COMPARATIVE STUDIES ON BUCKLING STRENGTH BETWEEN THE SWEDGE-STIFFENED AND THE RING-STIFFENED OF THE MIDGET TYPE SUBMARINE PRESSURE HULL

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ABSTRACT

The pressure hull on a submarine usually consists of a long cylinder reinforced with a ring-shaped frame, so as not to fail to withstand the hydrostatic pressure load when the submarine is submerged at the depth of the ocean. This study focused on pressure-hull construction buckling strength and weight of structures using a ring stiffened compare with swedge stiffened structure system. Investigation of buckling strength in both types of pressure hull is done by considering the position of transverse bulkhead, frame spacing and volume of compartment. Linear buckling analysis was performed using finite element method, and the laying configuration of transverse bulkhead and frame spacing has been proposed to determine the influence of each structural configuration on buckling strength and weight of pressure hull structure.

Keywords: pressure hull, ring stiffened, swedge stiffened, weight of structure.

INTRODUCTION

Yudo *et al.* (2017) has presented a paper on the investigation of buckling strength behavior in ring-stiffened submarine pressure. The results show that an appropriate arrangement of transverse bulkhead laying configuration on the ring-stiffened pressure hull can provide a suitable buckling strength. However, based on a numerical study using a mathematical model performed by Ross (1987), an alternative design of a swedge stiffened pressure hull structure is believed to have a more efficient structure compared to traditional ring-stiffened vessel design.

The traditional submarine pressure hull consists of a lengthened cylindrical shell, and if this vessel is not strengthened by a ring it will fail due to the compressive load applied to the structure. When the ring stiffeners are able to withstand sufficiently, providing the ring stiffeners will give an additional strength of buckling. However if the ring stiffeners is not strong enough, the combination of ring and shell structure will fail due to the occurrence of general instability. An alternative method to improve the pressure hull strength towards the incidence of general instability is to provide a swedge section on the pressure hull shell.

An additional advantage of the stiffened cylinder swedge is that it is more suitable for noise insulation. Axial swedge construction also provides an elastic structure compared to a stiffened pressure hull ring. This characteristic gives an added shock protection when a torpedo is launched. In very thin swedge-stiffened cylinder shells such as those used in missiles are easier to be assembled than ring-stiffened system.

Based on some of the ring-stiffened and swedge-stiffened characteristics which are described above, This study is focused on the comparison of buckling strength and structure weight between the ring-stiffened and swedge-stiffened pressure hull, considering the configuration of the transverse bulkhead position, frame spacing and the volume of pressure hull compartment.

DESIGN METHODS FOR PRESSURE HULLS

Traditional pressure hull design methods are deterministic in nature, and are based on classical elastic buckling and stress calculations. The strength-reducing effects - geometric imperfections, residual stresses, boundary conditions, etc. - are dealt with differently for inter-frame and overall collapse. Inter-frame collapse is predicted using empirical design curves, which inherently take the collapse mechanisms and fabrication processes of real pressure hulls into account. Analytical methods that account for geometric imperfections and residual stresses are used to calculate overall collapse pressures, based on the occurrence of first yield (P_y(n)) or by using the full elasto-plastic strength of the structure (P_{co}).

Zakki *et al*. (2017) have studied on the design of the swedge-stiffened pressure hull structures to redesign the pressure-hull construction of a midget type submarines owned by the Indonesian Ministry of Defense. Evaluation of buckling strength was investigated with the pressure load for submarine operations at depths of 100m, 200m and 500m.

Kendrick (1982) presented an overview of externally loaded pressure vessel design criteria based on the BS5500 design code (BSI, 1980). The design methods outlined by Kendrick (or a slightly modified version) were used in many contemporary codes (e.g. ECCS, 1988) and are still standard practice today (e.g. DPA, 2001).

