Design Aspects of a High Speed Monohull Ferry

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ABSTRACT

The paper describes the design process of a high speed Monohull RoPax Ferry which operates at a Froude number of 0.4. The design task was quite challenging, as two possible transport concepts were in principle possible: Two ships were needed with a total speed of 50 knots, which could result in a combination of a 30kn high speed Catamaran plus a conventional 20kn RoPax Ferry or alternatively in two identical sister vessels of 25kn each. The solution with a high speed catamaran plus the conventional RoPax-Ferry defined the total cost budget, which must not be exceeded by the design. This resulted in a tough boundary condition and made life cycle cost evaluations necessary. Due to harbor restrictions, the length of the ships was limited by about 110m, resulting in a Froude number of about 0.4. This resulted in high costs for the propulsion system. The ferries should have open RoRo-Cargo spaces for cost reasons, which made the stability requirements (Weather criterion plus Stockholm Agreement) quite challenging. This also strongly influenced the design of the final hull form. As the ship is very sensitive to weight, detailed steel structure optimizations had to be carried out to optimize the main grillage systems of the vehicle decks. The hullform and the appendage design required careful optimization to guarantee the required service speed with the engine power which was available in the price budget. As no vessel of comparison was available, the speed power estimation as well as all design tasks had fully to rely on numerical predictions. As the ship has further demanding requirements for course keeping and comfort in waves, the optimization of the hull form must include also these issues.

KEY WORDS

RoRo- Passenger Ferry, Fast Monohull

INTRODUCTION

In 2012, a shipping company located in South Europe contacted the German Shipyard Flensburger Schiffbau- Gesellschaft, Flensburg, for the Design of a 100m, 650Pax, 150 cars RoPax day ferry with a service speed of 20 knots. The design was intended as a part of the fleet renewal program the shipping company had to carry carried out, as the existing ships needed a replacement. During the initial design discussions which took place before the beginning of the initial design phase, it turned out that the business case of the shipping company actually required two RoPax vessels of the same capacity, where the intended service required a ship speed of in total 50 knots for the two vessels. The ship speeds were initially chosen as 30 knots and 20 knots. The 30 knot vessel should be a high speed catamaran which could be ordered more or less from the shelf. The 20 knot RoPax was then intended as a more or less conventional RoPax- ferry. A concept consisting of these two ships was proven to be able to fulfil the transportation requirements with respect to capacity and schedule. The shipping company had actually chartered in such kind of vessels temporarily and was looking now for new buildings. In this context, FSG was requested to develop a design for the 20 knot conventional RoPax ferry. This solution was favored by the owner because it seemed to be quite cost effective: The high speed catamaran represented – according to the understanding of the owner – a series product which could be ordered quite cost effectively. The remaining 20 knot RoPax seemed then to be not too challenging from a technical point of view, and it was expected that it should be possible that this ship could be ordered at a reasonably low budget, too. However, in principle it was also possible to fulfil the transportation task with two individually designed RoPax ferries of 25 knots service speed each, which would then be identical sister vessels. The challenging question now was whether a concept consisting of two identical 25 knot vessels could achieve the same cost effectiveness as the

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30 knot high speed catamaran and the conventional 20 knot RoPax. The initial budget estimation showed that the main cost driver of the sister vessel design task was the machinery plant. In this respect, the concept of the two 25 knot vessels turned out to be competitive with respect to new building costs only if the delivered propulsion power (trial conditions) could be kept below 15000 kW. If this was technically possible, the two ships with 25 knots were more competitive with respect to operational costs compared to the high speed catamaran and the conventional RoPax for the following reason: As the ships were not intended to operate in SECAS, the 25 knot sister ships could be operated with HFO, as medium speed diesels were used. In contrast, the high speed catamaran needed to burn MDO as it was driven by high speed diesel engines, and the conventional RoPax could burn HFO again. Based on these initial findings, a concept consisting of two identical RoPax-Ferries was regarded as a serious design alternative in case it was possible to find a solution for the hull form which required not more than 15000kW delivered power for a speed of 25 knots. Because new building costs were comparable (and not exceeding the owner’s budget limitations), but the operational costs were lower due to the intended use of HFO. Based on a propulsion efficiency of about 0.73-0.74 this requirement resulted in a still water resistance of the ship of not more than 850kN. Consequently, the project could be realized in case a hull form concept could be developed which required not more than 850kN still water resistance (equivalent to 15000 kW propulsion power).

