MODELING AND SIMULATING AXIAL VIBRATIONS ON THE CRARKSHAFT OF THE LARGE TWO-STROKE MARINE DIESEL ENGINE

Ngo D.Vuong[1], Lai H.Thien[3], Luu M.Hai[4,*], Hoang Q.Dong[3], Vu V.Tuyen[3], Do D.Luu[2]
1. Naval Technical Institute, Hai Phong City
2. Nam Can Tho University, Can Tho City
3. Vietnam Maritime University, Hai Phong City
4. Naval Academy, Khanh Hoa
Corresponding author: Luu M. Hai; Email: minhhaicdk46@gmail.com

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ABSTRACT

The axial vibration is the strong vibrated phenomenon occurring on the main propulsion using a large two-stroke marine diesel engine. This article researches to build a mathematical model of the axial stiffness coefficient and force generated by the geometric configuration of the crank pin-crank web assemblies and the combustion gas force. The numerical simulation is performed by finite element method, using ANSYS software, applied to the main engine MAN B&W 6G70ME.

Keywords: *Axial stiffness; crank pin–crank web set; axial force; diesel crankshaft; large two-stroke marine diesel engine; Axial vibration simulation.*

I. INTRODUCTION

On the main propulsion plant (MPP) of the motor vessel (MV.) using the two-stroke marine diesel (2-stroke MDE), the axial vibrations (AVs) are often monitored by engine builders and shipyards, as well as by eliminators of the axial movement, for examples, MAN-B&W MDE [**[5]**, **[6]**]. Therefore, on the series of the newly built MVs. and the currently operating systems designed in Japan, Korea are installed axial vibration monitoring (AMV, Axial Vibration Monitor).

Theoretically, many world scientists on MPP dynamics, such as Minchev N.D (Bulgaria, 1983,[3], Sevastakiev V (Bulgaria) [4]; Terskix V.P (Russia)… studied the AVs from 1970-1980. In monograph [1], Minchev modeled AVs of the axial system into a model consisting of n- lumped masses under the influence of forced longitudinal forces at the masses of the cylinders and propeller. Some research works of Terskix (with his PhD. students and colleagues) modeled the propeller shaft system by the finite element method. Some of the scientific works (in Bulgaria and Russia) have shown some results about the relationship between torsional and axial vibrations in the MPP. In the monographs [3, 4], Minchev and Sevastakiev gave a lot of empirical formulas to determine the axial stiffness coefficient of the crank pin–crank web set (CPCWS) for the calculation of the models of axial motion in diesel engines. However, the results given when applying the empirical formulas have very large deviations, so it is difficult to indicate which formulas can be used with permissible reliability.

In this study, the main research objective is to build a mathematical model to determine the axial stiffness coefficient of CPCWS by numerical simulation by finite element method.

II. MODELING AND CALCULATION MDE 2-STROKE CPCWS' AXIAL VIBRATION

1. Equivalent axial crankshaft force

The nature of generating axial oscillations of the CPCWS is described in following. The action of the gas force acting on the piston top is transmitted through the piston rod-crosshead-connecting rod and then acting on the crank pin. The gas force is a function of time variable, according to the crank-shaft angle (CA, $\phi(t)$) position. Due to the complex and frame-shaped structure of the CPCWS, when the crankpin is bent under the action of the bending force, the crank pin - crank web set will be deformed. The distance between two crank-webs appears "dynamic deflection". The dynamic contraction of CPCWS is shown in Figure 1.

The combustion force generated in the engine cylinder is simulated according to the working process of the internal combustion engine. The research object is a 2-stroke MDE used on ships with the following characteristics: for a working cycle, the crankshaft is running a revolution, that means 360 degrees (CA); the inside cylinder pressure is indicated and determined by Indicator Diagram.

The acting force on the cylinder's piston is denoted by P [unit: N or kN], which is analyzed into two force components: the force acting along the connecting rod and the friction force (at point B). The component force acting on the connecting rod is analyzed (at point A, the coupling between the connecting rod and the shaft) into two forces: radial (N) and tangential force (T). The T-force generates a torque. The radial force N is perpendicular to the shaft, but depending on the position of the crankshaft rotation angle causes the crankshaft to deform (figures 1, [1, 3]), which is equivalent to replacing it with a force F (vertical shaft) impact.

