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TRANSIENT TORSIONAL VIBRATION ANALYSIS OF MARINE PROPULSION PLANTS

Abstract

The classical approach to the torsional vibration analysis of the marine propulsion plants deals exclusively with steady-state vibration, analyzed in a frequency domain only. However, some phenomena cannot be properly addressed without carrying out the more demanding, transient torsional vibration analysis, performed in the time domain. Examples of such events include passing through a barred speed range, clutching heavy components, and propeller–ice interactions during a voyage on icy seas. In this paper, a mathematical model of transient torsional vibration analysis is presented. The system response is based on the modified standard computer code for torsional vibration analysis, and the time integration of the system response is performed by utilizing a fifth-order and sixth-order Runge-Kutta-Verner algorithm. Example analyses of passing through a barred speed range are also provided and the measurement results compared.

Key words: analysis, propulsion plant, torsional vibration, transient vibration

ANALIZA PROLAZNIH TORZIJSKIH VIBRACIJA BRODSKIH PORIVNIH POSTROJENJA

Sažetak

Klasični pristup analizi torzijskih vibracija brodskih porivnih postrojenja isključivo obrađuje stacionarne vibracije, analizirane u frekvencijskom području. Međutim, postoje pojave koje se ne mogu ispravno obraditi bez zahtjevnije analize prolaznih torzijskih vibracija, provedene u vremenskomu području. Neki primjeri takvih pojava su prolaz kroz zabranjeno područje brzina vrtnje, ukapčanje masivnih prigona i međudjelovanje brodskoga vijka i leda tijekom plovidbe zaleđenim morima. U ovomu radu dan je matematički model postupka analize prolaznih torzijskih vibracija. Odziv sustava se određuje dorađenim standardnim računalnim programom za analizu torzijskih vibracija, a njegova integracija po vremenu vrši se primjenom Runge-Kutta-Vernerova algoritma petoga i šestoga reda. U radu su dani primjeri analize prolaza kroz zabranjeno područje brzina vrtnje, kao i usporedba s rezultatima mjerenja.

Ključne riječi: analiza, porivno postrojenje, prolazne vibracije, torzijske vibracije

1. Introduction

Propulsion plant reliability is of the utmost importance for the safe operation of vessels. Where the design of marine propulsion plants is concerned, torsional vibration behavior is one of the most important factors that define its reliability.

The classical approach to the torsional vibration analysis (TVA) of marine propulsion plants deals exclusively with steady-state vibration, analyzed in a frequency domain only [1]. However, some phenomena cannot be properly addressed without carrying out the more demanding, transient TVA, performed in the time domain. Examples of such events include passing through a barred speed range [2], clutching heavy components [3], the ship's crash astern operation [4], and propeller–ice interactions during a voyage on icy seas [5, 6].

Transient TVAs are calculation-intensive, time-consuming processes that require multiple runs and that are greatly affected by the user's interaction. In general, such analyses can be performed independent of steady-state TVAs by using commercial software tools [5] or by utilizing customized software [2, 3, 7]. Various approaches have been applied in the literature, from modal analysis [7] and the Newmark method [2] to the specific application of bond graph modeling [3]. In addition, some authors have preferred simplified plant models [5], while others have employed the full one [2].

In this paper, a mathematical model of transient TVA is presented. The transient TVA module is built as a new feature of the existing TVA tool [8]. Therefore, they both share the same system models and same data files. In addition, for the time integration of the system response, a fifth-order and sixth-order Runge-Kutta-Verner algorithm is utilized.

The remainder of this paper is organized as follows. In Section 2, the basics of the TVA are presented. Then, in Section 3, the numerical procedure of the transient TVA is defined. The two application examples of the proposed procedure are described in Section 4. Then, in Section 5, questions arising from the study are discussed. Finally, in Section 6, conclusions are provided.

