Finite Element Analysis of a Catamaran Steel Hull

ทิพยวรรณ ดวงมณี[1], Bland, P.W.[2*], สุทัศน์ ลีลาทวีวัฒน์[3]
[1,2]ภาควิชาวิศวกรรมเครื่องกล คณะวิศวกรรมศาสตร์ มหาวิทยาลัยวิทยาศาสตร์และเทคโนโลยีแห่งเอเชีย,
89 หมู่ 12 ทางหลวง 331,อ.บางละมุง, จ.ชลบุรี, 20260
โทร 0-38754450 โทรสาร 0-38754460 E-mail: sap_green@hotmail.com[1], pwbland@asianust.ac.th[2]
[3]ภาควิชาวิศวกรรมศาสตร์โยธา คณะวิศวกรรมศาสตร์ มหาวิทยาลัยพระจอมเกล้าธนบุรี, 91 ถ. ประชาอุทิศ, เขต ทุ่งครุ, กทม 10140
โทร 0-2470-9141 โทรสาร 0-2427-9063 Email: sutat.lee@kmutt.ac.th

Tipayawan Duangmanee[1], P. W. Bland [2*], Sutat Leelataviwat[3]
[1,2]Department of Mechanical Engineering, Faculty of Engineering, Asian University,
89 Moo 12 Highway 331, Banglamung, Chonburi, 20260.
Tel: 0-38754450 Fax: 0-38754460 E-mail: sap_green@hotmail.com[1], pwbland@asianust.ac.th[2]
[3] Department of Civil Engineering, King Mongkut's University of Technology Thonburi
91 Prachauthit Road, Thung Khru District, Bangkok 10140.
Tel: 0-2470-9141 Fax: 0-2427-9063 Email: sutat.lee@kmutt.ac.th

Abstract

Some strength assessment criteria for ships' hulls have been established from experience or tests on full-scale craft, generating empirical design rules and then documented in standards. The main objective of this work was to develop an FE model to assess the strength of a benchmark design of a steel catamaran hull, and make comparisons to the relevant standards, allowing recommendations of further weight saving and/or increase in strength. The FE design and load assessment criteria applied to the benchmark design gave yield strength safety factors (SF) of 1.5 for the main transverse frames and 2.8 for the plates welded around the frames, showing potential for reducing weight. A reduction of plate thickness from 5 to 4mm saves 5.4% of the total ship weight, giving a SF of 2.1. Adding a radius to selected welded plates around the frame increased the frame SF to 1.7. Looking more widely, this project reflects the trend in growth of FE as a tool in all engineering industry.

1. Introduction

1.1 Project scope

This project performed a theoretical analysis, in parallel with full-scale prototype hull water tests carried out elsewhere. This paper reports the results of the theoretical analysis, with the following three main steps of work:

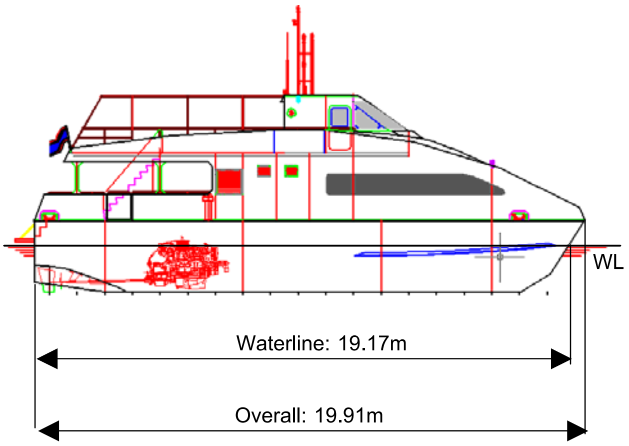
1. Assessment of the benchmark design (existing design of prototype hull): Using an appropriate standard [1] to compare recommended dimensions to the existing dimensions of key components.
2. Finite element analysis: Modelling of the benchmark design to predict the actual stresses in key components for different loading scenarios.
3. Weight reduction: Identify potential weight saving, with an estimate of the saving as a percentage of the total ships' weight.