**DESIGN CONSTRAINTS**

Besides 650 passengers, the ship should also be able to carry 150 private cars and (optionally) 16 trucks. This results in two vehicle decks of in total 700 lane meters and public spaces for the 650 passengers. For cost reasons, the ship must not have more than two main fire zones. Otherwise, the safe-return-to-port regulations must have been fulfilled which makes the ship significantly more expensive. Due to harbor restrictions, the length of the ship should not exceed 110 m. The maximum draft was governed by one specific port which allowed not more than 4.40m draft at the pier. Loading and unloading should be carried out by a single stern ramp. As the ship will be operated in Europe, it must fulfill the Stockholm Agreement damage stability as well as the SOLAS 2009. For cost reasons, the vehicle decks were not fully closed initially, which resulted in the situation that the buoyancy body consists only of the hull up the bulkhead deck plus the watertight double hull on the main garage deck. This resulted in significant challenges for the intact stability regulations and gave some severe boundary conditions for the initial hull form design. The main propulsion plant should consist of two medium speed diesel engines, which operate via a gear box on two conventional shaft lines with controllable pitch propellers. To avoid excessive vibrations, the CPPs were intended to operate in a combinatory mode. Two twisted full spade rudders in the propeller slipstream should guarantee course keeping as well as sufficient turning ability. All these findings resulted in a displacement of about 4800t and a design Froude number of about 0.4. Besides finding a hull form which can fulfill all the given requirements, it was of utmost importance to identify possible weight saving options, where aluminum was not considered due to cost reasons.

**INITIAL CONSIDERATIONS FOR THE HULL FORM**

From the initial design considerations, the requirements for the hull form design were the following:

- Total still water resistance less than 850 kn at 25knots
- Displacement about 4800 t
- Length not more than 110m, draft (A.P) not more than 4.40m
- Sufficient stability to fulfil the weather criterion
- Vehicle deck height adjusted to fulfil Stockholm agreement damage stability standard

From initial computations it was quite clear that a conventional displacement mono hull would never lead to the required target resistance. These initial computations were based on CFD- predictions, as data from comparable vessels were not available. Therefore, a new concept needed to be developed. When looking at the principal wave making of any displacement vessel at a Froude number of about 0.4, the following general conclusions can be drawn: Wave making is generally characterized by a large wave trough located at about L/2 due to the acceleration of the flow by the hydraulic blockage of the main section. This large trough coincides with the trough of the bow wave system, which leads to an amplification of that wave system in the wake. The ship further has a significant stern wave system which is a consequence of the full beam transom. As a consequence of these principal findings, the main goal of the hull form development was seen in the reduction of the hydraulic blockage by the ship’s main section. This means a drastic reduction of the main section area. As a consequence, the lost buoyancy needed to be placed somewhere else. When retrieving the literature, a principally comparable hull form concept has been proposed by Langenberg in 1988 [1] by empirical design methods (trial and error). Although the principal aim of that development was to recover energy from the ship’s wake, the concept had a main section which significantly reduced the main section generated wave trough. The hull form concept proposed by Langenberg is shown in...
Fig. 1 (left row, bottom) in comparison to a conventional twin screw hull (left row, top), which can be taken as a typical conventional hull of those days for the given Froude number.

