Calculation of Marine Propellers

MARCH 2012

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CHANGES

General
This document supersedes DNV-CN-41.5, April 2007.

Text affected by the main changes in this edition is highlighted in red colour. However, if the changes involve a whole chapter, section or sub-section, normally only the title will be in red colour.

Main Changes
• Sec.1.2
  — The references for \( K_Q \) and \( n_s \) have been corrected.
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1. Basic Principles

1.1 Scope and general instructions

The requirements given in this procedure apply to 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.

1.1.1 Propeller blades

Root section radius is to 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 Fig.1-1).

![Figure 1-1 Definition of root section](image)

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

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

Criteria for calculation of propeller blade strength according to the Rules for Classification of Ships Pt.4 Ch.5 Sec.1 B200 are given in paragraphs 2, 3, 4 and 5.

1.1.2 Propeller hub and pitch mechanism

Methods for how to assess relevant dynamic load conditions for propeller hub and pitch mechanism according to the Rules for Classification of Ships Pt.4 Ch.5 Sec.1 B300 are given in paragraph 6.

No detailed criteria for stress calculations are specified. Such calculations are to 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.

1.2 Nomenclature

- $a_r$: Skew coefficient at considered section [-], see 2.6
- $C$: Width of expanded section at blade root [m] (tunnel thrusters)
- $C_{QA}$: Maximum obtainable astern torque relative nominal torque [-], see 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]
- $e_r$: Distance between skew line and generatrix at the considered section [m], see 2.6
- $f$: Profile camber at considered section [mm]
- $F_{qf}$: Torque induced force [N], see 4.4.1 (tunnel thrusters)
- $h_m$: Mean pitch ratio [-], see 2.4.1
- $h_r$: Pitch ratio at considered section [-], see 2.4.1
<table>
<thead>
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<td>$H_r$</td>
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<td>$K_{BA}$</td>
<td>Astern bending moment coefficient [-], see 3.2.2</td>
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<td>$K_{bm}$</td>
<td>Moment arm factor at considered section [-], see 2.7.1</td>
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<tr>
<td>$K_{corr}$</td>
<td>Correlation factor [-] = 0.85</td>
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<td>$n_{BA}$</td>
<td>Maximum obtainable astern revolutions in bollard condition [1/s], see 3.2.3</td>
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<td>$n_{cav}$</td>
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<td>$n_p$</td>
<td>Propeller revolutions [1/s], see 4.5</td>
</tr>
<tr>
<td>$P$</td>
<td>Maximum continuous power [W]</td>
</tr>
<tr>
<td>$r$</td>
<td>Relative radius at considered section [-] = considered radius/R</td>
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<tr>
<td>$R$</td>
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</tr>
<tr>
<td>RPM</td>
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</tr>
<tr>
<td>$t$</td>
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</tr>
<tr>
<td>$t_{0.05c}$</td>
<td>Profile thickness at considered section, measured at a relative distance 5% from leading edge [mm]</td>
</tr>
<tr>
<td>$t_{0.95c}$</td>
<td>Profile thickness at considered section, measured at a relative distance 95% from leading edge [mm]</td>
</tr>
<tr>
<td>$t_{0.8,0.8}$</td>
<td>Profile thickness at 80% radius, measured at a relative distance 80% from leading edge [mm]</td>
</tr>
<tr>
<td>$t_c$</td>
<td>Profile thickness at considered section [mm]</td>
</tr>
<tr>
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<tr>
<td>$U_2$</td>
<td>Relative reduction of fatigue strength with increasing mean stress [-], See Rules for Classification of Ships or HSLC, Pt.4 Ch.5 Sec.1 Table B1</td>
</tr>
<tr>
<td>$V$</td>
<td>maximum ship speed [m/s], corresponding to $P$</td>
</tr>
<tr>
<td>$Δw$</td>
<td>effective wake variation [-]</td>
</tr>
<tr>
<td>$Z$</td>
<td>Number of blades [-]</td>
</tr>
<tr>
<td>$π$</td>
<td>= 3.1416</td>
</tr>
<tr>
<td>$α$</td>
<td>effective inflow angle [deg.]</td>
</tr>
<tr>
<td>$θ$</td>
<td>Pitch angle at root section [deg.] (tunnel thrusters), see 4.3 (2.7.2)</td>
</tr>
<tr>
<td>$θ_a$</td>
<td>Pitch angle at considered section [deg.], see 2.7.2</td>
</tr>
<tr>
<td>$ρ$</td>
<td>Density of water [kg/m³], = 1025 for sea water</td>
</tr>
<tr>
<td>$σ_{0.8}$</td>
<td>Peak stress [N/mm²] at 80% radius, see 3.2</td>
</tr>
<tr>
<td>$σ_A$</td>
<td>Dynamic stress amplitude [N/mm²]</td>
</tr>
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<td>$σ_{all}$</td>
<td>Allowable stress [N/mm²], see 4.1 (Tunnel thrusters)</td>
</tr>
<tr>
<td>$σ_m$</td>
<td>Mean stress [N/mm²], see 2.5</td>
</tr>
<tr>
<td>$σ_y$</td>
<td>Specified minimum yield strength [N/mm²], see Rules for Classification of Ships or HSLC, Pt.4 Ch.5 Sec.1 Table B1.</td>
</tr>
</tbody>
</table>
2. Calculation of High Cycle Stresses in Propeller Blades