1.2 The benchmark design

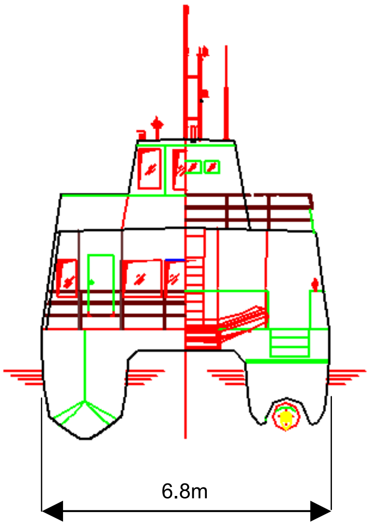
The ship is intended for scuba diving trips with a maximum speed of 24 knots and displacement of seawater of 60 tonnes. Fig.s 1-1a to 1-1c show a schematic representation of the ship with key dimensions. The ship consists of 19 transverse frames all 1m apart along its length, with a T-cross section, and 5mm plates welded around the edge of the frame forming the hull. Fig. 1-2 shows a single transverse frame with the location of the plates, labelled as: Deck plates (P1), side plates (P2&8), bilge plates (P3&7), bottom plates (P4&6) and keel plates (P5). Fig. 1-3 shows the T-cross section, view A-A from Fig. 1-2.

---

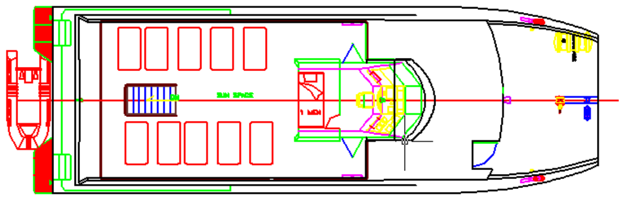
[*] Corresponding author



a) Side view from starboard



b) End view, split between bow and stern



c) Top view

Figure 1-1 Schematic representation of the subject catamaran

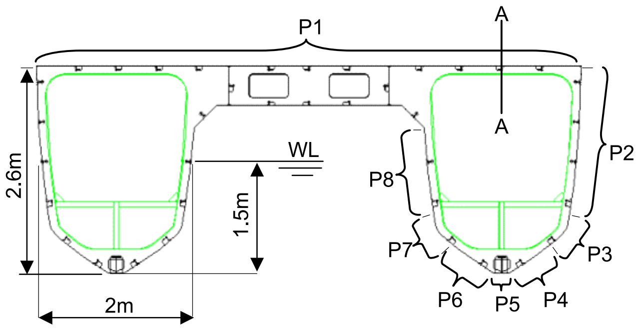


Fig. 1-2 Transverse frame showing plate locations. View A-A is shown in Fig. 1-3.

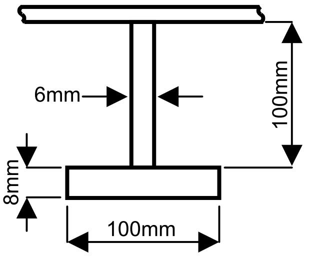


Fig. 1-3 Transverse frame T-cross section, shown with a length of plate, from view A-A in Fig. 1-2.

The structure is further reinforced by longitudinal stiffeners between every frame, with most stiffeners 0.5m apart. The three types of stiffener are; Type 1: L-angle 50×50×5 mm; Type 2: L-angle 65×50×5; Type 3: rectangular bar 75×8mm. Fig. 1-4 shows a larger view of half of a transverse frame, with the location of the stiffeners. A-H represents type1, I-J and R-T are type 2, and K-Q are type 3.