The BS5500 approach to design of pressure hulls is to proportion the structure such that: 1) inter-frame collapse is the critical failure mode, and 2) it is over-designed for overall collapse, which is difficult and computationally costly to predict accurately. Kendrick (1982) noted that the structural cost of avoiding failure by overall collapse is relatively small, and it is more economical to focus on predicting, and minimizing

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structural costs associated with, inter-frame failure of the shell.

The implementation of more rigorous overall elasto-plastic collapse methods (i.e. Kendrick's finite difference solution, P_{co}) has allowed at least one contemporary design code (DPA, 2001) to place roughly equal weight on inter-frame and overall collapse. This presents its own problems, as pressure hulls having similar predicted inter-frame and overall collapse pressures may have real collapse pressures that are significantly less than either of the calculated values. This so-called "failure mode interaction," or FMI, has been observed experimentally in at least one instance (Graham *et al.*, 1992).

FMI is attributed to large shell stresses due to overall bending of the combined ring-stiffener and shell plating that lead to shell yielding with the growth of overall displacements, and ultimately failure of the shell - collapse is "inter-frame, with overall tendencies" (Graham *et al.*, 1992). Graham *et al.* (1992) suggest that FMI can be avoided by ensuring that $P_{co} \geq 1.2$ P_{ci}. FMI has been addressed in SSP74 (DPA, 2001) by the inclusion of an interaction term, which reduces the flexural rigidity of the combined stiffener-shell based on P_{ci}, in the calculation of the overall elasto-plastic collapse pressure, P_{co}.

SWEDGE STIFFENED PRESSURE HULL DEVELOPMENT

The swedge stiffened pressure hull is proposed by applying a modification on the previous ring-stiffened design by creating the swedge section in the middle part of the pressure hull. The first swedge section is located 4.3m from the front-dome, while the second swedge is located 9m from the back-trail of the pressure hull, and the length between the two swedge is 9.5m or 41.6% of the overall pressure hull length. In the swedge section, the pressure hull diameter is increased of 1.0m or 3.3% of the middle part of pressure hull diameter. Furthermore the construction design configurations are made by modifying the position of the transverse bulkhead and the spacing of pressure hull stiffeners, as follow, see Figure-1:

- **Model 1:** First bulkhead (1.293 m from Bow), Second Bulkhead (5.501 from Trail), Frame Spacing on the swedge part 0.86 m, Frame spacing outside the swedge 0.5m, the weight 39.4 ton.
- **Model 2:** First bulkhead (1.293 m from Bow), Second Bulkhead (1.501 from Trail), Frame Spacing on the swedge part 0.86 m, Frame spacing outside the swedge 0.5m, the weight 39.4 ton.
- **Model 3:** First bulkhead (9.087 m from Bow), Second Bulkhead (1.501 from Trail), Frame Spacing on the swedge part 0.86 m, Frame spacing outside the swedge 0.5m, the weight 39.4 ton.
- **Model 4:** First bulkhead (9.087 m from Bow), Second Bulkhead (1.501 from Trail), Frame Spacing on the swedge part 0.86 m, Frame spacing outside the swedge 1.5m, the weight 36.3 ton.

THE SIMULATION MODEL AND CALCULATIONS

The full models of swedge stiffened pressure hull were used in FEA as shown in Figure-1. The configuration of the swedge stiffened pressure hull was defined as the variation of transverse bulkheads positions. The linear buckling calculations of swedge stiffened pressure hull under hydrostatic pressure are performed. In this model, the hydrostatic pressure of the buckling load, which can be obtained by the water depth where the submarine might be operated as the operational condition and the maximum depth which is the structure still not collapse and reliable is introduced.
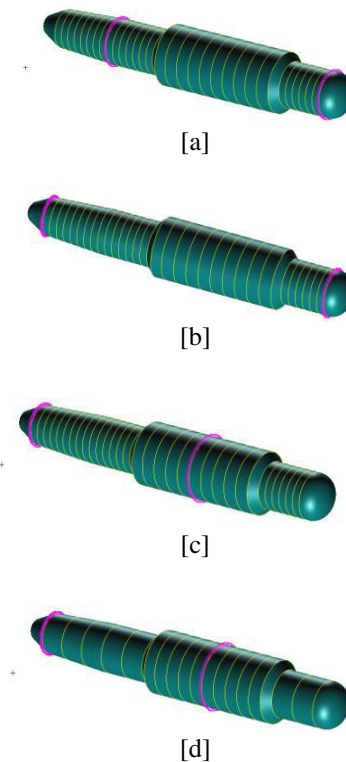