2. TVA

The basis of the propulsion plant TVA is an equation of motion, written in a matrix form as

$$\mathbf{J}\ddot{\boldsymbol{\varphi}} + \mathbf{C}\dot{\boldsymbol{\varphi}} + \mathbf{K}\boldsymbol{\varphi} = \mathbf{f} \,, \tag{1}$$

where $\mathbf{J}$ is the inertia matrix, $\mathbf{C}$ is the damping matrix, $\mathbf{K}$ is the torsional stiffness matrix, $\boldsymbol{\varphi}$ is the displacement vector, and $\mathbf{f}$ is the vibration excitation vector. Equation (1) is a non-homogeneous system of second-order linear ordinary differential equations with constant coefficients. The *i*-th equation in system (1) has the form

$$J_i \cdot \ddot{\varphi}_i - c_i \cdot \dot{\varphi}_i - k_i \cdot \varphi_i + \sum_j \left(\varphi_j - \varphi_i\right) \cdot k_{i,j} + \sum_j \left(\dot{\varphi}_j - \dot{\varphi}_i\right) \cdot c_{i,j} = f_i \,, \quad i = 1,2,...,n, \tag{2}$$

where J_i is the node inertia, c_i is the absolute damping, k_i is the absolute stiffness, $k_{i,j}$ is the shaft torsional stiffness, $c_{i,j}$ is the relative damping, and *i* and *j* are the node indices and adjacent node indices, respectively. The number of equations in the system, *n*, corresponds to the number of nodes (lumped masses) in the torsional vibration model.

2.1. Steady-state TVA

Where steady-state TVA is concerned, by utilizing the proper substitution (see [7, 9, 10]), Eq. (1) is transformed into a system of algebraic equations with complex coefficients:

$$-\Omega^2 \mathbf{J}\hat{\boldsymbol{\varphi}} + j\Omega \mathbf{C}\hat{\boldsymbol{\varphi}} + \mathbf{K}\hat{\boldsymbol{\varphi}} = \hat{\mathbf{f}} \,, \tag{3}$$

where Ω is the excitation frequency, j is the imaginary unit, and $\hat{\boldsymbol{\varphi}}$ and $\hat{\mathbf{f}}$ are the vectors of the complex angular displacement and excitation amplitudes, respectively.

The excitation frequency Ω is defined as a product:

$$\Omega = m \cdot \omega = m \cdot \frac{\pi \cdot n_r}{30}, \tag{4}$$

where m is the order of excitation, ω is the angular velocity, and n_r is the engine speed, min^{-1}.

After the determination of the natural frequencies and angular displacements of all nodes ($\varphi_i, i = 1,2,...,n$), the remaining process is straightforward [7, 10]. The analysis results are usually summarized in graphic form, as depicted in Fig. 1.

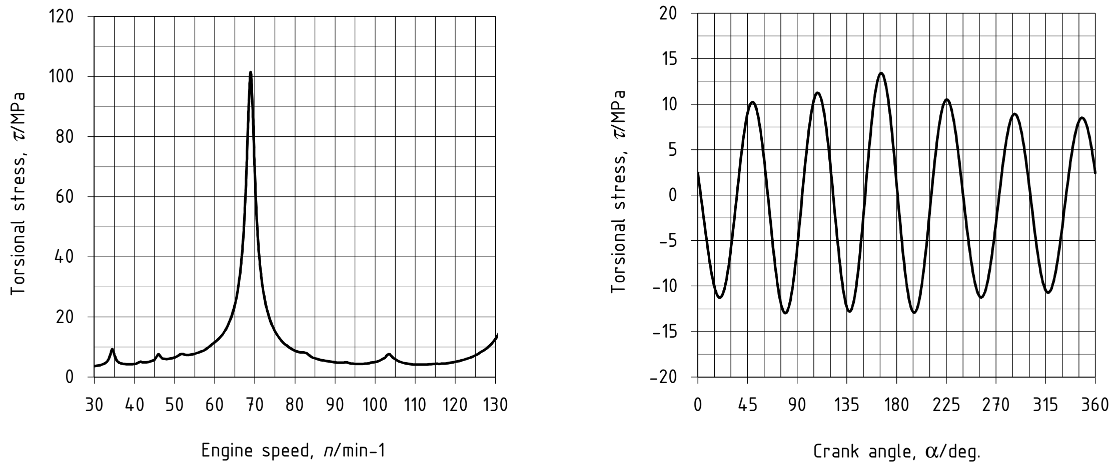

Fig. 1. Typical steady-state TVA results

Slika 1. Rezultati uobičajene analize stacionarnih torzijskih vibracija

2.2. Transient TVA

If the transient TVA is needed, the basis is the same as in the case of the steady-state TVA: the equation of motion, Eq. (1). However, the follow-up treatment is completely different. Since the vibration excitation $\mathbf{f} = \{f_1, f_2, ..., f_n\}$ is a function of time t,

