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Research Article

Prediction of Propeller Blade Stress Distribution Through FEA

Kiam Beng Yeo, Wai Heng Choong and Wen Yen Hau

ABSTRACT

The Finite Element Analysis (FEA) of marine propeller blade stress distribution due to hydrodynamic loading is presented and discussed. The analysis provided a better insight to complex marine propeller shape and interaction with hydrodynamic loadings. Stainless steel Wageningen B Series 3 blade propeller with 250 mm diameter, EAR of 0.5 and P/D ratio of 1.2 was adopted in the analysis. The propeller was subjected to the rotational speed of 0-6000 rpm. The pressure distribution demonstrated a positive pressure region on the face section and a negative region on the back section that produces the thrust generation. At 6000 rpm, a maximum positive pressure was achieved at 3225 kPa with a negative pressure of 7229 kPa. The hydrodynamic loading from the pressure distribution computation was applied to the stress distribution computation. From the analysis, the propeller blade stress distribution predicted a highly concentrated region near to the hub and decreasing with the growing value of the propeller radius. The highest stress value of 739 MPa at 6000 rpm was obtained at higher than the stainless steel yield stress (170 MPa) and the blade tip deflected towards the ship hull by 2.73 mm.

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INTRODUCTION

In marine propeller design, the propeller blade strength is always an important issue to the cavitation phenomenon for assurance of efficiency and functionality. The propeller blade is developed with an accountable strength based on the material and structural properties serving as the main propulsion component for converting mechanical power to thrust force propelling the ship to move forward through the water. Computational prediction of propeller blade stress distributions due to hydrodynamic loading has been important.

The stress analysis on marine propeller is complex and the solutions accuracy is critical in handling a curvature 3-D model of highly non-linear force, moment and pressure elements. The non-linear force or hydrodynamic loading elements are produced from the thrust and torque and the rotational motion inducing a centrifugal force. Direct analytical modeling of hydrodynamic loading is not a viable solution if realistic prediction is needed. Computational prediction is a significant approach to this problem (Bade and Junglewitz, 2010).

In Vidya Sagar *et al*. (2013) study, FEA based application was adopted for predicting marine propeller under steady state analysis with uniform thrust loading which or is a linear analysis. A clear methodology reported on utilizing the computational method coupling two types of numerical computational (**finite element** method-FEM and boundary element method-BEM) to predict marine propeller performance, hydrodynamic blade loads, cavitation patterns, stress distributions and deflection patterns was conducted by Young (2008). The BEM-FEM had demonstrated the transition of hydrodynamic loading into stress analysis. It provides realistic and accountable results for further application such as optimization of stress concentration region and propeller geometry parameters.

On the complex geometric model adoption in propeller blade, a better solution and shorter computation time for the stress analysis has yet to be reported by researchers. By the thumb of rules, the 2-D model is a much simplified model compared to the 3-D model in computational application. However, the 3-D model had its pros and cons. Chau (2010) stated that the type of numerical solution background theory critically affects the type of geometrical model adoption. Both of the computational 2-D model based on thin shell theory method and the 3-D model using classical theory of deformable solid had shown to produce acceptable similar results.

This study addressed a CFD-FEA prediction of the Wagenigen B Series 3 blade marine propeller pressure and stress distribution subjected to a steady state hydrodynamic loading. The prediction will provide an outcome for further understanding of the pressure and stress distribution on a marine propeller blade.

PROPELLER BLADE STATIC ANALYSIS MODEL

Idealized propeller structure can be simplified as a cantilever beam pivoted at the hub axis with a single loading on the free end or uniformly loaded along the beam. However, this ideal model does not include the highly non-linear wake field or external forces or moment such as the centrifugal forces. As more parameters and flow characteristics with different condition changes, more estimation shall be necessary to improve the effectiveness of theoretical analysis. As the propeller rotates about its central hub axis, each blade suffers different inflow field effect which causes various amplitudes of cyclic resultant moments and forces. Carlton (2007) suggested the general propeller blade stress equation as:

$$\sigma = \sigma_T + \sigma_Q + \sigma_{CBM} + \sigma_{CF} + \sigma_P \qquad (1)$$

where, σ_T, σ_Q, σ_{CBM}, σ_{CF} and σ_P are the stress components due to thrust, torque, centrifugal bending, direct centrifugal force and out of plane stress components, respectively.

FINITE ELEMENT ANALYSIS CODE

Finite Element Analysis (FEA) is one of the numerical computational methods in solving or obtaining approximation solution of boundary value problem in linear or non-linear application. The basic fundamental of the FEA is to represent the original model or shape with large number of **finite element** (triangular shape) that permit the finite solution to be carried out on each **finite element**. There are several linear or non-linear analyses that can be carried out by FEA, such as the material **mechanical properties** analysis, dynamic, fatigue, frequency, vibration, thermal and others. Generally, the linear static solution through displacement method in FEA can be described by matrix equation as:

$$[K]\{U\} = \{F\} = \{F^a\} + \{F^c\} \qquad (2)$$

where, [K] is the structural stiffness matrix, {U} is the vector of unknown nodal displacement and {F} is load vector ($\{F^a\}$ and $\{F^c\}$ of the applied and reaction forces). For $\{F^a\}$, it can be redefined to consider the loading as the mechanical $\{F^m\}$, thermal $\{F^{th}\}$ and gravitational load $\{F^{gr}\}$ and subsequently as:

$$\{F^a\} = \{F^m\} + \{F^{th}\} + \{F^{gr}\} \quad (3)$$

Table 1: Wageningen B series 3 blade propeller with P/D = 1.2

Parameters	Values
Diameter (D)	250 mm
Mean pitch, P_m	300 mm (P/D = 1.2)
Skew angle (θ_s)	0° (balanced skew)
Rake angle (θ)	0°
Propeller Expanded Area Ratio (EAR)	0.5