Figure-1. Boundary condition: [a] Model 1; [b] Model 2; [c] Model 3; [d] Model 4.

The boundary conditions and loading conditions

General purpose FE software is used for the linear buckling analysis in which the buckling mode shape is taken account. The quadrilateral 4 node element is used. The calculating pressure hulls consist of 4 models which is the variations of transverse bulkhead position is defined. The element number in the model is 30859 elements are used to maintain the calculation accuracy. The convergence of calculation by mesh division was confirmed.

The cylindrical coordinates were used. The boundary conditions are given at the mid span of the pressure hull at four points. The rigid body elements (RBE) are inserted at both transverse bulkheads in order to connect the center of bulkhead and the points on outer point of the circle as shown in Figure-1. The hydrostatics pressure is loaded at the wall of the pressure hull. The

rigid body elements (RBE) prevent the oval deformation of both transverse bulkheads, and keep the section in plane under translational and rotational deformation by the hydrostatics pressure. In the commercial software, a rigid link for either small deformation or large deformation can be implemented using RBE.

RESULTS AND DISCUSSIONS

Linear buckling behavior of swedge stiffened pressure hull

There are two major categories leading to the sudden failure of a mechanical component: material failure and structural instability, which is often called buckling. For material failures the yield stress is considered for ductile materials and the ultimate stress for brittle materials. Buckling refers to the loss of stability of a component and is usually independent of material strength. The load at which buckling occurs depends on the stiffness of a component, not upon the strength of its materials. When a structure whose order of magnitude of length is larger than either of its other dimensions, is subjected to axial compressive stress, due to its size its axial displacement is going to be very small compared to its lateral deflection is known as Buckling.