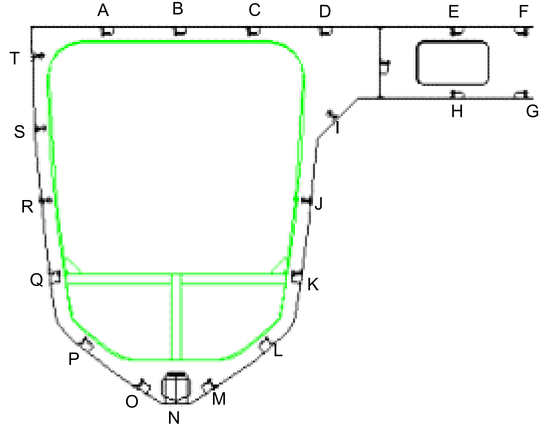
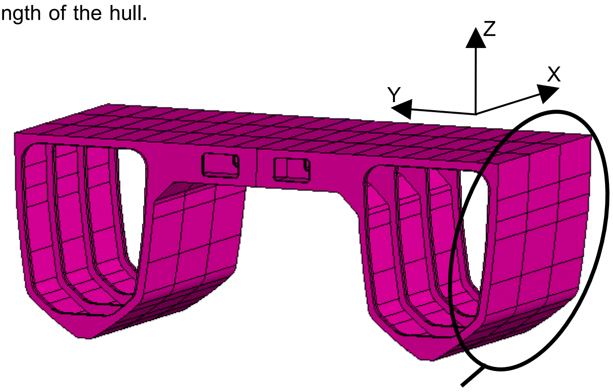


Fig. 1-4 Type and location of stiffeners

Fig. 1-5 gives a 3D view of four transverse frames and the plates and stiffeners between them. This is repeated along the length of the hull.



A larger view of highlighted side plates is shown in Fig 3-1

Fig. 1-5 3D view of hull structure

2. DNV Compliance

2.1 Component size

The standard used in the assessment [1] looks at the design speed, displacement, spacing between frames and stiffeners, various other dimensions such as overall length or draft, as well as material properties and other standard engineering variables. The standard uses empirical formulae, giving the required plate thickness and stiffener dimensions. The comparison of the DNV and benchmark's plate size is shown in table 2-1. The section modulus of the longitudinal stiffeners from DNV was 4.8 cm^3 whilst that of the 3 types of longitudinal stiffeners of the benchmark design were 3.2, 5.2 and 7.5 cm^3 for types 1, 2 and 3 respectively. In general, the required member sizes by the DNV are comparable to what have been provided in the existing design. The DNV standard appears to be more conservative requiring larger member sizes in locations where discontinuity may occur.

Table 2-1 Plate thickness (mm) of DNV and benchmark

	Keel	Bottom Bilge	Side	Deck
DNV	8	6	6	5
Benchmark	5	5	5	5

2.2 Safety factors

In this paper, the SF is defined as the calculated stress divided by the yield stress, taken as 250 MPa [3], The DNV standard does not explicitly quote a SF, but gives allowable stress (normal, shear and equivalent stress) as a function of the grade of steel used and also dependent on the component. For a grade of steel with a yield strength of not less than 235 MPa (closest match to that used in the benchmark design), the maximum allowable equivalent stress for girders is given as 180 Mpa. This could be interpreted as a safety factor of 1.3.

3. FE analysis

3.1 FE Models

3D shell and beam elements with six degrees of freedom were used in the FE analysis. The shell element was defined for the plate and frame components, capable of in-plane, normal loads and bending. The beam element represented the longitudinal stiffeners, defining the neutral axis, second moment of area, and cross sectional area. Two models were created. The first model (Model 1) was used to assess the side plate and the second model (Model 2) was used to assess the transverse frame. Two cases of boundary conditions, fully fixed and pinned, were assumed for the first model. These two boundary conditions were used to represent bounds of the real boundary condition.

3.2 Loading scenarios

Loading for Model 1 consisted of a slamming pressure applied to the side plates, as shown in Fig. 3-1. Loading for Model 2 consisted of applying forces to the frame at the location of the stiffeners under two load cases, such that there is a bending moment applied to the frame about the x-axis and symmetrical about the ship's centreline [2]. The first load case applied forces from the hull centreline to vessel centreline (stiffener locations G to N). The second load case applied forces from the hull centreline to the waterline (stiffener locations N to R). Maximum stresses in the structure were then identified for Model 1, Model 2 load case 1 and Model 2 load case 2.

3.3 Results

3.3.1 Model 1

For a slamming pressure of 18.35 kN/m^2, the stress in the side plates does not exceed 70 MPa for rigidly built in boundary conditions and 90 MPa for pinned jointed boundary conditions. The real boundary condition and hence actual maximum stress is somewhere between these two values. However, considering the worst case of 90 MPa, the plate still has a SF of 2.8 and thus gives the possibility for reduction of material which will be presented in section 4.