Then, the mechanical load vector $\{F^m\}$ is equal to the sum of applied nodal forces and moments and pressure elements as:

$$\{F^m\} = \{F^{nd}\} + \sum_{e=1}^{nel} \{F_e^{pr}\} \quad (4)$$

where, $\{F^{nd}\}$ is the applied nodal load vector $\{F_e^{pr}\}$ is the element of pressure load vector, e is the element number and nel is the number of element. Meanwhile the thermal and gravitational load vector can be solved as:

$$\{F^{th}\} = \{F^{nt}\} + \sum_{e=1}^{nel} \{F_e^{th}\} \quad (5)$$

$$\{F^{gr}\} = \left(\sum_{e=1}^{nel} \{M_e\} \right) \{a\} \quad (6)$$

where, $\{F^{nt}\}$ is the nodal temperature load vector, $\{F_e^{th}\}$ is the element of thermal load vector, $[M_e]$ is the element of mass matrix and $\{a\}$ is the acceleration vector.

PROPELLER BLADE MODEL

In this computational analysis, a Wageningen B Series propeller with P/D ratio value of 1.2 had been adopted. The propeller geometrical and particular details are tabulated in Table 1 and the Fig. 1 shows the virtual 3-D propeller model.

FEA BOUNDARY CONDITION AND MESHING

This analysis utilizes the SolidWorks Static Simulation FEA application tool to carry out the propeller blade stress distribution simulation. The propeller blade stresses are mainly due to the medium (water) head pressure and the resultant pressure and force and moment elements due to the geometry iteration with the medium when it been rotated. Computational Fluid Dynamics (CFD) application tool (SolidWorks Flow Simulation) was adopted to obtain the pressure, force and moment elements before transiting to the FEA application tool as external force or moment elements. Each of the propellers simulation underwent the rotating speed of 500, 1000, 2000, 3000, 4000, 5000 and 6000 rpm with non-cavitating flow condition.

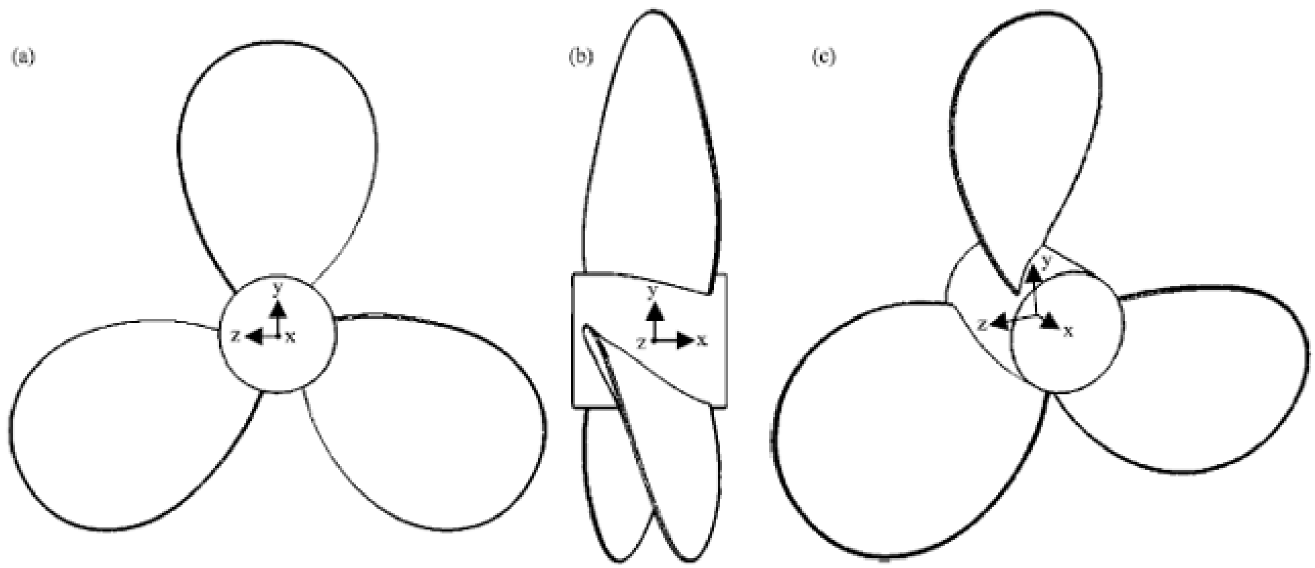


Fig. 1(a-c): Wageningen B series 3 blades propeller geometry (a) Front, (b) Side and (c) 3-D view

In order to perform a real world environment, the actual propeller material being assigned as stainless steel with the material properties: Young Modulus, E = 200 GPa, Poison's ratio, NUXY = 0.265, Mass density = 8027 kg m^{-3} and Yield strength = 170 MPa. The propeller model was fixed in the computation domain through the hub surface as shown in the Fig. 2. External loading elements included the pressure, force and moment elements were imported from the previous CFD analysis. The model meshed representing the original model with triangular tetrahedral element model was then generated as shown in Fig. 3. The FEA analysis had been concentrated on the propeller blade stress distribution and displacement due to the hydrodynamic loading.

