required propulsion power. It requires, however, a good aftship hull form to avoid any flow separation and additional viscous resistance. Pram type hull form (barge type) with a centre skeg offers a good basis. The centre skeg will only accommodate electrical propulsion motor with gearbox giving much more freedom for the skeg dimensions and for the aftship buttock angle than with a conventional and heavy slow speed engine. Propeller diameter can also be optimised.

Figure 3-2 shows a typical, simple pram type aft ship for a diesel-electric powered tanker.

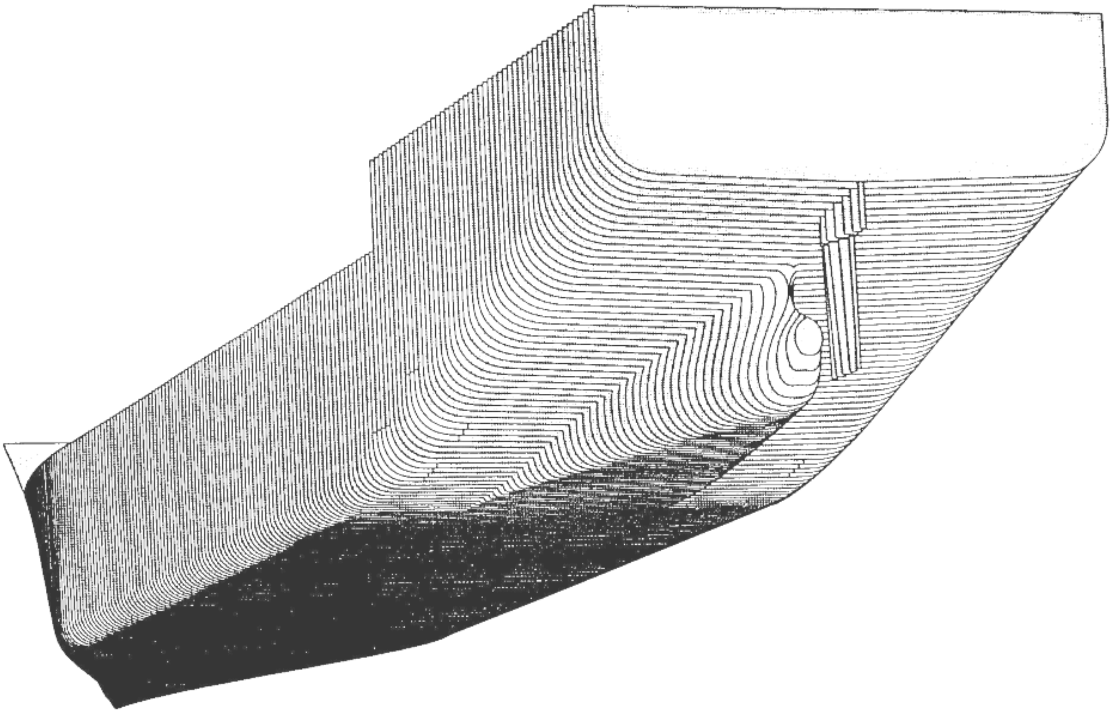

Figure 3-2 Typical pram type hull form for diesel-electric powered tanker, centre skeg designed to accommodate electrical propulsion motor

Speed-Power Performance

The possibility for hydrodynamically optimum longitudinal position of buoyancy (for speed and not only for loading of the vessel), propeller diameter and application of pram type hull form gives a power saving in a range from 5 up to 12 % in comparison with good optimised conventional machinery and hull form configuration. This is based on model tests and studies for typical product and chemical carrier from 7 000 dwt up to 40 000 dwt, bigger percentage typically for the upper range of vessel size.

Installed Power

The power plant principle makes it possible to utilise the installed power in the best possible way. Same diesel-electric power can rotate the propeller as well as cargo pumps. The number and size of diesel generators can be selected to meet the different power requirements at optimum loading in all prevailing service conditions. There is a clear saving in installed power which can be as much as 15-18 % for a 40 000 dwt product carrier. This is taking into account the lower required propulsion power, transmission efficiencies and having electric cargo pump drives.

It is also possible to use all the installed power for propulsion giving a possibility for 1 knot higher maximum speed than typical, a big commercial advantage. Or to reach a higher ice class than would be possible with conventional direct machinery.

Availability

Availability of power is very high, typically arranged with three diesel generators. Propulsion system is either a single motor arrangement with double winding or two motor arrangement, in both cases geared through dual pad / two step gearbox into single fixed pitch propeller. The single items are the shaftline and propeller.

High safety and redundancy is achieved through the arrangement of the power plant, generating sets located above waterline in separated fire safe compartments.

Building and Installation

The length of the diesel-electric engine room is mainly determined by the propulsion motor reduction gear package, which can now be placed in an aftermost location within the centre skeg. The philosophy of engine room design is to arrange the equipment so that short and straight routing of piping and cables, possibility of extensive prefabrication and modular construction, the shortest possible installation period and easy and fast maintenance can all be achieved.

With the modular construction principle when applied on functional basis, i.e. by manufacturing functional modules and connecting them with prefabricated piping packages (routes); it is possible to shift a big amount of difficult installation work from aboard the ship into workshops. The power plant machinery gives the possibility for fully modularised and prefabricated machinery.

Costs

Fuel cost is lower or the same due to optimised main parameters, hull form and propeller, in spite of the lower transmission efficiency with the diesel-electric configuration.

Building costs can be reduced with efficient arrangement allowing quick, late stage installation of machinery, modularised arrangement and reduced piping.

All existing requirements of maximum allowable exhaust gas emissions can easily be met without any additional heavy investment.

Future

The future design is obviously based on the good experience of diesel-electric machinery but replacing the propeller shaftline with rotatable thruster or pod drive. Comparative model tests show that applying thrusters with contra rotating propellers or pulling type pod propulsion gives further saving in required propulsion power. But the most important feature is to achieve complete operational redundancy, excellent manoeuvrability and extremely simple arrangement to build. Possible arrangement with two pods is shown in figure 3-3.

This arrangement allows additional cargo volume, 2-3 % compared to the conventional diesel-electric arrangement. But the main issue is the possibility to select completely new main dimensions, wide beam without any problem of water flow into the propellers, shallow draft enabling entrance into harbours with limited water depths. Length and cargo capacity are optimised together with the beam in accordance with required cargo carrying capacity and route and harbour restrictions. All this with highest possible power availability and redundancy and with excellent manoeuvring capabilities. The future of different size and type of tankers even up to VLCCs. This, of course, requires a new technical approach for cargo pumping system and arrangement.

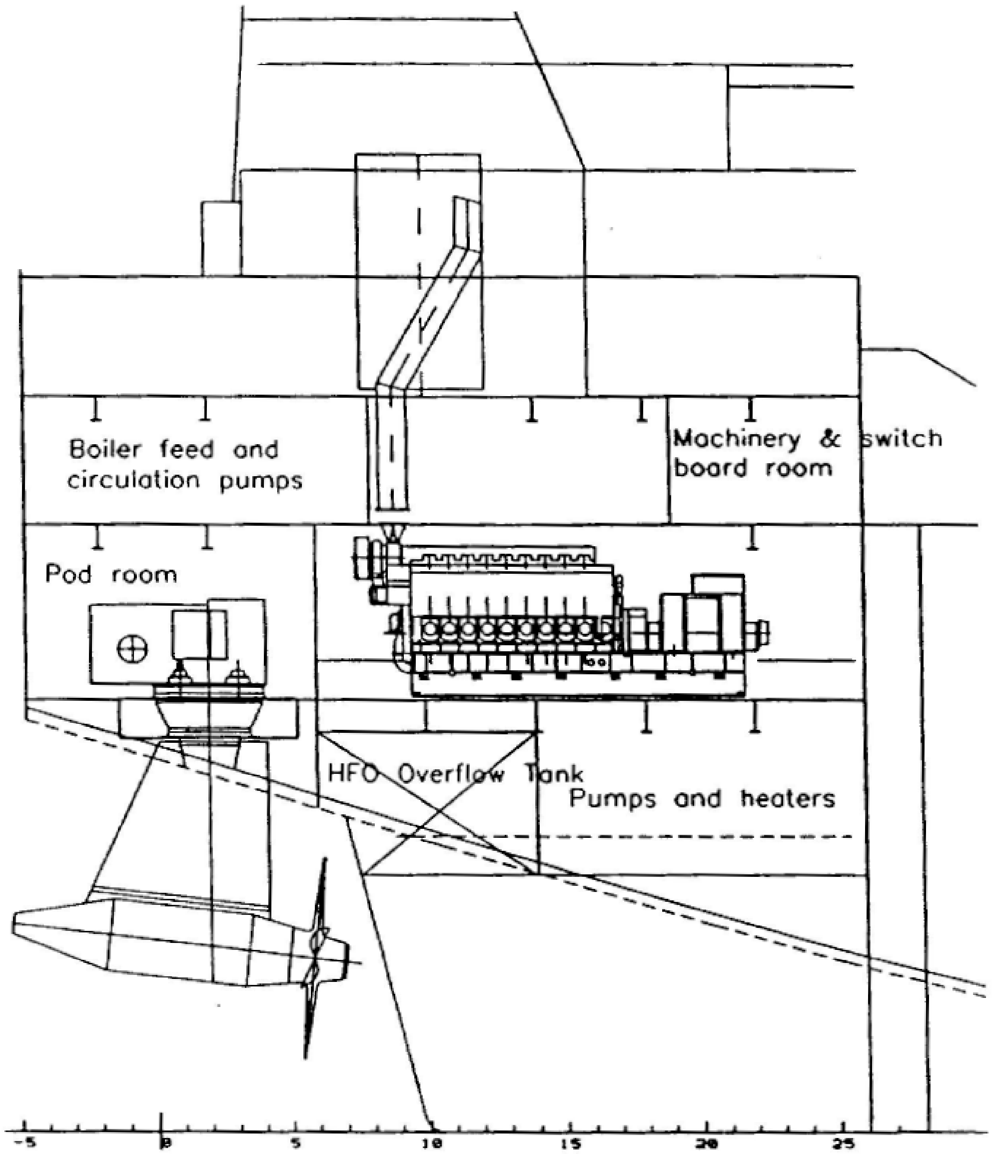

Figure 3-3 Future machinery configuration with two pulling pods

3.3 Gas Turbines in Passenger Cruise Vessels

Several studies for gas turbine propulsion in cruise ships have been carried out during the last decade. All these studies indicate clearly that power plant based gas turbine would bring sustainable advantages when applied together with electric propulsion.