The linear buckling analysis of the pressure hull models is made by extracting the hydrostatic pressure acting on the shell of the pressure hull and converting it to the compressive pressure. The compressive pressure is the applied on each configuration of the location of the pressure hull transverse bulkheads. The buckling analysis is done and the buckling load factor is obtained. The results of the buckling load factor for each model are shown in Table-1, while the buckling mode shape of the pressure hull buckling might be seen on the Figure-2, Figure-6.
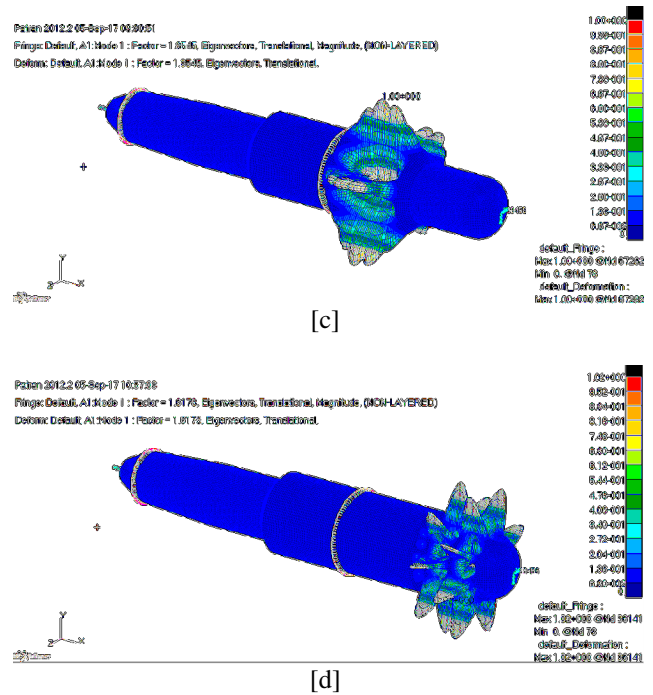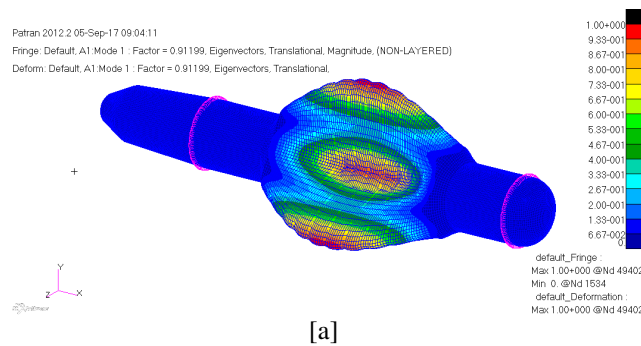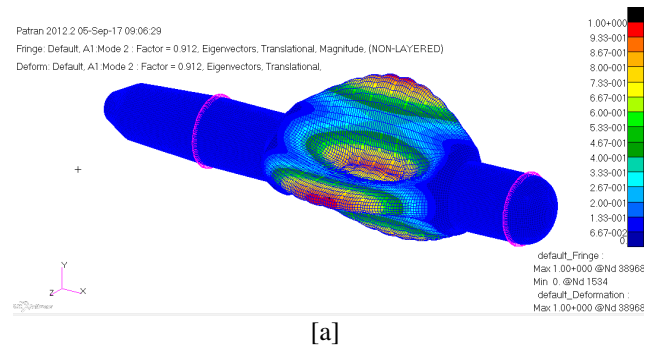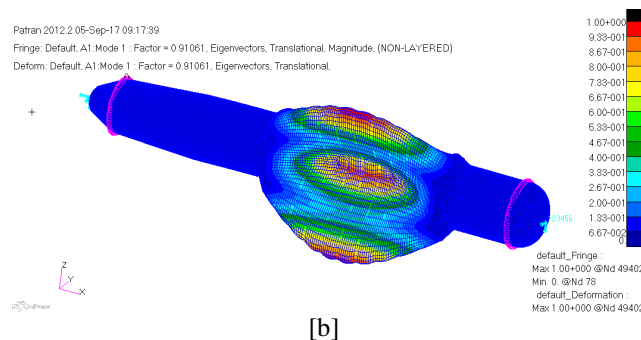
[c]

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Figure-2. 1st mode buckling mode shape: [a] model 1; [b] model 2; [c] model 3; [d] model 4.

[a]

[a]

[b]

[b]

