CLASS GUIDELINE

DNVGL-CG-0039 Edition December 2015

Calculation of marine propellers

DNV GL AS

FOREWORD

DNV GL class guidelines contain methods, technical requirements, principles and acceptance criteria related to classed objects as referred to from the rules.

Any comments may be sent by e-mail to *rules@dnvgl.com*

CHANGES – CURRENT

This is a new document.

CONTENTS

SECTION 1 BASIC PRINCIPLES

SECTION 1 BASIC PRINCIPLES

1 Scope and general instructions

This class guideline contains procedures and methods necessary for verification of all propellers of conventional design and arrangement. For propellers not recognised as conventional by the Society, e.g. surface piercing propellers, cycloidal propellers, etc. the approval will be based on special considerations.

Damage due to impact loads from objects in the water or grounding, etc. may reduce the strength of the propeller. However, this is not included in this procedure.

The risk for- and the effect from cavitation erosion are not considered. In general, cavitation that may be harmful with respect to erosion shall be avoided.

1.1 Propeller blades

Root section radius shall be taken at the termination of the root fillet, rounded upwards to the nearest 5% of propeller radius. If the fillets on pressure- and suction side do not terminate at the same radius, the outermost radius applies (see Figure 1).

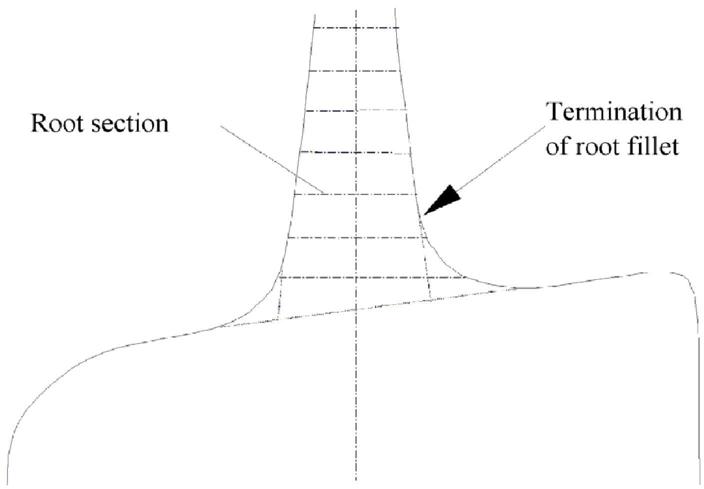

Figure 1 Definition of root section

At any section with radius less than the defined root section, bending strength shall be equivalent or higher, assuming resultant forces to act at 70% of the radius.

The radial distribution of section width and profile thickness shall follow a smooth curve.

Criteria for calculation of propeller blade strength are according to RU SHIP Pt.4 Ch.5 Sec.1 [2.2].

1.2 Propeller hub and pitch mechanism

Methods for how to assess relevant dynamic load conditions for propeller hub and pitch mechanism according to RU SHIP Pt.4 Ch.5 Sec.1 [2.3] are given in Sec.6.

No detailed criteria for stress calculations are specified. Such calculations shall be carried out according to sound engineering practice.

Fatigue properties for the applied materials shall be chosen on basis of recognised references, taking into account number of cycles, type of loading, size effects as well as notch sensitivity and influence of surface roughness.

