DET NORSKE VERITAS UND GERMANISCHER LLOYD AS RULES FOR CLASSIFICATION SHIP July 2016

Part 4 Chapter 2 Section 4

ALIGNMENT

1 General

1.1 Application

1.1.1 Scope

The shaft alignment rules are only applicable for propulsion plants. For geared plants, the calculations are only applicable for the low speed shaft line, which shall include the output gear shaft with radial bearings.

Vertical shaft alignment is always applicable, while horizontal alignment is applicable upon request.

1.1.2 Calculation versus specification

Propulsion plants as described in [1.3.2] require shaft alignment calculation.

All other plants need a shaft alignment specification only, see [1.3.3].

1.1.3 Aft most bearing

Acceptance criteria and modeling of aft most bearing are dependent of risk:

— White metal lined aft stern tube bearing which is either double sloped, or has a journal diameter 500 mm or greater, shall fulfill bearing lubrication requirement see [2.1.6].

— Other propulsion plants where alignment calculation is required shall fulfill requirements in [2.1.5].

Guidance note:

Aft most bearing is in most cases to be understood as aft stern tube bearing, but can also be other designs e.g. strut mounted bearings which are common in twin screw designs without skegs.

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1.3 Documentation

1.3.2 Systems requiring shaft alignment calculation

Shaft alignment calculation report shall be submitted for approval for propulsion plants with one out of the following criteria:

— minimum shaft diameters (low speed side) of 400 mm or greater for single screw and 300 mm for twin screw

— gear transmissions with more than one pinion driving the output gear wheel, even if there is only one single input shaft as for dual split paths

— shaft generator or electrical motor as an integral part of the low speed shaft in diesel engine propulsion.

Upon request, shaft alignment calculations may also be required for other plants when these are considered sensitive to alignment.

For required content of a shaft alignment calculation report, see [2.1.4].

1.3.3 Systems only requiring shaft alignment specification

For all propulsion plants other than those listed in [1.3.1], a shaft alignment specification shall be submitted for information. The shaft alignment specification shall include the following items:

— bearing offsets from the defined reference line

— bearing slope relative to the defined reference line if different from zero

— Installation procedure and verification data with tolerances e.g. gap and sag and jacking loads (including jack correction factors and jack positions) and verification conditions (cold or hot, propeller submersion, etc.).

2 Calculation

2.1 General

2.1.1 Calculation input data

The shaft alignment calculations shall at minimum include the following input data:

— propulsion plant particulars, e.g. rated power of main engine and propeller shaft rpm

— equipment list, i.e. manufacturer and type designation of prime mover, reduction gear (if applicable) and bearings

— geometry data of shafts, couplings and bearings, including reference to relevant drawings. For direct coupled plants, the crankshaft model shall be according to the engine designer's guidelines

— propeller data

— bearing clearances.

2.1.2 Alignment conditions

The shaft alignment calculations shall include the following conditions:
— alignment condition (during erection of shafting)
— cold, static, afloat, fully submerged propeller
— hot, static, afloat, fully submerged propeller
— hot, running with hydrodynamic propeller loads.
For geared shafting systems:
— running conditions as required to verify gear acceptance criteria
— all relevant combinations of prime mover operation
— Horizontal alignment is upon request.

2.1.3 Influence parameters

The shaft alignment calculations shall take into account the influence of:
— buoyancy of propeller
— thermal rise of machinery components (including rise caused by heated tanks in double bottom and other possible heat sources)
— gear loads (horizontal and vertical forces and bending moments)
— angular working position in gear bearings for gears sensitive to alignment, see guidance note 1
— bearing wear (for bearings with high wear acceptance e.g. bearings with water or grease lubrication)
— bearing stiffness (if substantiated by knowledge or evaluation, otherwise infinite)
— hull and structure deflections, see guidance note 2
— hydrodynamic propeller loads, see guidance note 3.

Guidance note 1:
For sensitive geared systems (e.g. gears with large face width or gears with more than one pinion driving the output wheel) even small alignment offsets may have large influence on the gear face load distribution. In such systems, angular position of the shaft has to be found by iteration. Vertical and horizontal offsets may be assessed by means of the vertical and horizontal forces in the previous iteration step. Bearing clearances have to be taken into account, but the oil film thickness can usually be disregarded (except for very light bearing loads). For fluid film bearings the angular working position may be estimated to 20 to 30° off the direction of the force (except for very light bearing loads).

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Guidance note 2:
Hull deflection is dependent of design, draught, trim, aft peak tank filling etc. Estimated deflections can be based on FEM calculations, experience from similar designs etc. Larger safety margins should be applied when these deflections are unknown and are expected to have influence on the alignment.

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Guidance note 3:
Rule paragraph [1.1.3] defines scope of aft most tail shaft bearing analysis dependent of risk. For propulsion plants where the aft most bearing lubrication criteria are required; see [2.1.6]. For other plants, the hydrodynamic loads can be applied by either verified measurements on similar designs or a default bending moment. The default bending moment should not be less than $0.05 \cdot T_0$ downward and $0.40 \cdot T_0$ upward, where T_0 is propeller torque at mcr. For twin screw plants a range of $\pm 0.3 \cdot T_0$ horizontally and $\pm 0.2 \cdot T_0$ vertical should be used.

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2.1.4 Results

The shaft alignment calculations shall at minimum include the following results:
— bearing offsets from the defined reference line
— calculated bearing reaction loads and pressures
— bearing reaction influence numbers
— graphical and tabular presentation of the shaft deflections with respect to the defined reference line
— graphical and tabular presentation of the shaft bending stresses as a result of the alignment
— nominal relative slope between shaft and bearing centrelines in aft most propeller shaft bearing (see [2.1.5]) and if applicable, details of proposed slope-bore — results from aft stern tube bearing lubrication criteria, see [2.1.6]
— a shaft alignment procedure with verification method and data with tolerances (e.g. aft bearing slope and geometry, reference line, stern tube bearing offsets, calculated gap & sag values and jacking loads including jack correction factors). The procedure shall clearly state at which vessel condition the alignment verification shall be carried out (cold or hot, submersion of propeller etc.). Positions of jacks and temporary supports have to be specified. The procedure shall be possible to use again when in service.

