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Small craft — Hull construction and scantlings —

Part 8: Rudders

Petits navires — Construction de coques et échantillonnage —

Partie 8: Gouvernails

Reference number
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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 12215-8 was prepared by Technical Committee ISO/TC 188, *Small craft*.

ISO 12215 consists of the following parts, under the general title *Small craft — Hull construction and scantlings*:

— *Part 1: Materials: Thermosetting resins, glass-fibre reinforcement, reference laminate*

— *Part 2: Materials: Core materials for sandwich construction, embedded materials*

— *Part 3: Materials: Steel, aluminium alloys, wood, other materials*

— *Part 4: Workshop and manufacturing*

— *Part 5: Design pressures for monohulls, design stresses, scantlings determination*

— *Part 6: Structural arrangements and details*

— *Part 8: Rudders*

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Introduction

The reason underlying the preparation of this part of ISO 12215 is that standards and recommended practices for loads on the hull and the dimensioning of small craft differ considerably, thus limiting the general worldwide acceptability of craft. This part of ISO 12215 has been set towards the lower boundary range of common practice.

The objective of this part of ISO 12215 is to achieve an overall structural strength that ensures the watertight and weathertight integrity of the craft.

The working group considers this part of ISO 12215 to have been developed applying present practice and sound engineering principles. The design loads and criteria of this part of ISO 12215 may be used with the scantling determination equations of this part of ISO 12215 or using equivalent engineering methods such as continuous beam theory, matrix-displacement method and classical lamination theory, as indicated within.

Considering future development in technology and craft types, and small craft presently outside the scope of this part of ISO 12215, provided that methods supported by appropriate technology exist, consideration may be given to their use as long as equivalent strength to this part of ISO 12215 is achieved.

The dimensioning according to this part of ISO 12215 is regarded as reflecting current practice, provided the craft is correctly handled in the sense of good seamanship and equipped and operated at a speed appropriate to the prevailing sea state.

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Small craft — Hull construction and scantlings —

Part 8: **Rudders**

1 Scope

This part of ISO 12215 gives requirements on the scantlings of rudders fitted to small craft with a length of hull, L_H, of up to 24 m, measured according to ISO 8666. It applies only to monohulls.

This part of ISO 12215 does not give requirements on rudder characteristics required for proper steering capabilities.

This part of ISO 12215 only considers pressure loads on the rudder due to craft manoeuvring. Loads on the rudder or its skeg, where fitted, induced by grounding or docking, where relevant, are out of scope and need to be considered separately.

NOTE Scantlings derived from this part of ISO 12215 are primarily intended to apply to recreational craft including charter craft.

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 8666, *Small craft — Principal data*

ISO 12215-5:2008, *Small craft — Hull construction and scantlings — Part 5: Design pressures for monohulls, design stresses, scantlings determination*

3 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

3.1
design categories
sea and wind conditions for which a craft is assessed by this part of ISO 12215 to be suitable, provided the craft is correctly handled in the sense of good seamanship and operated at a speed appropriate to the prevailing sea state

3.1.1
design category A ("ocean")
category of craft considered suitable to operate in seas with significant wave heights above 4 m and wind speeds in excess of Beaufort Force 8, but excluding abnormal conditions such as hurricanes

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3.1.2
design category B ("offshore")
category of craft considered suitable to operate in seas with significant wave heights up to 4 m and winds of Beaufort Force 8 or less

3.1.3
design category C ("inshore")
category of craft considered suitable to operate in seas with significant wave heights up to 2 m and a typical steady wind force of Beaufort Force 6 or less

3.1.4
design category D ("sheltered waters")
category of craft considered suitable to operate in waters with significant wave heights up to and including 0,3 m with occasional waves of 0,5 m height, for example from passing vessels, and a typical steady wind force of Beaufort Force 4 or less

3.2
loaded displacement mass
m_{LDC}
mass of the craft, including all appendages, when in the fully loaded ready-for-use condition as defined in ISO 8666

3.3
sailing craft
craft for which the primary means of propulsion is wind power, having $A_{\text{S}} > 0{,}07(m_{\text{LDC}})^{2/3}$ where A_{S} is the total profile area of all sails that may be set at one time when sailing closed hauled, as defined in ISO 8666 and expressed in square metres

NOTE 1 For the headsails, A_{S} is the area of the fore triangle.

NOTE 2 In the rest of this part of ISO 12215, non-sailing craft are called motor craft.

4 Symbols

For the purposes of this document, unless specifically otherwise defined, the symbols given in Table 1 apply.

NOTE The symbols used in the annexes are not listed in Table 1.