Table 3-1 shows the typical differences when comparing a vessel with aeroderivative gas turbines in combined cycle (COGES) to similar ship with diesel-electric machinery.

Table 3-1 Summary of the outcome from various studies carried out by Deltamarin about COGES application for a 60 000 - 80 000 GRT cruise ship with 40-60 MW installed power. Fuel consumption includes all fuel used on the vessel, both on engines and boilers. Different turbine configurations are also included..

	COGES SHIP	DIESEL-ELECTRIC SHIP
ENGINE ROOM AREA	900 -1800 m2 LESS	REF.
PASSENGER CABINS	23-41 MORE	REF.
FUEL OIL CONSUMPTION	7 % LESS - 40% MORE	REF.
FUEL + LUBE COST / YEAR	$M 0.7 - 2.8 MORE	REF.
MACHINERY FIRST COST	$M 1 - 7 MORE	REF.
MACHINERY WEIGHT	700 - 900 T LESS	REF.
MACHINERY MAIN COMPONENTS	40-50% LESS	REF.
MACHINERY CREW	3-6 LESS	REF.
NOx & SOx EMISSION	LESS	REF.
PARTICULATE EMISSION	LESS	REF.
CO2 EMISSION	LESS-MORE	REF.

Table 3-1 indicates the main advantages of the COGES-machinery: more passenger cabins and less emissions. Most of the studies include also the development of the general arrangement actually showing location of additional passenger cabins and related increase in other functions. Each passenger cabin brings additional space demand for crew area, public spaces, stairs, stores, etc. Based on statistics, some 54 m^2 is needed on an average cruise vessel to get one cabin of about 22 m^2 for two passengers.

Studies for the Millennium Class

The results of the earlier studies confirmed the owner, Royal Caribbean International (RCI), about the attractiveness of COGES machinery. Thus a specific study for the Millennium Class vessels was prepared.

The study was carried out at the beginning of the project and the results were evaluated together with the owner, yards and power plant supplier. The results were well in line with the earlier studies and the difference in cabin number was even higher than expected, as shown in table 3-2.

Table 3-2 Results from the COGES / Diesel study prepared for the Millennium Class for RCI

	COGES SHIP	DIESEL-ELECTRIC SHIP
PASSENGER CABINS	50 MORE	REF.
CREW CABINS	20 MORE	REF.
FUEL OIL CONSUMPTION	2 % LESS	REF.
FUEL + LUBE COST / YEAR	$M 2.7 MORE	REF.
NOx & SOx EMISSION	LESS	REF.
PARTICULATE EMISSION	LESS	REF.
CO2 EMISSION	LESS	REF.

The expected heat demand at sea was 22.6 MW with diesels and 20.7 MW with COGES and in port 10.9 MW with diesels and 10.0 MW with COGES.
The diesel ship has a higher value due to the use of HFO and also 500-700 kW higher electric power consumption due to the diesel ancillaries.

The principal configuration for the Millennium class cruise ships is presented in figure 3-4.

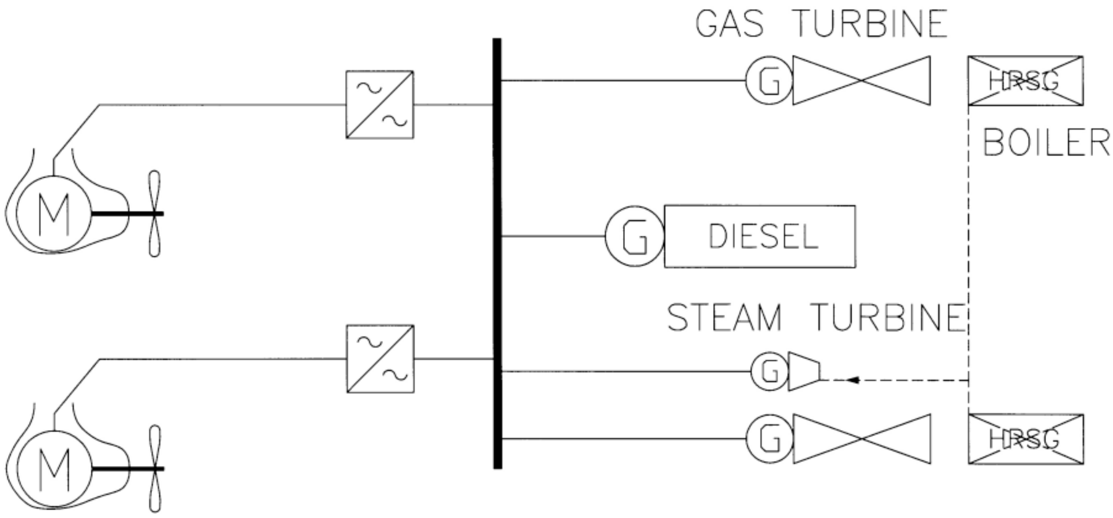

Figure 3-4 COGES plant as applied on the Millennium class vessels for RCI

Power plant configuration consists of two LM2500+ gas turbine generator sets, 25 MWe each, two heat recovery steam generators, 410^{o}C, 32.5 bar, 38 t/h each, steam drums, deaerator pre-heater, back pressure steam turbine generator 9 MWe exhausting at 3 bar$_a$, atmospheric condenser and a 3MW auxiliary diesel.

COGES Machinery for a Cruise Ship

Gas turbine propulsion is not just a single solution but a wide variety of possible solutions as shown in figure 1-25. This means that some evaluations are to be done for each specific project prior selecting the final configuration.

Actually the only drawback with the gas turbine solution is the need for clean and thus expensive fuel, and so the simple cycle gas turbine installation is out of interest on a big cruise ship, at least with aeroderivative engines. The fuel cost can be reduced by introducing gas turbine on combined cycle. Different solutions can be found feasible on other ship types, but this must be evaluated on a case by case basis.

For the time being there are only two mainlines to follow when improving fuel efficiency on gas turbine powered vessel: either cogeneration (COGEN) or combined cycle (COGES) application.
Both solutions are based on gas turbines producing electricity to ship network. In the COGEN plant the exhaust gas heat is used for process heat production only, whereas in the COGES plant the produced steam is primarily used on steam turbine for electric power production and secondarily to supply steam for ship purposes.

A typical installation is with two gas turbines. Operation of these units is arranged so that the other turbine is most of the time at stand-by, ready to start condition. In case three engines are selected, the negative impact on engine casing size is big and there are fewer possibilities to reach high space savings.

Although a cruise ship needs a lot of heat, mainly for fresh water production, the need is still so low that COGEN would not yet be a viable choice with aeroderivative engines, especially because it would call for higher installed gas turbine power. In some cases COGEN can give better results than COGES on small cruise ships, especially if colder, industrial type engines are applied. However, COGES would be a more attractive mainline to follow on big cruise ships.

Steam Turbine

Steam turbine can be of back-pressure type or condensing type. Condensing turbine exhausts to vacuum condenser while back pressure turbine exhausts to atmospheric condenser. Results are: back pressure turbine plant has less stages and lower efficiency leading to higher fuel consumption, but the benefit is the lower first cost. Condensing turbine acts controversially. The decisive criteria are the time spent at high cruise speed as well as the heat demand in port and at low cruising speed. The more time is spent at high speed, the better choice condensing turbine is and vice versa, see figure 3-5.

