

IMPLEMENTING SIMULATIONX IN THE MODELLING OF MARINE SHAFTING STEADY STATE TORSIONAL VIBRATIONS

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ABSTRACT

Marine propulsion shafting systems are exposed to torsional vibrations originating from excitations in their prime movers and propellers. It is essential to analyse their steady state response in the earliest stage of ship design. The paper describes the implementation of SimulationX software based upon simulation modelling for these calculations. This software can be used either by the design office of the shipyard or by the classification society for verification within the plan approval phase. Some specifics of the input data preparation are briefly discussed. In addition, the simulation results depend on the modelling approach chosen. For these reasons, the real two-stroke Diesel engine ship propulsion system was chosen and several different models were implemented for system modelling. SimulationX calculation results are compared with those of two well-known and field-proven programs that use an analytical approach. Finally, the results are compared with the measurements performed on the actual newly built ship. Discussion reviews the selected SimulationX model, and its verification and validation in the case of engine cylinders with normal ignition.

Keywords: mechanical dynamic system, critical speeds, torsional stress amplitudes, two-stroke marine Diesel engine, normal firing

INTRODUCTION

Marine propulsion shafting systems are exposed to dynamic excitations from their propulsion engines (usually diesel engines) and propellers. It is essential to analyse their torsional vibrations stationary response, as well as to verify calculated results within the plan approval process at the classification society in charge of classing the vessel. Furthermore, according to the classification rules, these results must also be validated by measurements of the torsional stress amplitudes of the shafts on board the ship over the operating speed range of the engine [1].

Marine classification societies are required to conduct a timely verification of the submitted calculations during the plan approval and to supervise validation of these verified results on board the first ship of the sister ship series. These

calculations result from the propulsion system design concept, the dimensions, the material properties and their dynamic loading. The results shall include natural frequencies, vibration modes, critical speeds, torsional stress amplitudes in the shaft sections, allowable stresses and barred speed ranges, if any. Verification of the results shall be based on the criteria specified in IACS Unified Requirements [2] as implemented in the technical rules of the classification society. Validation shall be performed by measuring the torsional stress amplitudes in an attainable section of the selected shaft and comparing them with the calculated values.

The first failures of marine propulsion systems shafting elements which can be attributed to the adverse effect of torsional vibrations were reported a long time ago, i.e. at the beginning of the 20th century, even before the first Diesel engine powered ocean-going ship entered into service [3].

These failures initiated the development of calculation and measurement methods for torsional vibrations. A systematic and methodical presentation of the calculation methods was published by Ker Wilson in 1935, with its 3rd edition in 1963 [4]. In 1968, Lloyd's Register published detailed instructions for calculating the torsional vibration characteristics of shafts within their classification rules [5]. This enabled the development of the computer program by Butković et al. [6], which was extensively used for these calculations in Croatia by Diesel engine manufacturers and the local classification society. An important contribution was the 1985 textbook by Hafner and Maass, which comprehensively presented the details of torsional vibration calculations and dedicated computer codes based on analytical methods [7]. Implementing his own analytical approach, Magazinović developed a powerful and user-friendly computer program [8] based on analytical methods, which is widely implemented in torsional vibration calculations in Croatia, especially in new constructions of the Croatian Shipyard Brodosplit. This program and its later extensions have been used and referred to in many publications, e.g. [9, 10, 11]. An important contribution to the analytical methods was brought out by Murawski and Charchalis, who proposed their simplified method for torsional vibrations calculations which was extensively elaborated in [12]. The last significant encirclement of the systematic analytical approach was brought out in the guidance VDI 2039 [13].

Simulation modelling approach for the calculations of torsional vibrations has been implemented by ESI-Group (formerly: ITI-Software) Dresden as the part of their platform for modelling, simulating and analysing of technical systems, SimulationX [14]. This software platform has been approved by several IACS classification societies by allowing the implementation of SimulationX for ships they classify. The basic implementation of SimulationX for these calculations was described in [15].