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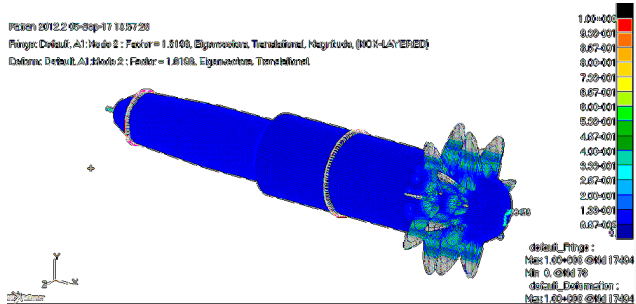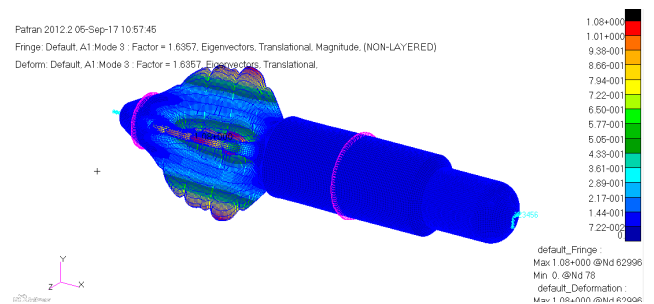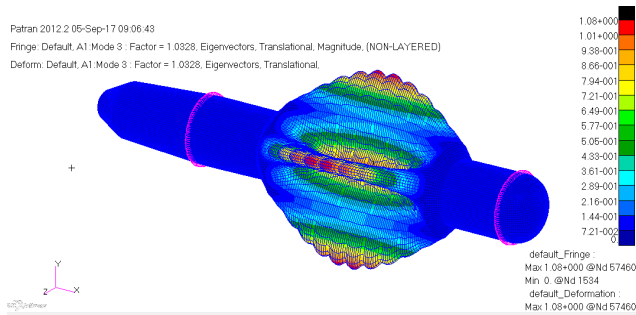
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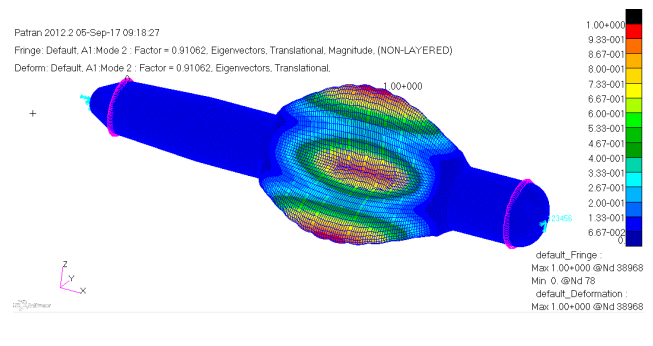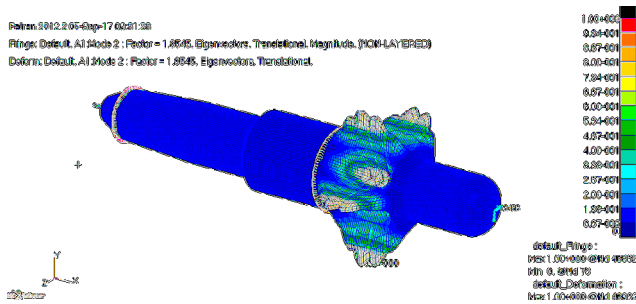
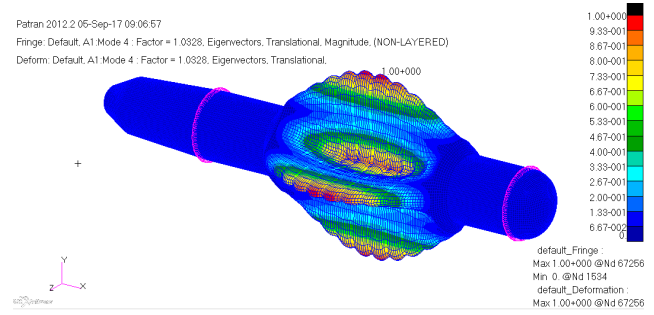
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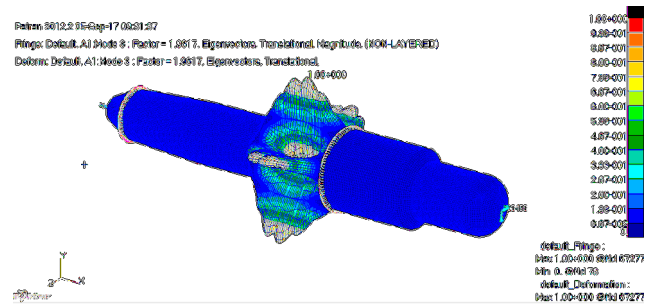
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Figure-3. 2nd mode buckling mode shape: [a] model 1; [b] model 2; [c] model 3; [d] model 4.