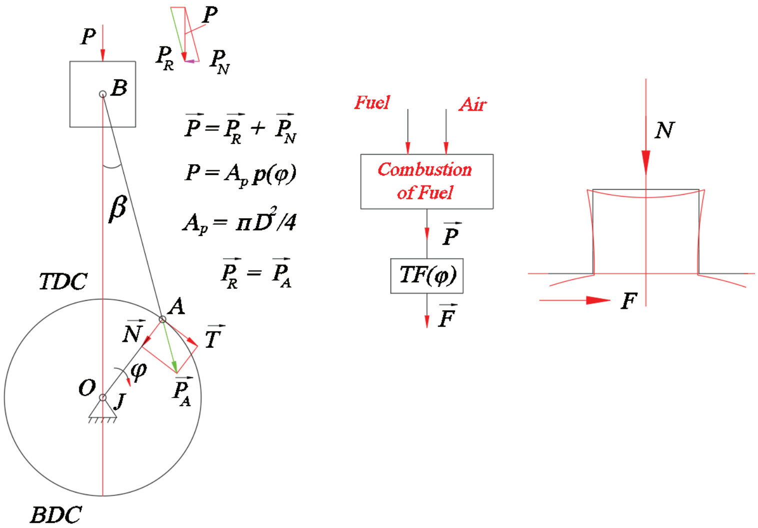

Figure 1. Relationship of combustion expansion force - axial force.

In Figure 1, assume that the angle formed by the connecting rod and the cylinder center line is β, and the position angle of the crankshaft - ϕ, deg.- the angle of rotation of the crankshaft. With the symbols: Radius of rotation R = S/2, (S is piston stroke, R is crankshaft radius, m), connecting rod's length L, m, cylinder's diameter D, m. Set λ = R/L –the crankshaft and connecting rod parameter. We see,

$$P_R = P_A = \frac{P}{\cos\beta}; N = P_A \cos(\varphi+\beta) = P.\frac{\cos(\varphi+\beta)}{\cos\beta} \quad (1)$$

With C_R transfer function for axial force

$$C_R = \frac{F}{N} = \frac{F}{P}\frac{\cos\beta}{\cos(\varphi+\beta)} \quad (2)$$

The axial forcing force is calculated by the empirical formula C_R [3]:

$$F = C_R P\frac{\cos(\varphi+\beta)}{\cos\beta} = p(\varphi)\frac{\pi D^2}{4}\frac{\cos(\varphi+\beta)}{\cos\beta}C_R \quad (3)$$

Using mathematical transformations and Taylor expansions, the converted axial force at a crankshaft is determined by the following formula:

$$F(\varphi)=C_R.\frac{\pi D^2}{4}\mathrm{p}(\varphi).(\cos\varphi\text{-}\lambda\sin^2\varphi) \qquad (4)$$

As a result, the problem is brought back to determine the axial force due to the gas force, which is determined according to the indicated pressure p(φ).

2. Calculation of equivalent stiffness

The calculation of axial vibration should be easy once the axial force in the previous section is completely determined. The problem here is to determine the equivalent stiffness of the shaft segment. In [3, 4], a variety of traditional formulas are given; however, they are significantly different.

According to the Dorei formula, we have:

$$C=\frac{E}{R^2}\left[\frac{1{,}65l_2}{J_{M.Ш}}(1-f)+\frac{R}{J_P}(\frac{2}{3}-f)\right]^{-1},\frac{N}{m} \qquad (5)$$

Where, R is the crankshaft radius; E - modulus of elasticity; l_2 - length of connecting rod-pin; J_w - moment of inertia of the crank web (m^4), and J_P -moment of inertia of the crank-pin, (m^4).

For the verified MDE 6G70ME-C9.2: R = 1.628m; $E_1 = E_2 = 200E+9$ Pa; $l_2 = 0.25$m; $J_w = \pi d_2^4(1-\eta^4)/64 = \pi \text{ x } 1.09^4(1-\delta_2^4/d_2^4)/64 = 0.138$ (m^4); , $J_P = 0.009$ (m^4).

It can be achieved the equivalent stiffness C= $1.4\text{x}10^9$ (N/m).

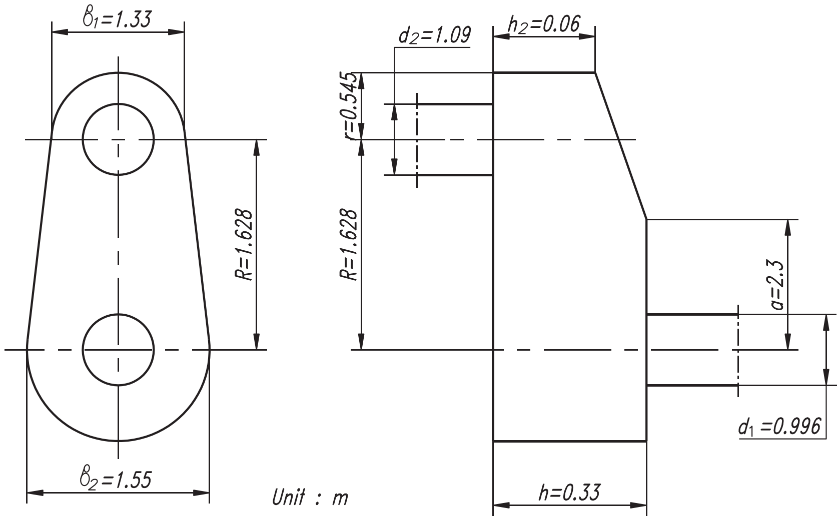

Figure 2. Crankshaft dimensions of the typical 2-stroke MDE.