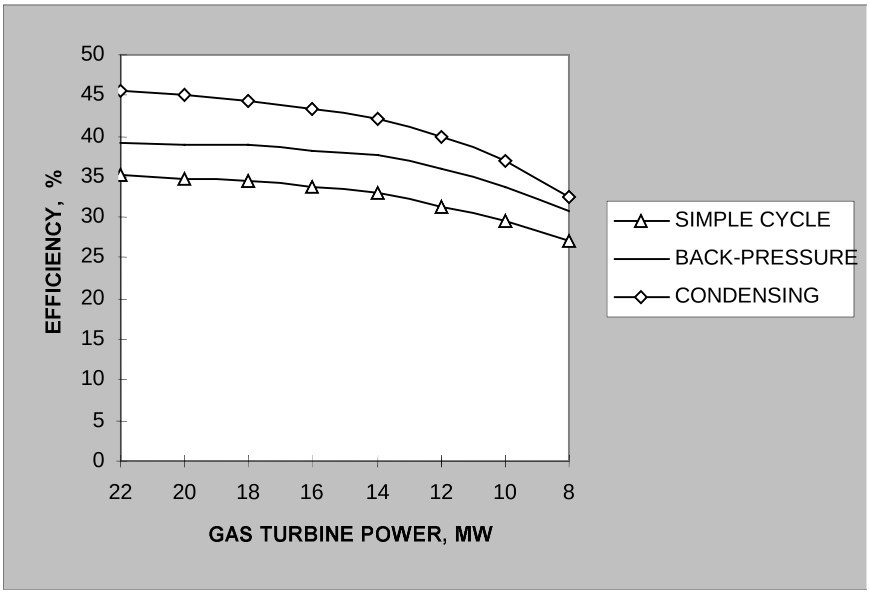

Figure 3-5 Example of thermal efficiency of gas turbine configurations based on GE LM 2500 as a function of gas turbine power. Note for certain power demand the gas turbine has different load on different options as the steam turbine covers part of the demand in the two COGES options 14 t/h 3 bar steam is extracted from the condensing turbine in this example.

In case the condensing turbine is selected the next choice is the number of boilers. The plant could be based on base load / booster power concept where only one of the gas turbines is provided with COGES while the other acts with simple cycle. The benefit is low first cost and, depending on the operation profile, about equal fuel consumption than with twin boiler installation. The major drawback is the fuel cost increase in case some component fails on the base load COGES plant and the power is produced by the simple cycle engine alone. Same happens if the operation profile of the vessel is changed so that the simple cycle engine must be connected in parallel with COGES.

In case ultimate fuel efficiency is targeted the plant would be based on condensing principle with two boilers and two turbines. This is also the most expensive solution. Two 5 MW steam turbines instead of one 10 MW unit would have better efficiency at low engine load (65% vs. 58%) but lower (77% vs. 81%) at high load. Although the ship operation consists of a lot of low speed operation, the twin turbine solution is seldom feasible on a cruise ship due to the high additional first cost.

Heat Recovery Steam Generators (HRSG)

The turbine choice affects also the boiler configuration. In case less efficient back-pressure turbine is selected, also the boiler should be made more simple. Four circuits are applied together with the condensing plant but three circuits is a more feasible arrangement for back pressure plant. The benefit is a less expensive boiler and some 800 mm lower boiler height.
Figure 3-6 shows the overall differences between a back-pressure plant and a condensing plant.

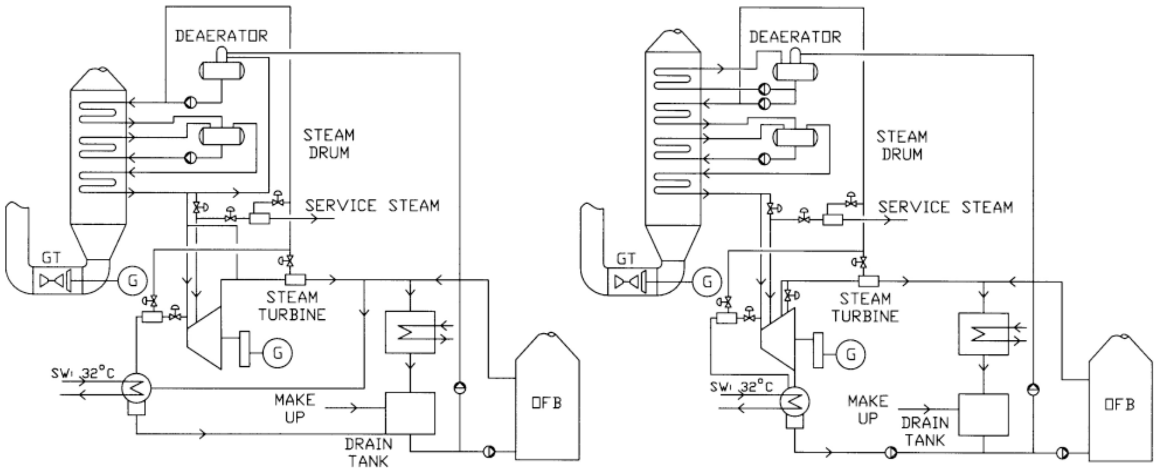

Figure 3-6 Principal system diagram for back-pressure COGES plant (left) and condensing COGES plant (right)

On a modern cruise ship the heat consumption is high and the total annual hours spent at low speed cruising or in port are also high. Thus if a condensing turbine is applied, most of the time considerable amount of heat demand would be covered by steam extracted from the turbine, and the last stages on steam turbine would create only negligible amount of power. In this condition the benefit of vacuum condenser is only marginal. Therefore less expensive back-pressure solution is typically feasible on a cruise ship. The condensing turbine option is more attractive on ships, which are operating most of the time at high-speed mode.

Operational Considerations

The primary reasons for selecting gas turbines are the increased passenger (or cargo) capacity and the low exhaust emissions.

On the other hand, high attention must be paid on engine starting and load acceptance functions when the power plant is based on only two main engines and most of the time only one engine is running.

Starting the gas turbine means that a huge amount of exhaust gas heat is released to the boiler. This leads to a situation where the temperature of cold boiler is rapidly increased to operation temperature. It would be possible to keep the stand-by boiler continuously heated but that would mean additional thermodynamical loss and lower efficiency for the whole system, which is not acceptable. Two possibilities remain: either installation of boiler by-pass line with diverter valves or applying boiler, which allows running in dry condition.
The first solution requires a larger casing size in the area of the boiler and on the other hand the diverter valve installation would lift the boiler position some two meters upwards.

Thus the latter option is typically selected. Although the application with dry running boilers is not new it calls for high quality materials to be adopted to accept up to 560°C exhaust gas temperature from e.g. LM 2500+ engine. Demanding task for a boiler supplier is also to get simultaneously compact boiler size and low pressure loss over the boiler. Total maximum target pressure loss on the exhaust ducting system is 400 mH_2O.

Although a steam turbine is principally installed in order to increase fuel efficiency, it is not allowed to cause secondary failures or trips in the power plant. Sudden failure on turbine or opening of generator breaker initiates immediate operation of quick closing valve to avoid turbine over-speeding. Then most of the produced steam must be dumped on the condenser and reliability of this device is of highest importance. The condenser must be rated to dump full steam production of two heat recovery boilers. Although the boilers are designed for running dry, this is not considered as a normal operation. System availability can be further increased by applying two continuously connected 50% condensers instead of a single 100% unit.

Another vital issue in the gas turbine plant is the high quality of combustion air. That is ensured by applying three stage filtration featuring moisture separation grids and pocket filters.

<u>Arrangement</u>

Cruise ships still today have arrangements with mixed location of systems, spaces and functions, e.g. engine casing in the middle of the best public spaces, machinery, service, crew and even passenger functions mixed on the freeboard deck. A lot of space is lost.

A typical diesel-electric power plant with ancillaries requires about 11.000-12.000 m^3 below freeboard deck space. The gas turbine power plant with ancillaries can be installed in a space of only 4000-6000 m^3, including the exhaust gas boiler and the space for combustion air intake arrangement, but depending on the selected system, COGEN or COGES.

Huge volumes are available for other functions with gas turbine plant but this volume is below the freeboard deck, which cannot directly be transferred into passenger spaces. New machinery configuration should be utilised in the general arrangement development already from drafting the first idea.

This is the approach for the All Aft Machinery configuration. The development started from the idea of separating machinery, service, crew and passenger functions by relocating them and utilising efficiently the additional volume offered by the gas turbine power plant.

<u>Advanced Panamax Cruise Ship</u>

By locating the power plant aft, close to the biggest power consumers, the pod propulsion motors, the whole midship area was released for other purposes. The idea was to have all machinery functions aft, all service functions down, freeboard deck forward for passengers and deck in between for crew.

To minimise the space required for machinery a COGEN type plant was selected, e.g. intercooled, recuperative (ICR) gas turbine at 25 MW known as WR-21 by

Rolls-Royce. The gas turbine plant was located on the freeboard deck in the aft corners of the vessel and just in front of the biggest consumers, i.e. the pod propulsion units; figure 3-7 shows the principal arrangement.

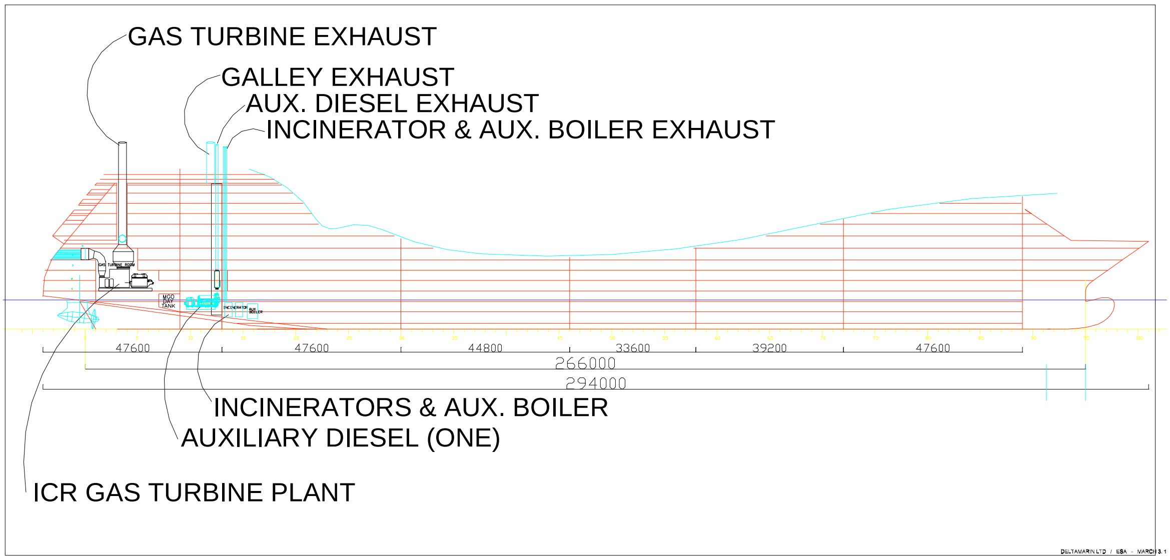

Figure 3-7 All Aft Machinery separates the machinery function from crew & passenger accommodation and services

The engine casing is divided, gas turbine plants having separate engine casings with air intakes in the aft corners of the ship and one casing is provided for incinerators, oil fired boiler and galley exhaust.

