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Capabilities of Transient Simulations in the Torsional Vibration Damper Selection

11 - Basic Research & Advanced Engineering - Technologies, Materials & Tools for Future Engines

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ABSTRACT

In many modern drivelines with combustion engines the protection against torsional vibrations (TV) by using a suitable damper is necessary. For the design of an optimal damper the calculation of vibratory loads must be carried out by using appropriate simulation and TV analysis techniques. Over the last decades, the torsional vibration analysis has been typically performed in the frequency domain with well-established models and software packages. These steady-state calculations have been proven to be an invaluable tool for protecting the components of a propulsion system against overload by torsional vibrations.

However, beyond the steady-state analysis, there is a constantly growing demand for time-dependent simulation techniques. For further optimizations a deeper understanding of a system's response to different kinds of short-term torsional loads is necessary. Such methods are used for example to optimize the driveline components when quickly passing through natural frequencies or the influence of a generator short circuit. Software tools for the steady-state analysis are very limited in their ability to assist in calculating the vibratory loads for such time-dependant excitations.

Under these aspects this paper illustrates how transient simulations of torsional vibrations in propulsion systems can be beneficial to the design and dimensioning of TV dampers.

Other very challenging tasks are ice class calculations according to the rules for designing ships when navigating them in ice covered areas. In this case a large amount of calculations and the computation of a worst-case response in a propulsion system is required.

The focus of this paper is to illustrate the influence of TV damper characteristics on the torsional response of the driveline under the excitation of propeller-ice interaction. Case studies of transient ice impact simulations show the benefits of applying steel spring dampers as well as viscous dampers to different ice class vessels. In this paper it is clearly illustrated that the selected size of ice blocks has a huge effect on the design of the TV damper. Nevertheless, it can be shown that for almost every application which can range from low level up to ice-breaker demand, a damper solution can be found.

1 INTRODUCTION

The simulation of torsional vibrations (TV) in modern drivelines with internal combustion engines is an invaluable tool for guaranteeing a safe and efficient operation over a long lifetime of the propulsion system. In many of these applications, TV dampers are the only option that can sufficiently protect the driveline against these torsional vibrations. For a long time, the dimensioning of TV dampers relied solely on simulations, for which it is assumed that the considered system has reached its steady-state. For such frequency-domain TV calculations there are well-established models and software packages available, with which the analysis of torsional loads can be performed effectively for long-term operations, see e.g. [1].

However, beyond those long-term operation modes, there are applications that require an analysis over a short time horizon, where fast changes in boundary conditions (such as abruptly arising excitations) must be covered.

There are numerous problems, where a detailed understanding of a system’s torsional response to a non-steady-state behaviour is very important, such as engine run-ups or generator short circuits. As illustrative examples two specific issues are investigated in this article that show how Geislinger spring and viscous dampers can be optimized by using transient simulations: the *passage of a barred speed range* (Section 2) and *ice impact simulations* (Section 3). For these case studies the focus is on the simulation of direct-coupled propulsion systems with two-stroke engines with bores ≥ 400 mm. They are typically used for driving bulk carriers, tankers or container vessels. A principle layout of the used simulation models can be seen in Figure 1.

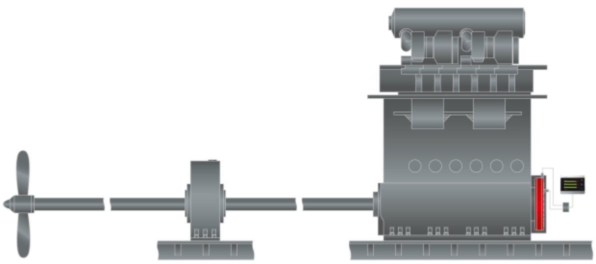

Figure 1. Principle layout of a two-stroke ship propulsion

2 BARRED SPEED RANGE PASSAGES

Typically for vessels equipped with two-stroke engines with 5-, 6- and often 7-cylinders, a strong first order resonance below 60% of the nominal speed is present.

For such propulsion systems, a barred speed range (BSR) is often introduced as a range of operation speed that must be passed in a rather short time. During such passages the critical resonant speed is passed, which causes high loads along the whole propulsion system.

