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〈Original Article〉

A study on the method vibration analysis of marine pump

Dong-Myung Bae, Da-long Qi*, Bo Cao and Wei Guo

Dept. of Naval Architecture and Marine Systems Engineering, Pukyong National University, Busan, 608-737, South Korea

Generally, there are many kinds of pumps used on a ship, which play an obbligato role to assist the ship's running. However, due to the pump's own motor, it becomes a main vibration resource that may cause serious problems in local. If the above situation happened, it may cause local resonance even structural damage. Also, the bigger the pump's size is, the more serious the vibration is. Therefore, it is very important for researchers to estimate the vibrations situation of pumps accurately and avoid or reduce the probability of resonance. Based on a real marine pump-HHG8065, this paper did a vibration analysis by FEM (Finite Element Method) and compared the results with actual test. Finally, it gave a discussion on the estimation of pump and provided an improvement way to the vibration level.

Keywords: Marine pump, FEM, Vibration estimation, Simulation

Introduction

Because the marine pump is an important piece of auxiliary machinery, which can ensure the conducive sailing activities of ships and meet the needs of the crew and passengers, prediction of the pump vibration is an essential step in the designing and manufacturing of the marine pump (Abdel-Rahman, 19879). It can affect the safety of marine shipping and reduce the military characteristics of the warships if the amplitude of vibration is too large. In order to reduce the vibration and noise of marine pumps, analysis of vibration characteristics is necessary. The marine pump is taken as the research objective and the vibration characteristic analysis as well as vibration modal analysis are performed (Liu, 1996).

The free vibration analysis of the overall pump are mainly calculated. The free mode is the inherent nature of structural free vibration and only related to the structure itself. The modal of natural frequency and vibration mode can be calculated by using the theory of finite element. Through the analysis of natural frequency and main modes, it can be seen that the influence of the whole machine vibration from the body's natural frequency. Through the fuel oil pump body in modal analysis of finite element method (Nicolas et al., 1999). Calculation free mode of the top 20 order, summarizes the vibration mode of the free mode. The simulation results will be compared with the actual test results on the verification analysis of the natural frequencies of the fuel pump, in order to seek a convenient, quick method for getting natural frequency of the fuel oil pump and the modal analysis (Lardies et al., 2001).

Material And Method

Basic theory of modal analysis

Natural mode is the natural vibration characteristics of the mechanical structure. Modal characteristics can be obtained from each mode which located in a susceptible frequency range of structure by using modal analysis method. The differential equations of motion with n degrees of freedom

*Corresponding author: qidalong@pknu.ac.kr, Tel: +82-629-6613, Fax: +82-629-6608

damped system is:

$$[M]\{\ddot{\delta}\}+[M]\{\dot{\delta}\}+[C]\{\delta\}=\{F(t)\} \quad (1)$$

Where, $[M]$ is the mass matrix; $[C]$ is the damping matrix; $[K]$ is the stiffness matrix; $\{\ddot{\delta}\}$, $\{\dot{\delta}\}$, $\{\delta\}$ are the response column vectors of acceleration, velocity, displacement in the system, they are of order n; $\{F(t)\}$ is the column vector of dynamic loads. Under the circumstances of free vibration, the equation (1) reduces to:

$$[M]\{\ddot{\delta}\}+[M]\{\delta\}=0 \quad (2)$$

$$[\delta]=[\delta_0]\sin\omega t \quad (3)$$

$$\{\ddot{\delta}\}=-\omega^2[\delta_0]\sin\omega t \quad (4)$$

$$[K]\{\delta\}-\omega^2[M]\{[\delta]\}=0 \quad (5)$$

$$[K][\delta_0]=\lambda[M][\delta_0] \quad (6)$$

Where, λ $(\lambda=\omega^2)$ is called the generalized eigenvalues, $[\delta_0]$ called the generalized eigenvectors. ω and $[\delta_0]$ can be determined by using the above equation, and obtained *n characteristic* solutions as results: $(\omega_1^2, \{\delta_0\}_1)$, $(\omega_2^2, \{\delta_0\}_2)$ ⋯, $(\omega_n^2, \{\delta_0\}_n)$.