2 Nomenclature

a_r	Skew coefficient at considered section [-], see Sec.2 [6]
C	Width of expanded section at blade root [m] (tunnel thrusters)
C_{QA}	Maximum obtainable astern torque relative nominal torque [-], see Sec.3 [2.3]
C_r	Width of the considered expanded cylindrical section [m]
C_{ratio}	Ratio between width of expanded sections at root and 60% radius [-] (tunnel thrusters)
D	Propeller diameter [m]
e_{root}	Distance between skew line and generatrix at the considered section [m], see Sec.2 [6]
f	Profile camber at considered section [mm]
F_{qf}	Torque induced force [N], see Sec.4 [4.1] (tunnel thrusters)
h_m	Mean pitch ratio [-], see Sec.2 [7.3]
h_r	Pitch ratio at considered section [-], see Sec.2 [7.3]
h_s	Submersion of shaft centre at maximum draft [-], see Sec.3 [2.5]
H_r	Pitch at considered section [m]
K_{BA}	Astern bending moment coefficient [-], see Sec.3 [2.2]
K_{bm}	Moment arm factor at considered section [-], see Sec.2 [7.1]
K_{corr}	Correlation factor [-] = 0.85. Adjustment factor found during calibration process between beam theory and finite elements analysis.
K_e	Effective section modulus coefficient at considered section [-], see Sec.2 [8]
K_f	Camber correction factor [-], see Sec.2 [8.1]
K_m	Bending moment coefficient at considered section [-], see Sec.2 [7]
K_{sk}	Skew correction factor at considered section [-], see Sec.2 [6]
K_{str}	Load correction factor, see Sec.2 [2]
K_T	Thrust coefficient [-], see Sec.2 [7.3]
K_{Tha}	Thrust coefficient at maximum astern bollard power [-], see Sec.3 [2.1]
K_{thick}	Correction factor for influence of thickness on fatigue strength [-], See Sec.2 [3] and Sec.4 [1] (tunnel thrusters)
K_Q	Torque coefficient [-], see Sec.2 [7.4]
K_{QA}	Astern torque coefficient [-], see Sec.3 [2.4]
Mt	Fluctuating blade load relative to mean load [-], see Sec.2 [4]
M_{th}	Bending moment due to propeller thrust [Nm], see Sec.4 [3] (tunnel thrusters)
M_q	Bending moment due to propeller torque [Nm], see Sec.4 [4] (tunnel thrusters)
n_{BA}	Maximum obtainable astern revolutions in bollard condition [1/s], see Sec.3 [2.3]
n_{cav}	Critical astern revolutions [1/s], see Sec.3 [2.5]
n_s	Propeller revolutions [1/s], see Sec.4 [4.1]
P	Maximum continuous power [W]
r	Relative radius at considered section [-] = considered radius/R
r_{root}	Relative radius at the root section [-], see Sec.2 [6]
r_t	Radial location of resulting load [-], see Sec.4 [3] (tunnel thrusters)
R	Propeller radius [m] = D/2
RPM	Propeller revolutions [1/min]
S	Safety factor [-] , see RU SHIP Pt.4 Ch.5 Sec.1 Table 5
t	Maximum profile thickness at blade root [mm] (tunnel thrusters)

$t_{0.05c}$ Profile thickness at considered section, measured at a relative distance 5% from leading edge [mm]
$t_{0.95c}$ Profile thickness at considered section, measured at a relative distance 95% from leading edge [mm]
$t_{0.8,0.8}$ Profile thickness at 80% radius, measured at a relative distance 80% from leading edge [mm]
t_r Profile thickness at considered section [mm]
Th Propeller thrust [N], see Sec.2 [7.3] and Sec.4 [3] (Tunnel thrusters)
U Fatigue strength amplitude [N/mm^2]
U_1 Fatigue strength amplitude at zero mean stress [N/mm^2], see RU SHIP Pt.4 Ch.5 Sec.1 Table 4.
U_2 Relative reduction of fatigue strength with increasing mean stress [-], see RU SHIP Pt.4 Ch.5 Sec.1 Table 4.
V Maximum ship speed [m/s], corresponding to P
Δw Effective wake variation [-]
Z Number of blades [-]
π ≈ 3.1416
α Effective inflow angle [deg.]
θ Pitch angle at root section [deg.] (tunnel thrusters), see Sec.4 [3] (Sec.2 [7.2])
θ_r Pitch angle at considered section [deg.], see Sec.2 [7.2]
ρ Density of water [kg/m^3], = 1025 for sea water
$\sigma_{0.8}$ Peak stress [N/mm^2] at 80% radius, see Sec.3 [2]
σ_A Dynamic stress amplitude [N/mm^2]
σ_{All} Allowable stress [N/mm^2], see Sec.4 [1] (Tunnel thrusters)
σ_m Mean stress [N/mm^2], see Sec.2 [5]
σ_y Specified minimum yield strength [N/mm^2], see RU SHIP Pt.4 Ch.5 Sec.1 Table 4.