RESULTS AND DISCUSSION

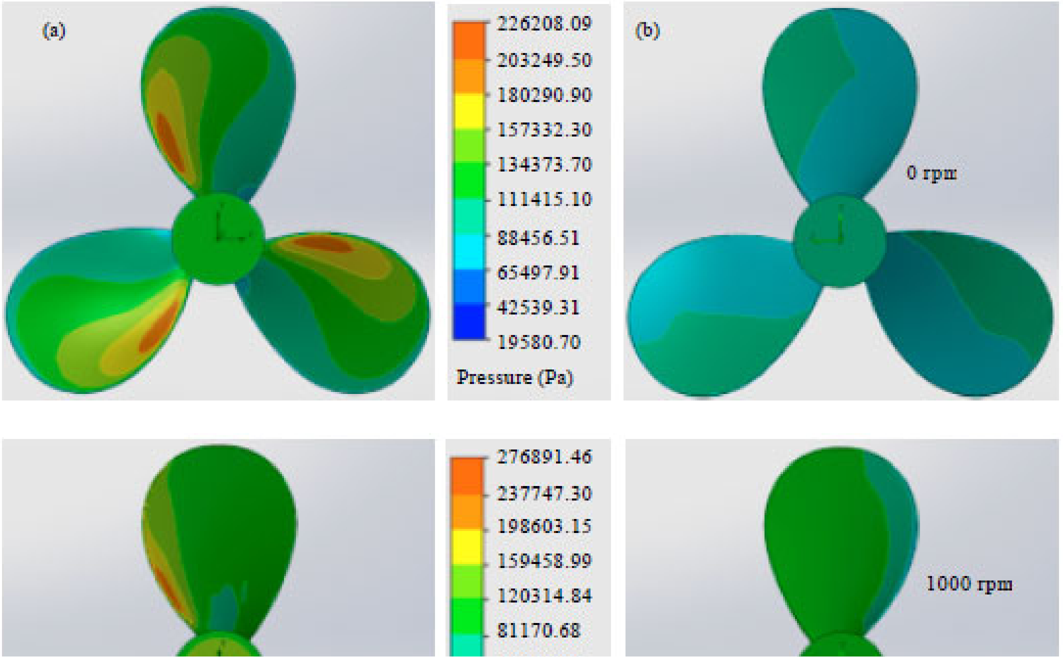
Based on the Eq. 2-5, the applied load $\{F^a\}$ for propeller blade stress distribution prediction without involving thermal loading through FEA method can be written as:

$$\{F^a\} = \{F^{nd}\} + \sum_{e=1}^{nel} \{F_e^{pr}\} + \left(\sum_{e=1}^{nel} \{M_e\} \right) \{a\} \tag{7}$$

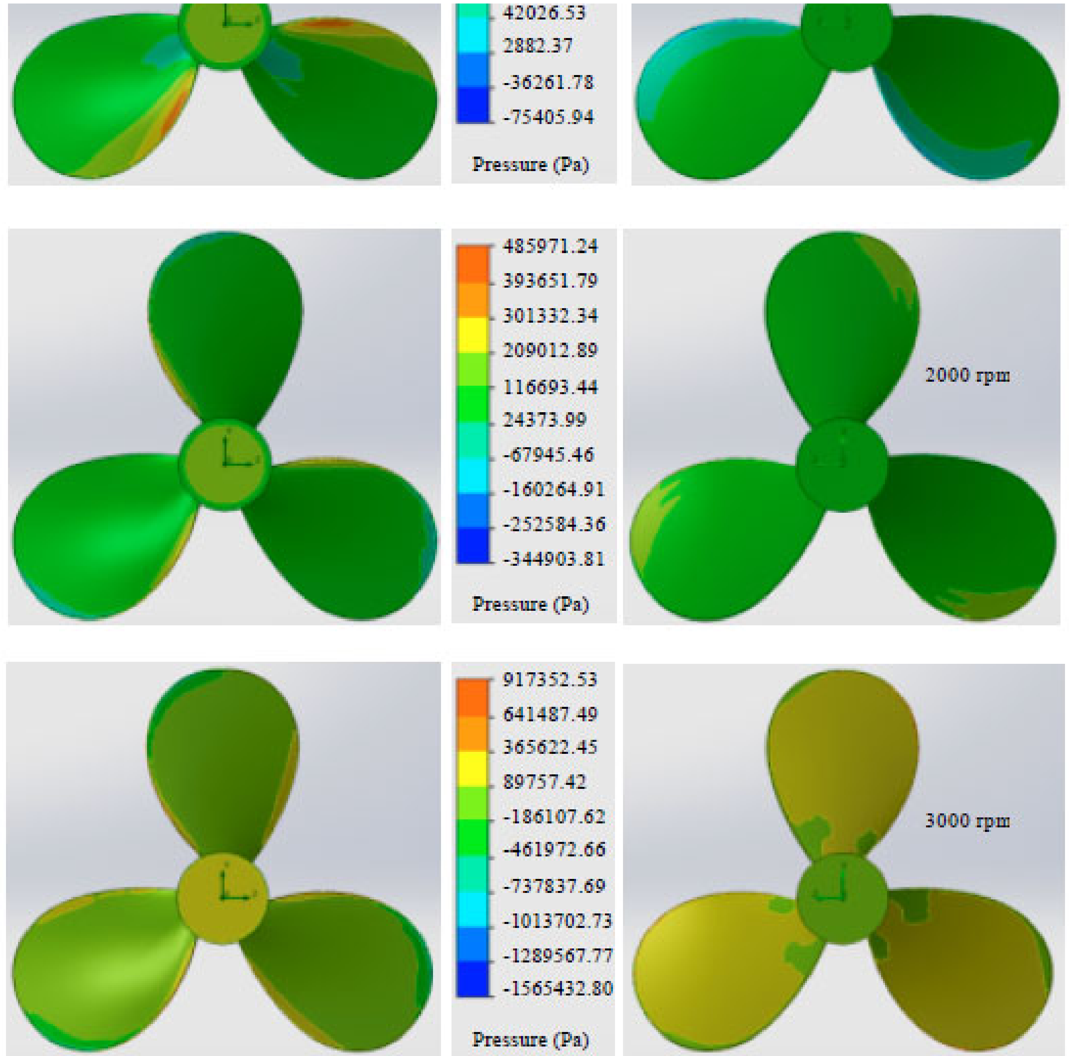
where, the element of pressure load vector $\{F_e^{pr}\}$ was preceded from the propeller blade pressure distribution study through CFD application. Wageningen B-Series 3 blade propeller with P/D ratio value of 1.2 was utilized to simulate the blade stress distribution due to the hydrodynamic elements.