SimulationX enables the modelling and simulation of complex multiphysics systems, e.g. mechanical translational or rotational, 2D or 3D systems and to perform virtual tests of these models by evaluating their response. The basic procedure is to assemble the system from the elements already available in SimulationX. Users' self-developed elements in Modelica, the object-oriented programming language for modelling of physical systems, may also be used. The way in which these elements are connected within the model determines whether a variable in the model will be input or output. The number of equations in these elements, describing the variables which may be either input or output values, is to be equal to the number of variables. It is irrelevant whether these equations are expressed in their implicit or explicit form, algebraic, differential or discrete. This enables the user to perform complicated types of linear or nonlinear analyses.

This paper aims to describe possibilities and advantages, with some possible ambiguities and disadvantages, of implementing the TVA module of SimulationX simulation modelling-based software in the procedure of torsional vibration calculations. Some essential details about the

modelling of damping and excitation and the preselection of the actual calculation model have already been published [16, 17]. Implementing simulation modelling to the problem of finding steady-state response of marine main propulsion shafting torsional vibrations compared with their solution in a closed analytical form (although the analytical solution is usually complicated and demanding) shows certain advantages, especially when any element in the system, e.g., a highly flexible coupling with a silicone elastic element or a spring-viscous torsional vibration damper, behaves nonlinearly in terms of its rotation-torque response. Simulation modelling software, such as SimulationX, is easy to use in the handling of these situations, because it is not critically important whether the problem is linear or nonlinear, just that the convenient solver has been selected. On the other hand, the disadvantage of SimulationX lies in the fact that its result is always expressed in the form of point values forming the final solution, without functional dependencies of the output values in analytical form. However, this is not a major problem, because the final results in the analytical approach are usually presented in the form of graphs consisting only of points showing the results. SimulationX software comprises its TVA module intended for torsional vibration analysis of rotational mechanical systems in general [14] and this module has been used for the modelling of the actual real systems presented hereafter.

MATERIAL AND METHODS

Torsional vibration simulation modelling starts from the actual propulsion shafting general arrangement drawing (prepared early within the ship machinery systems design phase). Its main objective is to create an equivalent discretised mechanical model composed of point inertias, massless elastic elements (torsional springs), massless damping elements (torsional dampers), and somewhat specific elements of engine cylinders together with the system excitation loads. These excitations comprise gas forces (due to combustion in the engine cylinders), inertia forces (due to accelerations of reciprocating components in the engines), and propeller forces (consisting of their steady-state and vibrational part).

In general, the torsional vibrations of marine propulsion shafting with n rotational inertias can be described by the following system of n ordinary differential equations of second order:

$$\mathbf{J}\ddot{\boldsymbol{\varphi}}(t) + \mathbf{C}\dot{\boldsymbol{\varphi}}(t) + \mathbf{K}\boldsymbol{\varphi}(t) = \mathbf{T}(t) \quad (1)$$

where:

\mathbf{J} , \mathbf{C} , \mathbf{K} – matrices of rotational inertias, damping (absolute and relative), stiffness;

$\boldsymbol{\varphi}$, $\dot{\boldsymbol{\varphi}}$, $\ddot{\boldsymbol{\varphi}}$ – vectors of angular displacements, rotational velocities, rotational accelerations;

\mathbf{T} – vector of time dependent excitation torques in the cylinders and on the propeller.

In general, depending on the form of the vector of excitation torques, an analytical solution of the system (Eq. 1) can be practically obtained when the elements of the matrices **C** and **K** are constant. However, if their dependence on, for example, the rotational velocity is nonlinear, the implementation of a simulation modelling approach would be easier than trying to solve the system of differential equations in an analytical closed form.

MATERIAL

The shafting general arrangement drawing, prepared by the shipyard, usually contains all of the information necessary to model shaft parts in terms of their discretised point inertia elements and the massless springs interconnecting these elements. However, the modelling of massless damping elements requires some additional knowledge, as their damping values are not always uniquely defined, but based on results from practice and previous cases. However, there are several references in the literature suggesting proper values to be used, e.g. [18].

Engine licensors generally provide sufficient data needed to model their Diesel engines, such as cylinder bore, stroke, connecting rod length, reciprocating masses, cylinder firing order, torsional elastic models of engine crankshafts, and engine cylinder pressures (or crank tangential forces) vs. crank angle. Propeller manufacturers provide documentation that contains the necessary information about the propeller, such as number of blades, propeller diameter and mean pitch, propeller mass, and its material. These propeller data are necessary to obtain the correct value of the mass moment of inertia of the propeller together with the entrained water vibrating with the propeller, which is used in the calculations.