$$f_i = \hat{f}_i \cdot e^{j\Omega t}, \tag{5}$$

the vibration response

$$\begin{aligned} \boldsymbol{\varphi} &= \{\varphi_1, \varphi_2, ..., \varphi_n\}, \\ \dot{\boldsymbol{\varphi}} &= \{\dot{\varphi}_1, \dot{\varphi}_2, ..., \dot{\varphi}_n\}, \\ \ddot{\boldsymbol{\varphi}} &= \{\ddot{\varphi}_1, \ddot{\varphi}_2, ..., \ddot{\varphi}_n\}, \end{aligned} \tag{6}$$

is a function of time, too. Therefore, in order to obtain the vibration response, the time integration of the differential equations system, Eq. (1), is needed. Unfortunately, analytical solutions can only be found for certain simple cases of Eq. (1) [11] and, therefore, a numerical procedure should be employed.

3. Numerical procedure

3.1. General procedure

By taking Eqs. (5) and (6) into account, system (1) can be viewed as a

$$\mathbf{F}(t,\boldsymbol{\varphi},\dot{\boldsymbol{\varphi}},\ddot{\boldsymbol{\varphi}})=\mathbf{0}, \tag{7}$$

i.e., as a system of n equations in $3n$ unknowns. In order to solve this system, the following three specific measures are required:

- *Initial conditions:* $\boldsymbol{\varphi}(t_0)=\boldsymbol{\varphi}_0$ and $\dot{\boldsymbol{\varphi}}(t_0)=\dot{\boldsymbol{\varphi}}_0$ ensure the viability of the foregoing procedure; if unknown, the simple condition $\boldsymbol{\varphi}_0=\dot{\boldsymbol{\varphi}}_0=\mathbf{0}$ can be applied.
- *New vector variable* (*additional set of equations*): by defining a new vector variable

  $$\mathbf{v}=\dot{\boldsymbol{\varphi}}, \tag{8}$$

  the second-order differential equation may be rewritten as a pair of first-order differential equations [12]. Then, Eq. (7) can be expressed by

  $$\mathbf{F}(t,\boldsymbol{\varphi},\mathbf{v},\dot{\mathbf{v}})=\mathbf{0}, \tag{9}$$

  while the number of equations rises to $2n$.
- *System acceleration*: the remaining n equations are obtained by substituting Eq. (8) into Eq. (2) and solving $\dot{\mathbf{v}}$ [10]:

  $$\dot{v}_i \equiv \ddot{\varphi}_i = \frac{f_i - c_i \cdot v_i - k_i \cdot \varphi_i + \sum_j (\varphi_j - \varphi_i)\cdot k_{i,j} + \sum_j (v_j - v_i)\cdot c_{i,j}}{J_i}. \tag{10}$$

  Equation (10) should then be evaluated by using the most recent values of φ_i and v_i.

3.2. Numerical integration

The numerical integration approach to solving the system of equations of motion, Eq. (1), belongs to so-called *time-marching schemes* [11], where the unknown analytical solution is approximated by a series of discrete responses obtained at the preset number of $m+1$ data points:

$$t_k = t_0 + k\cdot \Delta t,\quad k=0,1,2,\ldots,m, \tag{11}$$

where Δt is the time step and k is the time step counter.

Numerical integration by using a time-marching scheme is organized as follows. At the start of the process ($t=t_0$), two initial conditions ($\varphi_{i,0}$ and $v_{i,0}\equiv\dot{\varphi}_{i,0}$) enable us to determine the system acceleration, $\dot{v}_{i,0}\equiv\ddot{\varphi}_{i,0}$. Thereafter, we use a numerical integration algorithm to approximate the values of the displacements and velocities at the end of the time step interval ($\varphi_{i,1}$ and $v_{i,1}\equiv\dot{\varphi}_{i,1}$), which completes the computation of the first time step. The remaining time steps are then treated similarly. In this study, to integrate the differential equations, a fifth-order and sixth-order Runge-Kutta-Verner method is utilized (IMSL routine DIVPRK), as provided in [13].