Pressure distribution on the propeller blade: From the CFD analysis, each unit of area on propeller solid geometry has its own pressure magnitude which caused the stress distribution variation along the body. The pressure loading on the propeller surface involves suction on the back surface and a pressurized surface which faced the aft side to push the water back. Besides the numerical value due to different loadings, some phenomena in hydrodynamics that caused turbulence flow characteristics could be explained. The propeller blade with P/D = 1.2, pressure distribution for rotating speed 0-6000 rpm results were presented in Fig. 2. Overall pressure distributed on the face section was more uniform compared to the back section as influence by the interaction of hydrofoil geometry profile and the flow stream. The leading edge or hydrofoil stagnation point was subjected to higher pressure compared to the trailing edge. As the propeller rotates, the leading edge cut the flow stream and as a reaction, high pressure was developed at that point and subsequently distributed to the whole blade surrounding. It also shown that the propeller rotation speed greatly influences the blade pressure distribution; where the higher the rotation speed, higher the pressure surrounding.

At 0 rpm, the propeller blade was exposed to the water head pressure. The highest pressure region reach 226 kPa at the back section compared to the face region of 103 kPa. The pressure difference highly affected by blade geometry construction. As the propeller reach 1000 rpm, high pressure region was formed at the leading edge. At the back section, a small region of high pressure was formed as a stagnation region. At 2000 rpm rotation, encountering the initial fluid flow velocity boundary condition to produce torque and thrust, high pressure continuously occurred at leading edge with very small region of distribution and very low positive pressure had occurred at the large surface area.



Translate


42026.53
2882.37
-36261.78
-75405.94
Pressure (Pa)
485971.24
393651.79
301332.34
209012.89
116693.44
24373.99
-67945.46
-160264.91
-252584.36
-344903.81
Pressure (Pa)
2000 rpm
917352.53
641487.49
365622.45
89757.42
-186107.62
-461972.66
-737837.69
-1013702.73
-1289567.77
-1565432.80
Pressure (Pa)
3000 rpm