METHODS

This paper has been focused exclusively on marine propulsion systems with two-stroke diesel engines, due to some important specifics regarding the modelling of absolute damping, as described in [17]. Simulation modelling of the actual system is based on the elements available in the TVA module of the SimulationX software [14]. This software is based on Modelica, the object-oriented language for modelling of physical systems.

The modelling procedure consists of selecting the proper TVA module elements that will faithfully and correctly represent the behaviour of the real system. TVA L-cylinder elements represent engine cylinders and the marine propeller element represents the actual propeller. Shaft elements represent thrust shaft, intermediate shaft and propeller shaft. The next step is entering the input data into each element representing quantities relevant to inertia, elastic properties, damping properties and excitation. After that, model elements are to be properly connected the same way as in the real system. It is essential to transfer the actual position of the

crankshaft to the engine cylinder elements, to connect the injection of fuel to them properly, and to correctly correlate damping elements in the engine cylinders to their relevant inertia. Once the reference quantity, period variable and the compensation parameter have been set, and the required diagrams of the system response variables selected, the model is ready for the analysis of the torsional vibration steady-state response.

Several different SimulationX TVA models have been tested to find the one that is the most suitable for modelling of the actual system. The description of this procedure is too extensive and has already been partially presented in [17]. The two models of cylinder excitation gas forces finally remain: cylinder pressure excitation and crank torque excitation. The former uses actual cylinder pressure values over the entire crank angle ranges (-180 to +180 degrees) and the latter implements the forces acting on the crankpin in the tangential direction due to ignition in the cylinders over the same crank angle range. These two models produce slightly different results in SimulationX.

The reciprocating mass in the engine cylinders can be taken into account by means of either the physical model, or the nonreactive approach by crank angle. The latter model was considered in the calculations presented below.

The most convenient solution algorithm of the four available in SimulationX steady state simulation is the „Linear method with interpolation”. It was shown to produce final results quite quickly over the entire speed range of the engine without compromising quality.

RESULTS

Calculation results consist of the free and forced steady-state torsional vibrations response. Although possible, the analysis in the time domain was not found to be necessary.

The free vibration results can be expressed in the tables of natural frequencies, vibration modes and Campbell diagram. These results give an important insight into at which engine speeds the problems with resonance can be expected.

Far more important are the calculation results for forced vibrations expressed in the form of, for example, torsional stress amplitudes for the particular shafts within the drive system over the shaft speed ranges. These results also include the allowable stresses calculated in accordance with the IACS Unified Requirement M68 [2]. By comparing the actual calculated stresses with the allowable ones, the designer of the system decides whether it will be necessary to introduce barred speed ranges or even to introduce a torsional vibration damper.

CALCULATION EXAMPLE

To present torsional vibration simulation modelling results, the container/reefer vessel of 12,913 DWT with a propulsion shafting system composed of one two-stroke 7-cylinder