2.1 High cycle stress criterion

Dynamic stress amplitudes in the propeller blade are to fulfil the following criterion:

\[ S \leq \frac{U}{\sigma_A} \]

\( \sigma_A \) = 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:

\[ S \leq \frac{U K_{\text{str}} - U_1 \sigma_{m}}{\sigma_m K_{\text{thick}} M_t} \]

The formula is based on a fatigue diagram where:

\( S \) = safety factor (-), see Rules for Classification of Ships or HSLC Pt.4 Ch.5 Sec.1, Table B2
\( U_1 \) = fatigue strength amplitude (N/mm²) at zero mean stress (>10⁸ cycles), see Rules for Classification of Ships, Pt.4 Ch.5 Sec.1 Table B1
\( K_{\text{str}} \) = Load correction factor (-), see 2.2
\( K_{\text{thick}} \) = correction factor (-) for influence of thickness on fatigue strength, see 2.3
\( U_2 \) = relative reduction of fatigue strength with increasing mean stress (>10⁸ cycles), see Rules for Classification of Ships, Pt.4 Ch.5 Sec.1 Table B1
\( M_t \) = fluctuating load relative to mean load (-), see 2.4
\( \sigma_{m} \) = actual mean stress (N/mm²), see 2.5.

Allowable local stresses in fillets etc. are given in 5.1.

2.2 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, \( K_{\text{str}} \), which is to be taken according to the following empirical formula, unless otherwise is substantiated:

\[ K_{\text{str}} = 1.3 - 0.25 K_{sk} \]

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

Where \( K_{sk} \) is skew correction factor at considered section, see 2.6.

2.3 Correction factor for influence of thickness

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

Stainless steels:

\[ K_{\text{thick}} = 1.0 - 0.05 \ln \left( \frac{t}{25} \right) \]

Other materials:

\[ K_{\text{thick}} = 1.0 - 0.1 \ln \left( \frac{t}{25} \right) \]

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

\( K_{\text{thick}} \) is not to be taken higher than 1.0.

2.4 Fluctuating blade load relative to mean load

\( M_t \) due to hydrodynamic loads is to be taken from the following empirical relation, unless otherwise substantiated:

\[ M_t = \frac{Z^{0.74}}{10K_n D} Aw \]

Unless all significant low cycle load conditions are considered additionally (see also 3.1), \( M_t \) is not to be taken less than 0.50.

\( M_t \) 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 $M_t$ based on special consideration.

### 2.4.1 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, $\Delta w$ 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

$\alpha = \text{not to be taken less 7 deg. for azimuthing thrusters.}$

Otherwise, $\alpha$ is not to be taken less than 5 deg.

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

$$\Delta w = 0.4$$

All other propellers:

$$\Delta w = 0.5$$

**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.

---end---of---N-o-t-e---

### 2.5 Mean stress

Actual mean stresses in cylindrical sections at root and at 60% radius are to be found from empirically modified cantilever-beam theory as described in 2.5 to 2.8. The following applies for the calculation of actual mean stress:

$$\sigma_m = K_{sk} K_{n} K_{corr} \frac{\rho n^2 D^5}{K e C_l t_r^2}$$

### 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:

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

Where:

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

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

$e_{0.6} = \text{distance (m) between skew line (mid-chord line) and generatrix at the indexed radius, measured along the cylindrical section. Note that e is positive when skew line is forward of generatrix (see Fig. 2-1).}$

At 60% radius:

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

Where:

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

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

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

$r_{\text{root}} = \text{relative radius (as a fraction of propeller radius) at root section, (see 1 Basic principles).}$
2.7 Bending moment coefficient
Simplifying bending axis to be the chord line of each section, bending moment coefficient is found from:

\[ K_m = \frac{K_{bm}}{Z} \left( \frac{K_Q}{0.35} \sin \theta_r + K_i \cos \theta_r \right) \]

2.7.1 Moment arm factor
Moment arm factor for forces acting outside of section, \( K_{bm} \) is to be estimated from:

\[ K_{bm} = 0.35 r^2 - 0.73 r + 0.38 \]

2.7.2 Pitch angle
Pitch angle (\( \theta_r \)) of section at relative radius \( r \), is to be found from:

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

2.7.3 Thrust coefficient
Thrust coefficient, \( K_T \) is to be found from:

\[ K_T = \frac{Th}{\rho n_s^2 D^2} \]

\( Th \) = propeller thrust (N) corresponding to \( P \) and \( n_s \).
\( \rho \) = 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, \( h_m \) is to be taken from the following relation:

\[ h_m = 0.096 h_{root} + 0.666 h_{0.7} + 0.238 h_{1.0} \]

\( hr \) = pitch ratio (-) at relative radius \( r \), \( = Hr/D \)
\( Hr \) = pitch (m) of cylindrical section at relative radius \( r \).
2.7.4 Torque coefficient
Torque coefficient, $K_Q$ is to be found from:

$$K_Q = \frac{P}{2\pi \rho n_i D^2}$$

$P$ = maximum continuous power for which the installation is to be approved (W).

2.8 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
$t_{0.95c}$ 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 2.8.1.

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

Camber correction factor, $K_f$ is to be taken as 1.0 if $K_{sk}$ (see 2.6) is larger than 1.3 at the section in question. If $K_{sk}$ is not larger than 1.1, $K_f$ is to be estimated from:

$$K_f = 1.35 - 0.7 \left[ 0.5 - \frac{f}{t_r} \right]$$

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

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

2.9 Centrifugal stress
Centrifugal stress needs normally not be taken into account. However, for highly raked propeller blades and/or propellers with a tip speed exceeding 50 m/s, the term $U_2 \sigma_m$ in 2.1 needs to be corrected for centrifugal stress.

3. Calculation of Low Cycle Stresses in Propeller Blades

3.1 Low cycle stress criterion
Propellers for which the turning direction may be reversed are to be additionally checked for margins towards blade bending in astern operation (3.1 and 3.2). 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 Rules for Classification of Ships or HSLC Pt.4 Ch.5 Sec.1 Table B2

$\sigma_{0.8}$ = peak stress (N/mm²) at 80% radius, see 3.2

$K_{thick}$ = correction factor (-) for influence of thickness on bending strength of the section in question. $K_{thick}$ is to be taken as 1.0 for stainless steels, and as described for fatigue strength (see 2.3) for other materials on basis of actual thickness at 80% chord length at a relative radius of 80%

$\sigma_y$ = specified minimum yield strength (N/mm²), see Rules for Classification of Ships or HSLC, Pt.4 Ch.5 Sec.1 Table B1.

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

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

$K_{Tha}$ = thrust coefficient (-) at maximum bollard astern power, see 3.2.1
$K_{BA}$ = astern bending moment coefficient (-), see 3.2.2
$C_{0.8}$ = 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 3.2.3
$n_{cav}$ = critical astern revolutions (1/s = RPM/60), above which cavitation is expected to influence stress level, see 3.2.5.

3.2.1 Astern thrust coefficient
Thrust coefficient at maximum bollard astern power is to be found from:

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

3.2.2 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:

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

where the index refers to relative radius.

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

$$n_{BA} = \frac{C_{QA}}{2\pi \rho k_{QA} n_\rho D^5}$$

$C_{QA}$ = 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 3.2.4.
$n_{BA}$ need not be taken higher than $n_s$.

3.2.4 Astern torque coefficient
The following empirical expression is to be used for calculation of torque coefficient at maximum bollard astern power:

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

3.2.5 Critical astern revolutions
A stern number of revolutions above which cavitation is expected to have influence on stress level, is to be estimated from the following empirical expression:

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

$hs$ = 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.

$n_{cav}$ is not to be taken higher than $n_{BA}$. 

