Hydrodynamic effect on the sound pressure level around the marine propeller

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Received 27 January 2015; revised 30 October 2016

In order to predict non-cavitation noise in two operation point, unsteady Reynolds-Averaged Navier-Stokes (URANS) solver is used with Finite Volume Method (FVM) in ANSYS/FLUNT14.5 software. Time accurate calculations are made by applying the Moving reference frame (MRF) method and two-step Fillow Williams and Hawkings (FW-H) equations are used to calculate sound pressure level (SPL). Hydrodynamic performance investigation is carried out to obtain open water performance in five operation conditions.

[Keywords: Marine propeller; Sound pressure Level; Hydrodynamic performance]

Introduction

Propeller is one of the most important sources of noise. In design process specifically for marine propellers should be tried, lowest noise possible. Optimized propeller is important and that should be analyzed structural and hydrodynamics process before manufacturing. There are four main mechanisms through which a propeller can generate pressure waves in the water. First, displacement of water by the propeller blades, Second, The pressure difference between back and face blade, third, the cavity periodic change by acting on the blade wake and fourth, the sudden burst of the bubble process.

It is clear that first two reasons are in relation to both cavitation and non-cavitation propeller. But the second two reasons are related to the cavitation phenomena and therefore only occur when the propeller is experiencing cavitation.

Submarines and torpedoes are usually operated deep enough under the sea to evade cavitation. Compared with the extensive amount of literatures involving cavitation noise of propellers, works involving the non-cavitation noise of propellers are not easy to find. The non-cavitation noise of underwater propeller is numerically inspected in this study.

A small-scale part of turbulent eddies in the wake cause unsteady blade forces. Besides, the boundary layer separation and blades vortex shedding also causes fluctuating forces. On the other hand, non-uniformity inflow increases periodic forces. These periodic unsteady forces impose discrete tonal noise at the blade passing frequency (BPF). The Varying blades pitch, defects of manufacturing blades, corrosion of the blades during working, thickness of the blades, the asymmetry of hydrodynamic forces and the existence of asymmetric pressure field around the body can increase noise and vibration forces.

Conventional procedures to study the propeller unsteady force are the lifting surface and the panel methods. Kerwin³ applied the unsteady vortex lattice technique to formulate the unsteady propeller. Hoshino⁴ employed the panel method to simulate unsteady flow on propeller. These methods don't account viscous effects, such as the boundary layer and separation flow and usually repair results with empirical treatments. To conquer the deficiency of potential methods, URANS model have been successfully employed for marine propellers. Funeno⁵ studied unsteady flow around a high-skewed propeller in non-
uniform inflow. Hu applied URANS model to simulate the test case, DTMB 4119, propeller worked on non-uniform inflow conditions. Li investigated numerical prediction of flow around a propeller. His results were carried out using the commercial CFD software FLUENT.

The objective of the current study is to conduct a numerical simulation of the acoustic field generated by a marine propeller in uniform inflow. The far-field radiation is predicted by integral formula Ffowcs Williams-Hawkings (FW-H) equation, with the solution of the URANS solution. The hydro acoustic performances of test case propeller are compared with experimental test.

Materials and Methods

Acoustic equations

The first and most accomplished research in acoustic waves has been done by the Lighthill in 1952. He used continuity and momentum equations without simplification to obtain overall sound production relationship. By writing the continuity equation respect to time and the momentum equation as follows:

\[
\frac{Dp}{Dt} + \text{div}(\rho u) = \frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_j)}{\partial x_i} = q
\]

where \( q \) is the mass production rate per unit volume and the momentum equation as follows:

\[
\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial \rho}{\partial x_i} - \rho g \frac{\partial \tau}{\partial x_i} + f_i
\]

where \( f_i \) represents the body forces. Derivation the momentum equation respect to \( x_i \) and the continuity equation respect to time and subtracted from each other, the Equation 3 is retrieved.

\[
\frac{\partial^2 \rho}{\partial t^2} + C_0^2 \nabla \cdot \rho = \frac{\partial^2 (T_{ij})}{\partial x_i \partial x_j}
\]

In this equation \( C_0 \) is the sound speed and \( T_{ij} \) is the Lighthill stress tensor as Equation 4.

\[
T_{ij} = \rho u_i u_j + \delta_{ij} (p - \rho C_0^2) + \tau_{ij}
\]

In this equation the first term on the right Hand side is the turbulence velocity fluctuations (Reynolds stresses), the second term is due to changes in pressure and density and the third term is due to the shear stress tensor.