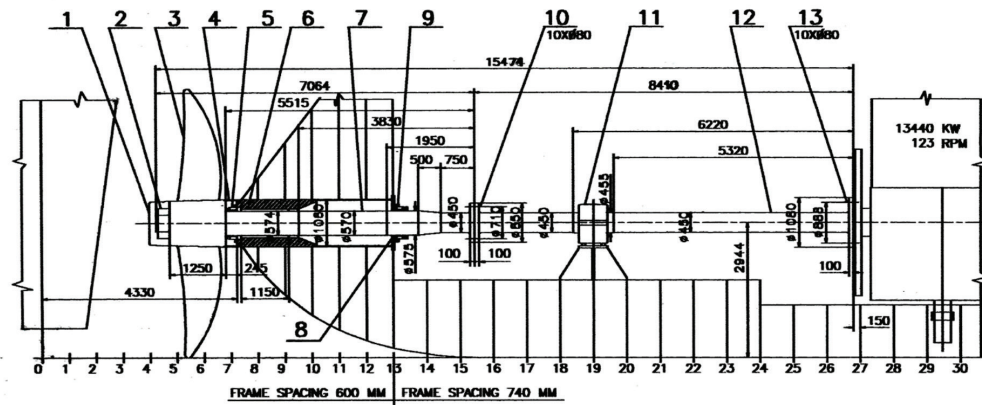


Fig. 1. Container/refer vessel propulsion system shaft line arrangement

slow-speed marine diesel engine with an MCR of 13,440 kW at 123 rpm, the intermediate shaft, the propeller shaft and the solid 4-bladed propeller was chosen as an example. Fig. 1 presents the part of the shaft line arrangement drawing of the ship propulsion system with all the required data, while Fig. 2 shows the SimulationX model for this system prepared for simulation modelling of torsional vibration steady state response.

It should be pointed out that the absolute damping elements (denoted as dynMagn in Fig. 2) are not part of the set of SimulationX TVA elements, so they had to be developed separately as compound elements based on the theory presented in [17].

FREE VIBRATIONS

Calculation results of the free torsional vibrations contain natural frequencies, mode shapes and Campbell diagrams for different modes. Table 1 shows the actual natural frequencies obtained by SimulationX expressed in different ways.

Table 1. SimulationX calculated natural frequencies (eigenvalues) for the free torsional vibrations

SimulationX, ver. 3.8, 7x TVA cylinders model, crank torque model								
natural frequency	mode 1	mode 2	mode 3	mode 4	mode 5	mode 6	mode 7	mode 8
f [Hz]	5.2367	25.055	48.989	70.349	88.807	93.34	104.95	117.57
ω [rad/s]	32.903	157.43	307.81	442.02	557.99	586.47	659.42	738.71
n [rpm]	314.2	1503	2939	4221	5328	5600	6297	7054

FORCED VIBRATIONS

Typically, the torsional vibrational response in the steady-state of the actual system contains the angular displacements of the point inertias in the model, as well as their rotational velocities and accelerations. Once the angular displacements of two points at the ends of a rotational spring element in the system at a given rotational speed have been calculated, the stresses arising from the internal torsional moments acting on that element causing those displacements are easily

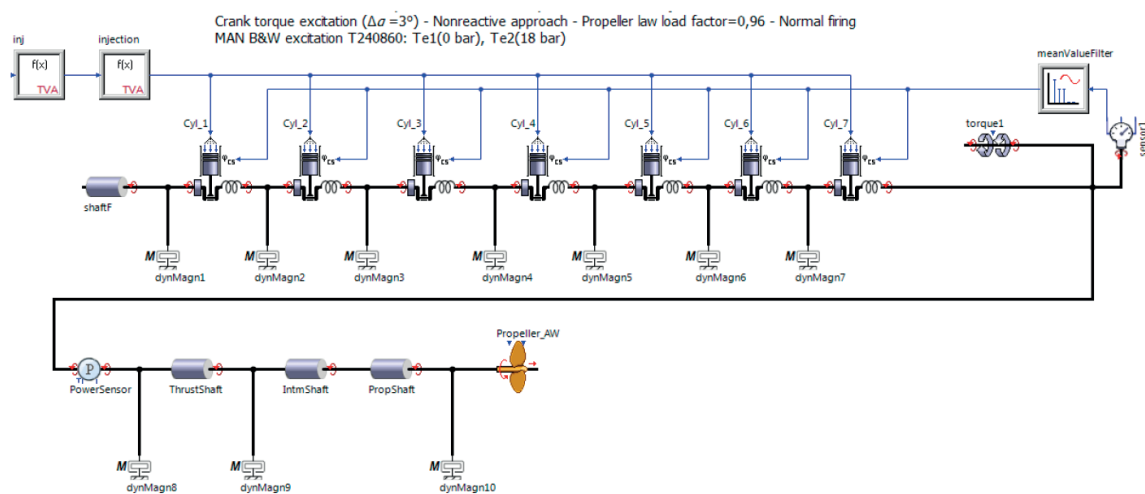


Fig. 2. SimulationX model for the calculation of torsional vibrations of the actual propulsion system

determined.

Fig. 3 shows the amplitudes of the torsional stresses in the elastic massless rotational spring element of the model representing the intermediate shaft.