Simulations can provide valuable information about the torsional response when passing such a barred speed range. It is very important to understand the transient torsional response of the system if the barred speed range cannot be passed quickly due to a lack of power margin, which is often the case if the upper bound of the BSR is comparably high with respect to the maximum continuous rating speed. Such long passage times may lead to a significant lifetime reduction of a propulsion system due to high loads on the shaft line.

2.1 Time domain simulation of a barred speed passage

In this part, the simulation of a BSR passage for a propulsion system with a slow speed, 6-cylinder two-stroke engine with a medium bore is investigated.

In Figure 2 the steady-state TV analysis of the vibratory stress on the intermediate shaft is shown for systems with the same engine and shaft line, but with different torsional countermeasures: the largest (engine) tuning wheel and two different spring dampers (D 260 and D 280), where 260 and 280 correspond to the outer diameter (in cm) of the TV dampers.

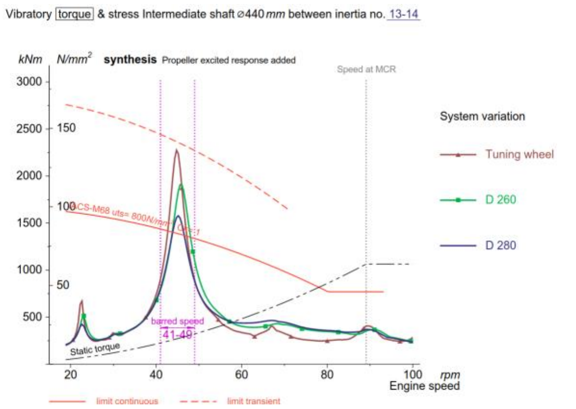

Figure 2. Steady-state TV analysis: Comparison of vibratory torque and stress responses for systems with tuning wheel and 2 different spring dampers

The steady-state TV analysis of the intermediate shaft stress shows a strong first resonance at approximately 44 rpm, such that a BSR from 41 to 49 rpm must be introduced. As some general guidance rules for BSR passage times stipulate a maximum of 5 seconds for a single passage, the red plot in Figure 3 shows the stress response for such an optimally fast BSR passage.

In contrast to the required BSR passage time, measurements show a significantly longer passage time, see e.g. [2, 3]. For this reason, the torsional stress response on the intermediate shaft of the 5-second-passage is compared in Figure 3 with a BSR passage that lasts for 25 seconds. Both simulations are performed with the identical torsional system with the same spring damper D 280 as in Figure 2. Please note that due to the large number of oscillations, the stress response curve is replaced by an area plot between the envelope curves of the corresponding stress response.

Based on the simulation results one can clearly see that the difference in the torsional load on the intermediate shaft is enormous. Hence, there are significantly less cycles of high stress levels for the short-time passage compared to the long BSR passage simulation.

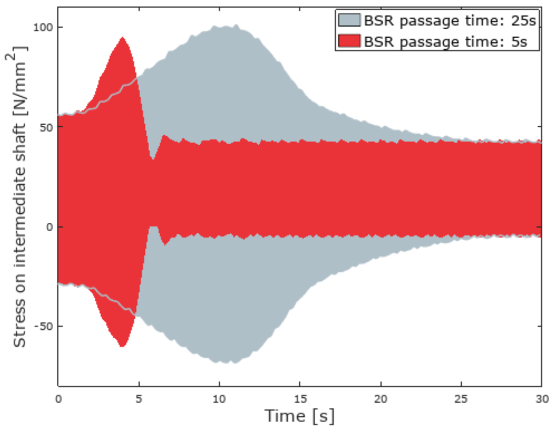

Figure 3. Stress response on the intermediate shaft over time for two different BSR passage times.

These results are essential for a detailed fatigue analysis of the shaft line and they have a substantial influence on the lifetime of the considered propulsion system. There are numerous methods to perform a fatigue analysis based on the above simulation results. An overview of different methods for an appropriate fatigue analysis can be found in [3].

Please note that the possibility of a 5-second-passage is restricted to systems with a comparably small resonance at a low speed range and a sufficiently large power margin at the upper end of the BSR. For systems with large stress amplitudes as e.g. the torsional system with the largest tuning wheel from Figure 2, the passage time in simulations might take much longer, since the occurring torsional loads cause major speed oscillations and consequently a prolongation of the BSR passage.

