M68
(Feb 2005) (Corr.1 Mar 2012) (Corr.2 Nov 2012) (Rev.1 Aug 2014)

Dimensions of propulsion shafts and their permissible torsional vibration stresses

M68.1 Scope

This UR applies to propulsion shafts such as intermediate and propeller shafts of traditional straight forged design and which are driven by rotating machines such as diesel engines, turbines or electric motors.

For shafts that are integral to equipment, such as for gear boxes, podded drives, electrical motors and/or generators, thrusters, turbines and which in general incorporate particular design features, additional criteria in relation to acceptable dimensions have to be taken into account. For the shafts in such equipment, the requirements of this UR may only be applied for shafts subject mainly to torsion and having traditional design features. Other limitations, such as design for stiffness, high temperature, etc. are to be addressed by specific rules of the classification society.

Explicitly the following applications are not covered by this UR:

- additional strengthening for shafts in ships classed for navigation in ice
- gearing shafts
- electric motor shafts
- generator rotor shafts
- turbine rotor shafts
- diesel engine crankshafts (see M53)
- unprotected shafts exposed to sea water

M68.2 Alternative calculation methods

Alternative calculation methods may be considered by the classification society. Any alternative calculation method is to include all relevant loads on the complete dynamic shafting system under all permissible operating conditions. Consideration is to be given to the dimensions and arrangements of all shaft connections.

Moreover, an alternative calculation method is to take into account design criteria for continuous and transient operating loads (dimensioning for fatigue strength) and for peak operating loads (dimensioning for yield strength). The fatigue strength analysis may be carried out separately for different load assumptions, for example as given in M68.7.1.

Notes:

1. This UR M 68 replaces URs M33, M37, M38, M39 and M48.

2. This UR M 68 applies to ships contracted for construction on or after 1 July 2006.

3. The “contracted for construction” date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of “contracted for construction”, refer to IACS Procedural Requirement (PR) No.29.

4. Rev.1 of UR M68 applies to ships contracted for construction on or after 1 July 2015.

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M68.3 Material limitations

Where shafts may experience vibratory stresses close to the permissible stresses for transient operation, the materials are to have a specified minimum ultimate tensile strength (σ_B) of 500 N/mm^2. Otherwise materials having a specified minimum ultimate tensile strength (σ_B) of 400 N/mm^2 may be used.

For use in the following formulae in this UR, σ_B is limited as follows:

- For carbon and carbon manganese steels, a minimum specified tensile strength not exceeding 600 N/mm^2 for use in M68.5 and not exceeding 760 N/mm^2 in M68.4.

- For alloy steels, a minimum specified tensile strength not exceeding 800 N/mm^2.

- For propeller shafts in general a minimum specified tensile strength not exceeding 600 N/mm^2 (for carbon, carbon manganese and alloy steels).

Where materials with greater specified or actual tensile strengths than the limitations given above are used, reduced shaft dimensions or higher permissible vibration stresses are not acceptable when derived from the formulae in this UR.

M68.4 Shaft diameters

Shaft diameters are not to be less than that determined from the following formula:

$$d = F.k.\sqrt[3]{\frac{p}{n_0}.\frac{1}{1-\frac{d_i^4}{d_o^4}}.\frac{560}{\sigma_B + 160}}$$

where:

d = minimum required diameter in mm

d_i = actual diameter in mm of shaft bore

d_o = outside diameter in mm of shaft. If the bore of the shaft is ≤0.40d_o, the expression $1-\frac{d_i^4}{d_o^4}$ may be taken as 1.0

F = factor for type of propulsion installation

= 95 for intermediate shafts in turbine installation, diesel installations with hydraulic (slip type) couplings, electric propulsion installations

= 100 for all other diesel installations and all propeller shafts

k = factor for the particular shaft design features, see M68.6

n_0 = speed in revolutions per minute of shaft at rated power

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p = rated power in kW transmitted through the shaft (losses in gearboxes and bearings are to be disregarded)

σ_B = specified minimum tensile strength in N/mm^2 of the shaft material, see M68.3

The diameter of the propeller shaft located forward of the inboard stern tube seal may be gradually reduced to the corresponding diameter required for the intermediate shaft using the minimum specified tensile strength of the propeller shaft in the formula and recognising any limitations given in M68.3.

M68.5 Permissible torsional vibration stresses

The alternating torsional stress amplitude is understood as $(\tau_{\max} - \tau_{\min})/2$ as can be measured on a shaft in a relevant condition over a repetitive cycle.

Torsional vibration calculations are to 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 are not to exceed the values given by the following formulae:

$$\pm\tau_C = \frac{\sigma_B + 160}{18}.C_K.C_D.(3 - 2.\lambda^2) \qquad \text{for } \lambda < 0.9$$

$$\pm\tau_C = \frac{\sigma_B + 160}{18}.C_K.C_D.1.38 \qquad \text{for } 0.9 \le \lambda < 1.05$$

where:

τ_C = permissible stress amplitude in N/mm^2 due to torsional vibration for continuous operation

σ_B = specified minimum ultimate tensile strength in N/mm^2 of the shaft material, see also M68.3

c_K = factor for the particular shaft design features, see M68.6

c_D = size factor

= $0.35 + 0.93 d_o^{-0.2}$

d_o = shaft outside diameter in mm

λ = speed ratio = n/n_0

n = speed in revolutions per minute under consideration

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n_0 = speed in revolutions per minute of shaft at rated power

Where the stress amplitudes exceed the limiting values of τ_C for continuous operation, including one cylinder misfiring conditions if intended to be continuously operated under such conditions, restricted speed ranges are to be imposed which are to be passed through rapidly.