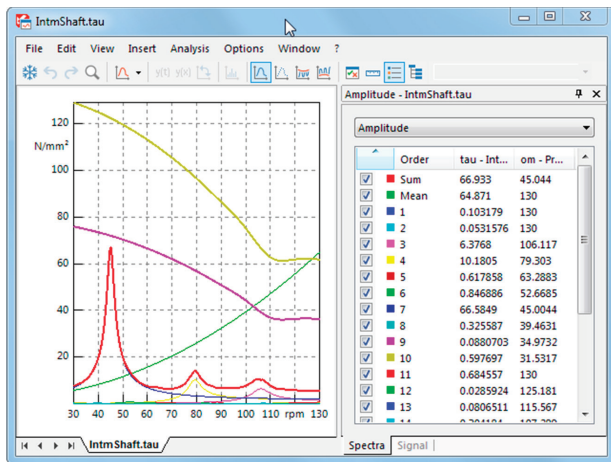


Fig. 3. SimulationX calculated and allowable torsional stress amplitudes in the intermediate shaft

Presented values are obtained by simulation modelling of the steady-state response of the system by SimulationX TVA module, together with the values of the allowable stress amplitudes based upon IACS UR M68 [2] obtained within the same calculation. Fig. 4 shows the same results for the propeller shaft.

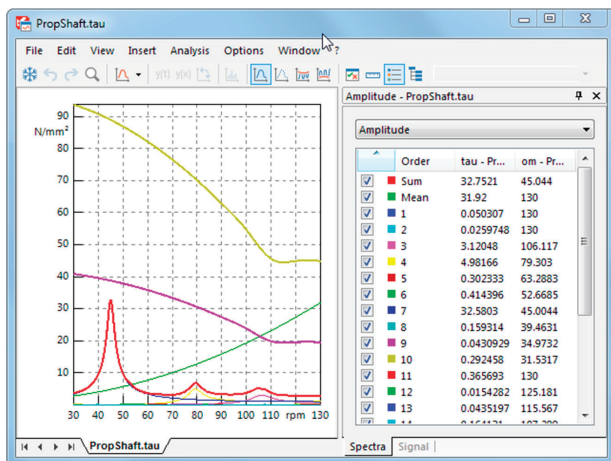


Fig. 4. SimulationX calculated and allowable torsional stress amplitudes in the propeller shaft

DISCUSSION OF THE RESULTS

The basic idea of this paper is to verify the calculation results obtained by the TVA module of SimulationX by means of the two selected well-known torsional vibration calculation programs based on conventional analytical approach and proven in practice. These programs are GTorsi, developed by MAN Energy Solutions from Copenhagen [19], and TorViC,

developed by CADEA from Split [8]. The former has been in practical use for a long time and implemented in many calculations of two-stroke propulsion systems. The latter is an excellent, reliable and user-friendly commercial program purchased long ago by the Croatian Register of Shipping (CRS), having also been implemented for a very long time within the CRS for the approval of marine propulsion and auxiliary systems torsional vibrations calculations.

Thus, if the simulation modelling by the SimulationX TVA module provides results that successfully pass verification by both GTorsi and TorViC (for this reason CRS gave the allowance to the research team to use the results obtained by TorViC within the CRS plan approvals), as well as the validation against measurement results on board, as normally required by classification rules, then the use of SimulationX TVA module in the plan approval of torsional vibrations calculations will be justified.

VERIFICATION OF THE FREE VIBRATIONS CALCULATION RESULTS

It has been found that the natural frequencies determined by means of SimulationX differ slightly from case to case, dependent even upon the actually selected model of excitation in the engine cylinders (cylinder pressure vs. crank torque model). This dependency is against the basic physical understanding of natural frequencies themselves, but it is present. On the other hand, when these natural frequencies are calculated by one of the two proven-in-practice programs (GTorsi and TorViC), the natural frequencies correctly remain the same from one calculation case to another and independent of excitation, which is correct. This finding would be a drawback of the implementation of SimulationX, though the differences found have been rather small.