**Figure 1: Conventional Twin Screw (Top, Left) and the Hull Form proposed by Langenberg [1] (Bottom, Left). Right: Two Versions of the EPROSYS HIGH-Speed ferry [2],**

During the German BMWI-funded research project EPROSYS [2], the Langenberg concept was adopted to the design of a 200m, 50knot RoPax Ferry and further developed. The concept was adjusted for twin screw propulsion, and a transom was developed which was compatible with conventional propulsion instead of water jets. Extensive use of potential flow computations was made to find the best hull form for conventional propulsion. Two versions of this hull form development are shown in Fig. 1, right column. During the design process, it became necessary to improve the numerical procedures of the TUHH-Potential flow solver KELVIN, which had difficulties to correctly predict the stern wave system. Wave cut measurements were taken during the model tests and they served as a validation basis for the improvements of the potential flow solver. The computation of the wave system of EPROSYS at 0.4 Froude number shows the pronounced stern wave system as well as the dominating trough generated by the main section, see Fig. 2.

**Figure 2: Wave system of the EPROSYS ferry [2] at a Froude Number of 0.4 computed by the potential flow solver KELVIN (TUHH).**

**INITIAL HULL FORM DESIGN PROCESS**

The numerical and experimental experiences of the EPROSYS-project were extremely useful for the design of our 25knot RoPax-Ferry. Initial calculations showed that the target resistance of 850kN at 25 knots could be achieved if the residual resistance of the ferry could be kept in the same (relative) magnitude as it was measured for the EPROSYS-hull. This target was challenging, because the main dimensions of EPROSYS (220m length, 24.7m breadth) were much more favorable with respect to wave making as the possible main dimensions of the 25 knot RoPax-ferry (110m Length, later increased to 112m and 20m breadth). Therefore, as a starting point, the following decisions were made:
• The flare of the main section was increased to the maximum possible, taking into account potential slamming effects.
• The profile of the ship was designed exactly to the bottom profile of the most critical harbor. This allowed a draft at the forward perpendicular of 6.0 m and a draft of 4.4 m at the aft perpendicular.

As a consequence of these decisions, the block coefficient of the ship could be reduced to about 0.43 (based on the breadth of the water line). However, initial computations of the maneuvering behavior showed a tendency of the ship to be not as directionally stable as desired. This needed to be compensated by future design measures. As the resistance prognosis was quite uncertain, it was decided to perform bare hull resistance tests as soon as possible when the initial design was sufficiently stable. This required of course a basic design which needed to be technically sound with respect to the main design drivers. During the different stages of the hull form design which were always supported by potential flow computations, it turned out that the application of the IMO-weather criterion in combination with the Stockholm-Agreement damage stability standard resulted in some challenging boundary conditions not only for the hull form, but for the design in total. This may be derived from Fig. 3.

![Figure 3: Main section design (left) and weather Criterion (top right), Initial Design. Red: Intact buoyant volume, Green: Damage buoyant volume (Initial). Bottom Right: KGmax-Curves for Intact and Damage Stability and the Design Loading Conditions for the Initial Design.](image-url)
which is then significantly over predicted. Due to the over predicted roll period, the roll back angle is also calculated by far too large (see Fig. 3, top right) with the consequence that the roll back area which has to be compensated is also too large. Due to this general malfunction of the weather criterion, the stability requirement is too high and it becomes at the same time the limiting stability criterion (the same principle holds for any other method which uses a linearized representation of the righting lever curve to predict roll periods).