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 are to enable safe navigation.

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

(a) The barred speed range is to cover all speeds where the acceptance limits (τ_C) are exceeded. For controllable pitch propellers with the possibility of individual pitch and speed control, both full and zero pitch conditions have to be considered. Additionally the tachometer tolerance has to be added. At each end of the barred speed range the engine is to be stable in operation.

(b) In general and subject to (a) the following formula may be applied, provided that the stress amplitudes at the border of the barred speed range are less than τ_C under normal and stable operating conditions.

$$\frac{16.n_c}{18-\lambda_c} \leq n \leq \frac{(18-\lambda_c).n_c}{16}$$

where:

n_C = critical speed in revolutions per minute (resonance speed)

λ_c = speed ratio = n_c / n_o

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

$$\pm\tau_T = 1.7.\frac{\tau_c}{\sqrt{c_K}}$$

where:

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

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M68.6 Table of k and c_k factors for different design features (see M68.7.2)

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 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

Note:

Transitions of diameters are to be designed with either a smooth taper or a blending radius. For guidance, a blending radius equal to the change in diameter is recommended.

Footnotes

1) Fillet radius is not to be less than 0.08d.

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 is to 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 $0.2d_o$ from the end of the keyway the shaft diameter may be reduced to the diameter calculated with k=1.0.

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

5) Diameter of radial bore (d_h) not to exceed $0.3d_o$.
The intersection between a radial and an eccentric (r_{ec}) axial bore (see below) is not covered by this UR.

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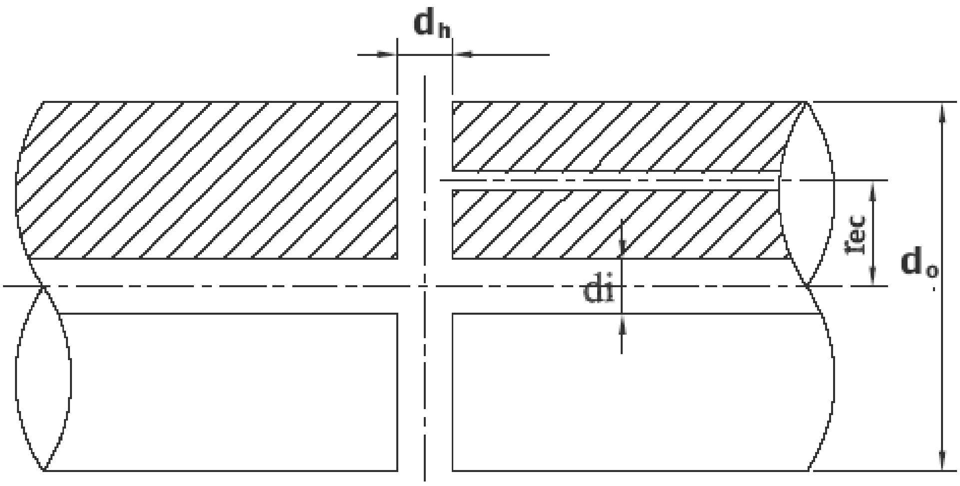

6) Subject to limitations as slot length (l)/outside diameter < 0.8 and inner diameter (d_i)/outside diameter < ~~0.8~~ 0.7 and slot width (e)/outside diameter > ~~0.10~~ 0.15. The end rounding of the slot is not to be less than e/2. An edge rounding should preferably be avoided as this increases the 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 respectively 120 degrees apart.

7) ~~c_K = 0.3 is a safe approximation within the limitations in 6). If the slot dimensions are outside of the above limitations, or if the use of another c_K is desired, the actual stress concentration factor (scf) is to be documented or determined from M68.7.3. In which case:~~

c_K = 0.3 is an approximation within the limitations in 6). More accurate estimate of the stress concentration factor (scf) may be determined from M68.7.3 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 √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.

M68.7 Notes

1. Shafts complying with this UR satisfy the following:

1. 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 M68.4.

2. High cycle fatigue criterion (typically >> 10^7), i.e. torsional vibration stresses permitted for continuous operation as well as reverse bending stresses.
The limits for torsional vibration stresses are given in M68.5.
The influence of reverse bending stresses is addressed by the safety margins inherent in the formula in M68.4.

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3. 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 in M68.5.

2. 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 $0.08d_o$ (geometric stress concentration of approximately 1.45).

$$C_K = \frac{1.45}{scf} \quad \text{and} \quad k = \left[\frac{scf}{1.45}\right]^x$$

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.

3. 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):

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

This formula applies to:

- slots at 120 or 180 or 360 degrees apart.

- slots with semicircular 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 as:

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

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

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