A generalization of Lighthill’s theory included hydrodynamic surfaces in motion, suggested by Ffowcs Williams & Hawking. He has prepared the basis for a important amount of analysis of the noise caused by rotating blades, including propeller, and fans. The Ffowcs Williams & Hawking (FW-H) theory contains surface source terms in addition to the quadrupole source presented by Lighthill. The surface sources are generally indicated thickness (or monopole) sources and loading (or dipole) sources. The FW-H equation is presented by relation (5).\(^\text{10}\)

\[
\frac{1}{c_0^2} \frac{\partial^2 p'}{\partial t^2} - \nabla^2 p' = \frac{\partial^2}{\partial x_i \partial x_j} \left[ T_{ij} H(f) \right] - \frac{\partial}{\partial x_i} \left[ \left( p_{ij} n_i + \rho u_i (u_n - v_n) \right) \delta(f) \right] + \frac{\partial}{\partial x_i} \left( \rho_0 v_n + \rho (u_n - v_n) \right) \delta(f)
\]

The terms in the right hand side are named quadruple, dipole and monopole sources, respectively. \( p' \) is the sound pressure at the far-field (\( p' = (p - p_0) \) ), while \( c_0 \)is the far-field sound speed and \( T_{ij} \) is the Lighthill stress tensor by equation 4. Also \( H(f) \) and \( \delta(f) \) are Heaviside and Dirac delta functions, respectively.\(^\text{10}\) By solving this equation pressure variation is found and sound pressure level calculate by

\[
SPL = 10 \log_{10} \frac{p_{rms}^2}{p_{ref}^2} = 20 \log_{10} \frac{p_{rms}}{p_{ref}} = 20 \log_{10} P_{rms} - 20 \log_{10} P_{ref}
\]

The sound pressure is a measured root mean square (rms) value and the internationally agreed reference pressure \( p_{ref} \).

Definition

The propeller model that was studied in this paper is shown in Fig.1. The model was generated by GAMBIT 2.4. The Main dimensions of the propeller are presented in Table 1.\(^\text{11}\) Also, operating conditions and its hydrodynamic characteristics results for five operation points are reported in Table 2. Incompressible finite volume method in commercial ANSYS Fluent 14.5 software is used.\(^\text{12}\)
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An unstructured hybrid mesh was applied for grid generation. Triangular cells are used for Blades and hub surfaces. A fine grid was used for near wall to capture the flow in this region. The grid aspect ratio gradually growing to decrease solution costs. A moving reference frame (MRF) was used to generate rotational speed around propeller. The generated grid is shown in Fig. 2. About two million cells have been produced for overall grid.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of blades</td>
<td>4</td>
</tr>
<tr>
<td>Propeller diameter (m)</td>
<td>2.100</td>
</tr>
<tr>
<td>Pitch ratio at 0.7R</td>
<td>0.8464</td>
</tr>
<tr>
<td>Expanded area ratio (EAR)</td>
<td>0.55</td>
</tr>
<tr>
<td>Hub ratio</td>
<td>0.276</td>
</tr>
<tr>
<td>Rake angle (deg.)</td>
<td>0</td>
</tr>
<tr>
<td>Skew angle (deg.)</td>
<td>40</td>
</tr>
</tbody>
</table>

Table 1- Main dimensions of the propeller

Table 2- Operating conditions and its hydrodynamic characteristics results

<table>
<thead>
<tr>
<th>State</th>
<th>Speed (Knot)</th>
<th>Nm (RPM)</th>
<th>J</th>
<th>KT</th>
<th>10KQ</th>
<th>η</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heavy load</td>
<td>1.3</td>
<td>197</td>
<td>0.1</td>
<td>0.34</td>
<td>0.46</td>
<td>0.11</td>
</tr>
<tr>
<td>Low heavy load</td>
<td>4</td>
<td>197</td>
<td>0.29</td>
<td>0.26</td>
<td>0.32</td>
<td>0.37</td>
</tr>
<tr>
<td>Moderate load</td>
<td>8.6</td>
<td>197</td>
<td>0.64</td>
<td>0.105</td>
<td>0.164</td>
<td>0.62</td>
</tr>
<tr>
<td>Low light load</td>
<td>10</td>
<td>197</td>
<td>0.74</td>
<td>0.051</td>
<td>0.104</td>
<td>0.58</td>
</tr>
<tr>
<td>Light load</td>
<td>12</td>
<td>197</td>
<td>0.9</td>
<td>0.03</td>
<td>0.07</td>
<td>0.43</td>
</tr>
</tbody>
</table>

In order to study the propeller performance in uniform flow and applying open water condition, a rectangular control volume was considered around propeller with velocity inlet and pressure outlet boundary conditions as Fig. 3.

The domain distances were considered large sufficiently to keep away from blockage effects on the propeller hydrodynamic performance characteristics.

The SIMPLE algorithm is used for pressure-velocity coupling equation and second order upstream discretization for momentum equations. Realizable k-e model is used to model turbulence with time step equal to 1e-4. To capture sound, the receivers are adjusted in z/D=0.5,1,1.5,2 and x/D=0.5,1,1.5,2 from center of the propeller as shown in the coordinate system of the propeller (Fig. 4).
Results and Discussion

For a better comparison, pressure coefficient distribution on the blade surface at normal and heavy loads is shown in Fig. 5 and Fig. 6 respectively. By growing propeller load, more thrust and torque are produced. At heavy loads (low advance ratios) the difference between pressure coefficients on both sides of the propeller increase significantly. Fig. 5 and Fig. 6 show that the range of the pressure coefficient on both sides propeller is much higher at heavy loads respect to normal loads. The pressure coefficient distribution on the suction side points out low pressure region at leading face. As seen, rotational blades lead to the existence of minimum pressure at this region and so highest pressure coefficient range at the pressure side.