Gas turbine casings up to lifeboat embarkation deck are to provide a space for the exhaust gas boiler and air intake arrangement. Above the exhaust gas pipe can be led just in exterior space without any casing with minimised disturbance for passenger decks.

Incinerator and galley exhaust casing location is naturally related to the location of the main galley and provision service flows. In this arrangement the location of the galley is aft and the dining rooms are forward of the galley.

When the main source of electrical power is moved aft the remaining machinery components such as fresh water generators, air conditioning cooling compressors and sewage treatment plant are left down on the double bottom.

The aft machinery section can be separated with watertight bulkhead from the other spaces on freeboard deck and the margin line may be defined to follow one deck higher up in the aft end and thus the stability performance is improved. This arrangement of watertight compartment on top of the freeboard deck is common also on a diesel-electric ship but the functional separation is more clear and easy with the aft machinery section.

The aft machinery section is not disturbing the passenger, crew and service flows. The machinery service and maintenance is easy to arrange with a direct access from quay to the machinery spaces.

All Down Service

Diesel powered vessels have traditionally the service corridor on the freeboard deck. Machinery space utilises the major portion of the midship area below the freeboard deck. Provision stores are partially located on and below the freeboard deck in aft ship and crew accommodation is provided in fore ship.

On decks 1 and 2 typical diesel-electric plant with ancillaries requires about 1500 m^2 per deck, that is about the same as maximum allowed area of main fire zone/one deck. With the all aft machinery totally 3000 m^2 is released for other functions.

Optimised passenger flow is guiding the design to have the additional passenger cabins directly linked to passenger elevators. When removing the crew accommodation and service corridor from the freeboard deck (deck 3) the area between passenger fore and aft elevator towers can be re-arranged for passenger cabins.

The most critical space is the service corridor connecting the fore and aft service elevators and stores. This is now moved down on deck 1, see figures 3-8 and 3-9. All stores and workshops are located beside the service corridor.

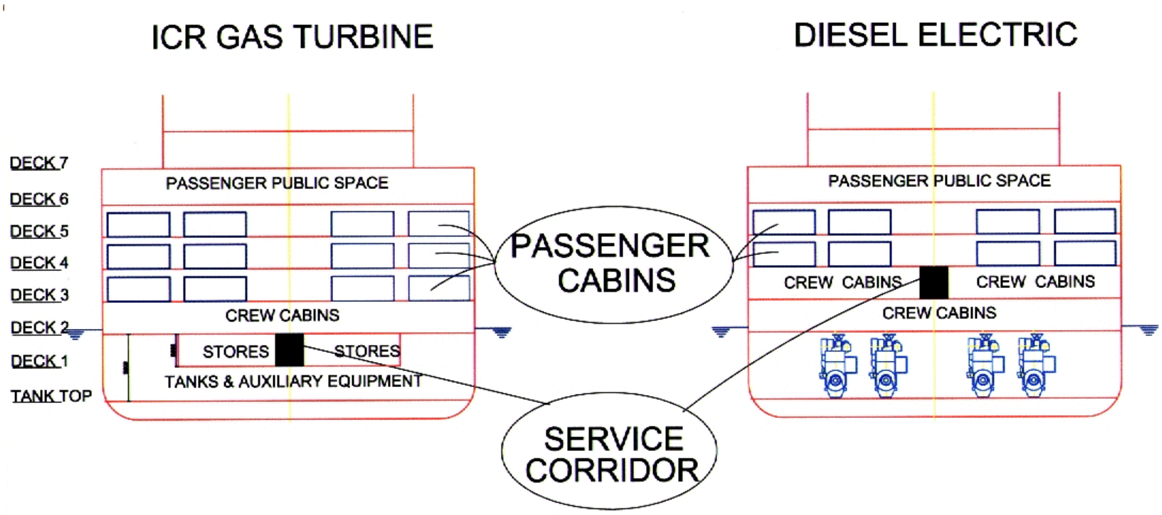

Figure 3-8 The major advantage of All Aft Machinery - All Down Service configuration is the increased passenger accommodation area

The deck above the stores, deck 2 is dedicated for crew. To decrease the number of watertight doors along the service corridor the store spaces are all located centrally inside B/5 longitudinal bulkheads.

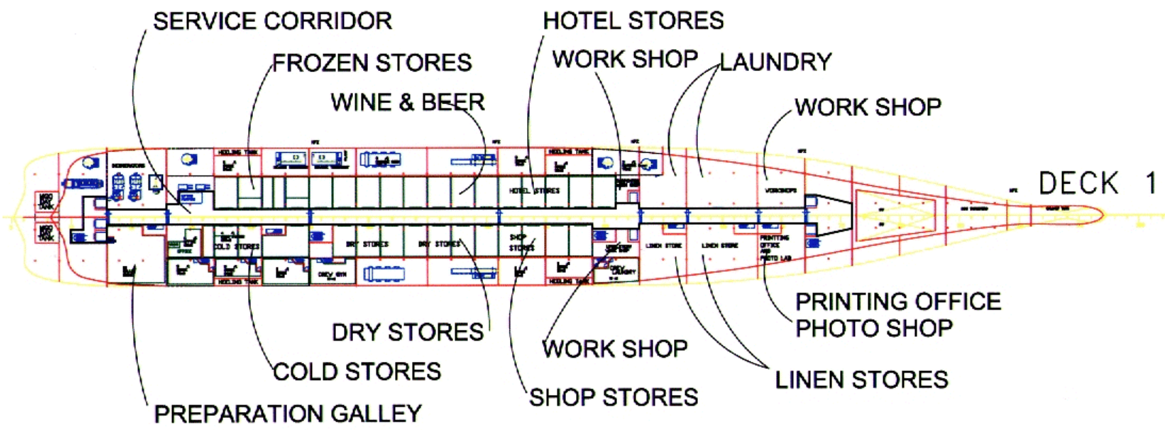

Figure 3-9 9 With the All Aft Machinery - All Down S configuration all main service functions are concentrated along the service corridor

When checking the damage stability with this configuration it was found out that when flooding into the longitudinal watertight compartment is allowed, the stability performance is better than with the conventional watertight compartment arrangement. This arrangement fulfils SOLAS 90 requirements easily for all SOLAS damages as well as damages penetrating inside B/5 limits. Service corridor has watertight doors at the main fire zone bulkheads only. Spaces outside of longitudinal bulkheads are used for various auxiliary machinery functions and for crew store rooms.

<u>Increased Passenger Capacity</u>

Typical diesel-electric powered Panamax passenger cruise ship has 1000 passenger cabins. With the all aft machinery and all down service configuration, Advanced Cruise Ship, the increase of cabins is +78 passenger outside cabins on deck 3, +50 passenger inside cabins on deck 3, +32 passenger inside cabins on deck 8, 9, 10 and 11; totally 160 additional passenger cabins and totally 12 additional crew cabins on decks 2, 3 and 4.

The freeboard deck (deck 3) amidships is a primary location for passenger accommodation, it is now far from any machinery and propulsion related noise and vibration sources, see figure 3-10.