2.1.5 Acceptance criteria

The shaft alignment has to fulfill the following acceptance criteria for all relevant operating conditions in [2.1.2]:
— acceptance criteria defined by manufacturer of the prime mover, e.g. limits for bearing loads, bending moment and shear force at flange
— acceptance criteria defined by the manufacturer of the reduction gear, e.g. limits for output shaft bearing loads and load distribution between bearings
— bearing load limits as defined by bearing manufacturer and Ch.4 Sec.1
— zero or very low bearing loads are only acceptable if these have no adverse influence on whirling vibration — tolerances for gap and sag less than 5/100 mm are not accepted.
Acceptance criteria for aft most tail shaft bearing:

- in hot static and hot running conditions the relative nominal slope between shaft and aft most propeller

2.1.6 Aft most bearing lubrication criteria

A white metal lined aft stern tube bearing which is either double sloped, or has a journal diameter 500 mm or greater, shall be designed to ensure hydrodynamic lubrication in all operational conditions. The minimum speed giving hydrodynamic lubrication (n_0), has to be lower than the actual shaft speed (n). Both low speed and full speed criteria have to be fulfilled, see guidance note 1.
For multi slope bearings the method applies to the bearing segment with highest nominal bearing pressure for each operational condition.

Low speed criterion:

The minimum shaft speed ensuring hydrodynamic lubrication ($n_{0,stat}$) is calculated for:

— Hot static condition: No hydrodynamic propeller loads, $n_{0,stat}$

$n_{min} \geq n_{0,stat}$

Full speed criterion:

The minimum shaft speed ensuring hydrodynamic lubrication ($n_{0,dyn}$) is calculated for the following conditions defined by the vertical hydrodynamic bending moment acting on the propeller, see guidance note 2 below:

— Hot running condition 1: 15% of full torque downwards, $n_{0,dyn1}$

— Hot running condition 2: 40% of full torque upwards, $n_{0,dyn2}$

$n_{full} \geq \max\{n_{0,dyn1}, n_{0,dyn2}\}$

The hydrodynamic propeller loads are defined as vertical bending moments as percentage of full speed torque, see guidance note 2.

— Calculation to be used for both criteria:

shaft bearing should not exceed $3 \cdot 10^{-4}$ rad (0.3 mm/m) and 50% of min. diametrical bearing clearance divided by the bearing length, whichever is less. For definition of relative nominal slope, see Figure 1. This criterion is only applicable for single slope or no-slope bearings.
A white metal lined aft stern tube bearing which is either double sloped, or has a journal diameter 500 mm or greater, shall fulfil requirements regarding hydrodynamic lubrication performance as stated in [2.1.6].

$$n_0 = \frac{28 \cdot 10^3\, C\, h_0\, p_{eff}}{\nu\, D\, L_{eff}}$$

$$h_0 = \frac{D^{0.43}}{760}$$

$$p_{eff} = \frac{10^6 W}{L_{eff} D}$$

$$L_{eff} = L\, K_D\, K_L \left[\left(0.1 + 0.17 \frac{W_{min}}{W_{max}}\right) - \left(0.32 - 0.02 \frac{W_{min}}{W_{max}}\right) \log(\alpha) \right] \quad , \quad L_{eff} \leq L$$

$$K_D = 0.53 \cdot 10^{-6} D^2 - 1.08 \cdot 10^{-3} D + 1.55$$

$$K_L = 0.33\left(\frac{L}{D}\right)^2 - 1.5\left(\frac{L}{D}\right) + 2.66 \quad , \quad \frac{L}{D} \leq 2$$

Table 3 Calculated parameters

n_0	minimum rotational shaft speed ensuring hydrodynamic lubrication [rpm]
h_0	minimum required lubrication film thickness [mm]
p_{eff}	effective bearing pressure [N/m^2]
L_{eff}	length of locally pressurized area [mm], $L_{eff} \leq L$
K_D	dimensionless size factor [-]
K_L	dimension less length to diameter ratio [-]

Table 4 Dimensions and physical parameters

n_{min}	actual shaft speed for continuous slow speed operation [rpm]
n_{full}	actual max shaft speed for continuous operation [rpm], typical at MCR
C	diametrical bearing clearance [mm]. Use nominal diameter for std. double slope machining in lower part of bearing, and actual diameter for trumpet shaped slope
L	bearing length, or segment length in case of multi slope bearing [mm]
v	the kinematic viscosity at 40°C [cSt] of the lubricant. To be used as minimum viscosity acceptable for the installation
D	bearing journal diameter [mm]

Table 5 Parameters from shaft alignment calculation, see Figure 2 and Figure 3

W	radial bearing load, W1 + W2 [N]
W_{max}	max value of W1 and W2 [N]
W_{min}	min value of W1 and W2 [N]
α	calculated relative slope between shaft and bearing at W_{max}, either α_1 or α_2 [mm/m], see Figure 3

White metal lined stern tube bearings shall be modelled in the shaft alignment calculation as presented in Figure 3. This is achieved by modelling the bearing with a support point at either bearing end (or at either segment end for multi slope bearings). The total bearing stiffness shall not be taken less than $5 \cdot 10^9$ N/m, and stiffness of each individual support point not less than $2 \cdot 10^9$ N/m, unless documented otherwise.

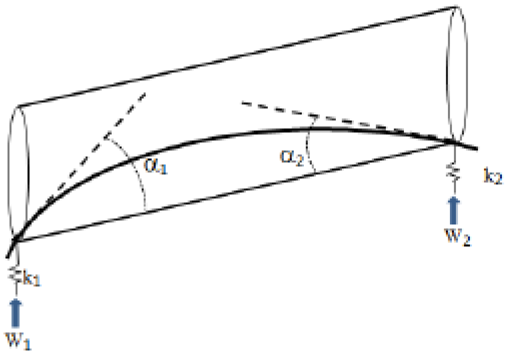

Figure 2 Model of a shaft resting in a single slope or no-slope bearing

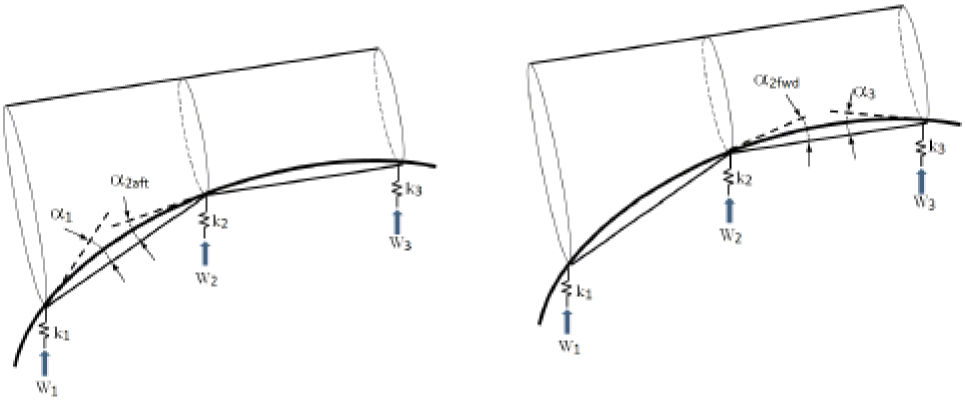

Figure 3 Model of a shaft resting in a double slope bearing

The following results from the calculation have to be presented:

$n_{0,stat}$, $n_{0,dyn1}$, $n_{0,dyn2}$, v

Guidance note 1:

The calculation of minimum speed ensuring hydrodynamic lubrication is based on a quasi-empiric solution of the Reynolds equation for journal bearings. Special conditions typical for stern tube bearings such as uneven load distribution and misalignment are implemented. This method shall ensure lubrication in areas with maximum bearing pressure. The method shall set a limit for the minimum continuous operational shaft speed and the minimum viscosity of the lubricant. Use of oil with high viscosity (above 200 cSt) generate viscous losses and heat, hence care has to be taken. The chosen viscosity (v) is the minimum value to be used as stern tube lube oil. The calculated oil film thickness (h_0) is a parameter to be seen as an integrated element of the calculation method, and shall not be understood as an acceptance of actual oil film thickness. (L/D) < 2 is a limitation in the calculation method, and not a limitation of the actual bearing length.

The centre load in a double slope bearing (see W2 in Figure 3) can be distributed to both bearing segments as there will be an oil film in both segments. The load shall then be distributed proportional to the end loads, from this it follows that W_{2aft} is $W_2 \times W_1/(W_1+W_3)$ and W_{2fwd} is $W_2 \times W_3/(W_1 + W_3)$.

Pt 4 Ch 4 Sec 1

2.9 Shaft bearings, dimensions

2.9.1 General

Radial fluid bearings shall be designed with bearing pressures and hydrodynamic lubrication thickness suitable for the bearing materials and within manufacturers specified limitations.

For shaft bearings with significant pressure in plants operating at very low speeds (e.g. electric drives, steam plants or long term running on turning gear), hydrostatic bearings may be required.

The length of the aft most propeller shaft bearing shall be chosen to provide suitable damping of possible whirling vibration.

2.9.2 Oil lubricated bearings of white metal

For the aft most propeller shaft bearing, the nominal bearing pressure (projected area) shall be below 8 bar for all static conditions.

For other oil lubricated white metal bearings, higher pressures can be accepted within the limits specified by the manufacturer. Compliance with geometrical tolerances and precision of alignment assumed in manufacturer's specification shall be verified in cases that the nominal pressure exceeds 12 bar in static condition.

The minimum length of the aft most propeller shaft bearing shall not be less than 1.5 times the actual journal diameter.
Minimum permissible diametrical bearing clearance for the aft most propeller shaft bearing:
$C \geq 0.001\, d + 0.2$
C = diametrical bearing clearance [mm]
d = shaft outer diameter [mm]

2.9.3 Oil lubricated synthetic bearings

The permissible surface pressures shall be especially considered, but not to exceed those for white metal.
For the aft most propeller shaft bearing the nominal surface pressure (projected area) shall be below 6 bar for all static conditions.
The minimum length of the aft most propeller shaft bearing shall not be less than 1.5 times the actual journal diameter.

2.9.4 Water lubricated synthetic bearings

The permissible surface pressures shall be especially considered, but not to exceed those for white metal.
For the aft most propeller shaft bearing the nominal surface pressure (projected area) shall be below 6 bar for all static conditions.
The minimum length of the aft most propeller shaft bearing shall not be less than 2.0 times the actual journal diameter.

2.9.5 Separate thrust bearings

For separate thrust bearings the smallest hydrodynamic oil film thickness, taking into consideration the uneven load distribution between the pads, shall be larger than the sum of the average surface roughness of the thrust collar and pad (Ra_collar + Ra_pad).

2.9.6 Ball and roller bearings

Ball and roller bearings shall have a minimum L_{10a} (ISO 281) life time that is suitable with regard to the specified overhaul intervals. The influence of the lubrication oil film may be taken into account for L_{10a}, provided that the necessary conditions, in particular cleanliness, are fulfilled.

6.2 Shafting arrangement

6.2.1 The machinery and shafting shall be arranged so that neither external nor internal (self generated) forces can cause harmful effects to the performance of the machinery and shafting.
If shaft brake is fitted, it shall be arranged so that in case of failure in the actuating system, the brake shall not be engaged.
6.2.2 The shafting system shall be evaluated for the influence of:
— thermal expansion
— shaft alignment forces
— universal joint forces
— tooth coupling reaction forces
— elastic coupling reaction forces (with particular attention to unbalanced forces from segmented elements)
— hydrodynamic forces on propellers
— ice forces on propellers, see Pt.6 Ch.6 of the Rules for Classification of Ships
— hydrodynamic forces on rotating shafts:
i) outboard inclined propeller shafts or unshielded impeller shafts, see [6.3.1] 1)
ii) mean thrust eccentricity caused by inclined water flow to the propeller, see [6.3.1] 1) (Applicable to HS, LC and NSC)
— thrust eccentricity in water jet impellers when partially air filled or during cavitation, see [6.3.1] 2)
— forces due to movements of resiliently mounted machinery (maximum possible movements to be considered)
— forces due to distortion or sink-in of flexible pads.