2.2 Effects of damper selection on the max. permissible barred speed passage time

Based on similar arguments as presented in Section 2.1, DNVGL released in July 2018 an amendment to DNVGL-RU Pt.4 Ch.2, [4], where the max. permissible time for a barred speed passage is given by

$$t_{BSR} = 5\left(\frac{\tau_{vmax}}{\tau_T}\right)^{-7.2} + t_{MR}\ [s],$$

where τ_{vmax} is the (calculated or measured) peak vibratory stress at resonant speed, τ_T is the corresponding IACS UR M68 limit for transient operation and t_{MR} ("*passing time allowance*") is an additional passage time extension of 10 seconds due to design features of considered shafts with stress concentration factors below 1.1 (for example for shafts with multi-radii flange fillets).

For the different systems from Figure 2, the new DNVGL rule restricts the max. permissible time for a BSR passage according to the calculated values t_{BSR} from Table 1 (without considering additional time extensions t_{MR}).

Table 1. Max. permissible time for BSR passages

Torsional countermeasure	Max. permissible BSR passage time t_{BSR}
Tuning wheel	6.3 s
D 260	22.1 s
D 280	87.5 s

As it can be seen from the calculated time limits above, the effects of applying a TV damper are remarkable and a safe passage through a barred speed range can be guaranteed by an appropriate spring or viscous damper. Please note that there is also the possibility of avoiding a BSR by applying specific TV damper solutions, with which there are no further operational restrictions due to torsional vibrations.

3 ICE IMPACT SIMULATIONS

For ice-going vessels, a special ice classification is necessary in order to prove sufficient protection against failures due to additional (torsional) loads caused by rough ice-covered sea conditions. Since real-life ice torque excitations are not considered to last for a long time, the frequency-domain TV analysis of ice-torque excited torsional response has not been widely established in the field of direct-coupled large-bore combustion engines, see e.g. DNVGL-RU Pt.6 Ch.6, [5], for a rather conservative frequency-domain TV analysis approach.

Besides the steady-state TV analysis, national agencies and the majority of classification societies have agreed upon the requirement of transient ice impact simulations in order to receive a corresponding classification. For these computationally demanding simulations, artificial torque excitation patterns consisting of half-sinusoidal curves are introduced to model the ice-torque excitation by propeller ice-interaction.

In literature and in classification rules there are many different ice class designations available. However, after considering major equivalences between the requirements there are basically 4 ice classes based on Finnish ice class rules acc. to the Finnish Transport Safety Agency (denoted by IC, IB, IA and IA+) and 7 polar ice class rules based on IACS Polar class rules (denoted by PC1 – PC7).

In this section the focus is on case studies of ice class projects with spring and viscous dampers. For the two-stroke engine market the smaller ice classes (IC–IA+) represent the majority of ice class projects. Nevertheless, an example of a PC4-classified (medium polar class) vessel with a 7-cylinder two-stroke engine is presented in Section 3.3.

3.1 Peak torque evaluation under worst case ice impact scenario

The reason for the high computational complexity of ice class calculations is due to the requirement of detecting the worst-case scenario of initial engine speed and phase angle between the ice excitation and the main engine. Additionally, the worst-case has to be found with respect to 4 different load cases, which raise the number of ice impact simulations to a large extent.

For initial speed variations, one has to keep in mind that after the ice impact, the simulated speed drops massively and in some cases the engine even stalls, see e.g. [6]. This typically causes a speed drop into or below the BSR and therefore, high torsional loads occur along the propulsion line by accelerating additionally in the main resonant speed range.

In Figure 4, the worst-case ice-torque excitation response (acc. to ice class IC) at the intermediate shaft is considered for a propulsion system equipped with a fixed-pitch propeller and a 6-cylinder two-stroke engine with a small bore. The worst-case scenario is calculated with respect to the phase angle, where the initial speed is taken into consideration in the range from 20 rpm to 117 rpm (nominal speed). For this case study two different types of torsional countermeasures are compared: the largest possible tuning wheel and a spring damper (D 220). Overall, the detection of the worst-case torque response on the intermediate shaft has been performed by 4560 transient ice impact simulations for each system.