Figure-4. 3rd mode buckling mode shape: [a] model 1; [b] model 2; [c] model 3; [d] model 4.

[a]

[a]

[b]

[b]

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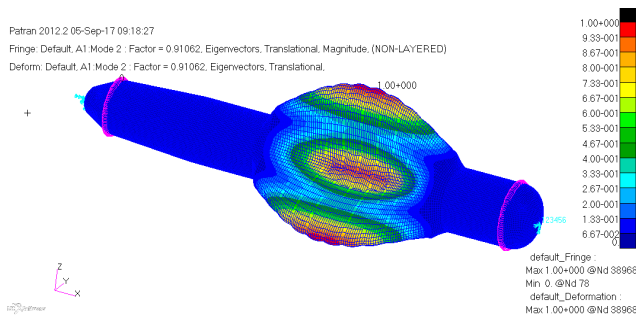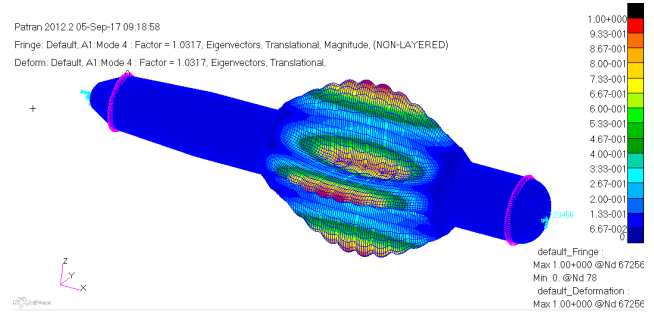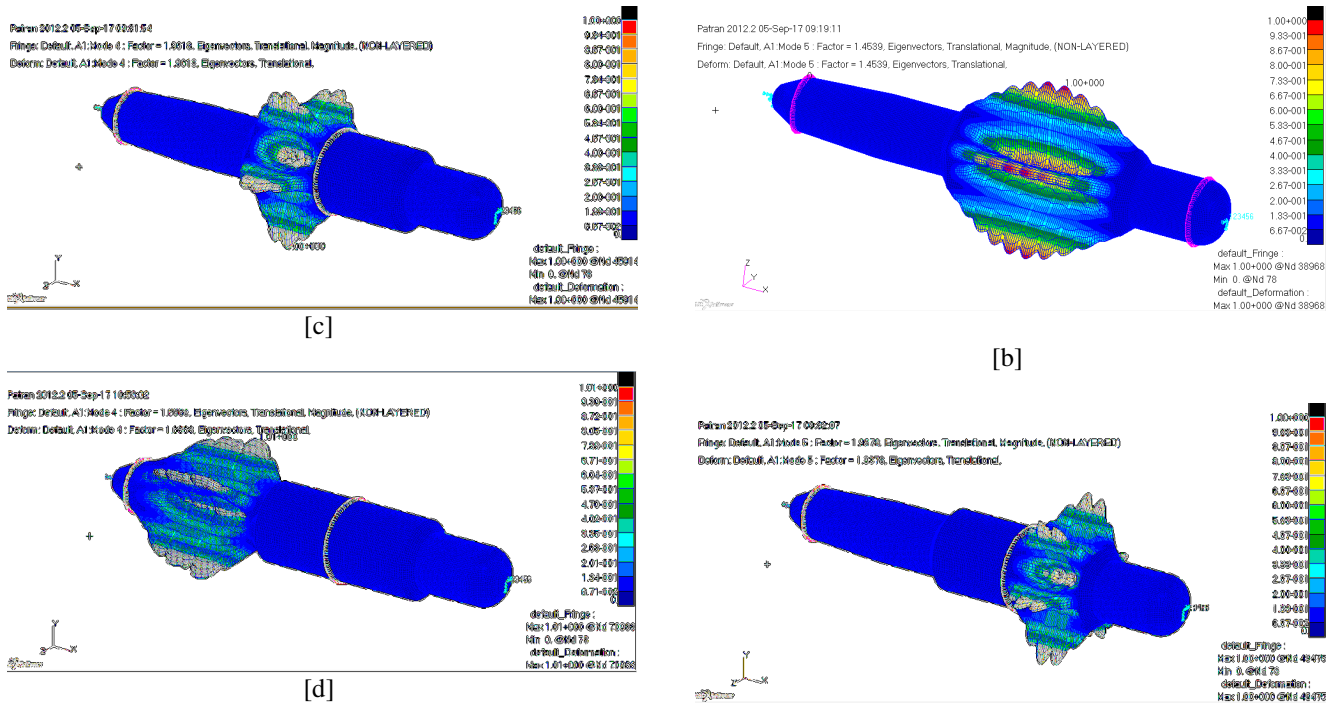
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[c]