4. Examples

The proposed procedure is tested against two transient torsional vibration measurement sets. These measurement results were compiled during official sea trials.

4.1. Example 1 - Engine startup transient motion

This example deals with the startup phase of a low-speed two-stroke propulsion plant, equipped with a CP propeller, set to zero-pitch during the run-up. The steady-state torsional vibration response is provided in Fig. 1, while the transient one is depicted in Fig. 2. During the timeframe shown, the propulsion plant was started up and run at 80 min^{-1}, some five seconds after the startup.

Fig. 3 presents the numerical simulation results obtained by using the proposed procedure. The simulation model comprised 13 nodes, and the majority of the data prepared for the standard steady-state analysis were used during the numerical simulation. The only exception was omitting the shaftline relative damping, since its inclusion caused numerical difficulties for the Runge-Kutta-Verner integration process (i.e. inability to satisfy the error conditions). The integration time step was 0,001 s.

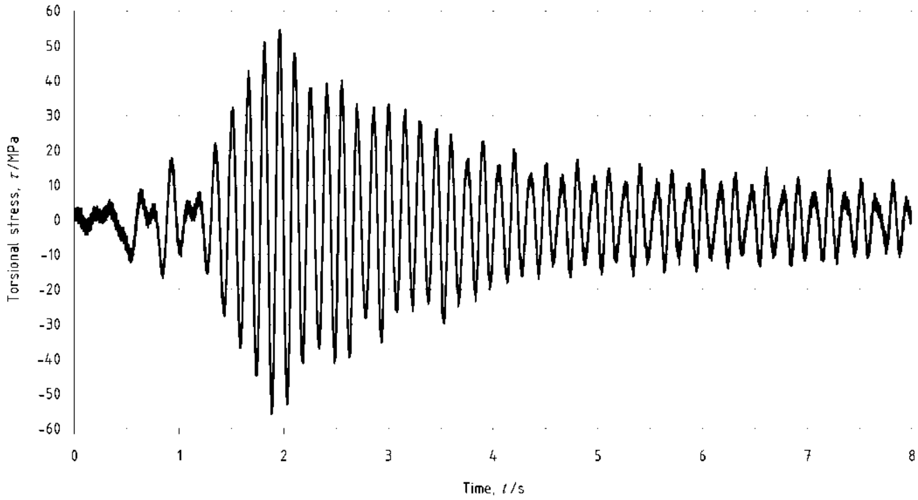

Fig. 2. Main engine startup measurement results

Slika 2. Rezultati mjerenja upućivanja glavnoga stroja

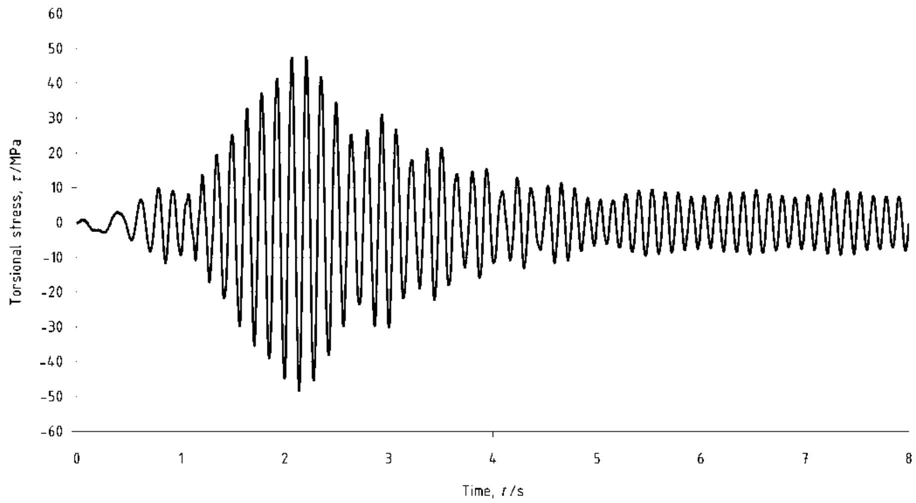

Fig. 3. Main engine startup simulation results

Slika 3. Rezultati simulacije upućivanja glavnoga stroja

It should be noted that the majority of the first second (from t = 0s to t = 1,3s) system response is mainly determined by an inflow of compressed starting air, rather than the ordinary diesel engine combustion process. Hence, it is reasonable to expect a lower simulated vs. measured correlation in that timeframe. When comparing the steady state (Fig. 1) with the transient vibration responses (Figs. 4 and 5), a significant stress level reduction is noted in the latter case, attributed to the rapid passing through of a barred speed range.