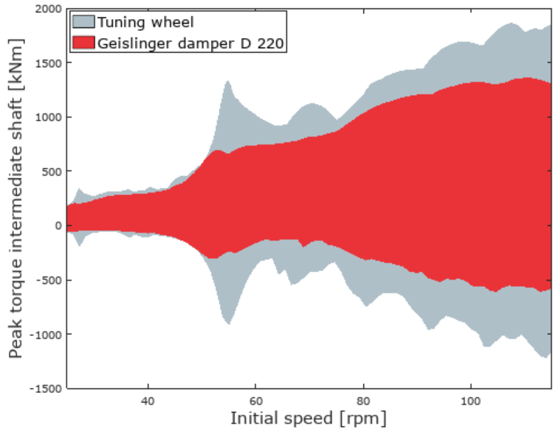

Figure 4. Worst case detection: Peak torque response at the intermediate shaft

Figure 4 illustrates that a TV damper can lower the torsional ice-impact loads effectively. In this particular example a reduction of at least 25% of the peak torque is achieved by applying the spring damper D 220 compared to any tuning wheel.

3.2 Influence of ice block thicknesses on TV damper selection

An essential parameter for the ice torque excitation definition of different ice classes is the ice block thickness H_{ice}, which is different for each ice class.

Increasing the parameter H_{ice} has two effects: firstly, the scaling of the ice-impact torque excitation is dependent on the ice block thickness and secondly, the number of propeller-ice block interactions depends not only on the number of propeller blades but also on the ice block thickness. In Figure 5 the effects of the parameter H_{ice} on the peak torque results on the propeller shaft is investigated.

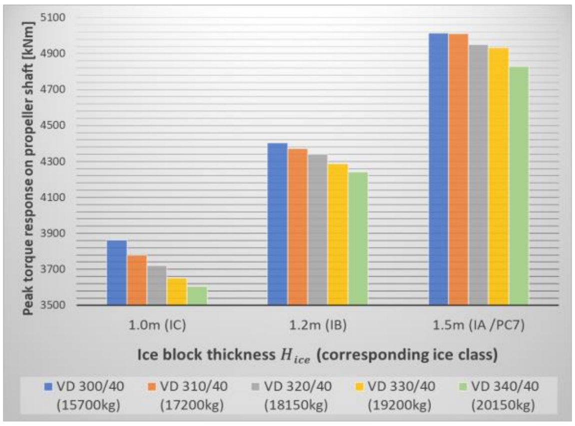

Figure 5. Comparison of peak torque responses on the propeller shaft for different viscous dampers

For this, a variation of a torsional system with a medium-bore, 6-cylinder two-stroke engine and 5 different viscous dampers is investigated. The considered viscous dampers are different with respect to the outer diameter, but they share similar stiffness and damping characteristics.

The results show the large impact of the parameter H_{ice} on the results of the worst-case simulation of the propeller shaft torque. Within each ice class, a reduction of the peak torque response could be achieved by applying a larger viscous damper in this specific example. However, one has to note that the possibility of arbitrarily lowering the effects of ice-induced torque loads is not given, since there are limitations on installation space and weight of the TV damper.

An advantage of an ice class damper solution with a viscous damper is that one does not have to consider a specific transient torque limit of the damper in the corresponding ice class calculations. On the other hand, viscous dampers are very likely to be significantly larger than steel spring dampers and therefore by the above-mentioned limitations, it is not always possible to provide a solution, where all ice class requirements are fulfilled.

3.3 Optimization of TV damper characteristics

Since every spring damper has a tailor-made design with respect to the specific demands of the considered propulsion system, it is possible to find ice class damper solutions for all kinds of ice classes - ranging from small ice-classed vessels up to ice breakers with medium or high polar classes.

The following example shows a parameter study of TV damper characteristics for a PC4-classified vessel with a 7-cylinder, medium-bore two-stroke engine. Originally, the system is considered with a spring damper D 340 with the initial stiffness of 44.5 MNm/rad and a relative damping of 500 kNms/rad. Based on this initial design, a variation of its stiffness (approx. ±8%) and of its relative damping coefficient (approx. ±12%) is considered.