Finite element method

In the finite element analysis, the body is approximated as an assemblage of discrete finite elements interconnected at nodal points on the element boundaries. The displacements measured in a local coordinate system *x, y, z* (to be chosen conveniently) within each element are assumed to be a function of the displacements at the *N* finite element nodal points. Therefore, for element *m* is shown as follow:

$$\delta^{(m)}(x,y,z) = H^{(m)}(x,y,z)\,\hat{\delta} \quad (7)$$

Where, is the displacement interpolation matrix, the superscript m denotes element m, and is a vector of the three global displacement components Ui, Vi, and Wi at all nodal points, including those at the supports of the element assemblage; i.e., Ui is a vector of dimension 3*N*,

$$\hat{\delta}^T = [U_1V_1W_1\ U_2V_2W_2\ U_NV_NW_N] \quad (8)$$

Results

This paper based on the actual project of fuel oil pump that was used on warship. According to the papers of the pump established can reflect the characteristics of the entity in the real 3D model, and become the reference for the simplified finite element model has been made. And the FE model is as follows.

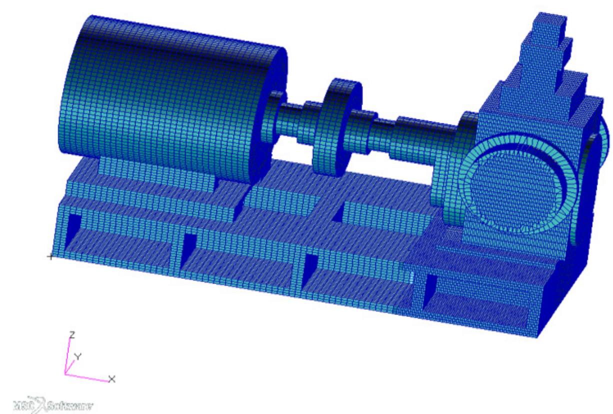

Fig. 1. The FE model of the pump body

As shown in the Fig. 1, the number of nodes are 418,682 and the number of elements are 210,474. The main dimension and data of the HHG8065 gear pump is shown in Table 1.

Table 1. The actual dimension and data

Item	Value of gear pump
Proportion	0.845
Shape size	933×431×656 MM
Total weight	262 KG±5%
Rotate speed of motor	1200 RPM
Rotate speed	1200 RPM

Pump natural vibration

From the study of modal simulation and the FE modal, the natural frequency and vibration modes in free vibration analysis are obtained. The results are presented in Fig. 2-7.

Through the analysis to get the 14 orders modal and also the natural frequency of the pump's values are shown in Table 2.