Increasing the form stability of the hull form would have meant to decrease the flare of the main section, but this was not possible due the large increase in ship resistance. On the other hand, systematic CFD computations have shown that it was favorable to increase the beam of the water line in the aft body. This resulted in a better hydrodynamic performance due to the reduction of the stern waves as well as in a better fulfilment of the weather criterion. Consequently, systematic design variations were carried out where the height of the bulkhead deck (main garage deck) was systematically varied. For each design variant, damage and intact stability was determined. Stability wise, the optimum height of the vehicle deck resulted in 7600mm above the base line with 1600mm residual freeboard at the design draft. Fig. 3, right, shows the limiting KGmax-curves and the loading conditions for this initial design variant. Now the boundary conditions for the preliminary design of the hull form were sufficiently settled in order to perform the necessary hydrodynamic optimizations. As the authors do not believe in automatic optimizations for complex problems, all hull form variants were designed manually after a careful inspection of the flow phenomena related to the previous variant. This approach requires of course powerful numerical tools for the variation of the hull form, automatic grid generation and for the post processing. Due to efficiency reasons, a potential flow solver was applied. During the fine tuning of the initial hull form, some 100 different variants were generated and carefully analyzed. The computed wave pattern of the most promising design variant at the end of this process is shown in Fig. 4. The comparison of Fig. 4 with Fig. 2 shows the improved resolution of the stern wave system at the high Froude number, which is a consequence of the numerical improvements of the flow solver.

![Figure 4: Wave Pattern of the most promising design variant (at a speed of 25 knots (left) and the body plan of this variant (right).](image)

From Fig. 4 the four important wave systems can be identified: The bow wave system is moderate and it does not interfere with the wave system generated by the mid ship. The stern wave system is also moderate. The dominating wave system is the transom generated wave system, but this is typically somewhat reduced in the propulsion condition. Optionally it is possible to further reduce these waves by the application of trim wedges. The design of the most efficient trim wedge can at present most efficiently be carried out during the experimental propulsion tests in the model basin. Numerical predictions of the optimum transom shape were in the past unfortunately not successful. Therefore, the shape of the transom was at this stage considered as preliminary only. The aft body of the ship is characterized by the large propeller tunnels which are required to achieve the required low pressure pulses. Due to the fact that the draft at the aft perpendicular could not be larger than 4.4m, the propeller clearance situation cannot be optimal. The fore body of the hull form is characterized by a small bulbous bow which was used to adjust the required LCB at low additional resistance. For this initial hull form, the bare hull resistance was estimated to about 720-750kN at 25knots. This left some room for the shaft line design as well as allowances for wind and the other appendages. At this design stage, bare hull resistance tests were conducted at FORCE Technology, Lyngby. These tests were found to be necessary to confirm the resistance prediction, which was at that moment only based on CFD-predictions. It must be mentioned again in this context that the whole project would have become obsolete due to cost reasons if the targeted resistance at 25knots would not have been met.
Figure 5: Wave pattern of the most promising initial design variant during the bare hull resistance tests at 25 knots.

The bare hull tests performed did in principle confirm the most important design assumptions. The resistance prediction was confirmed, which allowed to continue the project. Also the main characteristics of the wave pattern during the tests showed a good agreement with the calculations. As already indicated, some optimization potential for the flow clearing off the stern was indicated (see Fig. 5, right). Unfortunately, there was a significant difference between the dynamic trim predicted by the flow solver (abt. 1m down by head) and the model experiment (nearly even keel). This lead to the suspicion that due to the wrongly predicted trim, the position of the transom was in fact too low, and an additional test was performed with 1m trim down by head. Astonishingly, this 1m forward trim did neither show any measurable effect on the resistance nor on the transom flow characteristics. From the model test results, it became clear that our CFD-solver needed some improvement with respect to the correct prediction of then dynamic trim. Otherwise, it would not have been possible to further optimize the aft body. The improvement of the code could then be found by substituting the linearization of the hydrostatic stiffness matrix during the iteration of the dynamic equilibrium floating condition.