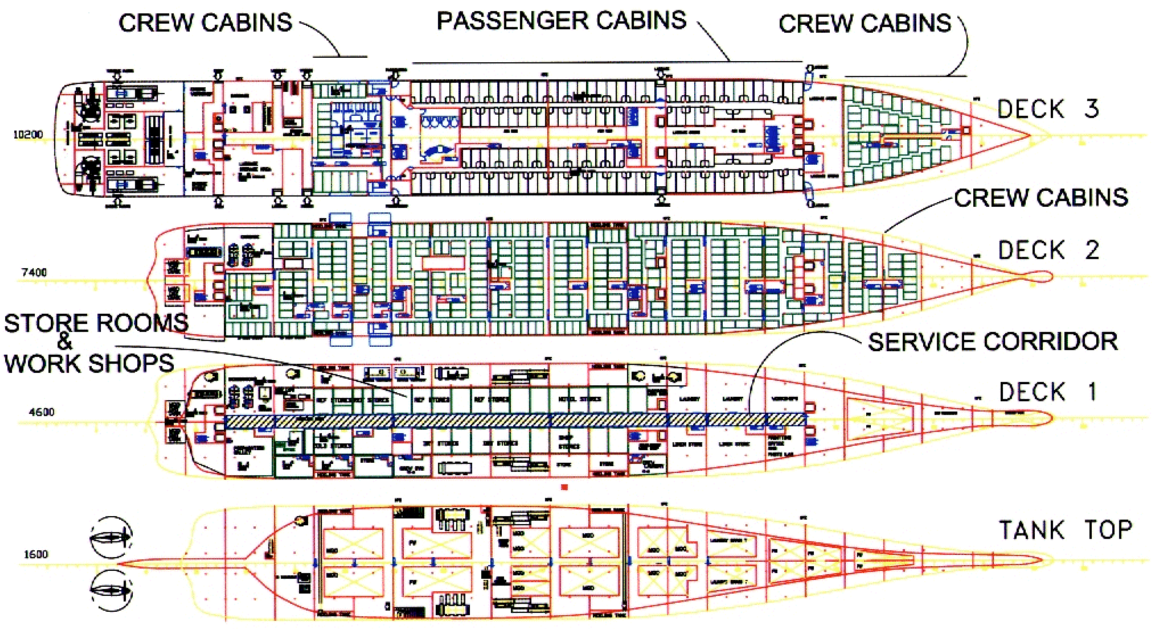

Figure 3-10 All Aft Machinery - All Down Service, Advanced Cruise Ship, configuration is clearly separating the passenger accommodation areas, service areas and machinery areas

Above the freeboard deck the major advantage of the all aft machinery configuration is the size of the engine casing. In the diesel-electric arrangement the engine casing is penetrating through the major passenger accommodation area causing loss of passenger cabins, as well as limiting the size of passenger public spaces and dominating the functional and architectural design of them, actually in the best area of the vessel with lowest accelerations in seaway.

All aft machinery has the engine casings in the aft corners of the hull and thus saves the valuable area for passengers.

On the upper passenger cabin decks the exhaust gas pipe is actually not taking any valuable passenger space as the space requirement is minimal and location can be arranged to match the cabin arrangement.

<u>Weight</u>

When comparing the diesel-electric and gas turbine powered ships it was assumed to have freeboard deck at the same height on both ships, even though no machinery unit is requesting similar heights for the all aft configuration.

With additional longitudinal watertight compartmentation it is also possible to have the freeboard deck about 500 mm lower for the all aft - all down gas turbine configuration. Ignoring this possibility the following results were reached.

The all aft machinery - all down service configuration is 1000 t lighter than traditional diesel-electric machinery. The difference is partly due to the lighter unit weight of ICR gas turbines and partly due to the location of major electric consumers just beside the power production plant.

The longitudinal centre of gravity is moved 1,1 m aftwards and vertical centre of gravity is about 0,45 m higher with the all aft machinery. Since the complete General Arrangement is different and stores are located below freeboard deck one needs to take into account the changes in deadweight distribution too.

Taking into account typical deadweight of 6000-6500 t, related tanks and equipment and their distribution the longitudinal centre of gravity for the all aft - all down configuration remains abt. 1,1 m more aft, loaded ship. But the vertical centre of gravity of loaded ship is about the same for both configurations, i.e. stores located all down reduced the height of vertical centre of gravity for loaded ship.

3.4 Fast Full Displacement Ferries

The definition of fast full (high) displacement ferries is not based on any code or regulation as the definition of fast ferries. However, it is useful to have a definition for this relatively new but today more and more popular type of ferry. We are using two criteria: speed above 25 knots or Froude number above 0,34. The first criteria are applicable for ferries above 160 m in length and the later one for smaller size ferries.

Today, spring 99, there are 27 ferries operating or on order fulfilling the above criteria, and 10 of them will be entering the market within the next two years. It is interesting to see that many owners are carefully considering the possibility and feasibility of investing on new fast full displacement ferries. The main criteria are the possibility to operate at high speed, 28-30 or even 32 knots without sacrificing too much the fuel efficiency and passenger comfort. The high-speed operation should also be possible on annual basis even in heavy weather conditions and on some routes in shallow water as well, without extreme wash effect. Good manoeuvrability is essential as well as fast cargo operation, otherwise the high steaming speed at sea becomes meaningless.

In the middle of the evolution process it is interesting to look at how quick the development has been; have we already learnt something and what could be the future.

Figure 1-19 in chapter 1.2.1 presents the Heickel coefficients for recently built ro-ro passenger ferries. The curves on the right hand side are for fast full displacement ferries with trial/operational speed between Froude numbers 0,34-0,38. The curve in the middle of these six curves is for a recently tested ferry with L_{PP} = 158,5 m, L_{WL} = 160 m, B = 24,8 m, T = 6,45 m, displacement 15.600 m^3, propeller diameter 5,0 m, trial speed 28 knots and maximum speed about 29 knots. A pram type hull form with ducktail and trim wedge was applied. It is interesting to compare this newbuilding with a four year old ferry, L_{PP} = 158,0 m, L_{WL} = 165,2 m, B = 24 m, T = 6,25 m, displacement 14.860 m^3, and propeller diameter 5,0 m. Comparing the model test performance at the same draft of 6,25 m, the newbuilding project having nearly the same displacement of 14.910 m^3, we can find rather big difference in the powering requirement at 27 knots: 32.000 kW / 28.300 kW, i.e. a difference of 13%. But even between similar size ferries model tested and built today differences from 8% up to 15 % can be found at similar displacement.

The learning curve has been tremendous. And it can be stated that we have certainly not yet reached the top, i.e. hull forms and propulsion arrangements can still be further optimised.

Lower required propulsion power means not only lower fuel consumption but also smaller main engines. The smaller required engine room space and also the weight are giving larger service and cargo (ro-ro/passenger) spaces and smaller investment cost.

With proper references it is easy to check whether the proposed design is of high quality or should it be reconsidered. It is advisable not to stick to predetermined configurations, open-minded approach gives typically better results, of course considering at the same time any possible risks.

The fast full displacement ferries are today built with L/B ratio not less than 6 and some even well above 7. Block coefficient is varying between 0,52 and 0,60; 0,57 being a typical value. Length is increased and width is restricted to get the block coefficient down and to reach a longer waterline, i.e. lower Froude number. Midship section coefficient is varying from 0,955 up to 0,988, the lower figure is from a rather short vessel (L_{PP} only 111,8 m), the longer vessels being between 0,98-0,988. LCB is varying from -2,6% with forward superstructure up to -3,6% with full length superstructure.

The tendency at this moment is towards longer vessels to increase the length / beam ratio and to decrease the block coefficient which is the easiest way to reduce the power requirement and to keep the Froude number below 0,40, i.e. not coming too close to the second wave hump.

The length is, however, a limitation in many harbours and the pressure is to find successful configurations which can reach the high speeds (Froude numbers) at reasonable length/beam ratio, about 6,0 or even below and to increase the deadweight meaning increase of block coefficient.

<u>Pod Propulsion</u>

The pod propulsion configuration offers an interesting possibility for ferries, especially for higher speeds, above 27 knots. Aftship hull form can be optimised for the speed without the obstructing shaft arrangement and machinery foundations. Pod hulls, when properly located and streamlined, operate as stern bulbs damping the aftship wave system, which typically forms a big part of the total resistance at higher speeds.

The propulsion power saving is at least between 10-15%, even differences up to 20-25% have been measured in comparable model tests between conventional twin screw shaft arrangement and twin pod arrangement. The difference in propulsion power is summed up from the optimised hull form, absence of shaftlines and rudders and optimised position and orientation of pods.

A disadvantage, at least for the time being, is the height of the pod units inside the hull. They tend to protrude through the main deck. Pod makers are working on this issue and it looks possible to get even rather high powers, up to 25-30 MW within 3…3,5 m. And the main interest is if we can change into short beamy vessels with the help of wave damping of pods.

Figure 3-11 presents an example ferry designed for 30 knots. A large lower hold is applied. Diesel generators are located besides the hold in separate fire safe compartments. The vessel can fulfil the SOLAS 90 damage stability requirements with all two compartment damages together with lower hold. Side casings are designed for the full length of the main deck. Diesel-electric machinery gives good fuel efficiency at different operational modes and less exhaust gas emissions than operation at variable engine speed.