Table 1 shows the comparison of the natural frequencies obtained by SimulationX with the results of GTorsi and TorViC. There are no differences between GTorsi and TorViC themselves, which proves that both of them perform a correct calculation of the natural frequencies. From Table 2, it can also be concluded that the differences of SimulationX natural frequencies compared to the two conventional programs are very modest (max. 0.5%), so this justifies the use of SimulationX to calculate the natural frequencies in the example presented here, regardless of its possible drawback as mentioned before.

VERIFICATION OF THE FORCED VIBRATIONS CALCULATION RESULTS

The proper selection of the calculation model for the forced torsional vibration response of the actual system in order to compare the results of SimulationX model with the analytical programs is also essential. For this comparison, the maximal torsional stress amplitudes in the thrust shaft, intermediate shaft (see Fig. 3) and propeller shaft (see Fig. 4) were selected as the main representatives with the results presented in Table 2.

Taking the TorViC results as a reference, with respect to

Table 2. Verification of the calculation results

SimulationX evaluation of TVA Normal firing			Comparison of results vs. TorViC Propeller law torque load factor: default = 0.96								
case	program	system	excitation	Natural freq.			Total torsional stress max. amplitude				
				n1 [rpm]	n2 [rpm]	n3 [rpm]	critical speed [rpm]	thrust shaft [MPa]	intm. shaft [MPa]	prop. shaft [MPa]	note
1	TorVic, v1.11	7x cyl.	tang. press. coeff. LARGE_2S	315.4	1501.7	2939.9	44.9	21.0	74.2	36.3	reference
2	GTORSI, v3.6.4	7x cyl.	tang. press. coeff. 240860	315.4	1501.7	2939.9	44.9	20.4	72.0	35.2	best match
3	SimX, v3.8	7x TVAcyl, cyl. press. nonreact.	excit 240860: p(0 18 bar)	313.9	1500.5	2939.3	44.8	18.5	65.3	31.9	
4	SimX, v3.8	7x TVAcyl, crnk. torq. nonreact.	excit 240860: Te(0 18 bar)	314.2	1503.3	2939.3	45.0	18.9	66.9	32.8	
5	SimX, v3.8	7x TVAcyl, cyl. press. nonreact.	excit LARGE_2S: p(0 18 bar)	313.9	1500.5	2939.3	44.7	18.9	66.7	32.6	
6	SimX, v3.8	7x TVAcyl, crnk. torq. nonreact.	excit LARGE_2S: Te(0 18 bar)	314.2	1503.3	2939.4	45.0	19.4	68.5	33.5	2 nd match
Relative values [%]											
case	program	system	excitation	n1	n2	n3	critical speed	thrust shaft	intm. shaft	prop. shaft	
1	TorVic, v1.11	7x cyl.	tang. press. coeff. LARGE_2S	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	reference
2	GTORSI, v3.6.4	7x cyl.	tang. press. coeff. 240860	0.0%	0.0%	0.0%	0.0%	-2.9%	-2.9%	-2.9%	best match
3	SimX, v3.8	7x TVAcyl, cyl. press. nonreact.	excit 240860: p(0 18 bar)	-0.5%	-0.1%	0.0%	-0.3%	-11.9%	-12.1%	-12.0%	
4	SimX, v3.8	7x TVAcyl, crnk. torq. nonreact.	excit 240860: Te(0 18 bar)	-0.4%	0.1%	0.0%	0.3%	-9.8%	-9.8%	-9.8%	
5	SimX, v3.8	7x TVAcyl, cyl. press. nonreact.	excit LARGE_2S: p(0 18 bar)	-0.5%	-0.1%	0.0%	-0.4%	-10.0%	-10.1%	-10.1%	
6	SimX, v3.8	7x TVAcyl, crnk. torq. nonreact.	excit LARGE_2S: Te(0 18 bar)	-0.4%	0.1%	0.0%	0.3%	-7.8%	-7.7%	-7.7%	2 nd match

the maximal torsional stress amplitudes, the best match is expectedly obtained with GTorsi (max. 2.9% of absolute difference), with the closest (2nd) match yielding the SimulationX crank torque excitation model, with exactly the same excitations as in TorViC. The maximum absolute difference of 7.8% may seem rather high, so an additional comparison was made in the form of comparing the graphs of calculation results over the entire engine speed range.