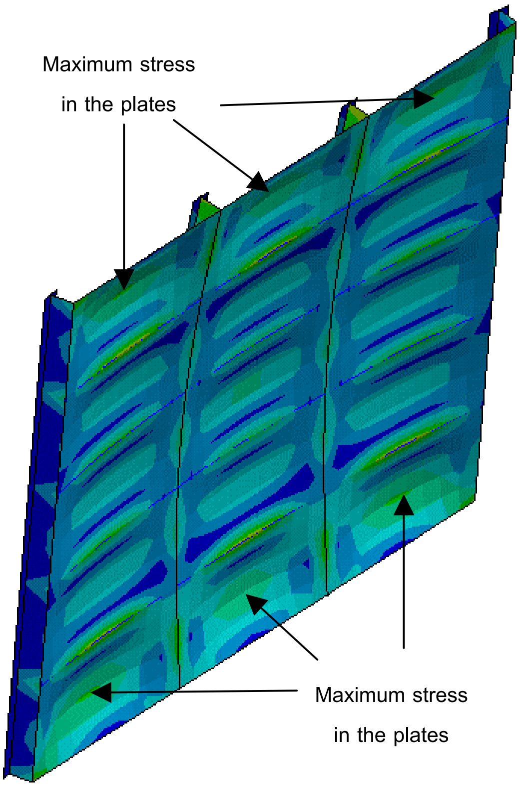


Fig. 3-1 Model 1 under uniform slamming pressure, pinned BC

3.3.2 Model 2

For an applied bending moment of 383 kNm, the first load case gives a maximum stress of 170 MPa and the second load case gives a maximum stress of 165 MPa, as shown in Fig. 3-2 and Fig. 3-3. The maximum stress for both load cases occurs at one of the corners in the frame, with most of the rest of the frame experiencing stresses not more than 80 MPa. Taking the maximum stress of the two load cases, the benchmark design transverse frame has a SF of 1.5.

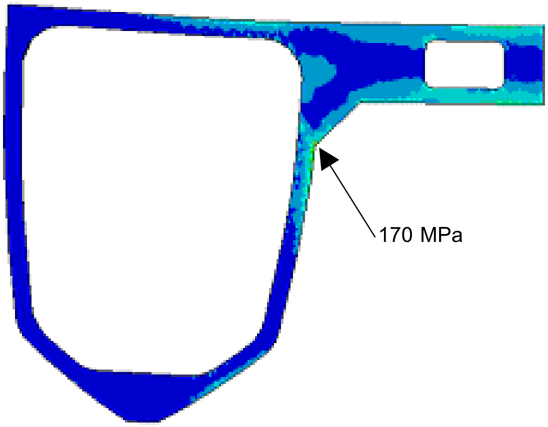


Fig. 3-2 Model 2 under 1st load case

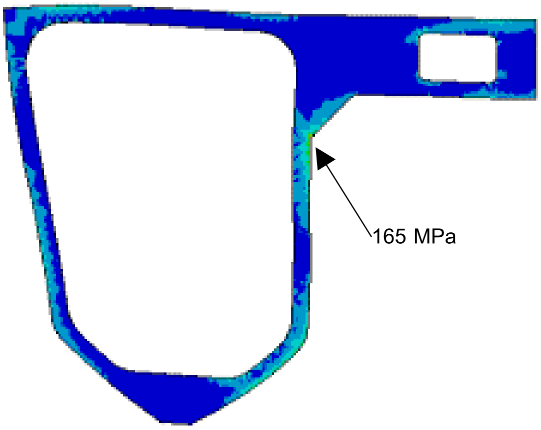


Fig. 3-3 Model 2 under 2nd load case

4. Weight Reduction

4.1 Plate thickness

Reducing the plate thickness from 5 to 4 mm gives an overall saving of 5.4% of the weight of the ship. For the worst case of assuming pinned jointed boundary conditions, the maximum stress increases from 90 to 120 MPa, hence decreasing the SF from 2.8 to 2.1. A thickness of 4mm also satisfies the buckling stress criteria stated in the standard [1].