6.3 Shaft bending moments

6.3.1 The shaft bending moments due to forces from sources as listed in [6.2.2] are either determined by shaft alignment calculations, whirling vibration calculations, or by simple evaluations. However, two of the sources in [6.2.2] need further explanations:
1) The hydrodynamic force F on an outboard shaft rotating in a general inclined water flow may be determined as
$F = 0.87 \cdot 10^{-4}\, \eta\, v\, n\, d^2 \sin\alpha$ (N/m shaft length)
d = shaft diameter (mm)
n = r/min of the shaft
v = speed of vessel (knots)
α = angle (degrees) between shaft and general water flow direction (to be taken as parallel to the bottom of the vessel)

η = "efficiency" of the circulation around the shaft. Unless substantiated by experience, it shall not be taken less than 0.6.
In order to determine the bending moments along the shaft line of an outboard shaft (as well as at the front of the hub), the bending moment due to propeller thrust eccentricity shall be determined e.g. as:

M_b=0.074 ɑ D T/H (Nm)

D= propeller diameter (m)

T = torque (Nm), which may be taken as the rated torque if low torsional vibration level

H= propeller pitch (m) at 0.7 radius

The bending moment due to the (horizontal) eccentric thrust should be directed to add to the bending moment due to the hydrodynamic force F in the first bearing span.
2) The stochastic bending moment due to thrust eccentricity in a water jet impeller during air suction or cavitation is based on the worst possible scenario:
50% of the normal impeller thrust (F_{TH} in N) applied at the lower half of the impeller, resulting in a bending moment as:

M_b=0.1 F_{TH} D (Nm)

D= the impeller diameter (m).

TORSIONAL VIBRATION

Part 4 Chapter 2 Section 2

1.1 Application

1.1.1 Scope

The rules in this section apply to all shafting used in rotating machinery for propulsion, power production, steering and manoeuvring independent of type of driver except auxiliary plants with less than 200 kW rated power.

1.1.2 Simplification

Only mechanical active systems shall be included in the analysis. De-clutched branches shall not be required in the model. Electric power transmission, hydrodynamic couplings and torque converters shall not be seen as components transferring torsional vibrations; consequently systems in both ends can be handled as independent mass elastic systems.

1.1.3 Acceptance criteria

Acceptance criteria are found in the respective rule chapters for the components.

2 Calculation

2.1 General

2.1.1 Analysis conclusion

All analysis reports shall have a conclusion. In case of forced vibration analysis the conclusion shall be based on a comparison between calculated dynamic response and the permissible values for all the sensitive parts in the plant. Assumptions, conditions and restrictions shall be presented.

2.1.2 Input data quality

General

Parameters of importance which are uncertain, varying or nonlinear are handled by use of extreme values.
It is not required to perform calculations with all combinations of these extreme data, but as a minimum the influence shall be quantitatively considered and also addressed in the conclusions.

Uncertain parameters

Variation of essential data such as dynamic characteristics of elastic couplings and dampers shall be considered. Especially rubber couplings and certain types of vibration dampers have wide tolerances of stiffness and damping.

Variation of parameter values

For components like couplings having stiffness with strong dependency on vibratory torque and/or temperature (as a consequence of power loss) calculation where these dependencies are included may be requested.

Nonlinear characteristics

Systems with components having a strong nonlinear characteristic within the operation range with large influence on the system dynamics shall be simulated in time domain.

Source of data

In vibration calculations the source of all essential data shall be listed. For data that cannot be given as constant parameters the assumed parameter dependency and tolerance range shall be specified.

2.2 Free vibration

2.2.1 Analysis content

Natural frequency calculations of the complete system are required. These shall include tables of relative displacement amplitudes, relative inertia torques, vector sums and, if used later, also their phase angles.

Specification of input data

Mass elastic system: Moments of inertia and inertia-less torsional elasticity/stiffness for each element in the complete system Components: List of components with technical data as found relevant.

Presentation of results

— Tables: Relative displacement amplitudes, relative inertia torques, vector sums and, if used later, also their phase angles.
— Graphs: Vibration mode shapes.

2.2.2 Calculation method

Calculation of relevant natural frequencies and their corresponding mode shapes shall be carried out by recognised calculation methods.

Guidance note:

Examples of recognised methods obtaining natural frequencies and their mode shapes are methodologies for direct matrix solutions calculating eigenvalues. Alternatively, approximate methods as the iterative Holzer's method can be used. Damping has very little effect on natural frequency of the system, and hence the calculations for natural frequencies may be made on the basis of no damping.

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2.3 Forced vibration frequency domain

2.3.1 Analysis content

Free vibration

Forced vibration shall include free vibration calculation see [2.2]

Specification of input data

Data to be specified as applicable:
— *Engine:* Engine type, rated power, rated speed, cycles per revolution, design (in-line/V-type), number of cylinders, firing order, cylinder diameter, stroke, stroke to connecting rod ratio, oscillating mass of one crank gear, excitation see [2.3.3].
— *Vibration damper:* Type, damping coefficient, moments of inertia, dynamic stiffness.
— *Elastic couplings:* Type, damping coefficient, moments of inertia, dynamic stiffness.
— *Reduction / power take off (PTO) gears:* Type, moment of inertia for wheels and pinions, individual gear's ratios per mesh, effective stiffness.
— *Shafting:* Shaft diameter of crankshafts, intermediate shafts, gear shafts, thrust shafts and propeller
shafts.
— *Propeller:* Type, diameter, number of blades, pitch and expanded area ratio, moment of inertia in air, moment of inertia of entrained water (for zero and full pitch for CP propellers).
— *Mass elastic system:* Values of all inertias, stiffnesses and damping values including propeller damping.

Presentation of results

— The results of the forced torsional vibration calculations shall be presented as relevant for the various components in the system.
— The results shall be presented as synthesis, including amplitude and phase from the orders representing the largest contributions.
— The results shall be presented by graphs including acceptance values, see [2.5].
— Where barred speed range is required, maximum time for passing shall be specified.

Guidance note:

Propeller moment of inertia for entrained water shall be specified by propeller designer Ch.5 Sec.1 [1.2.5].

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Method and mass elastic system

The forced torsional vibration shall be calculated by means of linear differential equations, one for each lumped mass. Each mass shall be described by its inertia, connected by torsional springs to adjacent masses, damping described as absolute (mass) damping and relative (shaft) damping, and excitation applied on mass. Other recognized methods may be accepted upon request.

Representative parameter values

The parameters used in vibration calculations shall be representative for the actual speed, mean torque, frequency, temperature, and vibratory torque. The latter implies that if an element is strongly dependent on the level of the vibratory torque and used in a linear vibration calculation, then the whole calculation may have to be made by iteration.

Two-stroke engine

Engine designer's model and parameters shall be applied.

Propeller damping

In order to best represent the damping properties of a propeller, the Archer's or Frahm's approach with torque dependent damping coefficients should be used. Alternative methods using a dynamic magnifier or Schwanecke's empirical approach or other approaches shall be subject to special consideration. For planning crafts damping shall be based on derivation of the actual torque characteristics, see guidance note.