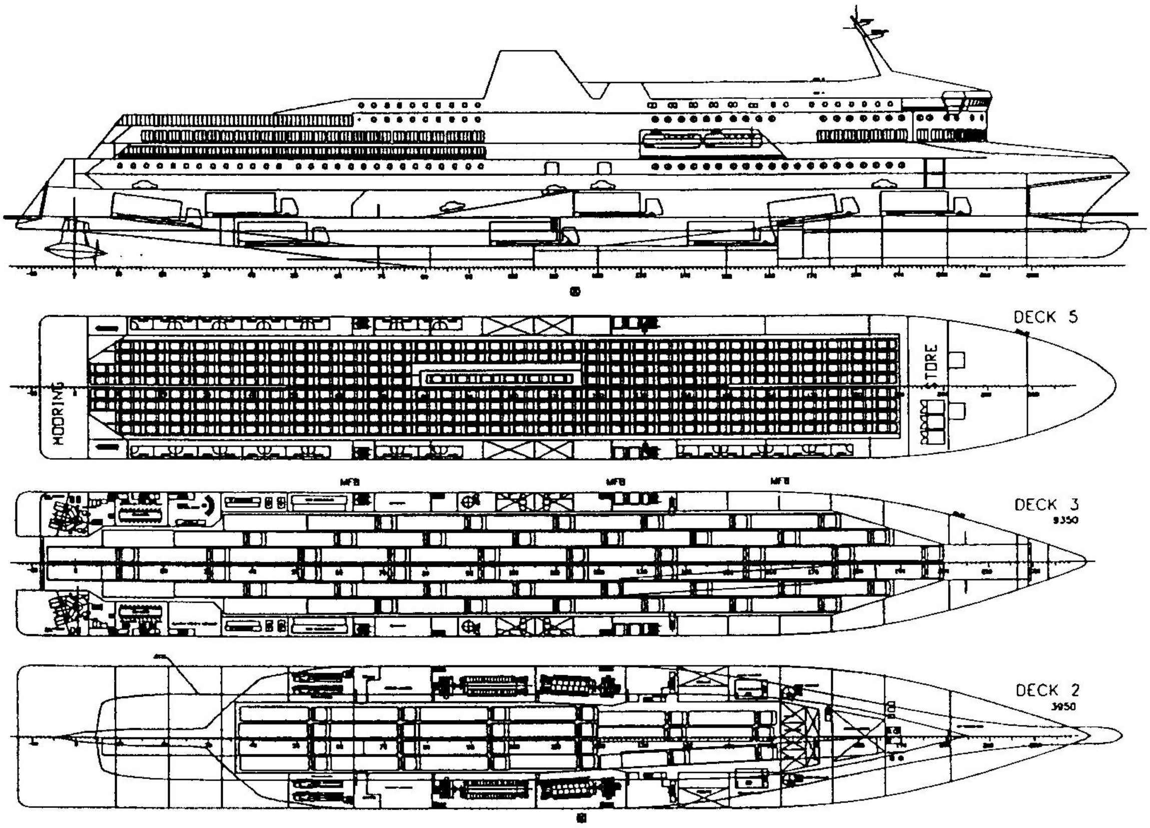

Figure 3-11 Example of a fast large size ferry with pod propulsion

4. Design Methods

General

Design and engineering methods can be subdivided in accordance with the different phases of a typical newbuilding project. It all starts with the first feasibility ideas and through project development phase we end up into the shipbuilding contract. The ship is defined with specification, arrangement drawings, principal system diagrams and descriptions, architect specification and documents, and with other possible technical documentation. A lot of different kind of documentation is produced, but mainly without any simulations of the ship's main function, such as cargo handling, passenger flows, service and maintenance flows, safety simulations and similar.

The next phase after the contract is basic design and coordination engineering together with procurement handling, master scheduling and build procedure planning. A lot of new parties are introduced into the process and the problem seems to be the coordination. Even within the design phase there are several parties involved and everybody is working within the same ship. Coordination becomes the major issue not only technically but it is also time consuming. On the other hand procurement requires good definitions of systems, areas, etc. to be purchased and design work cannot proceed without information of these systems. Efficient coordination and timing is required.

The next phase is detail engineering, work planning and preparation including not only the yard's own work but also the work of different subcontractors and suppliers.

There are no general nor specific tools and methods available which could be used throughout the different design and engineering phases, and even within each phase many different systems are used without proper link and coordination between each other. A lot of time consuming coordination problems arise.

There are generally several computer systems at shipyards and within the industry serving shipping companies and shipyards. They are, however, typically tailormade systems and they lack integration. Product data is spred out between different systems and companies without common product model and thus the data in each system lives its own life. Coordination takes time and in most cases the final coordination takes place only during the building and installation phase, sometimes leading to rather costly solutions. There is also high bureaucracy within the different systems and they support bureaucracy, not flexibility.

3-D Computer Modelling

3-dimensional computer models are still mainly prepared to compensate the actual plastic design models, i.e. the 3-D model is prepared to produce only workshop drawings. Most of the models, modelling techniques, are specific for structural design or piping design.

Models of complete vessels with all disciplines included are still rare. 3-D computer modelling technique is not used at the project design stage. 3-D computer modelling technique should be applied in accordance with the actual design procedure, i.e. starting from the project design phase before the shipbuilding contract is even signed. The same model should then be extended into a real product when design and engineering are proceeding.

Project Design with Virtual Reality Modelling

Possibilities for increased efficiencies in the newbuilding process are continuously searched not only by the shipyards but even shipping companies and suppliers are looking for more efficient products and processes.

Figure 4-1 Early project phase virtual model of a passenger cruise ship

In today's market situation shipyards as well as shipping companies are concentrating on their core business and outsourcing most of the support functions. This is to reduce costs but also to balance the use of capacity in the fluctuating market.

Extensive use of consultants, subcontractors and suppliers is a typical situation in the shipbuilding but is not only plain sailing. For the shipowner the product definition becomes of essential nature. Typical contract documents may be enough for the yard but when all the major systems and spaces on the ship are subcontracted proper and adequate definition of the ship at the earliest possible stage becomes of vital importance if not critical. At a later stage coordination between the different parties must be fluent and efficient to keep the schedule and, of course, all parties should understand the final end product in the same way, in accordance with the shipowner's original plan. This is, however, sometimes difficult when several different parties are involved and the end product is not always exactly as originally ordered.

To be able to serve the ship development process from the first idea up to the commissioning, a product model is required including all the necessary information for the complete process. The time in the process is also essential to be included in the model leading us to a 4D Product Model.

We have worked for about two years with the idea of introducing a 4D model at the earliest possible stage, and to make it as a generic model suitable for all the different tasks in the process.
Very soon the virtual reality became an essential part of the process and today we speak of Virtual Mock-ups in 4D where the 4th dimension is the project/process time.

Figure 4-2

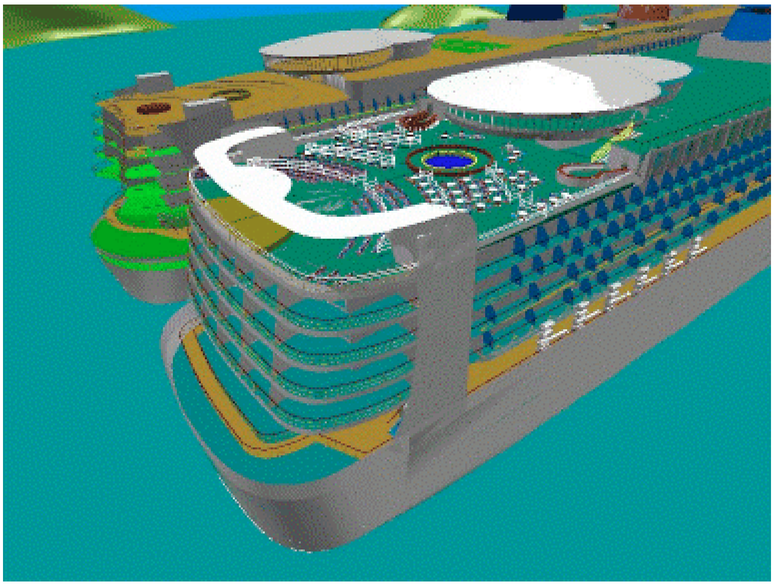

Figure 4-3

First virtual reality ship models were prepared spring-summer 1998, for a passenger cruise ship outline project. The first test case worked better than expected and the yard requested a full virtual reality model to be prepared including exterior, all public spaces, example cabins and selected service spaces. The main issue was to visualise the design of the ship exterior and public spaces. It was easy for everyone in project meetings and presentations to understand the specific and special features of for example unsymmetrical cone type atrium, front bulkhead passenger balcony arrangement and similar. It was also interesting to see how easily and quickly alternative solutions could be prepared and compared during the presentations. Figures 4-1 to 4-3 present photos of typical early project phase models.

The first full design coordination model was prepared for the Royal Caribbean International "Vantage" class vessel, and later on detail navigation bridge and engine room coordination models, both including also virtual reality simulations were added. Figure 4-4 presents the exterior model.

Figure 4-4 RCI´s "Vantage" class exterior model

Several conclusions can already today be made out of these projects. The number of required man-hours in preparing a virtual model is not more than is required for preparing typical arrangement drawings. Drawings can be extracted directly from the 4D model.

Visual presentation of any new idea, space or arrangement is easy and much more efficient than with drawings, renderings or even with cartoon models.

Figure 4-5 Typical project phase model

The ship can be projected in large scale on a screen which simplifies presentations and common meetings with the owner, yard, architect, consultant, subcontractor, etc. Fly-around and walk-through of the virtual ship in accordance with a predetermined route and spaces or as required in the meetings is possible. Different alternatives can be on display at the same time and fly-around or fly-in can be made for both.

Changes can be made on the spot in the meeting concerning colours, lighting, furniture, as well as arrangement, structures, equipment, furniture, etc. Major alternatives may require working 'overnight', but only overnight.
Visualisation of changes and alternatives is immediately available, decision making becomes easier and more reliable.