4.2. Example 2 - Escape from the barred speed range

This example deals with a bulk carrier FP propeller propulsion plant running at the major resonance at 60 min^{-1}. At t = 16 s, the engine speed ramps to 65 min^{-1}. The measured transient response is provided in Fig. 4, while the corresponding numerical simulation is shown in Fig. 5. The simulation model consists of 11 nodes, resembling exactly the model used during the steady-state analysis. The integration time step was again 0,001 s.

When comparing the simulated with the measured stress amplitudes, a steeper stress reduction is noted in the former case as well as more rendered resonance in the region of t = 19 s. Otherwise, the simulated transient vibration amplitudes agree with those of the steady-state TVA (data not shown).

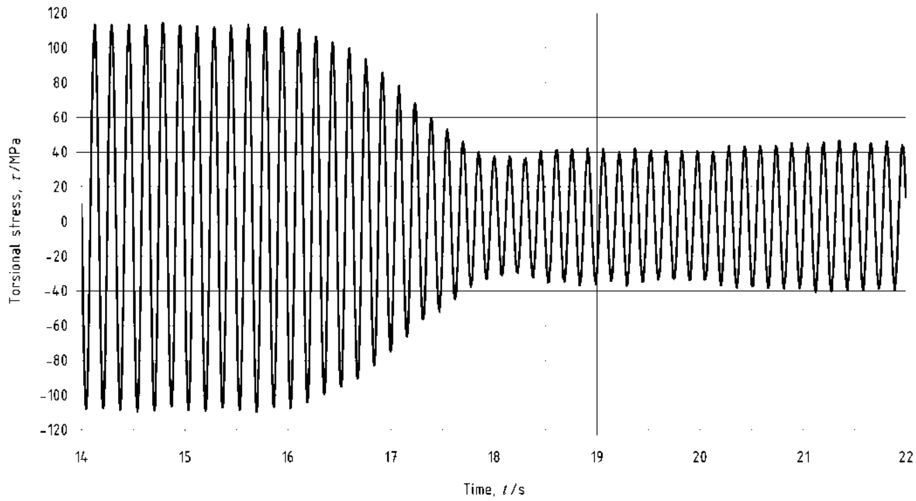

Fig. 4. Main engine escape from the barred speed range measurement results

Slika 4. Rezultati mjerenja izlaska iz zabranjenoga područja brzina vrtnje

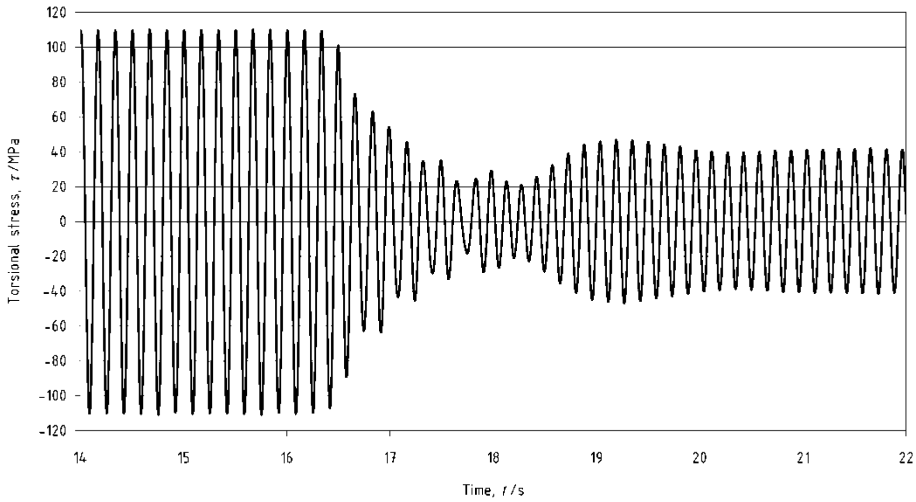

Fig. 5. Main engine escape from the barred speed range simulation results

Slika 5. Rezultati simulacije izlaska iz zabranjenoga područja brzina vrtnje

5. Discussion

The performed numerical simulations showed certain basic features of the proposed procedure. The proposed numerical procedure is capable of reasonably rendering the transient phenomena inside the propulsion plant. The full propulsion plant model may also be safely used as a basis for performing such simulations (as found by [2]), despite contrary views expressed in [5, 14]. Further, the numerical computations showed robust behavior and they were found not to be too sensitive to the time step size, as shown in Table 1.

Table 14. Influence of the integration step size on the simulation results (Example 1)

Tablica 1. Upliv veličine koraka integracije na rezultate simulacije (Primjer 1)

Δt /s	τ_{max} /MPa	τ_{min} /MPa
10^{-1}	38,67	−33,96
10^{-2}	42,86	−42,02
10^{-3}	43,64	−42,80
10^{-4}	43,64	−42,81
10^{-5}	43,64	−42,81

In accordance with the findings expressed in [5], the simulation results were found to be greatly affected by phasing the crank angle. This finding means that different system responses can be obtained by changing the starting moment of the simulated transient phenomena. In addition, it means that multiple simulations are necessary in order to find the most adverse transient response.

6. Conclusions