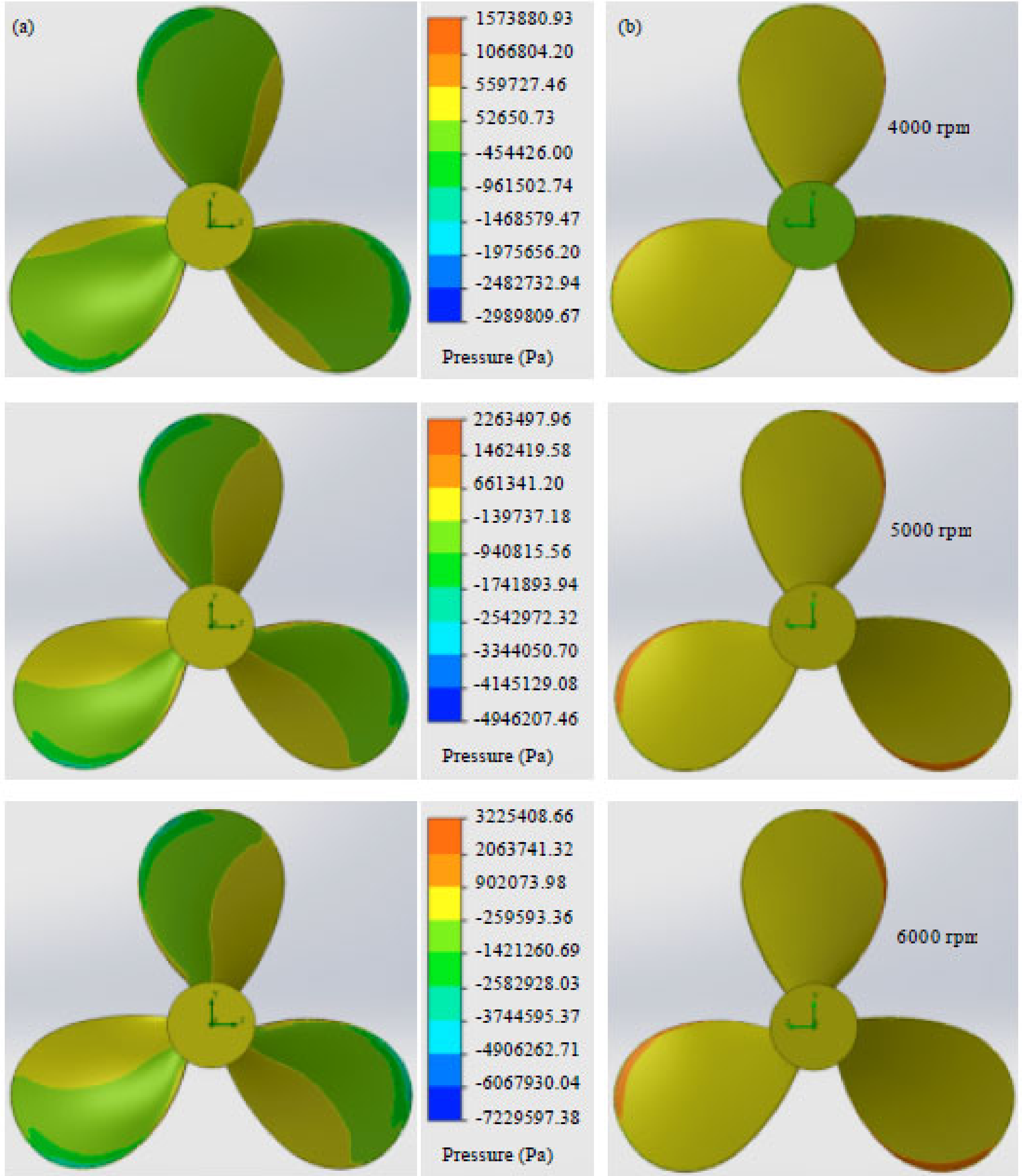


Fig. 2(a-b): Propeller blade pressure distribution at various rotational speed (0-6000 rpm). Static analysis on propeller blade due to hydrodynamic loading (a) Back (suction) and (b) Face (pressure)

