[1] IMO. Imo train the trainer (ttt) course on energy efficient ship operation.
[2] Dahler, Iseskär, Holm. Fuel efficient propeller design and torsional vibrations in propulsion machinery. TVS, 2014
[3] Wilson, W.K. Practical solution of torsional vibration problems, Vol 1. 1956

Dynamic Analysis and Vibration Reduction in Slow Rotating Propulsion System

NTNU
DNV·GL

Master Thesis Specifics

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- Field of specialization: Marine Engineering
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- The current deadline of this master thesis is the 16th of July. This poster presents the work and results obtained so far

Objectives

1) Create a 'good' mathematical formulation of a slow rotating propulsion system onboard a state-of-the-art eco-ship
2) Obtain a valid computer software model of the same system. Verification through comparison to real measurements from DNV GL
3) Perform dynamic analysis of the selected subject-propulsion system to identify critical components and external excitations
4) Recommend torsional vibration reduction measures for slow rotating propulsion systems based on the obtained results from point 3)

Motivation

Focusing on the impact from shipping towards the environment have led to stricter emission regulations in recent years. Related to marine activities, one example is the Energy Efficiency Design Index (EEDI) launched by IMO in 2012. The EEDI is defined as environmental impact in terms of carbon dioxide emissions, divided by the transportation work.

To meet the regulatory limit of this index, and to save and reduce fuel consumption, ship owners tend to install smaller main engines with less power [1]. The result has been a new type of vessels called 'eco-ships', which are operated by a slow rotating direct-drive 2-stroke engine.

Less available power lead to poorer manoeuvrability in adverse conditions, but also larger torsional vibrations due to longer operation within the engine's range of critical speeds. This operation window, called the barred speed range (BSR), is associated with resonance. There is consequently a trade-off between reduced emissions and the life time of a vessel drive train on an eco-ship. Figure 1 illustrates potential hazards related to this issue, in this case fracture of the propulsion shafting.

Figure 1: Torsional vibration damage of propulsion shafting

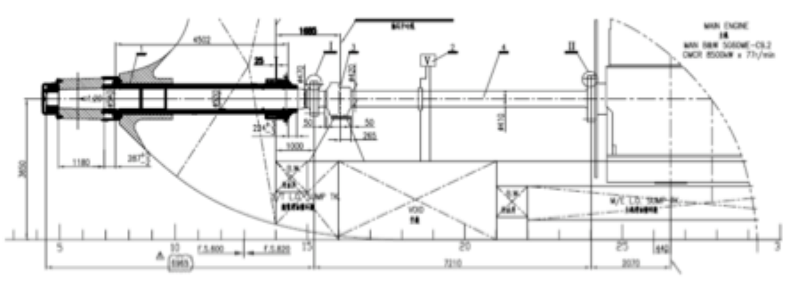

Figure 2: System layout of case-study propulsion drive train

Case Study Propulsion System

- 5-bladed fixed pitch propeller (full pitch)
- 2-stroke slow rotating 5-cylinder MAN & Turbo diesel engine ('Green' ultra long stroke-to-bore ratio)
- MCR condition: 77rpm and 8500 kW engine power
- Barred speed range: 40-51 rpm
- Direct drive (no gears)
- Torsional vibration damper (Geislinger)
- Mass elastic data is provided by DNV GL

Methodology

The lumped mass modelling approach (Objective 1)

- The rotating components of the drive train are modelled as rotating discs/inertias with one degree of freedom (angular displacement about the shaft, θ). See example in Figure 3.
- Shaft segments are modelled as mass less springs and dampers
- The case-study propulsion system will be modelled as a torsional lumped mass system in both of the applied software

Introducing Damping - The dynamic magnifier model

- Divides damping into internal/relative/shaft damping and external/absolute/mass damping.
- Introduces the dimension less parameter *M*, called the 'magnifier number'. Damping expressed in terms of percentage of critical damping [2]

$$M_i = \frac{100}{2 \cdot \zeta_i} \quad [\%]$$

- Alternatively introduce damping in the unit of Nms/rad as a traditional damping factor *C*. A function of the magnifier number *M* [2]

$$C_i = \frac{J \cdot \omega_n}{M_i} \quad [Nms/rad]$$

- The most prevalent first mode of natural frequency (ω_{n1}) is used as basis for calculation a constant damping factor

Computer software (Objective 2)

(Created models are presented in Figure 4)

1) 'Nauticus Machinery – Torsional Vibrations'
- A computer software from DNV GL, to perform torsional vibration calculations of shafting systems
- Damping introduced through the magnifier number *M*
- Solves the equations of motion based on analytical algorithms (not time domain)

2) 'Simpack Multi Body Simulation Software'
- Damping introduced as damping factors C_i
- Time domain simulation, can therefore evaluate e.g. fatigue life
- Co-simulation with MATLAB – Simulink to apply control algorithms for controlling angular velocity and torque
- Utilized a traditional proportional-integral-derivative (PID) controller

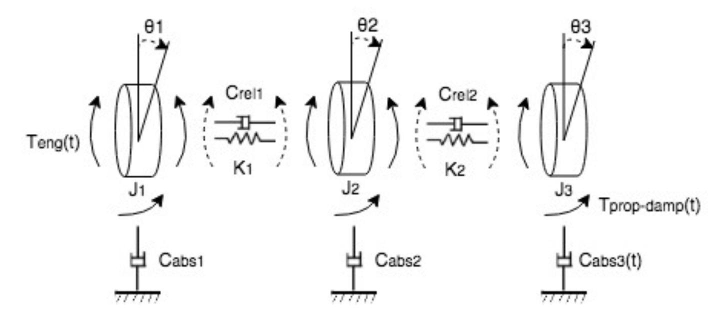

Figure 3: Free body diagram of torsional lumped mass system of a simplified vessel drive train