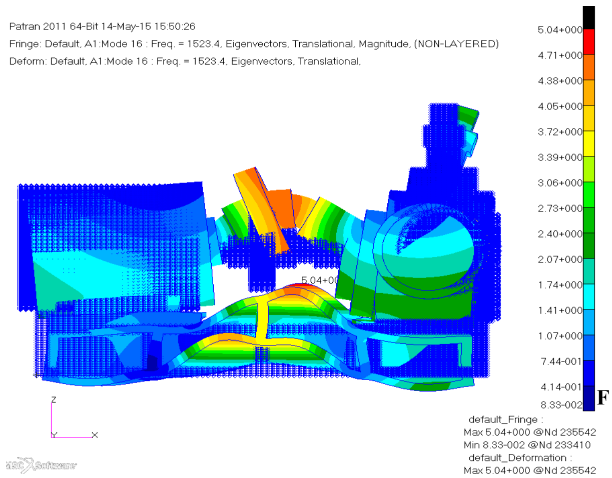

Fig. 2. The 1st vertical vibration mode

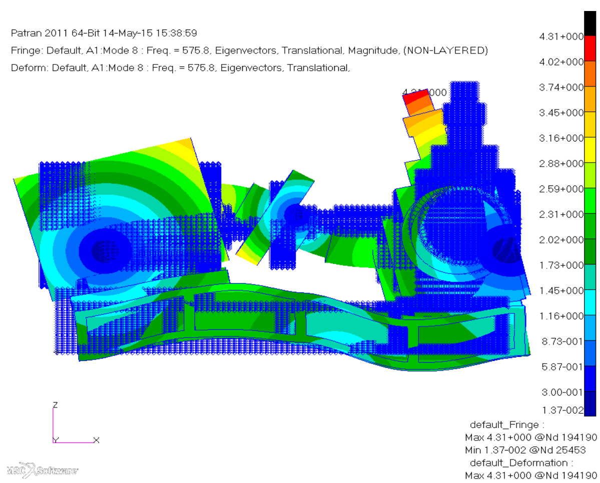

Fig. 3. The 2nd vertical vibration mode

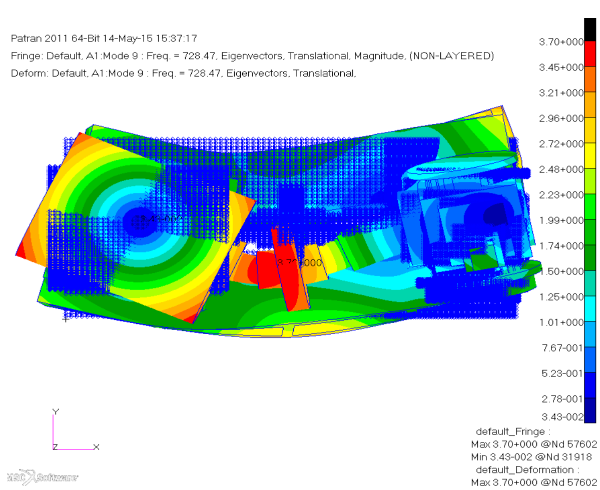

Fig. 4. The 1st horizontal vibration mode

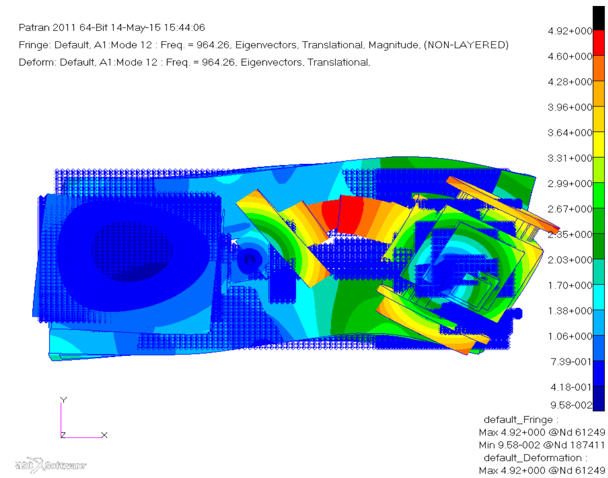

Fig. 5. The 2nd horizontal vibration mode

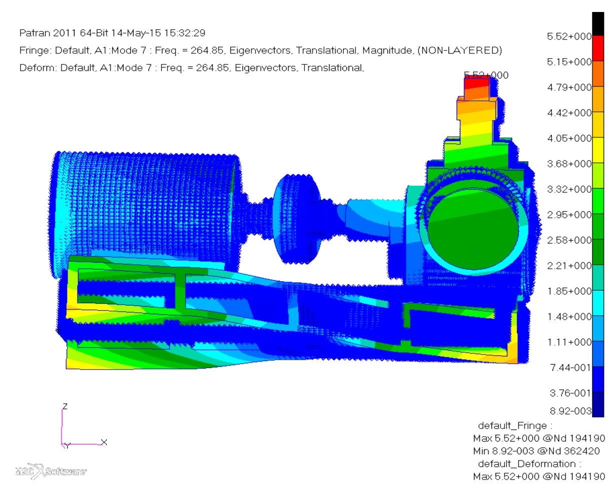

Fig. 6. The 1st torsional vibration mode

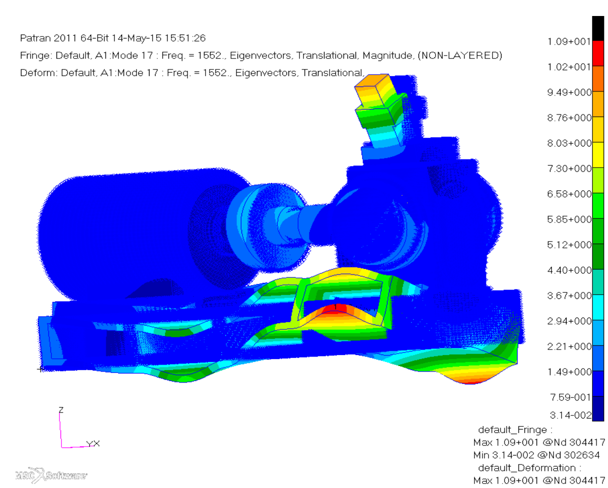

Fig. 7. The 2nd torsional vibration mode

Table 2. The HHG8054 natural frequency calculation results of the FE model

Order	Frequency (Hz)	Order	Frequency (Hz)
1	264.9	8	1362.8
2	575.8	9	1397.8
3	728.5	10	1523.4
4	796.1	11	1552.0
5	917.1	12	1632.7
6	964.3	13	1846.5
7	1121.5	14	1897.0

The modal analysis of the pump system which the natural frequency in 1000 Hz-2000 Hz interval has a greater growth and through the modal analysis of pump, it is found that the pump body of high order natural frequency occurred in the middle part of the chassis.

Response to excitation

The frequencies of the motor:

From the Table 1, it can be known the speed of the motor is 1200 RPM, therefore, the frequency of the motor is:

$$f = 1200/60 \times 6 = 120 \text{ Hz}$$

The gear mesh frequency of the pump:

According to the pump its gear mesh frequency $f = nz / 60$, when rotational speed $n = 1200$ r/min, the gear teeth $z = 16$, and the pump has gear mesh frequency $f = 320$ Hz.

Though the modal analysis can give information about the natural frequency and vibration mode of structure as well as take corresponding measures according to the results of modal analysis. It can avoid unnecessary loss caused by the structure resonance during the course of work. The frequency of the motor and pump is calculated by the third chapter shows that, the frequency of the motor is 120 Hz and the pump its gear mesh frequency $f = 320$ Hz, so the resonance will not occur when the motor and pump running.

Local vibrations

Due to the actual vibration test points of the pump at four points near the chassis bolt, this analysis through to the nephogram of the chassis as the center for research and analysis. The Fig. 8-10 represent the nephogram about pump's chassis.