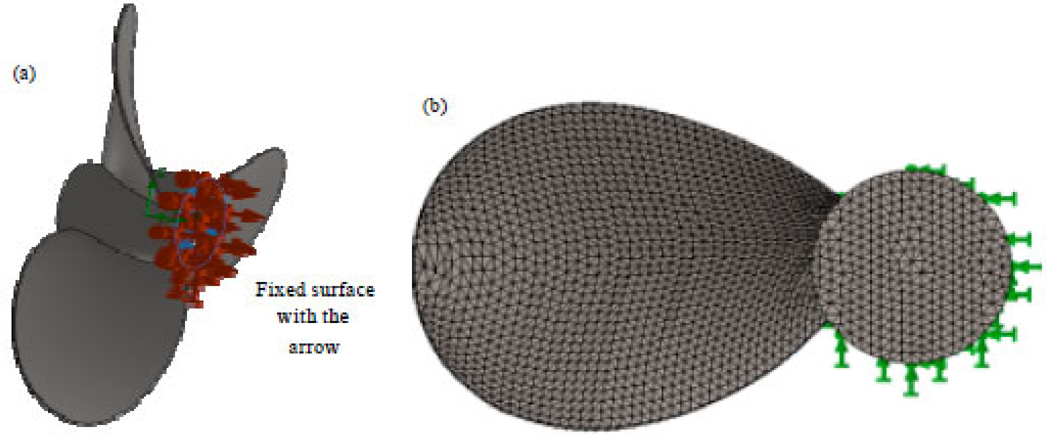


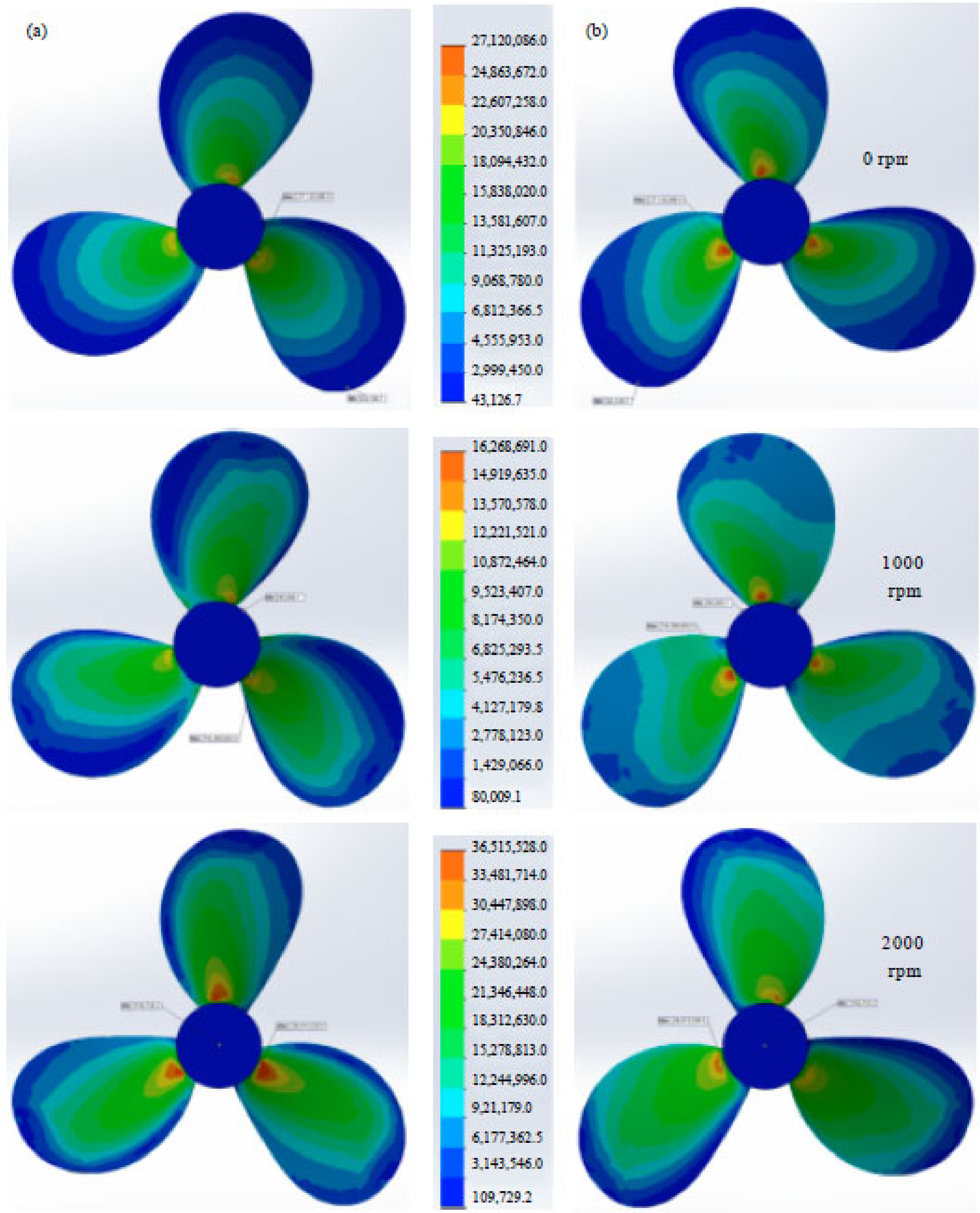
Fig. 3(a-b): Unmeshed and meshed model (a) Unmeshed model and (b) Generated tetrahedral element model

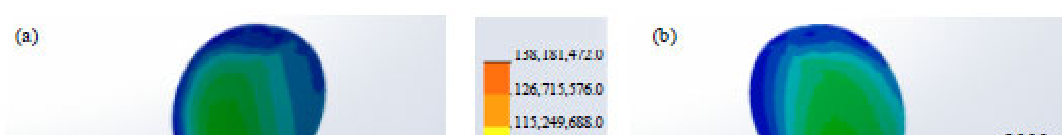
A small negative pressure region had also occurred at the back trailing edge. The negative pressure magnitude direction being against the propeller blade satisfied the hydrofoil suction fundamental.

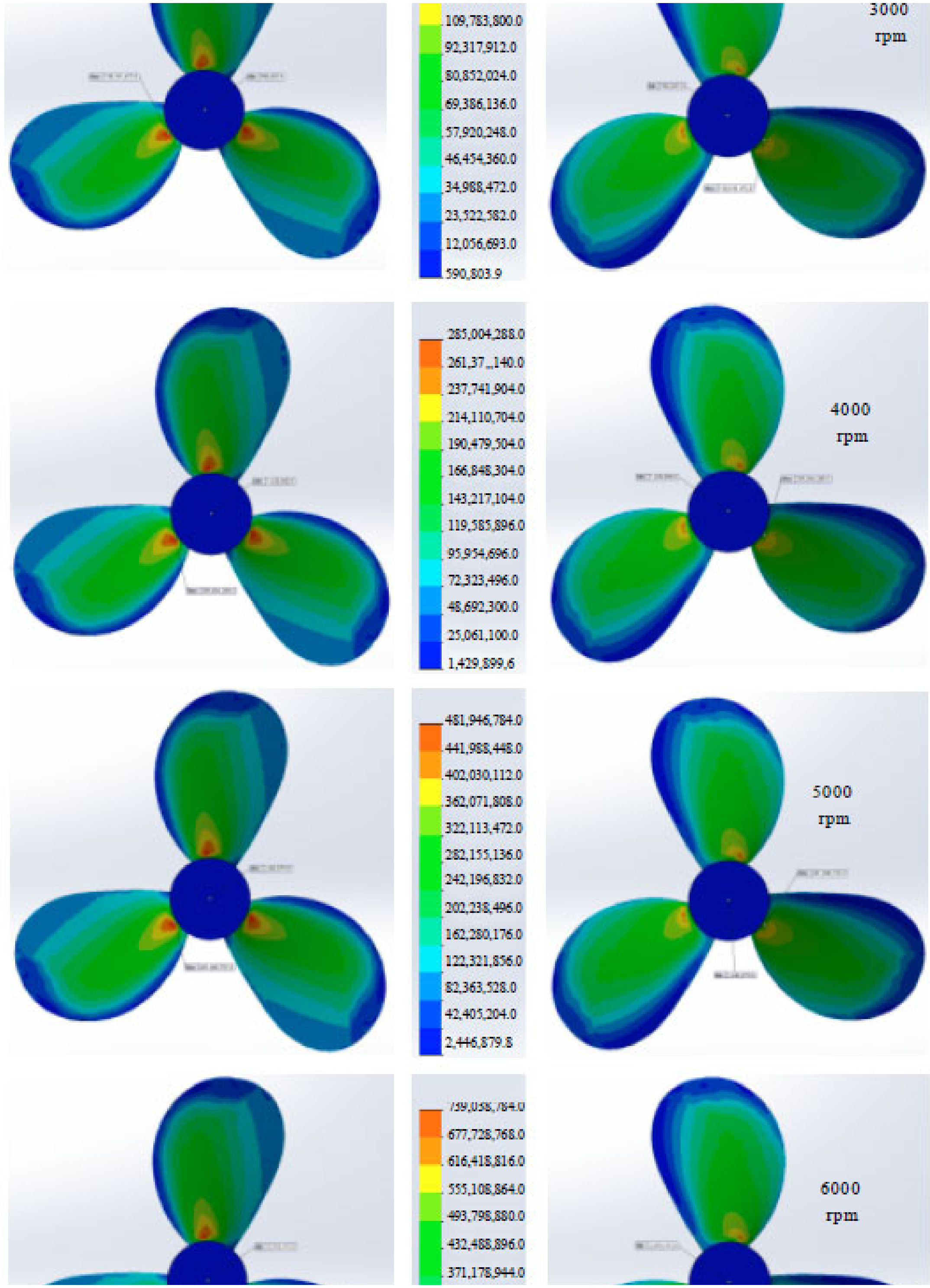
For high rpm performance of marine propeller of 3000-6000 rpm and higher, the negative pressure continued to increase and expand at the back region. At 6000 rpm, positive pressure reached 3225 kPa and negative pressure reached 7229 kPa. The pressure magnitude difference was required for the thrust force generation or forcing the water medium towards to the ship hull for forward movement. This interaction also led to the cavitation phenomenon which required further analysis and validation.