The equivalent stiffness based on Gulialmoti - Machot is given by

$$\frac{1}{C}=\frac{64R^2l}{\pi Ed_2^4}+\frac{7{,}2R(R-0{,}44d_2)^2k_5}{E\textit{в}h^3},\frac{N}{m} \qquad (6)$$

In which, l - length of crankshaft, m; - width of web, m; h - thickness of web, m; d_2 - diameter of the connecting rod's-crank pin, m; The coefficient is determined by

$$k_5=\frac{2}{3}-\frac{1}{3}\cos\alpha_{cp}$$

And, α_{cp} is the mean angle of deviation of crankshaft.

For the verified MDE 6G70ME-C9.2: l = 0.34 m; = (1.55+1.33)/2 = 1.44m; d_2 = 1.09 m; $\alpha_{cp} = 60^0$.

Then, we have: C= $1.214\text{x}10^9$ (N/m).

According to the calculation Skorchev, we have:

$$C=\frac{E}{k_\varphi R^2}\left[\frac{R}{J_\omega}k_0+\mathrm{Z}_0\frac{l}{J_p}\right]^{-1},\ \frac{N}{m} \qquad (7)$$

where

$$k_{\varphi} = 0.75\frac{\varphi_{cp}}{180^0} + 0.33 = 0.58; k_0 = 0.222(\frac{R}{d} + 1,17) = 0.591; z_0 = 0.25$$

l and *d* are the length and diameter of connecting rod's neck, m (l = 0.25m; d = 1.09m)

$$J_p = \frac{\pi d^4}{64} = 0.0693; \varphi_{cp} = \frac{\varphi_1 + \varphi_2}{2} = 60^0$$

φ_1, φ_2 is the angle between the segments of crankshaft ($\varphi_1 = \varphi_2 = 60^0$);

$$J_{\varpi} = \frac{в_e h_e^3}{12} = 0.00431m^4, \text{ với:}$$

$в_e, h_e$ and the equivalent width and thickness of the web, $в_e = в$, and $в$ = 1.44m, h_e = h=0.33m.

Eventually, the equivalent stiffness is calculated: C = 5.735x10^9 (N/m).