DESIGN OF MAIN SECTION AND BASIC STEEL STRUCTURE DESIGN

As the model tests have verified the principal assumptions, the design could be continued and needed the finalization. Unfortunately, at that stage the potential owner had put forward the request for additional passenger cabins in the accommodation without increasing the total cost budget. Technically, this was not possible with the existing design due to stability reasons. Therefore, it was decided to close the superstructure in the aft body, which then required ventilation (extra costs) and about 40 tons of extra steel. But the integration of the additional passenger cabins required an increased beam of the vessel, which resulted in additional steel weight of about another 40 tons. To keep the ship in cost budget, it was decided to replace two auxiliary diesel engines by shaft generators. Due to pressure pulse considerations it was then necessary to operate the CPP in a strict combinatory mode in the open sea which made adjustments at the main engine turbo chargers necessary. The only problem then remained that the total weight of the ship was increased by about 120 tons. It was found hardly possible to increase the buoyancy of the ship without any detrimental effect on the resistance, as the existing hull had already been thoroughly optimized. From the previous CFD-investigation, the general trend was identified that the wave making of the hull form was not sensitive to a local increase of the transom´s with.

Figure 6: Automatically generated steel structure model for rule scantlings determination (here: GL-POSEIDON, left) and automatically generated grillage model of the vehicle deck 4 including the design loads (middle) and computed deflections (right).
Additional buoyancy could then be generated by increasing the transom width of the design water line in the aft body, which lead to a delta- shaped water line of the aft body. Further, it was decided to add a 2m duck tail to gain some savings from the stern waves. However, all these measures were considered not to be sufficient to achieve the required resistance under the new boundary conditions. Therefore it was decided to step into detailed optimizations of the steel structure to analyze further potential weight savings. Most relevant for the steel structure optimization was the determination of the optimal web frame distance. Increasing the web frame distance will result in a decreased weight contribution from the transverse members, but it will increase the weight contribution from the longitudinal members. Rox [3] has proposed a design methodology for the main section of RoRo- Vessels where the layout of the web frames and longitudinal girders is determined by grillage computations which are automatically generated from the steel structure design model. At the same time, the scantlings of the longitudinal members can be automatically determined by electronic rule scantlings (e.g. POSEIDON or NAUTICUS) if the steel structure design model is automatically transferred into these rule scantling tools. Further, all design loads required for the determination of the rule scantlings need to be defined and to be automatically transferred into the rule scantling determination software. As the scantlings of the longitudinal members do depend on their unsupported length, which is the web frame distance, the whole problem can easily be automated with respect to this web frame distance. If once a web frame distance is defined, a grillage model (see Fig. 6) can be derived from the steel structure model to optimize the scantlings of the web frames and the longitudinal girders from the direct calculation of the maximum stresses.

Each deck is then represented by an individual grillage, and each grillage can be optimized in such a way that the maximum stresses under the assumed loading situation are the same for the web frames and the longitudinal girders. From these calculations, the scantlings of the web frames and the longitudinal girders can be obtained. If the web frame distance is varied, the scantlings of the longitudinal members can be obtained for each web frame distance from automated rule scantling computations. The result of this optimization process is shown in figure 7. As expected, the weight per m of the transversal members decreases with increasing web frame distance, whereas the weight per m of the longitudinal members shows the opposite behavior. The optimum was found at a web frame distance of 3200 mm. The frame spacing was adjusted to 800mm with a web at every 4th frame. The watertight bulkheads were then adjusted to fit into the new frame spacing. Due to this optimization, about 90 tons in steel weight could be saved. These 90 tons result from the optimized deck grillages (50t) and from the optimized web frame spacing (40t). Figure 7 does also show that the initially selected web frame spacing was already quite good, but not optimal. These computations together resulted in the situation that the displacement of the hull form needed to be increased by about 60 tons. Unfortunately, the LCB had additionally to be moved 35cm forward, which was counteracted by the modification of the design waterline of the aft body. These circumstances made a new hull form design necessary, where the beam had to be increased by 0.5m to and the LCB had to be moved forward. Some additional buoyancy was also required to compensate the increase in mass.