SECTION 2 CALCULATION OF HIGH CYCLE STRESSES IN PROPELLER BLADES

1 High cycle stress criterion

Dynamic stress amplitudes in the propeller blade shall fulfil the following criterion:

$$S \leq \frac{U}{s_{A}}$$

σ_A = dynamic stress amplitude
S = safety factor
U = fatigue strength amplitude.

The stresses referred to are principal stresses.

In normal ahead operation, the high cycle stress criterion may be written:

$$S < \frac{U_1 K_{thick} \quad U_2 \sigma_m}{\sigma_m K_{str} Mt}$$

The formula is based on a fatigue diagram where:

S = safety factor (-), see RU SHIP Pt.4 Ch.5 Sec.1 Table 5
U_1 = fatigue strength amplitude (N/mm^2) at zero mean stress (>10^8 cycles), see RU SHIP Pt.4 Ch.5 Sec.1 Table 4
K_{str} = load correction factor (-), see [2]
K_{thick} = correction factor (-) for influence of thickness on fatigue strength, see [3]
U_2 = relative reduction of fatigue strength with increasing mean stress (>10^8 cycles), see RU SHIP Pt.4 Ch.5 Sec.1 Table 4
Mt = fluctuating load relative to mean load (-), see [4]
σ_m = actual mean stress (N/mm^2), see [5].

Allowable local stresses in fillets etc. are given in Sec.5 [1].

2 Load correction factor

The relative dynamic variation in local stress may differ from the relative dynamic load variation because of variation in pressure distribution. This is in particular relevant for skewed propellers. The stress calculation model takes this into account using the load correction factor, K_{str}, which shall be taken according to the following empirical formula, unless otherwise substantiated:

$$K_{str} = 1.3 - 0.25 K_{sk}$$

Not to be taken less than 0.7 nor higher than 1.0.

Where K_{sk} is skew correction factor at considered section, see [6].

3 Correction factor for influence of thickness

Unless otherwise documented, correction factor, K_{thick} for influence of thickness on fatigue strength of the section in question shall be found from the two empirical formulae:

Stainless steels:

$$K_{thick} = 1.0 - 0.05 \ln\left(\frac{t_r}{25}\right)$$

Other materials:

$$K_{thick} = 1.0 - 0.1 \ln\left(\frac{t_r}{25}\right)$$

where t_r is actual profile thickness (mm) for the referred location at the section in question.

K_{thick} shall not be taken higher than 1.0.

4 Fluctuating blade load relative to mean load

The fluctuating blade load due to hydrodynamic loads shall be taken from the following empirical relation, unless otherwise substantiated:

$$\mathrm{Mt} = \frac{Z^{0.7} V}{10 K_T n_s D} \Delta w$$

Unless all significant low cycle load conditions are considered additionally (see also Sec.3 [1]), Mt shall not be taken less than 0.50.

Mt needs not to be taken higher than 1.0.

For directly coupled diesel engines running sub-critically in torsion at full speed, it is necessary to increase the value of Mt based on special consideration.

4.1 Effective wake variation

Effective wake variation expresses the change in effective wake between the peripherical locations at which the propeller blade is exposed to maximum and minimum load, respectively.

Unless otherwise is documented, effective wake variation, Δw shall be taken as follows:

Propellers where wake variation is dominated by homogenous oblique inflow (pulling thrusters, high speed vessels with inclined propeller shaft, etc.):

$$\Delta w = 0.05\alpha$$

where:

α = effective inflow angle, not to be taken less 7 deg. for azimuting thrusters.

Otherwise, α shall not be taken less than 5 deg.

Twin screw propellers (others than mentioned above) with shaft brackets:

$$\Delta w = 0.4$$

All other propellers:

$$\Delta w = 0.5$$

Guidance note:

Effective wake variation comprises the three dimensional wake variation. Prediction of effective full scale wake variation from a nominal model scale wake field should take into account scale effects, as well as the influence of the working propeller.

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

5 Mean stress

Actual mean stresses in cylindrical sections at root and at 60% radius shall be found from empirically modified cantilever-beam theory as described in [5] to [8]. The following applies for the calculation of actual mean stress:

$$\sigma_m = K_{sk} K_m K_{corr} \frac{\rho\, n_s^2 D^5}{K_e C_r t_r^2}$$

6 Skew correction factor

Relative increase in local maximum stress at a section is expressed by the skew correction factor, K_{sk}.