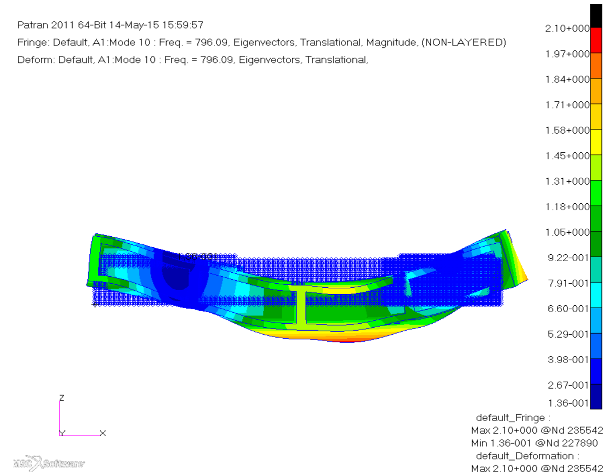

Fig. 8. The vertical local vibration mode

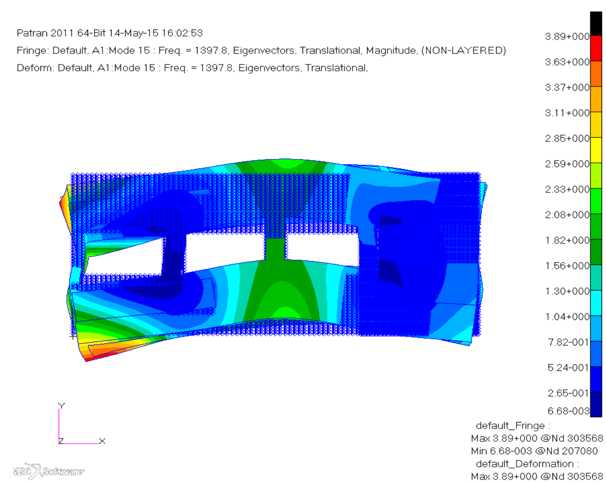

Fig. 9. The horizontal local vibration mode

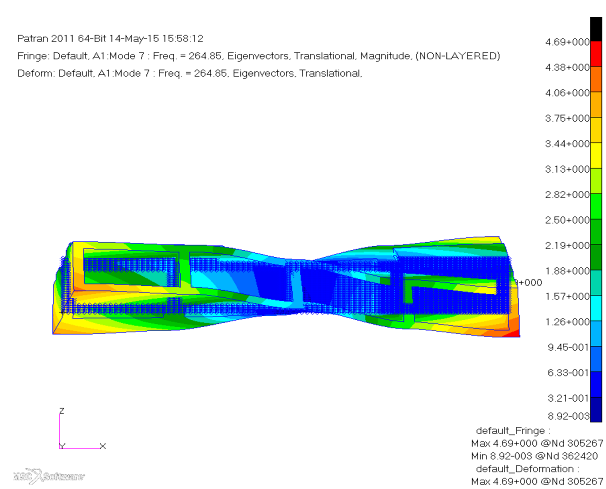

Fig. 10. The torsional local vibration mode

Through the analysis, the results of the vibration diagram of the pump's chassis are obtained, following conclusion to analyze the vibration mode of the pump's chassis are concluded.

Actual test

In this paper, based on the fuel oil transfer pump as the research object, through the actual experiments to understand the vibration of the pump. This chapter presents the testing instruments, equipment, and the vibration test method and summarize the vibration of the pump.

(a) Vibration measurement method based on : MIL-STD-740-2(SH):1986 ; The experiment evaluation standard based on (MIL-STD-740-2(SH):1986 Type).

(b) Measurement instrument system

The measurement instrument system are shown in Fig. 11.

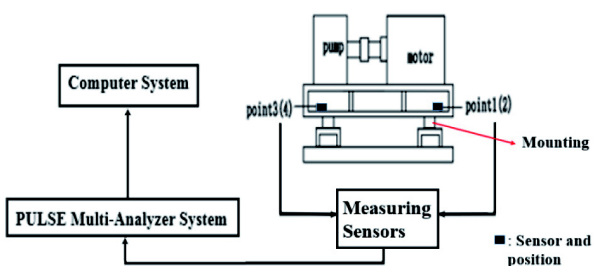

Fig. 11. The measurement instrument

As shown in Fig.11, the measuring position are four corners of the pump chassis.

The average of actual field testing is presented in Fig. 12.

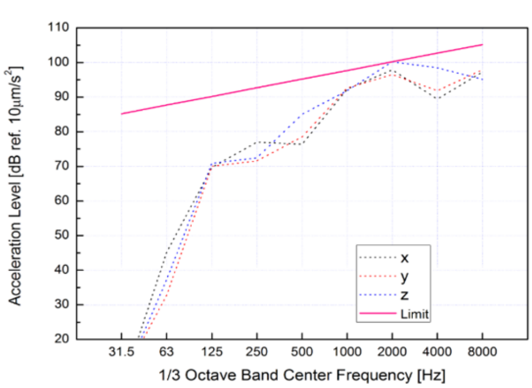

Fig. 12. The actual vibration frequency

The actual test data showed that between 1000 Hz and 2000 Hz, the pump is most likely close or exceed to the standard line, and reduce the vibration of the pump is still the main problems need to solve.

Discussion

The modal analysis of the pump system which the natural frequency in 1000 Hz-2000 Hz interval has a greater growth and at the same time the displacement of intermediate position of the chassis is large. Through the modal analysis of pump, it is found that the pump body of high order natural frequency occurred in the middle part of the chassis.

In order to improve the large displacement problem in the middle of the chassis of the pump, thickening of the middle portion at the bottom of the entity of the pump and at the same time broaden the width of the chassis can be taken as solution step for the displacement problem. The modal analysis is conducted to obtain the results of this solution step. The result of modal analysis for this setting is presented in Fig. 13 and Table 3.