Fig. 5 shows the amplitudes of the torsional stresses in the thrust shaft, intermediate shaft and propeller shaft over the whole speed range (30 to 130 rpm) obtained by SimulationX compared to the analytically calculated results by TorViC. These results show a very good agreement justifying the implementation of the SimulationX TVA module.

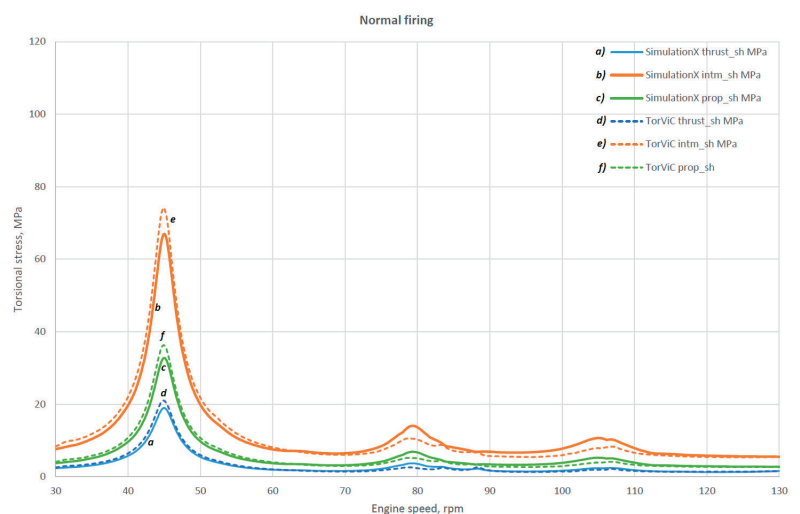


Fig. 5. Verification of the obtained SimulationX calculation results in terms of torsional stress graphs

VALIDATION OF CALCULATED RESULTS

Verification of any simulation modelling results by comparing them to other calculation results obtained by any means does not mean much unless these calculated results are somehow validated by actual practical measurements on a real propulsion system. This means that the only proper confirmation of a certain calculation approach is its validation by measurements of the stresses on the actual propulsion system on-board the ship itself.

Classification rules always prescribe that the calculations of torsional vibrations of ship propulsion shafting systems must be verified not only during the plan approval phase, but also validated by the measurements on-board the first ship in the series of identical new ships (so-called sister ships).

Validation results from the measurements actually performed on the first ship in the series of the two sister ships in the analysed example have been taken from [20]. These results were obtained by strain gauge measurements on the reachable surface of the intermediate shaft.

The measurement results in terms of torsional stress amplitudes in MPa over the actual engine operating speed range (30 to 110 rpm) are presented by circles in Fig. 6 together with the lines representing the results calculated by SimulationX, TorViC and GTorsi. It is obvious that this comparison of the calculated vs. measured torsional stress amplitudes shows excellent agreement, which further justifies the implementation of SimulationX for the presented calculations.

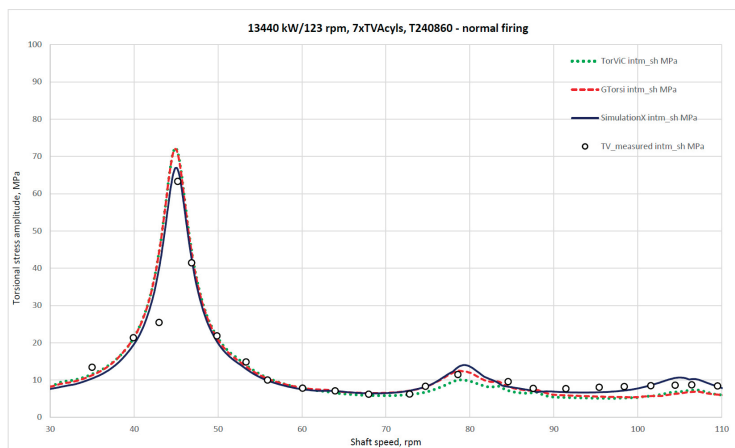


Fig. 6. Validation of SimulationX calculation results vs. engine speed compared with the measurements on-board ship

CONCLUSIONS

This research was initiated with the aim of judging whether the TVA module of simulation modelling based software SimulationX can be correctly applicable within a classification society plan approval process of the torsional vibrations calculations for marine propulsion shafting

systems and under what conditions. The basic idea of the paper is to present the methodology and results obtained by SimulationX software TVA module to the calculation of torsional vibrations steady-state responses. In addition, the SimulationX results have been verified by means of the calculation results obtained using the two computer programs already proven in practice for this purpose. These have been in use for a long time and validated by the available results of shipboard measurements.