At root section:

$$K_{sk} = 1 + 15 a_{root}^2$$

where:

a_{root} = skew coefficient (-) at root section, given as

$$a_{root} = \frac{1}{D}\left(0.15 e_{1.0} + e_{0.6} - 2 e_{root}\right)$$

e_{root} = distance (m) between skew line (mid-chord line) and generatrix at the indexed radius, measured along the cylindrical section. Note that e is positive when skew line is forward of generatrix (see Figure 1).

At 60% radius:

$$K_{sk} = 1 + 2.5 a_{0.6}^{1.5}$$

where:

$a_{0.6}$ = skew coefficient (-) at the 60% radius, given as

$$a_{0.6} = \frac{1}{D}\left[\frac{C_{0.9} - C_{root} - 2(e_{0.9} - e_{root})}{(0.9 - r_{root})} - \frac{C_{0.6} - C_{root} - 2(e_{0.6} - e_{root})}{(0.6 - r_{root})}\right]$$

For negative values of $a_{0.6}$, a value of 0.0 shall be used.

r_{root} = relative radius (as a fraction of propeller radius) at root section, (See Figure 1).

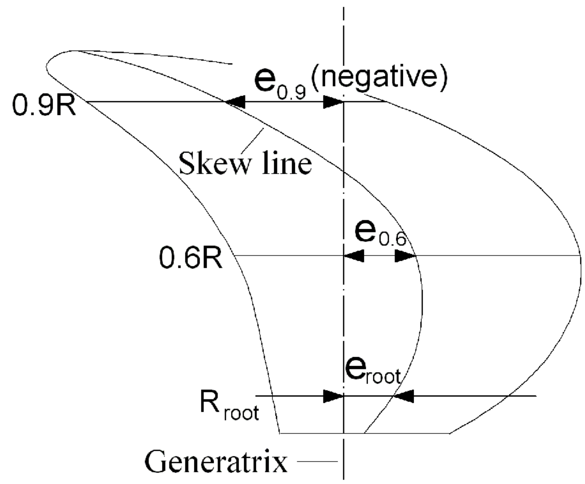

Figure 1 Skewed propeller (expanded outline)

7 Bending moment coefficient

Simplifying bending axis shall be the chord line of each section, bending moment coefficient is found from:

$$K_m = \frac{K_{bm}}{Z}\left(\frac{K_Q}{0.35}\sin\theta_r + K_T\cos\theta_r\right)$$

7.1 Moment arm factor

Moment arm factor for forces acting outside of section, K_{bm} shall be estimated from:

$$K_{bm} = 0.35\,r^2 - 0.73\,r + 0.38$$

7.2 Pitch angle

Pitch angle (θ_r) of section at relative radius r, shall be found from:

$$\theta_r = \tan^{-1}\left(\frac{h_r}{\pi\ r}\right)$$

7.3 Thrust coefficient

Thrust coefficient, K_T shall be found from:

$$K_T = \frac{Th}{\rho\ n_s^2 D^4}$$

Th = propeller thrust (N) corresponding to P and n_s.
ρ = density of water (kg/m^3), = 1025 for sea water.

If Th is not known, K_T may be estimated from:

$$K_T = 7.7\frac{K_Q}{h_m} - 0.06$$

where mean pitch ratio, h_m shall be taken from the following relation:

$$h_m = 0.096 h_{root} + 0.666 h_{0.7} + 0.238 h_{1.0}$$

h_r = H_r/D = pitch ratio (-) at relative radius r,
H_r = pitch (m) of cylindrical section at relative radius r.

7.4 Torque coefficient

Torque coefficient, K_Q shall be found from:

$$K_Q = \frac{P}{2\pi\ \rho\ n_s^3 D^5}$$

P = maximum continuous power for which the installation shall be approved (W).

8 Effective section modulus coefficient

Effective section modulus coefficient is estimated from:

$$K_e = K_f\left[0.045 + 0.06\left(\frac{t_{0.05c}}{t_r} + \frac{t_{0.95c}}{t_r}\right)\right]$$

$t_{0.05c}$ = profile thickness (mm), measured at a relative distance 5% from leading edge
$t_{0.95}$ = profile thickness (mm), measured at a relative distance 95% from leading edge
K_f = camber correction factor (-), expressing increase in section modulus due to profile camber, see [8.1].