Guidance note:

Propeller damping is a consequence of the propeller's torque absorption characteristics, defined as $C = dT/d\omega$, where T is absorbed torque and $\omega = \frac{2\pi n}{60}$. The torque characteristic for non-planning vessels can be formulated as $T = n^m$, where μ is 2 in steady state condition, but is somewhat higher due to the superimposed vibratory torque. The *Archer number* is defined as $a = \mu\left(\frac{60}{2\pi}\right)$. The corresponding Frahm number is $Q \approx a/9{,}545$. Archer number is depending on the actual propeller design and load, but is typically in the range 24-30 for conventional propellers. Dynamic magnifier for absolute damping is defined as $M = J\omega/C$, where J is propeller inertia and ω is actual vibration frequency. The corresponding relative damping is $\zeta = (\omega/\omega_n)/2M$. Dynamic magnifier or relative damping should only be applied based on experience from measurements of similar plants.

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2.3.3 Excitation

Two-stroke engine

Engine excitation shall be based on harmonic tables of tangential crank pressure from engine designer relevant for the actual engine with respect to type approval. Alternatively it can be based on measurements of cylinder pressure for the actual engine.

Four stroke engine

In addition to the methods for two-stroke engines, simplified methods with generic predefined pressure-time characteristics based on main engine data may be accepted.

Propeller excitation

Propeller excitation can be taken as a percentage of the actual mean torque according to Table 5 unless other values are substantiated by the propeller manufacturer. The values are representative for max continuous forward operation. Propeller excitation for extreme steering manoeuvres of azimuth thrusters shall be takenas 3 times the excitation in Table 5, unless other figures can be documented.

Table 5 Propeller excitation as percent of mean torque

Number of blades	*Blade frequency*	*Double blade frequency*
3	8%	2%
4	6%	2%
5	4%	1.5%
6	4%	1.5%

Other excitations

Other excitation sources as electric drive control system, water jet impeller pulses, universal joints (second order), etc. may have to be taken into accountwhen it influences the system behaviour.

2.3.4 Conditions

— Normal operation. For engines this shall be applied as uniform pressure distribution over all cylinders

— Misfiring operation, only applicable for engines

— Where the installation allows various operation modes, the torsional vibration characteristics shall be investigated for all possible modes, see guidance note.

Guidance note:

Examples of designs to investigate are installations fitted with controllable pitch propellers for zero and full pitch, power take off gear integrated in the main gear or at the forward crankshaft end for loaded and idling generator, clutches for engaged and disengaged branches.

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Selection of misfiring cylinder

For calculation in misfiring condition the misfiring cylinder shall be selected as follows:

for vibration modes and orders with vector sums almost equal zero, any cylinder may be selected

for vibration modes with significant vector sums (e.g. > 0.1 relative to maximum cylinder amplitude) either

- the cylinder which has the opposite phase angle of the vector sum should be selected or - calculating all combinations and presenting the worst.

2.4 Forced vibration time domain

2.4.1 Analysis content

Free vibration

Forced vibration shall include free vibration calculation, see [2.2].

Specification of input data

Engine data to be specified as applicable; Brand, model, bore, stroke, piston rod length, number of cylinders, V-angle, firing sequence and max rpm.

2.4.2 Calculation method and model

Method and mass elastic system

The forced torsional vibration shall be calculated by numerical integration of differential equations as found relevant for the system modelled.

Simplified model

The mass elastic system for numeric simulation can be simplified in order to remove high natural frequencies.

It is required to verify by natural frequency calculations that the simplified system has approximately the same lower (only the important) frequencies as the detailed system.

Presentation of results

Simulation results shall be presented by graphs. Resolution and choice of parameters shall reflect the intention of the simulation.

2.4.3 Relevant cases for simulation

Passing through a barred speed range

Simulation of fixed pitch propeller plants shall take into account the most important properties of the propulsion, the ship mass and resistance (fully loaded) and the rpm control.

The result of transient vibration documentation shall contain the peak vibration level and an estimation of the equivalent number of cycles. The acceptance criterion is the peak torque (or stress) and the corresponding equivalent number of cycles that shall be used for the shaft calculations.

The equivalent number of cycles is defined as the number that results in the same accumulated partial damage (Miner's theory) as the real load spectrum. This equivalent number of cycles for passing up and down through the barred speed range shall be multiplied with the expected number of passages during the foreseen lifetime of the ship. A detailed method for evaluating the equivalent number of cycles and expected number of passages is presented in class guideline DNVGL-CG-0038.

Ice impact loads

Response of non-harmonic impact loads from ice as described in the ice rules (see [1.3]) shall be simulated in the time domain when shaft speed cannot be maintained due to ice loads. Frequency domain calculation in resonant speed can be used as an option.

Large inertia loads

For plants that have a major critical resonance below idling speed and a low ratio of engine inertia to driven machinery inertia, the transient vibration torque shall be considered. This applies e.g. to diesel generator sets with highly elastic couplings and similar propulsion plants without clutch.

Clutching-in

The calculation of the system shall determine:

— the peak torque in couplings and gears
— the first decreasing torque amplitudes
— the heat developed in the clutch
— the flash power in the clutch.

The clutch parameters such as the actuation pressure-time characteristics and if necessary also the changing coefficient of friction shall be used in the calculation.

The results are not to exceed the permissible peak torques and amplitudes in couplings and gears in addition to the permissible heat (J) and flash power (W) in the clutch.

Torque measurements during the clutching-in may be required. This applies when calculations indicate peak torques or amplitudes near the approved limits.

Short circuit in PTO driven generators

A possible short circuit in a generator is not to be detrimental for the power transmitting elements such as couplings and gears. The purpose of the calculation shall determine the peak torques and amplitudes that occur before the safety system (circuit breaker) is in action. The duration to be considered is 1 s.

Guidance note:

If the excitation torque (in the air gap between rotor and stator) is not specified, it can be assumed as:

$T = T_0\,[10\, e^{-t/0.4} \sin(\Omega\, t) - 5\, e^{-t/0.4} \sin(2\Omega\, t)]$

where:

$\Omega/2\pi$ = the electric net frequency (50 or 60 Hz)

t = time in s.

---e-n-d---o-f---g-u-i-d-a-n-c-e---n-o-t-e---

Influence of speed governor

When the speed governor influence has be taken into account it shall be done in the time domain.

2.5 Acceptance criteria

If any result is close to the acceptance limit and there are uncertainties in the calculations, vibration measurements may be required, see [4].