III. SIMULATION OF AXIAL VIBRATION AND ANALYSIS

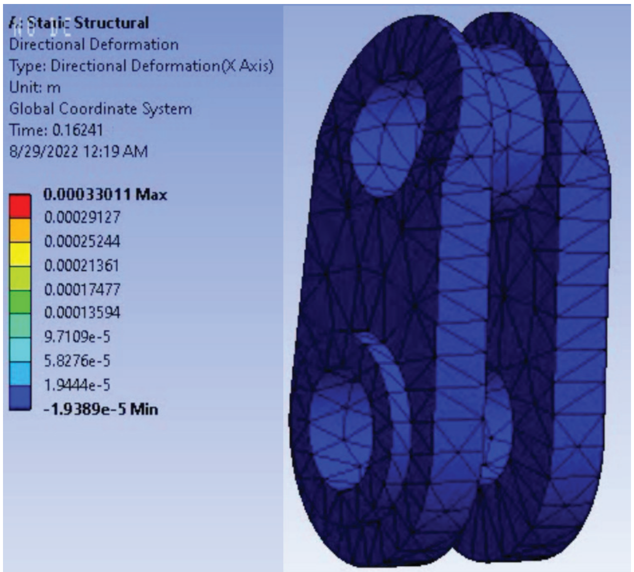

Figure 3. 3D meshed model of an shaft segment in ANSY

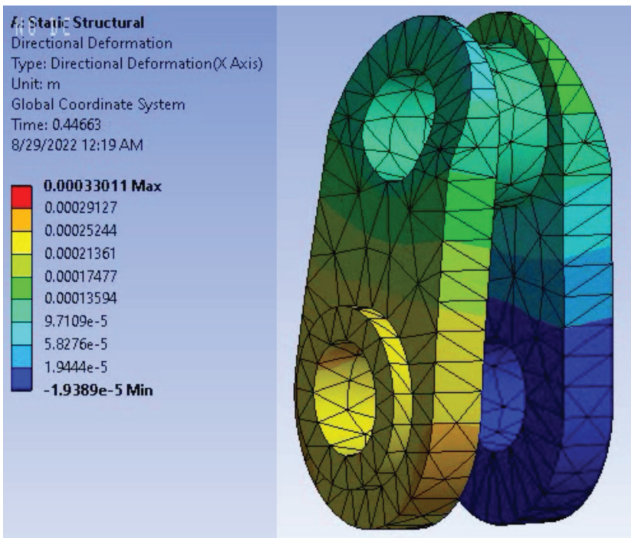

Figure 4. Crank-shaft deformation

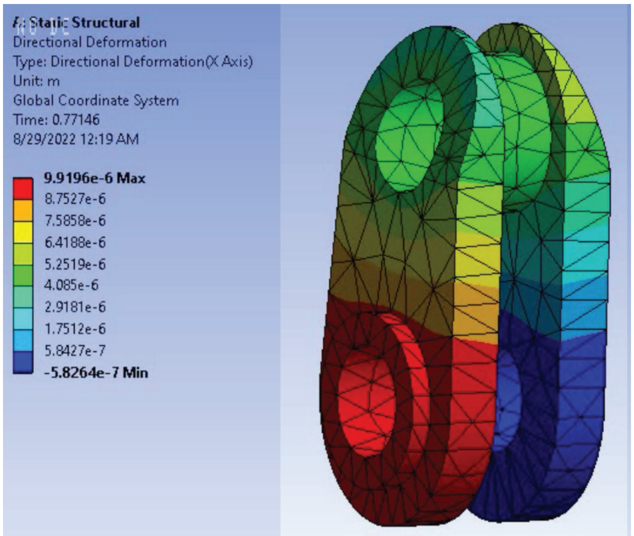

Figure 5. Shaft deformation values.

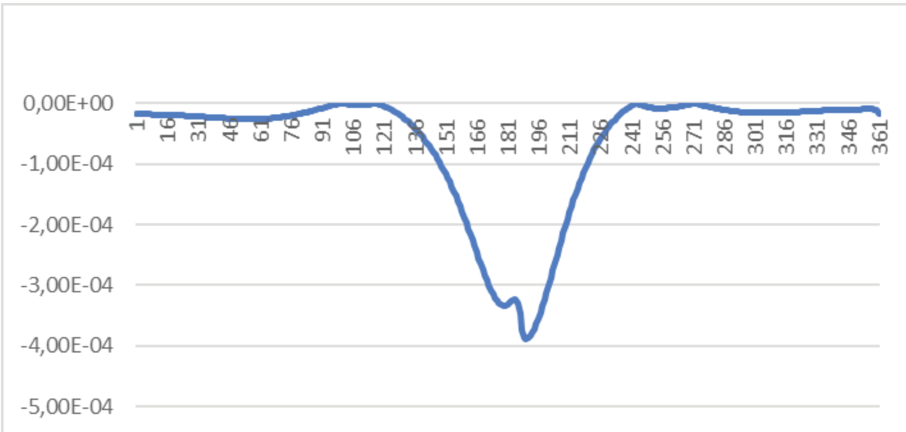

Figure 6. Shaft deformation graph.

The numerical simulation results performed on ANSYS software by FEM method are given from Fig. 3 to Fig. 6. In Fig. 3, the 3D-model of the crankshaft is built and meshed. All dimensions of the model are taken based on Fig. 2. By setting boundary conditions, force conditions, material declarations, and axial vibrations are calculated with given strain states, as shown in Fig. 4. In Fig. 5÷6, the deformation values are given. From these calculation results, the equivalent stiffness is determined by C=1.39E+9 (N/m). From that, it can be concluded that the formula (5) given by Dorei gives the closest results to the simulation results. Although it is not exactly the same as the simulation results, the Gulialmoti - Machot swimming calculation results give relatively accurate results. In another dimension, Skorchev gave computations far different from those calculated by the finite element method on ANSYS.

IV . CONCLUSION

Taking the verified MDE's parameters (MAN B&W 6G70ME), the article gives calculations for the most important parameters related to axial vibrations of the MDE crank-shaft, such as axial stiffness coefficient and equivalent axial force. The numeric experiments using the FEM and ANSYS software for the longitudinal vibrations of the CPCWS are developed under the combustion processes in side the MDE cylinder. Through simulation results and comparing with traditional formulas, conclusions can be drawn about the accuracy of these recommended formulas as well as of the accuracy of vibration analysis through the calculation of equivalent axial stiffness.

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