[b]

[d]

[c]

Figure-5. 4th mode buckling mode shape: [a] model 1; [b] model 2; [c] model 3; [d] model 4.
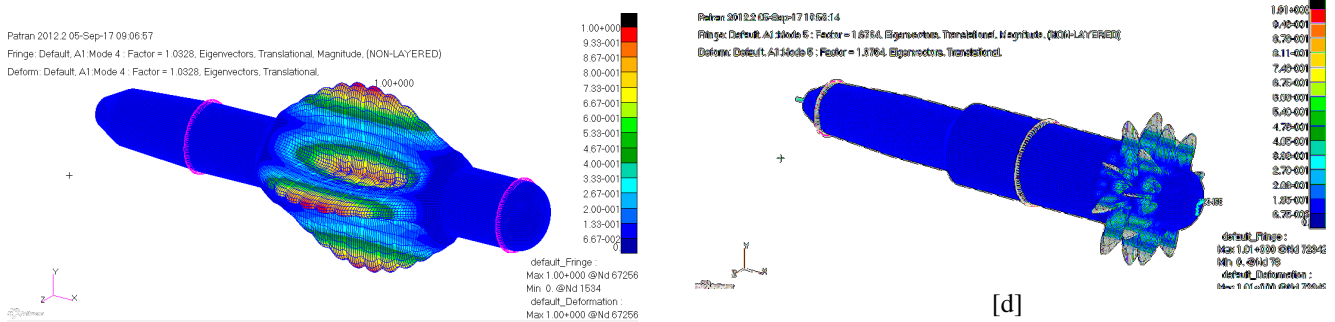

[a]

[d]

Figure-6. 5th mode buckling mode shape: [a] model 1; [b] model 2; [c] model 3; [d] model 4.

Buckling strength comparison of Swedge-Stiffened and Ring Stiffened

Since the numerical analysis to determine the magnitude of the buckling strength of the swedge-stiffened pressure hull was made, the comparison of buckling strength is done by comparing the buckling load factor of the swedge-stiffened with the ring-stiffened pressure hull buckling load factor has been calculated in the previous study, Yudo (2017). The comparison analysis is done by comparing the design of model 3, model 4 and the ring-stiffened which is using uniformly spaced transverse bulkhead. The variables which is used for the buckling strength comparison includes: buckling load factor (mode 1-mode 5), weight of pressure hull structure and compartment volume in pressure hull. The results can be seen in Table-2.

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Table-1. Buckling load factor of the swedge stiffened pressure hull.