The research is focused on the propulsion systems with the two-stroke slow-speed marine diesel engines with the fixed pitch propellers, which are common in bulk carriers and tankers. The important fact is that the stated problem can be expressed in an analytical form of the system of ordinary linear differential equations and solved analytically in closed form. However, these calculations are extensive, especially with respect to the input data preparation and require dedicated technical specialists in order to implement such programs in a proper way. On the other hand, the TVA module of SimulationX does not require deep skills and extensive training to be used, especially when the software is already available in the company, such as a classification society, for different other purposes. There are some important items to be taken care of, so the implementation of SimulationX had to be properly verified and validated. This was the reason and the background for selecting the proper comparable systems to be analysed by SimulationX, for which the actual validation measurements have also been already performed in the shipyard on board the newly built vessels.

The simulation modelling computer program selected for this task is SimulationX, developed by ESI Group, Dresden. It is based on Modelica, a unified object-oriented language for modelling of different physical systems. Important issues related to the selection of appropriate modelling elements have been described. The discretisation of the real system into „lumped masses” and „massless shafts” was briefly explained. It was practically impossible to obtain correct results by SimulationX in terms of stress amplitudes in the shafts, because the absolute damping element as available in SimulationX does not provide proper outcome. For the correct modelling of damping based upon the dynamic magnification values, a special self-developed element relating the damping dynamic magnification factor to a particular mass had to be developed and implemented in SimulationX calculations, with all the details that have already been presented in [17].

Furthermore, it was also important to present modelling of the excitation forces, originating from the gas and inertia forces in the engine cylinders. Modelling of the loading caused by the propeller also had to be considered.

The selection of the actual model from the various models available in SimulationX was not presented in detail, but only the finally selected model, to avoid too extensive a presentation.

Calculation results were presented using a real example of a two-stroke, 7-cylinder, slow-speed marine diesel engine

propulsion system, in terms of free and forced vibrations basic calculation results. These results were then compared for verification with the results of two world-renowned calculation programs based on an analytical approach that has long been proven in practice.

Free vibrations are expressed by the calculated natural frequencies (eigenvalues), where the SimulationX results show excellent agreement with the analytical results. It should be noted that the eigenvalues obtained by SimulationX are different for various excitations implemented in selected model, contrary to the definition and physical meaning of natural frequencies. However, they follow those of the two analytical programs very well.

Forced vibrations are presented in terms of maximal amplitudes of the steady-state torsional stresses in the thrust shaft, intermediate shaft and propeller shaft of the actual system and also show very good agreement with the values obtained by the two analytical programs.

Finally, the validation of the calculation results by comparing them with the measured results, expressed in the stresses in the intermediate shaft over the achievable engine speed range finally justifies that SimulationX proves to be a suitable tool for modelling of steady-state torsional vibrations.

However, in the case of the misfiring of one of the engine cylinders (which is also a common case to be considered in these calculations), the implementation of SimulationX requires special attention and a somewhat different approach than one might expect. This will be a matter for future work of the research group, together with the extensive analyses of four-stroke engine-based systems with all of the possible causes that will force the use of nonlinear elements within SimulationX models.

ACKNOWLEDGEMENTS

Part of the research for this article was performed using the equipment and software obtained within the scope of the scientific research project named: "Functional integration of the University of Split, PMF-ST, PFST and KTF-ST through development of scientific and research infrastructure in the building of the three faculties", contract number KK.01.1.1.02.00182.2.

The authors would like to express sincere thanks to the Croatian Register of Shipping (CRS) for the given allowance to use the calculation verification results obtained within the CRS plan approval process of the newbuilding in the presented example by means of the CRS owned program TorViC, developed by CADEA, Split.

The authors also want to thank DIV Brodosplit, Split for providing the results of validation measurements obtained during the sea-trials of the newbuilding in the presented example.

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