Stress distribution on the propeller blade: Static stress analysis was conducted to predict and verify the propeller blade structure behavior as subjected to hydrodynamic loading or thrust initiated loaded. The analysis is carried out using the SolidWorks Static Simulation FEA application tool.

The obtained propeller blade stress distribution results were shown in Fig. 4.


(a)
(b)
27,120,086.0
24,863,672.0
22,607,258.0
20,350,846.0
18,094,432.0
15,838,020.0
13,581,607.0
11,325,193.0
9,068,780.0
6,812,366.5
4,555,953.0
2,999,450.0
43,126.7
0 rpm
16,268,691.0
14,919,635.0
13,570,578.0
12,221,521.0
10,872,464.0
9,523,407.0
8,174,350.0
6,825,293.5
5,476,236.5
4,127,179.8
2,778,123.0
1,429,066.0
80,009.1
1000
rpm
36,515,528.0
33,481,714.0
30,447,898.0
27,414,080.0
24,380,264.0
21,346,448.0
18,312,630.0
15,278,813.0
12,244,996.0
9,21,179.0
6,177,362.5
3,143,546.0
109,729.2
2000
rpm


(a)
(b)
126,715,576.0
115,249,688.0


109,783,800.0
92,317,912.0
80,852,024.0
69,386,136.0
57,920,248.0
46,454,360.0
34,988,472.0
23,522,582.0
12,056,693.0
590,803.9
3000
rpm
285,004,288.0
237,741,904.0
214,110,704.0
190,479,504.0
166,848,304.0
143,217,104.0
119,585,896.0
95,954,696.0
72,323,496.0
48,692,300.0
25,061,100.0
1,429,899.6
4000
rpm
481,946,784.0
441,988,448.0
402,030,112.0
362,071,808.0
322,113,472.0
282,155,136.0
242,196,832.0
202,238,496.0
162,280,176.0
122,321,856.0
82,363,528.0
42,405,204.0
2,446,879.8
5000
rpm
677,728,768.0
616,418,816.0
555,108,864.0
493,798,880.0
432,488,896.0
371,178,944.0
6000
rpm

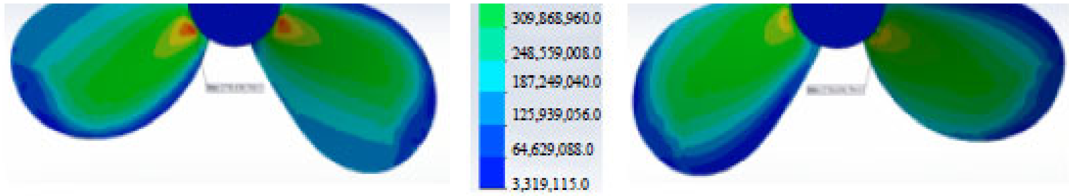


Fig. 4(a-b): Propeller blade stress distribution at various rotational speed (0-6000 rpm), (a) Back (suction) and (b) Face (pressure)