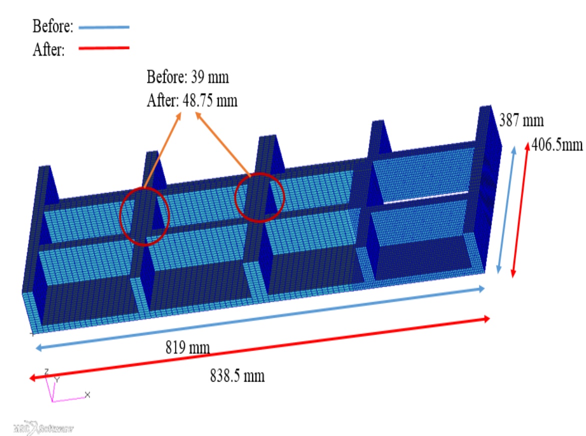

Fig. 13. The FE model after improvement

Table 3. The natural frequency calculation results of the FE model

Order	Frequency (Hz)	Order	Frequency (Hz)
1	262.4	8	1327.8
2	563.8	9	1391.7
3	712.5	10	1467.4
4	768.0	11	1498.0
5	870.3	12	1570.6
6	950.9	13	1658.4
7	1082.1	14	1697.6

It can be seen from Table 3 that the natural frequency of the pump is reduced after thickening and widening the chassis of pump, especially in the fifth, tenth, twelfth, thirteenth, fourteenth orders are decreased significantly. The natural frequency of the second, eighth increase, but the maximum displacement occurs at the uppermost part of the pump. It does not affect the connection of the pump and ship, so it can be ignored.

This condition also helpful to reduce the noise and vibration of pump, prolonging the life of the pump and the boat deck. However, while the option for thickening and widening chassis, the overall quality of the pump also increased. This will reduce the economy of the pump.

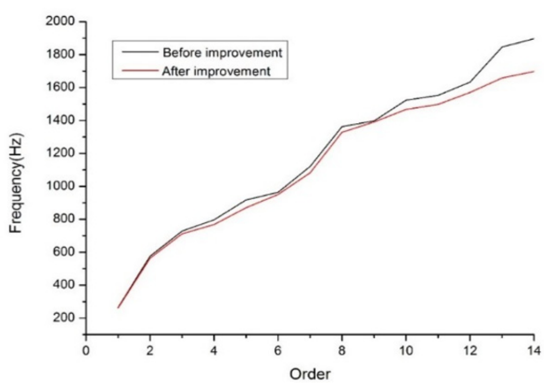

Fig. 14. The results of free vibration after improvement

Fig. 14 presents the companion between before and after improvement is taken. With the option for thickening and widening the chassis of the pump are taken, the top 14 natural frequency have been decreased, especially in the higher modes. For these nodes, the natural frequency reduce significantly.

The maximum displacement of the chassis is also reduce significantly, especially in the higher modes. After the chassis widening and thickening, the natural frequency of the pump can be reduced to some extent, and reduce the vibration value and amplitude of the pump during the actual operation of the process. The solution for widening and thickening can be decreasing the vibration damage to other equipment and the noise impact on the environment arising.

CONCLUSION

This thesis takes the marine pump as the research objective. The vibration characteristic analysis is conducted. According to the free vibration analysis the following results were obtained. The study of the vibration characteristics of the marine pump system has important theoretical value and innovative significance.

It is a convenient way to get the natural frequency of the fuel pump by using FEM, which lays the foundation for further analysis of the vibration characteristics of the pump, especially in the process of the virtual prototype design. It provides a feasible and convenient way to do analysis on the vibration characteristics of other machine parts.

Receiving the top 14 order constraint modes by calculating the FE model. It provides a theoretical basis for further improvement of the rigidity and the structure design of the pump body. Judging from the results, reasonable improvement schemes can reduce the natural frequency of the original pump by changing the distribution of the weight of the pump body’s structure. This can keep the natural frequency of the pump within the scope of the standard and it is a convenient way to manufacture the pump.

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