![Figure 7: Steel weight per m of the longitudinal and transversal members as a function of the web frame distance.](image)

**FINALIZATION OF THE HULL FORM**

The forward shift of LCB could be obtained from an increased volume of the bulbous bow. From systematic CFD-computations it was found that the bulb could tolerate more displacement at no or very small costs in resistance. An increase of the transom beam resulted in a slight reduction of the stern waves and gave some additional displacement. The additional displacement from these two measures allowed to further reduce the flare of the main section. Due to the improvements in our flow solver, the transom could now better be adapted to the dynamic sinkage at the A.P. The comparison between Fig. 4 and Fig. 8 shows that despite the increase in buoyancy, the wave resistance could again be reduced. But one also has to
mention that about 50 further individual additional hull forms needed to be designed and numerically tested to achieve the improvements from Fig. 4 to Fig. 8. But for sure numerical methods are useful tools to improve the quality of the ship design.

Figure 8: Wave pattern of the finalized design at a speed of 25 knots (left), computed by the potential flow solver KELVIN (left) and the body plan of the finalized design (right).

APPENDAGE DESIGN AND DESIGN FINALIZATION

Figure 9: Profile of the finalized ferry design

The propulsion plant consists of two medium speed diesel engines which directly operate via a conventional shaft line on the CPPs. The electric power is supplied by two shaft generators and an auxiliary diesel engine. The controllable pitch propellers are operated in a combinatory mode to avoid the onset of face side cavitation within in the operational profile on sea. Nevertheless it was found necessary to carefully design the shaft line layout and the brackets to achieve a wake field which allows for the desired CPP operation. The additional resistance of the appendages should also be kept as small as possible. Unfortunately, whirling calculations of the long shaft lines (see also Fig. 9) have shown that a second pair of brackets became necessary. It was decided to arrange one pair of brackets horizontally, whereas the other pair of brackets was arranged in the 2o’clock position. The strong center skeg could carry the required transversal forces of the horizontal brackets. This initial arrangement was carefully fine-tuned with RANS- calculations, see Fig. 10. The RANS- calculations included also computations of the total resistance of the ship which indicated that the target resistance would most probably be met. The computations also showed a sensitive behavior with respect to the layout of the transom. This general trend was also found by the potential flow computations carried out before.

Figure 10: RANS- mesh of hull and appendages (top left), and computation of the wake field (bottom, left). The final layout of the appendages is shown on the right side.
As hull form and appendages were finalized, it was possible to run resistance and propulsion tests to verify the required propulsive power. As all previous computations have indicated that the hull form was sensitive with respect to the final shape of the transom, it was decided to test three alternative transom designs during the propulsion condition. With the best variant, the whole resistance and propulsion program was carried out. Figure 11, top left, shows the different arrangements of the transom tested during the propulsion condition in the large towing tank of HSVA. The best results were achieved with transom 10010, although the results did not differ very much from those obtained with transom 10000. The largest power demand was obtained from the version with the smallest trim wedge. This effect could not be foreseen by the CFD-calculations, but these calculations have shown an unstable behavior of the transom flow and have therefore initiated these additional tests. Based on the best variant, the resulting power demand was slightly below the initially targeted value of 15000 kW. It can be expected that some smaller savings might be possible if the tests will be repeated with the final propeller design. But the tests have confirmed that the design has performed successfully.

Figure 11: Shapes of transoms tested (top left), speed power results (top right) and wave elevation along the hull during the HSVA-tests (bottom).

CONCLUSIONS

An unconventional design task has been presented, which basically required two passenger ships with a speed of together 50 knots. Initially, this resulted in a solution of one high-speed catamaran of 30 knots and a conventional RoPax of 20 knots. This initial solution has defined the budget for the design of two identical 25 knot RoPax ferries, which had to achieve the 25 knots at a propulsion concept within the limitations of the budget. This resulted in an unconventional twin screw hull form which was developed in order to achieve the given resistance. The final design fulfilled the design requirements, and it was obtained after a sophisticated design process which was strongly supported by direct calculation methods, which are nowadays useful tools to support the design of modern ships.

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