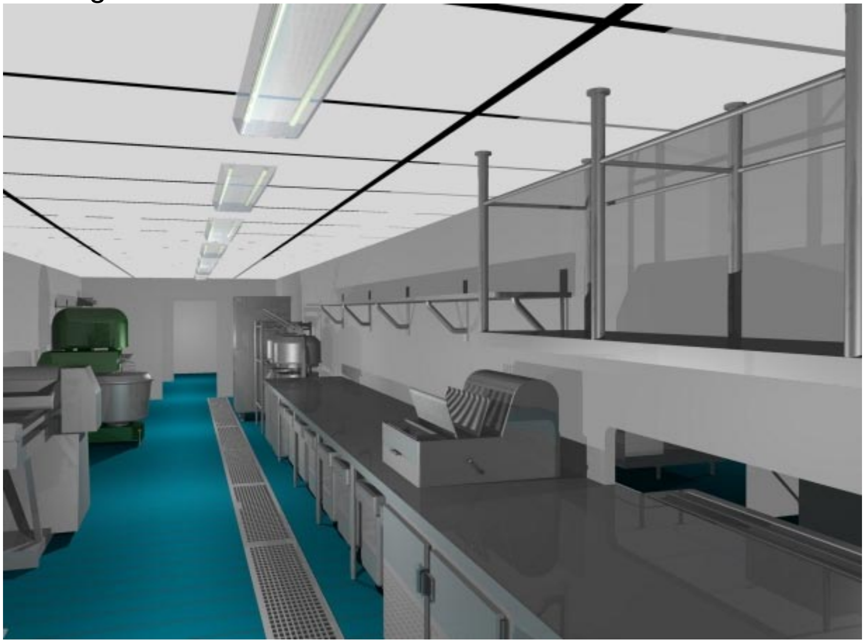

Figure 4-6 Early phase galley model

Functionality of spaces can also be easily checked with performance simulations in the virtual model utilising efficient simulation solutions. These can be ro-ro deck operations, or any kind of cargo operations, passenger flows, and escape simulations, luggage handling, galley, catering operations and similar.

The model can also be connected with virtual navigation simulator with models of any required harbour together with the mathematical of the ship's manoeuvring characteristics.

All this can be done already before the shipbuilding contract is signed, with one single product model well coordinated all the time.
A lot of typical misunderstandings and mistakes are avoided at the earliest possible stage.

Figure 4-7 Virtual bridge model

Practically this means that it is realistic and possible to have a good, well coordinated model available upon the signing of the newbuilding contract. This will save a lot of later coordination time and reduce remarkably discussions of contents and actual meaning of the newbuilding contract and design.

The same model can be used for calculating essential parameters in the design, i.e. areas, volumes, weight, centre of gravity, materials and, of course, costs.
Subcontractor and supplier inquiry models can be extracted as well and subcontractor work can be better defined before the shipbuilding contract is signed.

Basic Coordination Phase

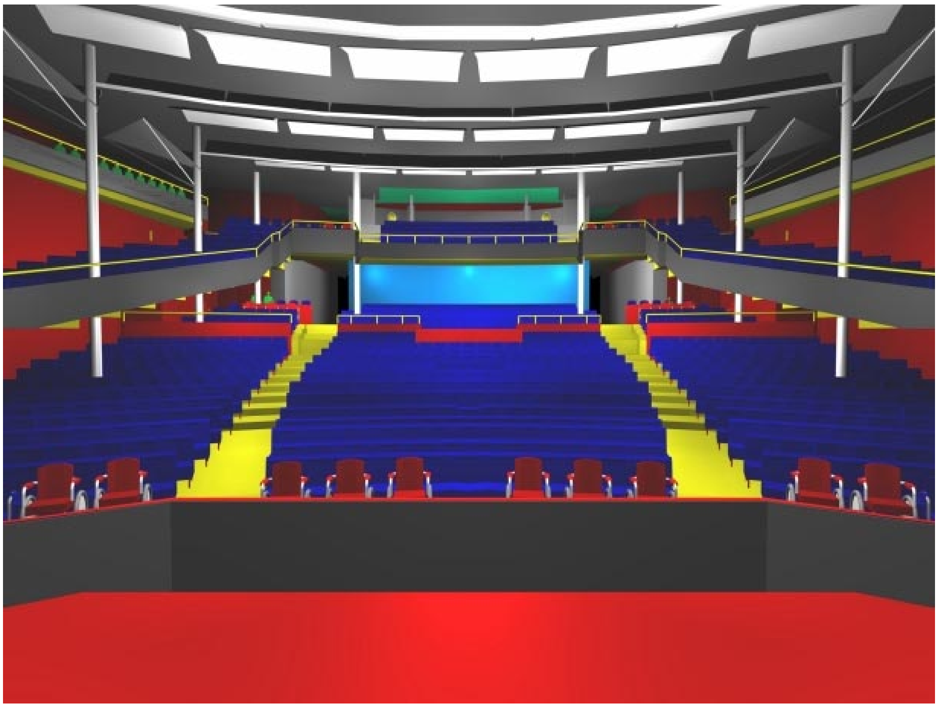

Figure 4-8 Virtual early stage theatre model for functionality checking

A detailed well defined virtual reality product model as contract document (CD-rom) of the newbuilding enables a quick start for the coordination, design, procurement and planning phase immediately after the contract is signed.

The same model can be used for technical design as basis for architectural design, as inquiry specification (or part of it), and as yard building procedure planning model. This reduces the risk of modifications saving both man-hours and, particularly, lead time.

The structural basic design can be completed in the same model and direct link into the major classification societies' models is studied at this moment. System diagrams will be prepared in the model including both system characteristics and important space reservations for ducting, piping and cable trays.

When the model is properly completed with piping, ducting, cable trays, main components, equipment and systems it forms a basis for turnkey contracting, detail production planning and workshop drawings.

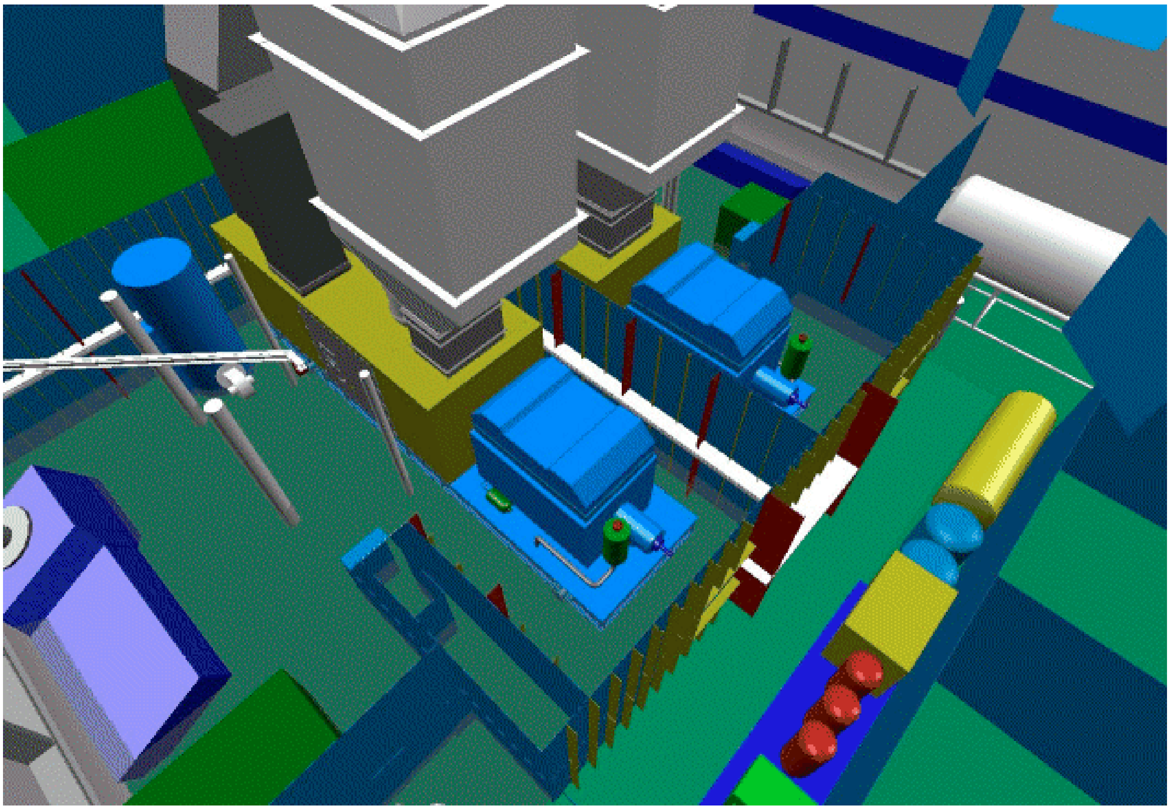

Figure 4-9 Engine room model for functional checking

Process Simulations

These virtual ship models are available as basis for process, safety, ro-ro and any kind of simulations as required and essential for a specific project.
First shipyard has been modelled and building procedure with block assembly, block and grand block outfitting and dry dock building stage are being simulated, checking critical phases, areas, material flows etc.

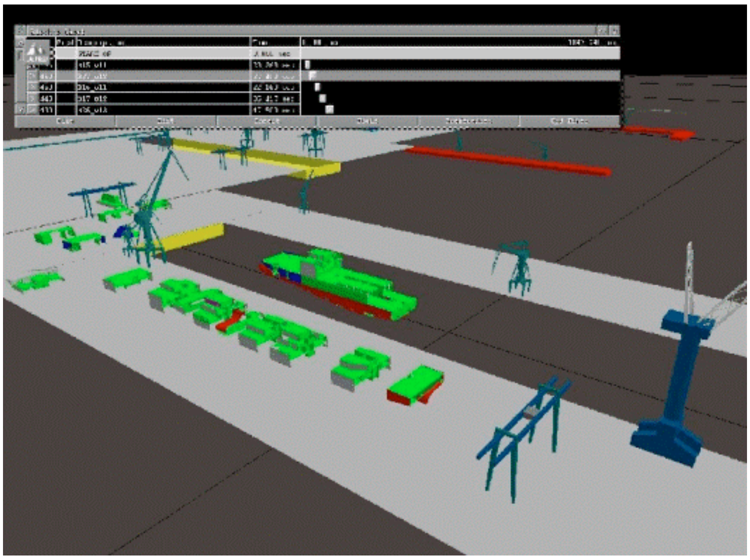

Figure 4-10 Virtual simulation of the Hellenic Shipyards in Greece

Safety simulations for a passenger vessel are required by IMO at least for the passenger evacuation process. No standard tools are yet available.
We decided to test the virtual ship models and process simulation tools available. First test cases have already proved that it is possible to simulate passenger evacuation process and related possible problems.
A ro-ro passenger ferry newbuilding for Strintzis Line was selected as a basic test case, having a rather high passenger capacity of 2000 in a handy size vessel of overall length of only 136 meters.