The parameter study reveals that the simulated torsional responses along the different components respond in a different way to the stiffness-damping-variation of the TV damper. Whereas on some components the worst-case ice-torque excitation behaves almost linear (e.g. for the propeller shaft in Figure 6 and the damper in Figure 8), other components show a nonlinear behavior with respect to a variation of the damper characteristics, see e.g. Figure 7 for the peak torque on cylinder 1.

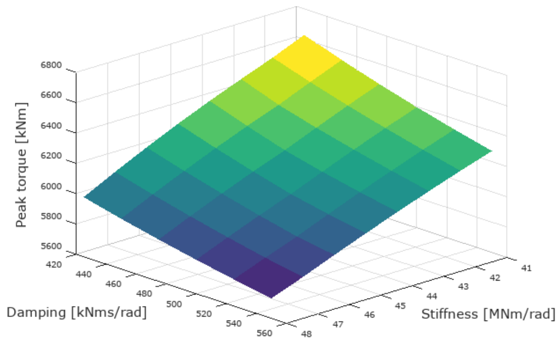

Figure 6. Peak torque response at propeller shaft

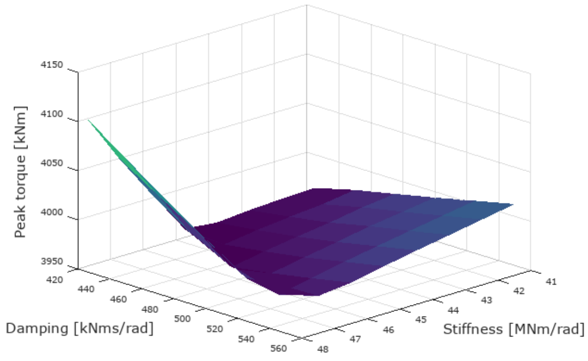

Figure 7. Peak torque response at cylinder 1

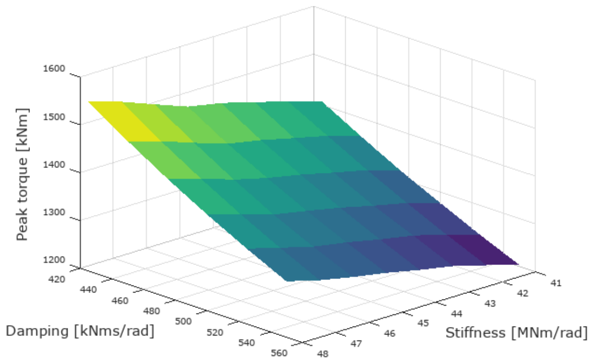

Figure 8. Peak torque response at damper D 340

In addition, the variation of one of the parameters lead on the one hand to a reduction of torsional loads in certain propulsion parts, whereas the peak torque is increased at other system components.

Hence for finding an optimal solution, the challenge is to equilibrate all those effects to find the best possible damper design for each application. Moreover, by such dimensioning methods, solutions can be provided with a spring damper, where none of the other torsional countermeasures, such as tuning wheels, are sufficient.

4 CONCLUSIONS

This article contains investigations on how transient simulations can help to find an optimal TV damper solution if short-time torsional load peaks on the driveline are present. For various aspects of (transient) BSR passage and ice impact simulations it is analysed how a time-domain TV analysis is beneficial for dimensioning spring- and viscous-type dampers.

More specifically, the effectiveness of transient BSR passage simulations for the selection of a proper torsional countermeasure is illustrated by the comparison of different TV countermeasures. The benefits of applying a TV damper lead to a reduction of torsional loads per BSR passage and therefore to an expected extension of the lifetime of a propulsion system. Simulations as presented in Section 2.1 can effectively assist in the selection process of an optimal damper solution.

Furthermore, some TV-related topics concerning ice impact simulations are discussed. First, the worst-case peak torque evaluation is highlighted by an example, where the peak torque response on the intermediate shaft could be significantly reduced by applying a steel spring TV damper. Afterwards, the effects of the ice block thickness H_{ice} on the torsional response for systems equipped with viscous TV dampers are discussed. It is shown that the variation of parameter H_{ice} has a major influence on the torsional response and therefore also for the optimal TV damper layout. Finally, a brief insight into the TV damper optimization process based on ice impact simulations is provided by a TV damper parameter study for a PC4-classified vessel.

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