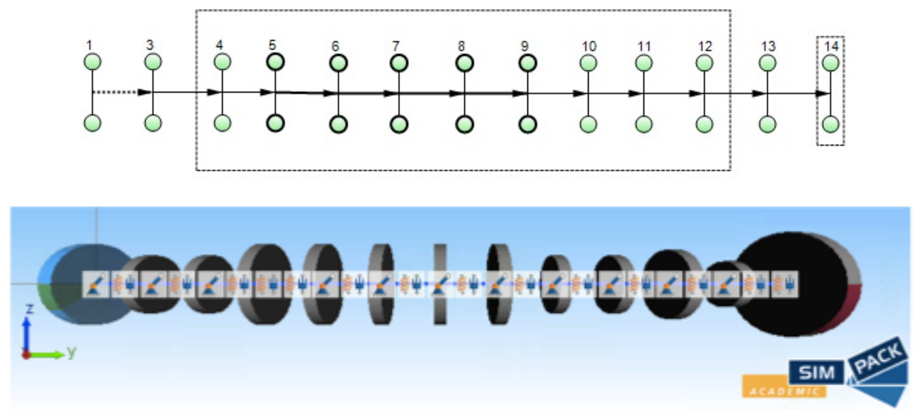

Figure 4: Software models of the propulsion system in Nauticus Machinery (upper) and Simpack (lower)

Dynamic Analysis – Procedure

(Objective 3)

1) Free vibrations
- Set up equations of motion (eom) for damped free vibrations
- Solve eom and determine system characteristics such as natural frequencies, mode shapes, critical damping and thereby damping ratio ζ

2) Forced vibrations
- Apply harmonic external excitation from the diesel engine and set up forced eom

$$J\ddot{\theta} + C\dot{\theta} + K\theta \quad = \quad T_0 \; sin(\omega t)$$

- Determine critical speeds and identify dangerous excitation frequencies through Campbell diagram

Vibration Reduction – Procedure

(Objective 4)

- Sensitivity analysis: Change multiple system parameters (e.g. stiffness, damping, introduce additional torsional vibration dampers) and evaluate the effects (in Nauticus Machinery)
- Introduce magnetic bearing and control stiffness and damping continuously (in Simpack MBS Software)

Results *(To be continued)*

Free vibration analysis

- Natural frequencies of the propulsion system have been found in both Nauticus Machinery and Simpack. They have further been verified through comparison to a study performed by DNV GL. The first three modes are presented in Table 1.
- Mode shape analysis of 3rd mode oscillations reveals that the crank throw between the 3rd and 4th cylinder (mass ID 7 and 8) is critical. See Figure 5

Forced vibration analysis

- Critical speeds are retrieved as the intersection points between the blue lines (engine excitation) and the horizontal natural frequencies in Figure 6. The engine's BSR is in between the two red lines.
- Engine excitation is modelled as multiples of the engine speed n_e [3]:

$$f_e = \frac{n_e}{60}, \frac{2n_e}{60}, \frac{3n_e}{60}, \frac{4n_e}{60}, \dots$$

- The Campbell diagram (Figure 6) verifies that the BSR indeed is a critical speed
- The BSR of the engine will be verified by modal energy analysis later on

Vibration reduction

1) Sensitivity analysis (Nauticus Machinery)
- Increasing stiffness with 10% and damping through parameter *M* with 100% has relatively little effect (max. 10% decrease) on the drive train's torsional stress levels.
- Applying a second torsional vibration damper in between the engine and propeller reveals on average a 45% decrease of torsional stress in the shaft segments.

2) Magnetic bearings
- Applying and testing the effects of magnetic bearings have not yet been approached.

Table 1: First three modes of natural frequencies

Mode no.	Cyclic frequency [Hz]
1	3.75
2	5.04
3	26.13

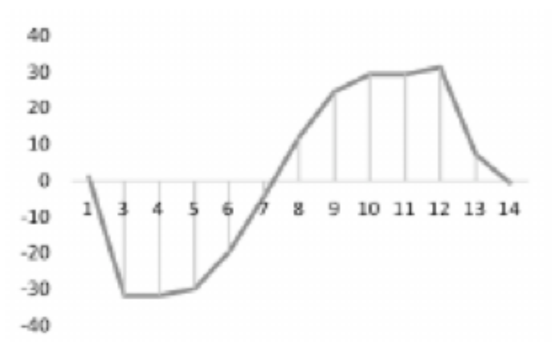

Figure 5: 3rd mode shape (results from Nauticus Machinery)

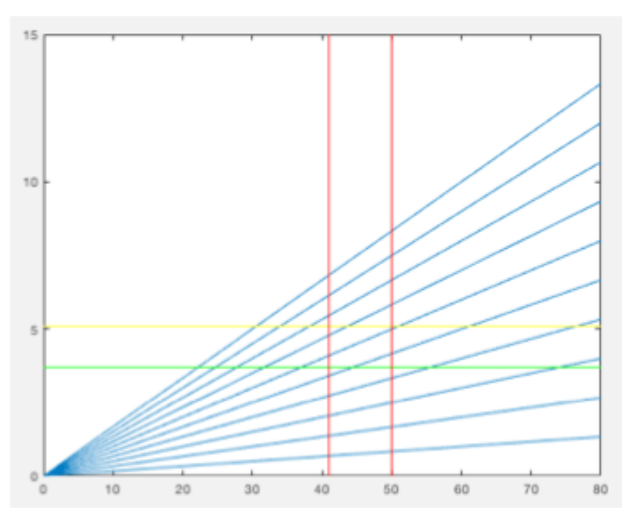

Figure 6: Campbell diagram of engine excitations (engine rpm on x-axis and the system's frequency in hertz on y-axis)