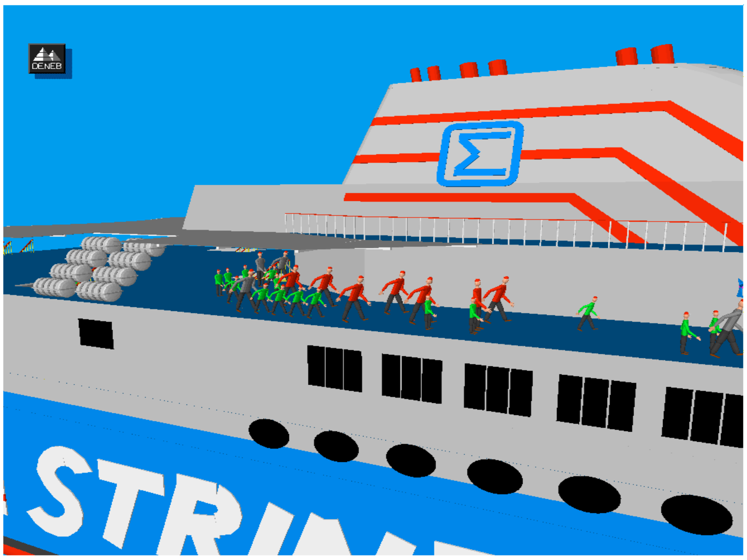

Figure 4-11 Virtual passenger evacuation simulation

A basic ship model was prepared with Catia Ship Solutions and the simulations were carried out with Deneb Robotic´s Quest simulation program. All spaces, corridors, cabins, public spaces, doors, tresholds etc. with their exact shape and locations can be read from the model database or from the configuration file. The number of passengers and their location are given to the model as parameters.

The number of passengers can be a constant value or it can be a common distribution to be varied in the simulation runs. Each passenger has an individual evacuation speed (walking, running, climbing etc.) The speed can be constant or distributed for different passenger groups such as adults, juniors, seniors etc.

When the basic ship model is ready, even a simple first stage outline project model can be used, and all parameters are set, simulations can be carried out.

The simulator locates single passenger randomly to the selected area in the ship, e.g. night versus day time situation. When all passengers are randomly set to their positions the alarm goes, and passengers start to react on the evacuation alarm and start to move to the muster station. Passengers can by pass each other when having different speeds and crowded and narrow passage ways will reduce the speed accordingly.
Simulations can be repeated with different distributions of speeds, passenger locations etc. to find out the most critical cases.

Simulation run gives the time required for the passengers to gather into the muster stations. A dedicated muster station can be assigned for each passenger simulating a mixed flow, panic situations can be created as well by forcing some of the passengers to stop or to go against the main stream.

Required evacuation times are received with possible bottle necks shown. An efficient tool for early project checking as well as for final engineering stages is developed. The next step is to include fires and flooding and to develop the models and simulations into training tools.

Complete 4D Virtual Product Model including structure, main piping, ducting, cable trays and main components within the basic design stage is split into hull sections, blocks and main construction phases. A virtual model of the shipyard with all the production lines, cranes, outfitting spaces, etc. is available. The construction process is easily simulated. Variations can be studied in a transparent mode. All parties involved can quickly understand the process and their specific part of it. Effect of block outfitting, required outfitting areas, use of modular construction and turnkey supplies can be easily demonstrated.
Figure 4-12 presents an example of build procedure description.

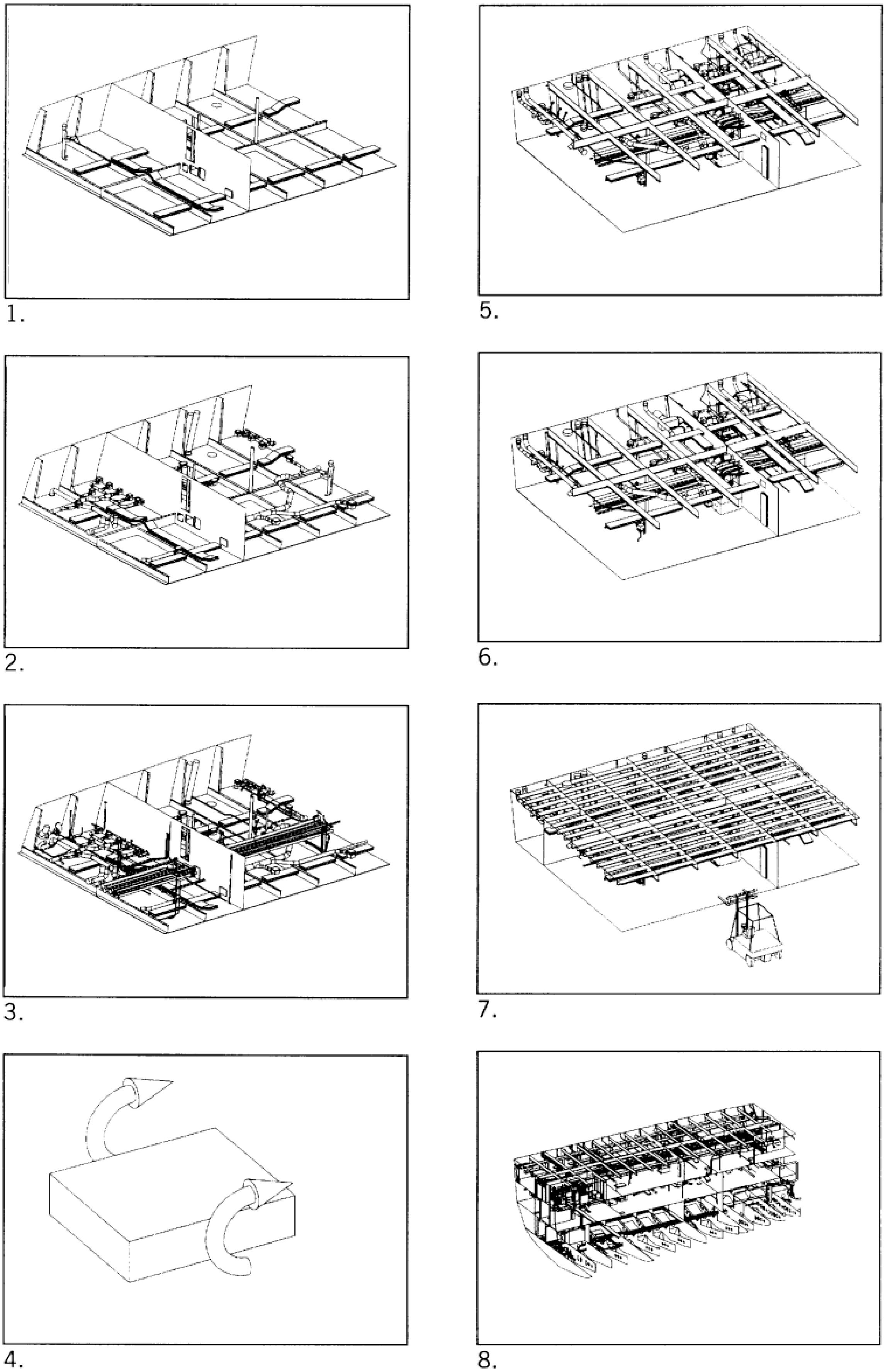

Figure 4-12 Build procedure description with different installation phases

The benefit for the turnkey supplier is obvious, he will get adequate technical data and installation process description even before the contract is signed, and it allows proper planning and detail design.
Normal planning, scheduling and follow-up tools are linked with the 4D Virtual Product Model and updating of schedule goes together with the model.

A systematic approach into ship newbuilding project is possible for the first time with the 4D Virtual Product Modelling technique.

The quantum leap in the information flow can be compared with what happened when the industry turned over from hand drafting into CAD and later on into 3D CAD some 10-15 years ago.

Today for a passenger cruise ship the total number of design man-hours can vary with 100.000 man-hours (difference) depending on the level and quality of basic design. In dollars this is 5-10 MUSD. The effect of using a virtual reality product model properly from the first project stage is leading to even bigger impacts. This is a huge potential, taking into account that the design activity forms about 10% of the total newbuilding costs, and the saving effect goes through the complete building process and not only the design.

The saving potential in the lead time can be from two months up to six months depending on the type of the vessel, yard, owner and procedures.

The vessel has a perpendicular length of 185,0 metres, beam of 26,80 metres and draught of 6,85 metres, and the required power for 28 knots is 31 200 kW.

The coming new probabilistic damage safety approach will give us an opportunity to disregard the B/5 lines and optimise the size of the lower hold and machinery arrangement.