Pressure coefficient contours at x/D=0.5,1,2,4 in the propeller upstream are shown in the Fig.7 and Fig.8 for normal and heavy load conditions respectively. Also in Fig. 9 and Fig. 10, pressure coefficient plots for back and face on blade in r/R=0.3,0.5,0.7,0.9 are presented.

The differences between pressure coefficients in leading edge to the trailing edge are mainly because of the variation in the flow regime and boundary layer influence in the direction of chord line.
Fig. 7 - Pressure coefficient contour on normal load condition in 4 section,
a) x/D=0.5  b) x/D=1  c) x/D=2  d) x/D=4

Fig. 8 - Pressure coefficient contour on heavy load condition in 4 section,
a) x/D=0.5  b) x/D=1  c) x/D=2  d) x/D=4
By comparing Fig. 9 (a) to (d), it is concluded that the difference between the pressure coefficient is much greater on external sections ((c), (d)) respect to internal sections ((a),(b)). In addition, according to Fig. 9 and Fig. 10 (a) to (d), the range of the pressure coefficient on both sides of heavy load propeller is much greater than normal load propeller, which show more thrust production at low advance ratios.

![Fig.9](image_url)  
**Fig.9-Pressure coefficient distribution on blade section for normal load condition,**

- a) \( r/R = 0.3 \)
- b) \( r/R = 0.5 \)
- c) \( r/R = 0.7 \)
- d) \( r/R = 0.9 \)

![Fig.10](image_url)  
**Fig.10-Pressure coefficient distribution on blade section for heavy load condition,**

- a) \( r/R = 0.3 \)
- b) \( r/R = 0.5 \)
- c) \( r/R = 0.7 \)
- d) \( r/R = 0.9 \)

The numerical result of the open water characteristics is shown in Fig.11. The definitions of the principle performance parameters of the propellers are explained as

\[
J = \frac{V_A}{nD} \quad K_Q = \frac{Q}{\rho n^2 D^5} \\
K_T = \frac{T}{\rho n^2 D^5} \quad \eta_0 = \frac{K_T}{K_Q} \frac{I}{2\pi} \tag{7}
\]
where $J$ is the advance ratio and is varied by raising the advance velocity, $K_T$ the thrust coefficient, $K_Q$ the torque coefficient and finally $\eta_o$ the propeller efficiency. $V_a$, $n$ and $D$ are advance velocity, propeller rotating speed and diameter of the propeller, respectively.

![Graph](attachment:image1.png)

**Fig. 11-** Open water characteristics of the propeller

Furthermore, the distributed axial velocity at $x/D=0.5,1,2,4$ in propeller upstream shows that velocity peak happen at the blade tip which is clear caused by the further distance from the propeller centre. Region with lowest velocity occurs at the blade root and there is a striking growth in the velocity from the root to the tip of the blades (Fig.12).

![Graph](attachment:image2.png)

Frequency spectrum was extracted from experimental result and compare with numerical results in $z/D=1$ for normal load condition as Fig.13. There is a good agreement between the numerical and experimental results. In this diagram for estimated frequency 80(Hz), the peak noise level equal to 143 (dB) is obtained. Noise at frequencies between 1,000 to 3000 drop to 100 dB. In frequency above 3000 Hz a few increasing sound power is observed.

Noise spectrum for further receivers in $z/D=0.5,1,1.5,2$ and $x/D=0.5,1,1.5,2$ is shown in Fig.14 and Fig.15 for normal loaded condition. Far away from propeller the sound power level is reduced. Also noise spectral in Heavy loaded condition in $z/D=0.5,1,1.5,2$ and $x/D=0.5,1,1.5,2$ is shown in Fig. 16 and Fig. 17.

![Graph](attachment:image3.png)

**Fig.12-** axial velocity contour in 4 section for normal load condition
a)$x/D=0.5$  b)$x/D=1$  c)$x/D=2$  d)$x/D=4$
Conclusions

In order to estimate the propeller noise spectrum a test case model was introduced and various investigations have been done on this sample test. The analysis of the results show that Reynolds average method offer acceptable accuracy with more efficient number of grid. It is clear that the use of the filtering methods, like LES and DES, gives better solution at near wall modeling as long as the meshing size is fine enough that it’s impossible in complex 3D geometry. But it can be fitted to the course grid to model with Reynolds average method that with respect to results achieved, the acceptable range will be extracted.
Acknowledgement

The numerical computations presented in this paper have been performed on the parallel machines of the High Performance Computing Research Centre at Amirkabir University; their support is gratefully acknowledged. The authors received no direct funding for this research.

Reference