Pressure hull design	Mode	Buckling load factor	Acceptance criteria	Status
Model 1	Mode 1	0.9119	1.00	Not passed
	Mode 2	0.9120	1.00	Not passed
	Mode 3	1.0328	1.00	passed
	Mode 4	1.0328	1.00	passed
	Mode 5	1.0328	1.00	passed
Model 2	Mode 1	0.91061	1.00	Not passed
	Mode 2	0.91062	1.00	Not passed
	Mode 3	0.91062	1.00	Not passed
	Mode 4	1.03170	1.00	passed
	Mode 5	1.45390	1.00	passed
Model 3	Mode 1	1.8545	1.00	passed
	Mode 2	1.8545	1.00	passed
	Mode 3	1.9617	1.00	passed
	Mode 4	1.9618	1.00	passed
	Mode 5	1.9878	1.00	passed
Model 4	Mode 1	1.6178	1.00	passed
	Mode 2	1.6196	1.00	passed
	Mode 3	1.6357	1.00	passed
	Mode 4	1.6368	1.00	passed
	Mode 5	1.6764	1.00	passed

Based on the results of the analysis it appears that the stiffened pressure hull ring has a better buckling strength when compared to the proposed swedge stiffened pressure hull. In the load factor mode 1, the buckling load factor ring of stiffened pressure hull has a load factor of 2.511, which is larger by 55.2% and 35.4% than swedge model 4 and swedge model 3, respectively. However, according on the acceptance criteria, it is indicated that the design of the swedge model has met the acceptance criteria; while the ring-stiffened design has a tendency to over design compared to the proposed design of swedge-stiffened pressure hull.

According on the weight structure, it is indicated that the ring-stiffened has a smaller pressure hull weight compared to both swedge stiffened pressure hull types. The weight of the swedge model 3 is 9.44% larger than the ring-stiffened pressure hull. In the case of the swedge model 4, although it has an increased weight, however the additional weight is not significant, therefore it can be considered as large as ring-stiffened weight. The increase of weight in both types of swedge can be explained because of the addition of steel weight due to pressure hull modification process is done by increasing the diameter of pressure hull in the middle part of the cylinder. However, in the case of the swedge model 4 the additional of steel weight can be reduced, by increasing the frame spacing distance, so that the number of the frame components on the swedge model 4 is reduced.

Despite the increased in the steel weight and the decreased in the buckling strength, however, in the both proposed swedge designs, the compartment volume have increased significantly compared to the volume compartment of the ring-stiffened pressure hull. In the both of the swedge pressure hull, the magnitude of compartment volume is 198.52 m^3. This is 33.08% larger than the compartment of the ring-stiffened pressure hull which has a volume of 149.2 m^3.

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Table-2. Comparison of pressure hull using the swedge-stiffened and the ring-stiffened.

Comparison criteria	Swedge model 3	Swedge model 4	Ring Stiffened	Acceptance criteria
Load factor mode 1	1.8545	1.6178	2.5114	1.50
Load factor mode 2	1.8545	1.6196	2.5114	1.50
Load factor mode 3	1.9617	1.6357	2.8238	1.50
Load factor mode 4	1.9618	1.6368	2.8238	1.50
Load factor mode 5	1.9878	1.6764	3.0560	1.50
Structure Weight	39.4 Ton	36.3 Ton	36.0 Ton	-
Vol. of Pressure Hull	198.52 m^3	198.52 m^3	149.2 m^3	-

CONCLUSIONS

Based on the results of comparative studies on the strength of buckling between swedge-stiffened and ring-stiffened pressure hull, it is indicated that the ring-stiffened pressure hull has a better buckling strength compared with the proposed swedge-stiffened pressure hull. However, according to the acceptance criteria, the ring-stiffened pressure hull design shows an overdesign tendency than the both of proposed swedge-stiffened design.

According to the weight of the pressure hull structure, it shows that the ring-stiffened pressure hull has a smaller structure weight compared to the proposed swedge-stiffened design, however the additional weight which is caused by the increased of the diameter at the middle part of the pressure hull can be reduced by decreasing the number of the frame components with an increase of frame spacing distance.

The modification process by increasing the diameter at the middle part of the pressure hull, gives a significant increase in the volume of compartment. Although this additional diameter may cause a decreased buckling strength and weight gain, however it can be optimized by using an appropriate structural arrangement by providing optimum frame spacing and a suitable transverse bulkhead position.

ACKNOWLEDGEMENT

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