8.1 Camber correction factor

The introduction of skew tends to move the point where maximum stresses occur from mid-chord towards trailing edge. Therefore the combined effect of skew and camber shall be taken into account calculating the camber correction factor.

Camber correction factor, K_f shall be taken as 1.0 if K_{sk} (see [6]) is larger than 1.3 at the section in question.

If K_{sk} is not larger than 1.1, K_f shall be estimated from:

$$K_f = 1.35 - 0.7\left|0.5 - f/t_r\right|$$

f = maximum profile camber (mm) at section in question.

For intermediate values of K_{sk} (from 1.1 to 1.3), K_f shall be found by linear interpolation.

9 Centrifugal stress

Centrifugal stress needs normally not be taken into account for fatigue analysis. However, for highly raked propeller blades and/or propellers with a tip speed exceeding 60 m/s, the term $U_2\ \sigma_m$ in [1] needs to be corrected for centrifugal stress.

SECTION 3 CALCULATION OF LOW CYCLE STRESSES IN PROPELLER BLADES

1 Low cycle stress criterion

Propellers for which the turning direction may be reversed shall be additionally checked for margins towards blade bending in astern operation ([1] and [2]). The following criterion is, as a minimum, to be fulfilled in astern operation:

$$S \leq \frac{\sigma_y \, K_{thick}}{\sigma_{0.8}}$$

S = safety factor (-), see RU SHIP Pt.4 Ch.5 Sec.1 Table 5

$\sigma_{0.8}$ = peak stress (N/mm^2) at 80% radius, see [2]

K_{thick} = correction factor (-) for influence of thickness on bending strength of the section in question. K_{thick} shall be taken as 1.0 for stainless steels, and as described for fatigue strength (see Sec.2 [3]) for other materials on basis of actual thickness at 80% chord length at a relative radius of 80%

σ_y = specified minimum yield strength (N/mm^2), see RU SHIP Pt.4 Ch.5 Sec.1 Table 4.

The stresses referred to are equivalent stresses.

2 Peak stresses

Peak stresses, $\sigma_{0.8}$ are assumed to act in the region of 80% radius. Stresses shall be calculated from the following empirical formula:

$$\sigma_{0.8} = \frac{D^5 \, \rho \, K_{Tha} \, K_{BA}}{C_{0.8} \, t_{0.8,0.8}{}^2} \left(2 n_{BA} \, n_{cav} - n_{cav}{}^2\right)$$

K_{Tha} = thrust coefficient (-) at maximum bollard astern power, see [2.1]

K_{BA} = astern bending moment coefficient (-), see [2.2]

$C_{0.8}$ = chord length (m) at relative radius of 80%

$t_{0.8.0.8}$ = profile thickness (mm) at 80% chord length (measured from leading edge), at a relative radius of 80%

n_{BA} = maximum obtainable astern revolutions (1/s = RPM/60) in bollard condition, see [2.3]

n_{cav} = critical astern revolutions (1/s = RPM/60), above which cavitation is expected to influence stress level, see [2.5].

2.1 Astern thrust coefficient

Thrust coefficient at maximum bollard astern power shall be found from:

$$K_{Tha} = \left(0.19 + 0.084\,Z\,\frac{C_{0.8}}{D}\right)\,h_m$$

2.2 Astern bending moment coefficient

Bending moment at 80% relative radius is represented by the astern bending moment coefficient and found from the following empirical expression:

$$K_{BA} = \frac{C_{0.95}\sqrt{\left(\dfrac{0.5C_{0.95} - 0.3C_{0.8} + e_{0.95} - e_{0.8}}{D}\right)^2 + 0.0056}}{0.047\,Z\,(D - e_{0.95} + e_{0.8})}$$

where the index refers to relative radius.

2.3 Maximum astern number of revolutions

Maximum obtainable astern number of revolutions in bollard condition shall be found from the empirical expression, if not otherwise is substantiated:

$$n_{BA} = \sqrt{\frac{C_{QA}\ P}{2\pi\ \rho\ K_{QA}\ n_s\ D^5}}$$

C_{QA} = maximum obtainable astern torque as a fraction of maximum forward torque at MCR (to be taken as 1.0 if not known).