4.2 Frame modifications

It is possible to reduce the weight of the frame by removing material from low stressed regions. As an example, a hole 300 mm in diameter with a 50×5 flange was added as shown in Fig. 4-1 and Fig. 4-2, without increasing the maximum stress. The maximum stress can be reduced by smoothing the sharp corners between welded plates, shown in close up in Fig. 4-3. This may be done in practise as part of the manufacturing process, and is a reminder of the difference between the FE model and the real structure. As a result, the highest stress was reduced to 145 MPa for model 2 load case 1, giving an increased SF of 1.7. However the extra manufacturing process of adding the hole and flange may not be cost effective, compared to the weight saved.

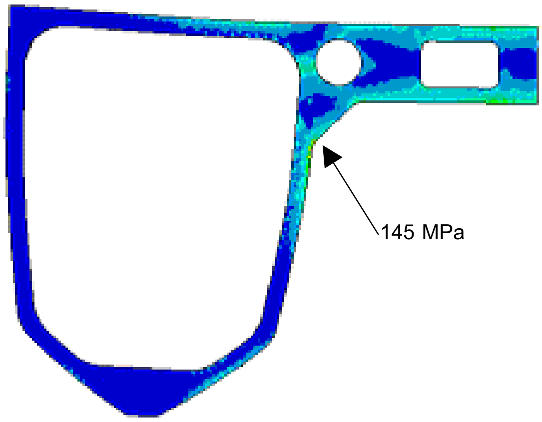


Fig. 4-1 Modified Model 2 under 1st load case

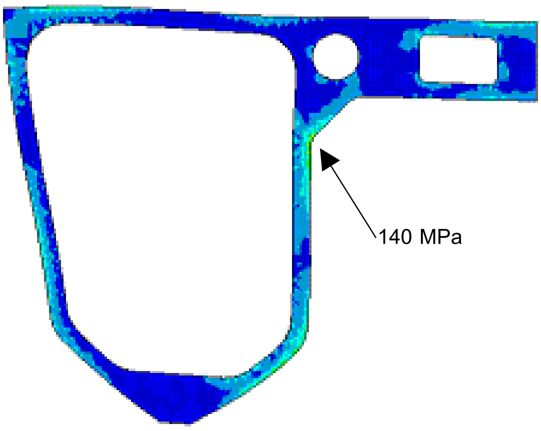


Fig. 4-2 Modified Model 2 under 2nd load case

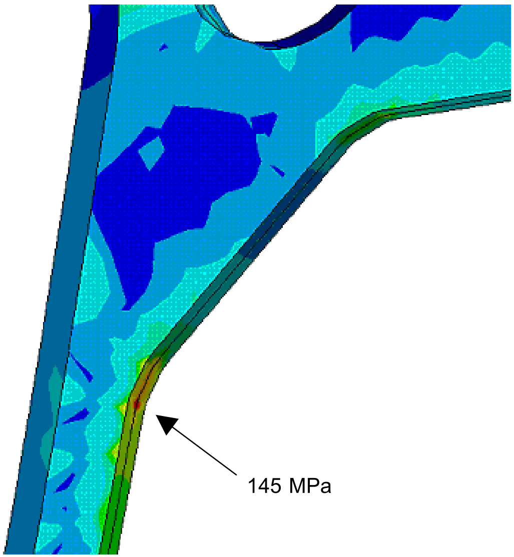


Fig. 4-3 Modified model 2 under 1st load case

5. Conclusion

This study has shown that, for this particular catamaran, a marine design standard using empirical formulae appears to be conservative in terms of plate thickness and section modulus of key structural components. This was verified by an FE model of a benchmark design, giving satisfactory yield strength safety factors from 1.5 to 2.8. Small modifications to the benchmark design were made to demonstrate the possibility of weight saving and to even out the range of safety factors, giving a weight saving of 5.4% and a safety factor range of 1.7 to 2.1.

6. Acknowledgements

The authors gratefully acknowledge Mr. Jirasak Jariyakosol, formerly a naval architect with Asian Marine Services, for his generous assistance in carrying out this research.

References

[1] Det Norske Veritas, “Rules for Classification of High Speed and Light Craft”, Part 3 Chapters 1 and 2.

[2] R.G. Latorre and P.P. Herrington, “Design of a 33-Knot Aluminum Catamaran Ferry”, Marine Technology, Vol.37, No.2, Spring 2000, pp.88-99.

[3] R.C. Hibbeler, “Mechanics of Materials”, ISBN 0-13-120200-6, 2003.