2.5.1 Availability of main functions

In specifying prohibited ranges of operation it has to be observed that the navigating and manoeuvring functions are not severely restricted.

2.5.2 Determination of barred speed range

Speed ranges or operating conditions where the following acceptance criteria are exceeded, shall be barred for continuous operation. Corresponding signboards shall be fitted at all manoeuvring stands and all tachometers marked with red. The tachometers shall be accurate within the tolerance +/-0.01 n_0. A barred speed range above λ = 0.8 is not permitted.

The width of a barred speed range shall be determined as follows:
— range where permissible values are exceeded
— extend with tachometer tolerance in both ends
— further extension in case of unstable engine operation at any end of the barred range.

Guidance note:

For 2-stroke fixed pitch plants the width of the barred speed range should not be made unnecessary wide because this can result in a too slow passage with the consequence of higher vibratory stress level and increased number of cycles with high stress level.

---e-n-d---o-f---g-u-i-d-a-n-c-e---n-o-t-e---

2.5.3 Misfiring condition

Exceeding the acceptance limits in misfiring condition shall result in:
— Restricted (e.g. < 0.5 hours) operation when the vibration level is acceptable for limited time (slow heating of rubber elements) — Restricted driving or load condition (barred speed range or speed reduction etc.)
— Rejection when the vibration level may be critical as e.g. speed governor response, heating of rubber elements causing damping and stiffness to alter to further increase the vibration level, hard gear hammer, etc.

2.5.4 Shafts

Design requirements with acceptance criteria for shafts are found in Ch.4 Sec.1.
For plants with gear transmissions, the shafts (inside as well as outside the gearbox or thruster) shall be designed for at least the same vibration level as the gearing. Unless significantly higher vibration are expected to occur somewhere in the shafting, documentation of the vibration levels in the shafts is not required.
For direct coupled plants the vibration level (τ_v) is not to exceed the values used for the shafting design with regard to continuous operation. Alternatively, the calculated vibration for continuous operation may be used for the shafting design.
For shafts that are designed on the basis of transient vibration, the torque amplitudes as well as number of equivalent cycles per passage are not to exceed the prerequisites for the shaft design
Extended documentation to be submitted for designs where it is likely to expect high cycle fatigue due to passing of barred speed range (See guidance note below).

Guidance note:

In this context high cycle fatigue is expected when high transient stress amplitudes are combined with a large number of cycles. Total number of cycles is dependent of cycles for each passing of barred speed range (BSR) and the vessel's operation profile. A large number of cycles are to be understood as above 10^5 cycles. Extended documentation shall contain fatigue analysis supported by engine and propeller curves as relevant. Classification guideline DNVGL-CG-0038 *Calculation of shafts in marine applications* can be used for fatigue analysis. DNVGL-CG-0038 calculates fatigue capacity based on Wöhler curve (S-N curve) and Miner sum.

---e-n-d---o-f---g-u-i-d-a-n-c-e---n-o-t-e---

2.5.5 Crankshafts

Design requirements and acceptance criteria for crankshafts are found in Ch.3 Sec.1.
The permissible vibration torque (or shear stresses) and peak torque (only applicable to semi-built shafts) are determined in connection with the engine approval. Other criteria may also apply, such as acceleration at mass for cam drive branch or journal movements in bearings.

2.5.6 Vibration dampers

Design requirements and acceptance criteria for dampers are found in Ch.3 Sec.1.
Depending on the type of damper (viscous, rubber, steel spring) the following shall be considered:
— dissipated power (all kinds)
— vibration torque (rubber type and some steel spring types)
— vibration angle (some steel spring types).
The limits specified in the respective type approvals apply.
Design requirements and acceptance criteria for torsional elastic couplings are found in Ch.4 Sec.5.
Torsional elastic couplings have design limitations with respect to:
— dissipated power
— vibration torque.
These limits are for continuous operation. Higher values may be accepted for a limited time of operation if twist amplitudes are monitored.
Transient vibration which occur occasionally (i.e. less than 50 000 times) such as clutching-in is not to exceed neither T_{Kmax1} nor ΔT_{Kmax}.
Transient vibration which occur very infrequently indeed such as short circuit [2.4.3] are not to exceed T_{Kmax2}.
Power loss need not be considered for transient operation.

2.5.8 Other couplings

Design requirements and acceptance criteria for actual components are found in Ch.4 Sec.4.
For other couplings and similar components such as membrane couplings, universal joints, link couplings, elements of composite materials, etc. the approved vibration torque shall not be exceeded.
Tooth couplings are limited with regard to cyclic torque reversals. The negative torque is not to exceed 20% of T_0 unless especially approved.

2.5.9 Gear transmissions

Design requirements and acceptance criteria for gear transmissions are found in Ch.4 Sec.2.
The permissible vibration torque in gear transmissions is limited as:

1) In the full speed and load range (> 90% of rated speed and load) the vibration torque is not to exceed (K_A - 1)·T_0 where K_A is the application factor used in the gear transmission approval.
2) The vibration torque is limited to 35% of T_0 throughout the entire operation range.
3) Gear hammer (negative torque) is not permitted except in unloaded power take off branches, where 10% of T_0 (referred to the subject shaft speed) and 15% short duration misfiring is permitted.
4) Transient vibrations shall not cause negative torques of more than 25% of T_0.
5) Transient peak torques shall not exceed T_0.
6) Transient peak torques shall not exceed the approved (K_{AP} T_0) or (1.5 T_0).

2.5.10 Shrink fits including propeller fitting

Design requirements and acceptance criteria for shrink fits are found in Ch.4 Sec.1.
The estimated vibration torque shall not exceed the value used in the approval of the shrink fit connection.
Permissible vibration torque in shrink fit connections shall be considered for direct coupled plants and when the peak torque in a barred speed range exceeds the peak torque at full load. Peak torque values during misfiring operation shall be subject to special consideration.

2.5.11 Propellers

Design requirements and acceptance criteria for propellers are found in Ch.5 Sec.1.
No specific limitations apply unless especially mentioned in connection with the propeller approval.

2.5.12 Thrusters

See Ch.5 Sec.3.

2.5.13 Electric rotating machines generators, pumps, compressors etc.

The vibration level shall not exceed any limitation specified by designer of the electric generator or motor.

2.5.14 Speed governor

The vibration levels at the sensor location of flexibly coupled propulsion engines shall not exceed the value specified by the engine manufacturer. If no value is specified and approved, tests and measurements shall be made in order to verify that the governor response is insignificant.