K_{QA} = torque coefficient at maximum bollard astern power, see [2.4].

n_{BA} need not be taken higher than n_s.

2.4 Astern torque coefficient

The following empirical expression shall be used for calculation of torque coefficient at maximum bollard astern power:

$$K_{QA} = \left(0.034 + 0.017\,Z\frac{C_{0.8}}{D}\right)\,h_m$$

2.5 Critical astern revolutions

Astern number of revolutions above which cavitation is expected to have influence on stress level, shall be estimated from the following empirical expression:

$$n_{cav} = \frac{1}{D}\sqrt{26.0 + 2.5\left(h_s + \frac{D}{2}\right)}$$

h_s = submersion of shaft centre (m) at maximum draft. If not known, h_s may be taken as 5D for thrusters, 2D for other propulsion systems.

n_{cav} shall not be taken higher than n_{BA}.

2.6 Other low cycle dynamic stresses

Other low cycle dynamic stresses (see RU SHIP Pt.4 Ch.5 Sec.1 [2.2]) shall be specially considered when applicable. This normally requires detailed information regarding expected load profile as well as detailed stress calculations.

SECTION 4 SIMPLIFIED CRITERIA FOR PROPELLER BLADES WORKING IN A TUNNEL

1 Blade bending stresses

Propellers working in a tunnel are normally not exposed to significant fatigue loads. This is provided that the propellers are not in use at significant ship speeds. Additionally, tunnel openings are assumed to be provided with a reasonable fairing. On this basis, only margins towards permanent blade deformation need to be checked.

If windmilling of the propeller during sea passages may occur, it shall be taken into account as an additional load case. Otherwise effective countermeasures shall be introduced to avoid windmilling, e.g. shaft brake.

The following criterion applies:

$$S \leq \frac{\sigma_{All}\, K_{thick}}{\sigma_m}$$

S = safety factor, see RU SHIP Pt.4 Ch.5 Sec.1 Table 5

σ_m = mean stress in blade root section, see [2]

σ_{All} = allowable stress, to be taken as specified minimum yield strength, or 50% of specified minimum tensile strength, whichever is the least (see RU SHIP Pt.4 Ch.5 Sec.1 Table 4)

K_{thick} = correction factor for influence of thickness on bending strength of the section in question. K_{thick} = 1.0 for stainless steels, and as described for fatigue strength in Sec.2 [3] for other materials.

Allowable local stresses in fillets etc. are given in Sec.5 [1].

2 Mean stress

Mean stress at blade root may be calculated from cantilever-beam theory and found from:

$$\sigma_m = \frac{M_{th} + M_q}{0.09 C\, t^2}$$

M_{th} = bending moment due to propeller thrust (Nm), see [3]

M_q = bending moment due to propeller torque (Nm), see [4]

C = width of expanded section (m) at blade root

t = maximum profile thickness (mm) at blade root.

In the formula above it is assumed that section modulus may be expressed by 0.09 C t^2.

3 Bending moment due to propeller thrust

Bending moment due to propeller thrust is found from:

$$M_{th} = \frac{D\ Th}{2Z}(r_t - r_{root})\ \cos q$$

D = propeller diameter (m)
Th = propeller thrust (N). Th may be taken as 0.12 · P, if not known
P = maximum engine power (W) for which the installation shall be approved
Z = number of blades
r_t = radial location of resulting load (as a fraction of propeller radius), to be taken as 0.82 for controllable pitch propellers and 0.75 for fixed pitch propellers
r_{root} = relative radius at root section (-)
θ = pitch angle (deg.) at root section (see Sec.2 [7.2] for definition).

4 Bending moment due to propeller torque

Bending moment due to propeller torque is found from:

$$M_{th} = \frac{D\ F_{qf}}{2Z}(r_t - r_{root})\sin\theta$$

F_{qf} = torque induced force (N), see [4.1].

4.1 Torque induced force

The propeller torque may be replaced by the torque induced force acting at a distance from the rotational centre. Torque induced force is found from:

$$F_{qf} = \frac{P}{2\pi\ n_s\ r_t \frac{D}{2}}$$

n_s = propeller revolutions (1/s =RPM/60).