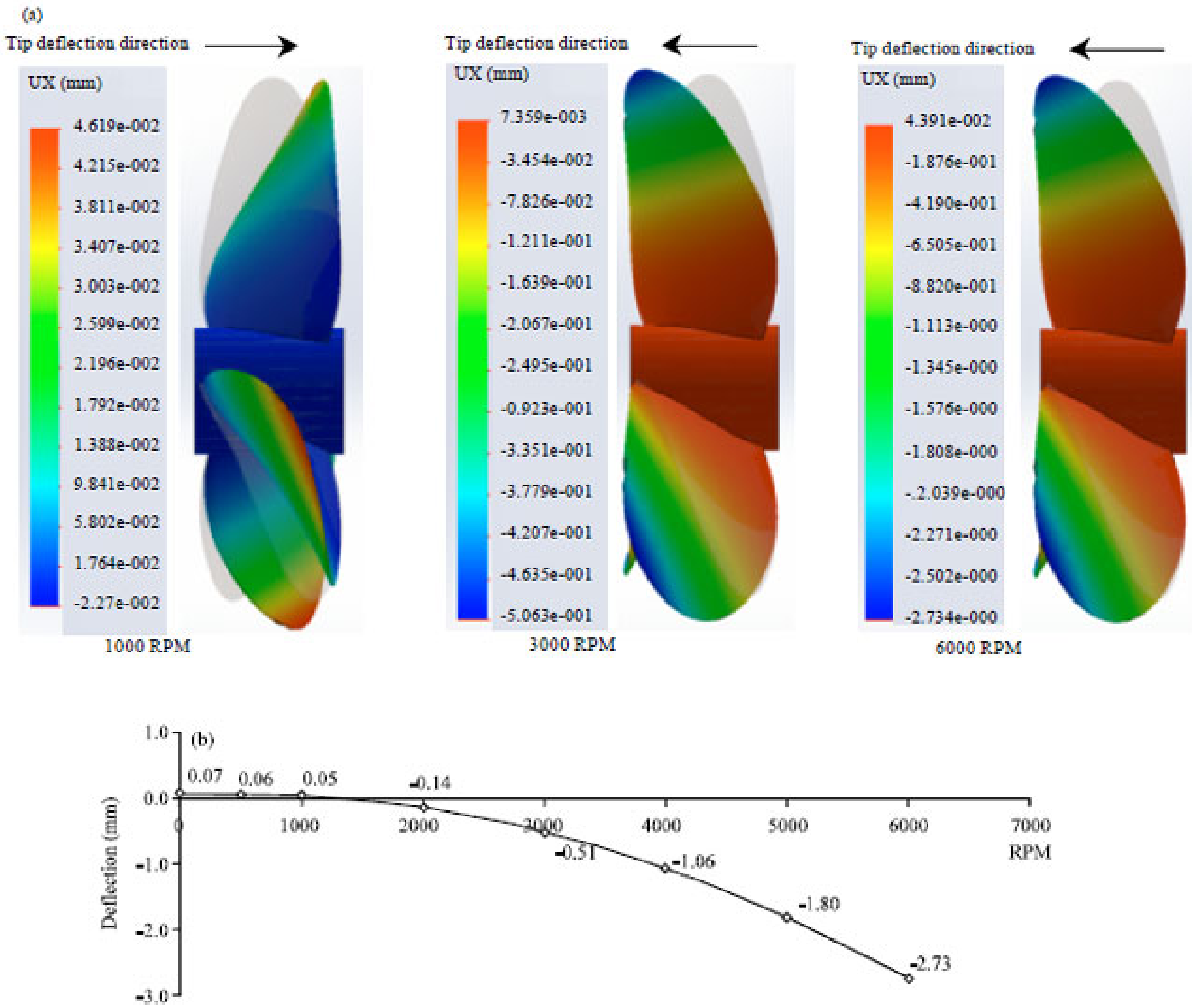


Fig. 5(a-b): Propeller blade (a) Displacement due hydrodynamic loading and (b) Deflection direction

Generally, all profile had achieved the similar trend of stress distribution with each respective rpm value. As expected, the stresses were concentrated near to the blade root area and it was ideal for the stresses value to decreases with increasing value of radius as a consequence of fixing the blade at the hub. As the propeller was rotating at 4000 rpm, the maximum stress value reach 285 MPa which further increased up to 739 MPa at 6000 rpm. The propeller will be subjected to face material failure as the stainless steel material yield stress is designed for a 170 MPa.

Deflection distribution on the propeller blade: The propeller blade had deflected due to the hydrodynamic loading as shown in Fig. 5. The maximum deflection occurred at the blade tip free end section. At 0-1000 rpm, there minor tip deformation occurred. As the rpm increases, the tip is deflected towards to the ship hull. The displacement direction concurs with the thrust force and hydrodynamic loading. The degree of deflection had shown to influence by the rpm. At 6000 rpm, tip displacement value reached 2.73 mm. The tip displacement towards the ship hull had increased the pitch value and subsequently allows higher thrust generation. This condition had been a ongoing research focus to develop a composite marine propeller with flexible pitch in-lab.

CONCLUSION

The present analysis demonstrated that marine propeller blade pressure and stress distribution can be predicted through Computational Fluid Dynamic and Finite Element Analysis application tool (SolidWorks Flow and Static Simulation) is significant to marine propeller design and development. The Wagenigen B-Series 3-blade marine propeller with 250 mm diameter and P/D ratio value of 1.2 had been adopted in this analysis. The high pressure and stress concentration regions had been successfully identified for future optimization of propulsion efficiency.

At speed of 2000 rpm, negative pressure form on the back section satisfies the hydrofoil fundamental for thrust generation. The pressure region further developed and at 6000 rpm maximum positive pressure reached 3225 kPa and a negative pressure at 7229 kPa. Cavitation is suitable at low pressure regions of vapor pressure (3.8 kPa at 28°C). Results shown that the propeller is subjected to material failure at 4000 rpm as high stresses (285 MPa and at 6000 rpm is 739 MPa) developed surpassed the stainless steel material yield strength value of 170 MPa. The blade structure displacement had also predicted and the maximum tip displacement value towards the ship hull reached 2.73 mm at 6000 rpm.

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