Pt 4 Ch.4 Sec.1

2.2.8 *Simplified calculation method for shafts in direct coupled plants.*

1) This method may also be used for other intermediate and propeller shafts that are mainly subjected to torsion. Shafts subjected to considerable bending, such as in gearboxes, thrusters, etc. as well as shafts in prime movers are not included.
….
4) Permissible torsional vibration stresses:
The alternating torsional stress amplitude shall be understood as ($\tau_{max}-\tau_{min}$)/2 measured on a shaft in a relevant condition over a repetitive cycle.
Torsional vibration calculations shall include normal operation and operation with any one cylinder misfiring (i.e. no injection but with compression) giving rise to the highest torsional vibration stresses in the shafting.
For continuous operation the permissible stresses due to alternating torsional vibration shall not exceed the values given by the following formulae:

$$\pm\tau_C = \frac{\sigma_B + 160}{18} \cdot c_K \cdot c_D \cdot (3 - 3 \cdot \lambda^2) \quad \text{for } \lambda < 0.9$$

$$\pm\tau_C = \frac{\sigma_B + 160}{18} \cdot c_K \cdot c_D \cdot 1.38 \quad 0.9 \le \lambda < 1.05$$

where:

τ_C = stress amplitude (MPa) due to torsional vibration for continuous operation
σ_B = specified minimum tensile strength (MPa) of shaft material, see item 2)
c_K = factor for particular shaft design, see item 5)
c_D = size factor, = $0.35 + 0.93 \cdot d_0^{-0.2}$
d = actual shaft outside diameter (mm)
λ = speed ratio = n/n_0
n = speed (rpm) under consideration
n_0 = speed (rpm) of shaft at rated power.

Where the stress amplitudes exceed the limiting value of τ_C for continuous operation, including one cylinder misfiring conditions if intended to be continuously operated under such conditions, restricted speed ranges shall be imposed, which shall be passed through rapidly.
If this is exceeded, flanged shafts (except propeller flange) shall be designed with a stress concentration factor less than 1.05, see Guidance note below. Alternatively, a calculation method which is taking into account the accumulated number of load cycles and their magnitude during passage of the barred speed range, may be used, see Guidance note to [2.2.1].

Guidance note:

This may be obtained by means of a multi-radii design such as e.g. starting with r_1 = 2.5 d tangentially to the shaft over a sector of 5°, followed by r_2 = 0.65 d over the next 20° and finally r_3 = 0.09 d over the next 65°(d = actual shaft outside diameter).

---e-n-d---of---g-u-i-d-a-n-c-e---n-o-t-e---

Restricted speed ranges in normal operating conditions are not acceptable above λ= 0.8. Restricted speed ranges in one-cylinder misfiring conditions of single propulsion engine ships shall enable safe navigation.

The limits of the barred speed range shall be determined as follows:

— the barred speed range shall cover all speeds where τ_C is exceeded. For controllable pitch propellers with the possibility of individual pitch and speed control, both full and zero pitch conditions have to be considered

— the tachometer tolerance (usually $0.01 \cdot n_0$) has to be added in both ends

— at each end of the barred speed range the engine shall be stable in operation.

For the passing of the barred speed range the torsional vibrations for steady state condition shall not exceed the value given by the formula:

$$\pm\tau_T = 1.7 \cdot \tau_C / \sqrt{c_K}$$

where:

τ_T= permissible stress amplitude in N/mm^2 due to steady state torsional vibration in a barred speed range.

5) Table 6 shows k and c_K factors for different design features.

Transitions of diameters shall be designed with either a smooth taper or a blending radius.

Guidance note:

For guidance, a blending radius equal to the change in diameter is recommended.

---e-n-d---of---g-u-i-d-a-n-c-e---n-o-t-e--

Table 6 k and c_K factors for different design features

Intermediate shafts with						*Thrust shafts external to engines*		*Propeller shafts*		
Integral coupling flange 1) and straight sections	Shrink fit coupling 2)	Keyway, tapered connection 3)4)	Keyway, cylindrical connection 3)4)	Radial hole 5)	Longitudinal slot 6)	On both sides of thrust collar 1)	In way of bearing when a roller bearing is used	Flange mounted 1) or keyless taper fitted propellers 8)	Key fitted propellers 8)	Between forward end of aft most bearing and forward stern tube seal
k = 1.0	1.0	1.10	1.10	1.10	1.20	1.10	1.10	1.22	1.26	1.15
c_K = 1.0	1.0	0.60	0.45	0.50	0.30 7)	0.85	0.85	0.55	0.55	0.80

Footnotes

1) Fillet radius shall not be less than 0.08 d.

2) k and c_K refer to the plain shaft section only. Where shafts may experience vibratory stresses close to the permissible stresses for continuous operation, an increase in diameter to the shrink fit diameter shall be provided, e.g. a diameter increase of 1 to 2% and a blending radius as described in the table note.

3) At a distance of not less than 0.2 d from the end of the keyway the shaft diameter may be reduced to the diameter calculated with k = 1.0.

4) Keyways are not to be used in installations with a barred speed range.

5) Diameter of radial bore not to exceed 0.3 d.

The intersection between a radial and an eccentric axial bore (see Figure 1) is not covered by this method.

6) Subject to limitations as slot length (l)/outside diameter < 0.8, and inner diameter (d_i)/outside diameter < 0.7 and slot width (e)/outside diameter >0.15. The end rounding of the slot shall not be less than e/2. An edge rounding should preferably be avoided as this increases the stress concentration slightly. The k and c_K values are valid for 1, 2 and 3 slots, i.e. with slots at 360°, respectively 180° and 120° apart.

7) c_K = 0.3 is an safe approximation within the limitations in 6). More accurate estimate of the stress concentration factor (scf) may be determined from the formulae in item 6 or by direct application of FE calculation. In which case:

c_K = 1.45/scf.

Note that the scf is defined as the ratio between the maximum local principal stress and $\sqrt{3}$ times the nominal torsional stress (determined for the bored shaft without slots).

8) Applicable to the portion of the propeller shaft between the forward edge of the aftermost shaft bearing and the forward face of the propeller hub (or shaft flange), but not less than 2.5 times the required diameter.