3.2.6 Other low cycle dynamic stresses
Other low cycle dynamic stresses (see Rules for Classification of Ships or HSCLC, Pt.4 Ch.5 Sec.1 B201) are to be specially considered when applicable. This normally requires detailed information regarding expected load profile as well as detailed stress calculations.

4. Simplified Criteria for Propeller Blades Working in a Tunnel

4.1 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.

The following criterion applies:

\[ S \geq \frac{\sigma_{\text{All}} K_{\text{thick}}}{\sigma_{m}} \]

- \( S \) = safety factor, see Rules for Classification of Ships or HSCLC, Pt.4 Ch.5 Sec.1 Table B2
- \( \sigma_{m} \) = mean stress in blade root section, see 4.2
- \( \sigma_{\text{All}} \) = allowable stress, to be taken as specified minimum yield strength, or 50% of specified minimum tensile strength, whichever is the least (see Rules for Classification of Ships or HSCLC, Pt.4 Ch.5 Sec.1 Table B1)
- \( K_{\text{thick}} \) = correction factor for influence of thickness on bending strength of the section in question. \( K_{\text{thick}} = 1.0 \) for stainless steels, and as described for fatigue strength in 2.3 for other materials.

Allowable local stresses in fillets etc. are given in 5.1.

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

\[ \sigma_{m} = \frac{M_{\text{th}} + M_{q}}{0.09 C t^2} \]

- \( M_{\text{th}} \) = bending moment due to propeller thrust (Nm), see 4.3
- \( M_{q} \) = bending moment due to propeller torque (Nm), see 4.4
- \( C \) = 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 \( C t^2 \).

4.3 Bending moment due to propeller thrust
Bending moment due to propeller thrust is found from:

\[ M_{\text{th}} = \frac{D Th}{2Z} (r_{1} - r_{\text{root}}) \cos \theta \]

- \( D \) = propeller diameter (m)
- \( Th \) = propeller thrust (N). \( Th \) may be taken as 0.12\( \cdot P \), if not known
- \( P \) = maximum engine power (W) for which the installation is to be approved
- \( Z \) = number of blades
- \( r_{1} \) = 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
- \( r_{\text{root}} \) = relative radius at root section (-)
- \( \theta \) = pitch angle (deg.) at root section (see 2.7.2 for definition).

4.4 Bending moment due to propeller torque
Bending moment due to propeller torque is found from:

\[ M_{q} = \frac{D F_{qf}}{2Z} (r_{1} - r_{\text{root}}) \sin \theta \]

- \( F_{qf} \) = torque induced force (N), see 4.4.1.
4.5 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).

4.6 Profile thickness at 60% radius
At the 60% radius, the profile is not to be less than derived from the following expression:

\[
t_{0.6} = 0.7 t \sqrt{C_{\text{ratio}} K_{\text{thick}}}
\]

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

5. Fillets and Tip Thickness of Propeller Blade

5.1 Blade root fillets
The local stresses in the blade root fillets are not to 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 are to be considered, allowing local stress levels as for the blade root fillets.

5.2 Tip thickness

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

\[t_{0.6} = \text{required profile thickness at 60\% radius. If the section at 60\% radius fulfils the high cycle criterion in 2.1, } t_{0.6} \text{ may be taken as actual profile thickness at 60\% radius. In general } t_{0.6} \text{ may be derived from 2.5, as the minimum thickness for which the high cycle stress criterion is fulfilled.}

\[t_{0.9} = 2.25 D + 0.25 t_{0.6}\]

6. Dynamic Loading on Propeller Hub and Pitch Mechanism

6.1 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 number 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.
6.2 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 number of cycles are normally considered as realistic:

- propellers on ships where propeller pitch is used as load control system of prime mover (See E102): \(10^{10}\) cycles
- propellers included in DYNPOS system: \(10^8\) cycles
- other propellers: \(10^7\) cycles.

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6.3 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:

\[0.2 \cdot M_t \cdot C_{0.6} \cdot F_{\text{mean}} \quad \text{(kNm)}\]

Where:

- \(M_t\) = fluctuating blade load relative mean load, see 2.4
- \(C_{0.6}\) = width of expanded cylindrical propeller blade section at 60% radius (m), see also 2.5.
- \(F_{\text{mean}}\) 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 6.2).

**Guidance note:**
The following number of cycles are normally considered as realistic:

- propellers on ships where propeller pitch is used as load control system of prime mover (See E102): \(10^{10}\) cycles
- other propellers: \(10^8\) cycles.

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