In this paper, a numerical procedure for the transient TVA of marine propulsion plants was proposed and examined. The performed numerical simulations showed reasonable agreement between the simulated and measured transient responses. In addition, the proposed procedure was found to be stable and not too sensitive to the selected integration time step size.

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References

[1] ...: "Dimensions of Propulsion Shafts and Their Permissible Torsional Vibration Stresses", M68, International Association of Classification Societies, London, 2012.

[2] LEE, D-C., KIM, S-H., YU, J-D.: "Theoretical Analysis on Transient Torsional Vibrations of Two Stroke Low Speed Diesel Engines", Journal of the Korean Society of Marine Engineering, 31(2007)3, 207-214.

[3] BRUUN, K., PEDERSEN, E., VALLAND, H.: "Modelling for Transient Torsional Vibrations in Marine Power Train Systems", Proceedings of International Conference on Bond Graph Modeling and Simulation, New Orleans, 2005.

[4] TAKAGI, M., SASAKI, S.: "Some Consideration for Behavior of Propulsion Diesel Engine in Transient Condition", Bulletin of the M.E.S.J., 24(1996)2, 57-63.

[5] DAHLER, G., STUBBS, J. T., NORHAMO, L.: "Propulsion in Ice - Big Ships", Proceedings of 9th International Conference and Exhibition on Performance of Ships and Structures in Ice, ICETECH, Anchorage, 2010.

[6] ...: "Ice Class Regulations and the Application Thereof", Transport Safety Agency, Helsinki, 2010.

[7] WALKER, D. N.: "Torsional Vibration of Turbomachinery", McGraw-Hill, New York, 2004.

[8] MAGAZINOVIĆ, G.: "*TorViC v2.2* - Program for Propulsion System Torsional Vibration Analysis", User's Guide (in Croatian), CADEA, Split 2014.

[9] HAFNER, K. E., MAASS, H.: "Torsionsschwingungen in der Verbrennungskraftmaschine", Springer Verlag, Wien 1985.

[10] VANCE, J. M.: "Rotordynamics of Turbomachinery", John Wiley & Sons, New York, 1988.

[11] DE JALON, J. G., BAYO, E.: "Kinematic and Dynamic Simulation of Multibody Systems: The Real-Time Challenge", Springer Verlag, New York, 1994.

[12] CARNAHAN, B., LUTHER, H. A., WILKES, J. O.: "Applied Numerical Methods", John Wiley and Sons, New York, 1969.

[13] ...: "IMSL − Fortran Subroutines for Mathematical Applications", Math/Library, Volumes 1 and 2, Visual Numerics, Houston 1997.

[14] ...: "Ice Strengthening of Propulsion Machinery", Classification Notes No. 51.1, Det Norske Veritas, Høvik, 2011.