6) Notes:

A. Shafts complying with this method satisfy the load conditions in [2.2.2].

a) Low cycle fatigue criterion (typically < 10^4), i.e. the primary cycles represented by zero to full load and back to zero, including reversing torque if applicable. This is addressed by the formula in item 3).

b) High cycle fatigue criterion (typically >10^7), i.e. torsional vibration stresses permitted for continuous operation as well as reverse bending stresses. For limits for torsional vibration stresses see item 4).

The influence of reverse bending stresses is addressed by the safety margins inherent in the formula in item 3.

c) The accumulated fatigue due to torsional vibration when passing through a barred speed range or any other transient condition with associated stresses beyond those permitted for continuous operation is addressed by the criterion for transient stresses, item 4).

B. Explanation of k and c_K.

The factors k (for low cycle fatigue) and c_K (for high cycle fatigue) take into account the influence of:

— The stress concentration factors (scf) relative to the stress concentration for a flange with fillet radius of 0.08 d (geometric stress concentration of approximately 1.45).

$$c_k \approx \frac{1.45}{scf} \quad \text{and} \quad k \approx \left(\frac{scf}{1.45}\right)^x$$

— where the exponent x considers low cycle notch sensitivity.

— The notch sensitivity. The chosen values are mainly representative for soft steels (σ_B < 600), while the influence of steep stress gradients in combination with high strength steels may be underestimated.

— The size factor C_D being a function of diameter only does not purely represent a statistical size influence, but rather a combination of this statistical influence and the notch sensitivity.

The actual values for k and c_K are rounded off.

C. Stress concentration factor of slots

The stress concentration factor (scf) at the end of slots can be determined by means of the following empirical formulae using the symbols in Footnote 6) in Table 5:

This formula applies to:

$$scf = \alpha_{t(hole)} + 0.8\left(\frac{(l-e)/d}{\sqrt{\left(1-\frac{d_i}{d}\right)\cdot\frac{e}{d}}}\right)$$

— slots at 120°, 180° or 360° apart

— slots with semi-circular ends. A multi-radii slot end can reduce the local stresses, but this is not included in this empirical formula.

— slots with no edge rounding (except chamfering), as any edge rounding increases the scf slightly.

$\alpha_{t(hole)}$ represents the stress concentration of radial holes (in this context e = hole diameter), and can be determined from:

$$\alpha_{t(hole)} = 2.3 - 3\cdot\frac{e}{d} + 15\cdot\left(\frac{e}{d}\right)^2 + 10\cdot\left(\frac{e}{d}\right)^2\cdot\left(\frac{d_i}{d}\right)^2$$

or simplified to: $\alpha_{t(hole)} = 2.3$.

LATERAL VIBRATION

Pt 4 Ch 2 Sec 3

1 General

1.1 Application

1.1.1 Scope

The rules in this section apply to all shafting used in rotating machinery for propulsion, power production, steering and manoeuvring independent of type of driver except auxiliary plants with less than 200 kW rated power.

1.1.2 Vibration regimes

The following vibration regimes are covered within this section:

— lateral vibrations are handled as whirling, see [2]

— axial vibration see [3].

1.1.3 Acceptance criteria

Acceptance criteria for components are found in the respective rule chapters for the components.

1.1.4 Coupled vibrations

Axial vibrations initiated by torsional vibrations can be handled as independent, but with the radial component of excitation from the cylinder forces.

1.1.5 Forced vibration analysis

Time domain simulation may be requested in addition to forced vibration calculation in the frequency domain for transient analysis.

2.1 Analysis

2.1.1 Extent and method of calculation

As a minimum, the calculations shall include the natural frequencies and mode shapes of the relevant vibration modes.

2.1.2 Uncertain and variable parameters

A variation of parameters shall be included in the analysis in case of uncertain or variable important parameters.

Guidance note:

Important but uncertain parameters as stiffness of aft stern tube bearing, resulting bearing load position, bearing load distribution over length (if calculating with distributed bearing reaction), entrained water on propeller, etc. shall be varied within their probable range and natural frequencies to be presented as corresponding graphs.

---e-n-d---o-f---g-u-i-d-a-n-c-e---n-o-t-e---

2.1.3 Entrained water to propeller

Calculation of entrained water shall be presented.

2.1.4 First order acceptance criteria

Resonance with the shaft speed (1st order forward whirl) shall have a separation margin of at least 20% to the operating speed range, guidance note.

Guidance note:

Example: A system with idle at 20 rpm and MCR at 100 rpm shall not have a 1st order fwd whirl in the range 16 to 120 rpm.

---e-n-d---o-f---g-u-i-d-a-n-c-e---n-o-t-e---

2.1.5 Higher order acceptance criteria

Resonance caused by propeller blade passing (blade order) shall be avoided in the upper operating speed range unless it is substantiated that resonance will not cause harmful response, see guidance note.

Guidance note:

Approval is based on an over-all evaluation, and resonance should be avoided above 80% of max rpm. Exceptions may be given: A long shafting with many bearings is not found sensitive if the propeller is the main excitation source and the mode shape indicates resonance in forward end of shafting. Also bearing designs where good damping is expected, e.g. high bearing length to diameter ratio combined with a bouncing vibration mode, may justify acceptable resonance response at high rpm.

---e-n-d---o-f---g-u-i-d-a-n-c-e---n-o-t-e---

2.1.6 Presentation of results

Results shall be presented both in way of numerical values and graphical e.g. as a Campell diagram. Mode shapes of the natural frequencies shall be presented.

AXIAL VIBRATION

Pt 4 Ch 2 Sec 3

3.1 Analysis

3.1.1 Extent and method of calculation

As a minimum, the calculations shall include the natural frequencies and mode shapes of the relevant vibration modes.

3.1.2 Uncertain and variable parameters

If the lowest vibration mode (with the node in the thrust bearing) is of significance to the conclusion, the calculations shall be made with various thrust bearing stiffness in order to see the influence of an estimation error.

3.1.3 Acceptance criteria free vibration

If major critical resonance occurs near or in the operational speed range and no damper is foreseen, forced axial vibration calculations shall be required.

3.1.4 Acceptance criteria forced vibration

In crankshafts, the stresses due to axial vibration shall not exceed the values used in connection with the engine approval. The amplitudes at the front end of the crankshaft shall be within the engine designer’s specified limit.

3.1.5 Presentation of results

Results shall be presented both in way of numerical values and graphical of the mode shapes of the natural frequencies.

4 Measurements

4.1 Axial vibration

Measurements of axial vibrations shall be required if major critical resonance occurs near or in the operational speed range and no damper is foreseen.