5 Profile thickness at 60% radius

At the 60% radius, the profile shall not be less than derived from the following expression:

$$t_{0.6} = 0.7 \cdot t\sqrt{C_{ratio} \cdot K_{thick}}$$

t = required profile thickness (mm) at root
C_{ratio} = the ratio between width of expanded sections at root and 60% radius (-).

SECTION 5 FILLETS AND TIP THICKNESS OF PROPELLER BLADE

1 Blade root fillets

The local stresses in the blade root fillets shall not exceed 1.2 times the nominal allowable stresses for the defined root section. For single radius fillets this may be obtained by a fillet radius not less than 75% of the required thickness of the root section.

For built-up propellers, the stresses in the area between recessed bolt holes shall be considered, allowing local stress levels as for the blade root fillets.

2 Tip thickness

Guidance note:

For propellers intended for propulsion, profile thickness (mm) at 90% radius shuld not be less than:

$t_{0.6}$ = required profile thickness at 60% radius. If the section at 60% radius fulfils the high cycle criterion in Sec.2 [1], $t_{0.6}$ may be taken as actual profile thickness at 60% radius. In general $t_{0.6}$ may be derived from Sec.2 [5], as the minimum thickness for which the high cycle stress criterion is fulfilled.

$$t_{0.9} = 2.25\,D + 0.25\,t_{0.6}$$

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

SECTION 6 DYNAMIC LOADING ON PROPELLER HUB AND PITCH MECHANISM

1 Start and stop of propeller

Considering start and stop of propeller, servo force shall vary between zero and the force corresponding to maximum predicted operating servo pressure (excluding extreme conditions), unless otherwise is substantiated.

If not known, maximum operating pressure may be taken as the maximum of:

— 80% of design pressure
— design pressure - 15 bar.

Guidance note:

The following numbers of cycles are normally considered as realistic, depending on operational profile:

— propellers on ships intended for short distance voyages, such as shuttle ferries, or propellers included in DYNPOS system: 10^6 cycles
— propellers on ships intended for long distance voyages, such as large tankers and container ships: 10^4 cycles
— other propellers: 10^5 cycles.

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

2 Change of pitch setting

Considering change of pitch setting, the force needed to overcome the frictional resistance in the hub and pitch mechanism bearings shall be taken as dynamic load amplitude.

Mean load in this condition shall normally correspond to predicted servo pressure in the normal operating condition (free running at MCR, except for propellers included in DYNPOS system, where servo pressure corresponding to zero-pitch condition shall be used, if higher).

If not known, mean servo pressure may be taken as the maximum of:

— 50% of design pressure
— 75% of maximum operating pressure.

Guidance note:

The following numbers of cycles are normally considered as realistic:

— propellers on ships where propeller pitch is used as load control system of prime mover: 10^{10} cycles
— propellers included in DYNPOS system: 10^8 cycles
— other propellers: 10^7 cycles.

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

3 Dynamic loads from propeller blades

Dynamic loads from propeller blade spindle torque variations during normal ahead operation, can normally be assumed to be carried by frictional resistance in the blade bearings. However, during effectuating of a pitch change, these dynamic loads may be transmitted into the pitch mechanism and shall be considered.

Unless otherwise is substantiated, dynamic spindle moment transmitted into the pitch mechanism in this condition shall be taken as:

$$\Delta Q_{spindle} = 0.2 \cdot Mt \cdot C_{0.6} \cdot F_{mcan} \text{ (kNm)}$$

where:

Mt	=	fluctuating blade load relative mean load, see Sec.2 [4]
$C_{0.6}$	=	width of expanded cylindrical propeller blade section at 60% radius (m), see also Sec.2 [5].
$\Delta Q_{spindle}$	=	fluctuating spindle torque (kNm)

F_{mean} is resulting hydrodynamic blade force (kN), to be taken as the vector sum of axial and transverse mean blade force in normal, ahead operation.

Mean load on the pitch mechanism in this condition shall correspond to predicted servo pressure in the normal operating condition (see [2]).

Guidance note:

The following numbers of cycles are normally considered as realistic:

— propellers on ships where propeller pitch is used as load control system of prime mover: 10^{10} cycles

— other propellers: 10^{8} cycles.

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