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Table 1 — Symbols, coefficients, parameters

Symbol	Unit	Designation/meaning of symbol	(Sub)clause/table concerned
A	m^2	Total area of the moving part of the rudder	6.2.1, 6.2.3
A_0	m^2	Rudder effective area (Types II to IV)	6.2.3
A_1	m^2	Rudder blade area (Types II to IV) or top blade area (Type V)	6.2.3
A_2	m^2	Bottom rudder blade area (Type V)	6.2.3
A_3	m^2	Rudder skeg area [only used to determine type (see Figure 3)]	6.2.3
c	m	Rudder chord length at centre of area level	6.2.1, 6.2.2
c_1	m	Length of the top chord (Type I)	6.2.1
c_2	m	Length of the bottom chord (Type I)	6.2.1
co_1	m	Compensation at top chord (distance from LE to rotation axis) (Type I)	6.2.2
co_2	m	Compensation at bottom chord (distance from LE to stock CL) (Type I)	6.2.2
d	mm	Required solid stock diameter	10.4
d_i	mm	Inner diameter of tubular stock	10.6
d_o	mm	Outer diameter of tubular stock	10.6
F	N	Final side force on rudder	7.1
F_1	N	Side force on rudder in design category sea state	7.2
F_2	N	Side force on rudder during a turn at speed in slight sea	7.3
h_b	m	Height between rudder top and centre of hull bearing	6.2.1
h_c	m	Height between rudder top and centre of area	6.2.1
h_d	m	Height between rudder top and centre of skeg bearing (Type V)	6.2.3
h_e	m	Height between rudder bottom and centre of skeg bearing (Type V)	6.2.3
h_{in}	m	Height between centre of upper bearing and a point inside the hull (Type I)	6.2.1
h_{ou}	m	Height between bottom of spade and a point outside the hull (Type I)	6.2.1
h_r	m	Average height of rudder blade (see Figure 1)	6.2.1
h_s	m	Height of skeg from hull attachment to skeg bearing (Types II to V)	6.2.3
h_u	m	Height between centres of hull (lower) bearing and upper bearing	6.2.1
k_b	1	Rudder bending coefficient	6.2.1
k_{FLAT}	1	Coefficient lowering force for flat or wedge rudder blade shape	7.3
k_{GAP}	1	Coefficient lowering force due to gap hull/rudder top	7.2
k_{LD}	1	Length displacement coefficient	7.2
k_S	1	Coefficient for skeg deflection	8.3.4
k_{SEA}	1	Coefficient considering extra load due to sea in design categories A and B	7.2
k_{SERV}	1	Coefficient considering lower required service in design categories C and D	7.3
k_{SIG}	1	Coefficient lowering design stress for F_2	7.3
k_{USE}	1	Coefficient considering lower usage of craft with damage survey	7.2
k_5	1	Fibre type factor	13.3.1.2
L_S	m	Effective length of the skeg	8.3.4

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Table 1 (*continued*)

Symbol	Unit	Designation/meaning of symbol	(Sub)clause/table concerned
L_{WL}	m	Length at waterline, according to ISO 8666 in m_{LDC} conditions	7.2
M	Nm	Bending moments on the rudder stock or skeg	8
M_S, M_H	Nm	Bending moments at skeg or hull	8.3.4
m_{LDC}	kg	Loaded displacement mass	3.2, 7.2
r	m	Horizontal distance from rudder force to stock axis	6.2.1
r_{min}	m	Minimum value of r	9
R_U, R_H, R_S	N	Reaction force at upper bearing, hull bearing, skeg bearing, respectively	8
t	mm	Skin thickness of tubular or hollowed closed section	Table 6
T	Nm	Torque (twisting moment) on the rudder stock	9
u	m	Longitudinal distance from leading edge to stock axis at centroid chord	6.2.1
V_{MAX}	knots	Maximum speed of craft in calm water, m_{LDC} conditions	7.3
w	kg/m^2	Minimum fibre mass per area of rudder blade	13.3.1.2
z_b	m	Effective bending moment lever $z_b = k_b \cdot h_r + h_c$	8.2.1
z_{eq}	m	Equivalent bending moment lever	10.4
α	1	Tip chord to root chord ratio (c_2/c_1)	6.2.2
Λ	1	Geometric aspect ratio of the rudder	6.2.1, 6.2.3
σ	N/mm^2	Direct stress (ultimate, yield, design)	5
τ	N/mm^2	Shear stress (ultimate, yield, design)	5
χ	1	Ratio between reaction at skeg and rudder force	8.3.2

5 Design stresses

5.1 Rudder material

Values of design stresses shall be taken from Table 2

Table 2 — Values of design stresses

Stresses in newtons per square millimetre

Material	Direct stresses			Combined stresses
	Tensile/compressive σ_d	Shear τ_d	Bearing σ_{db}	
Metals[a]	$\min(\sigma_y ; 0,5\ \sigma_u)$	0,58 τ_d	1,8 σ_d	$\sqrt{\sigma^2 + 3\tau^2} \leqslant \sigma_d$
Wood and fibre-reinforced plastics (FRP)	$0,5 \times \sigma_u$	0,5 τ_u	1,8 σ_d	$\left(\frac{\sigma}{\sigma_u}\right)^2 + \left(\frac{\tau}{\tau_u}\right)^2 < 0,25$

[a] Steel, stainless steel, aluminium alloys, titanium alloys, copper alloys (see Annex A). In welded condition for welded metals.

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In Table 2,

— σ_d is the design tensile, compressive, or flexural strength (as relevant);

— σ_u is the ultimate tensile, compressive, or flexural strength (as relevant);

— σ_y is the yield tensile, compressive, or flexural strength (as relevant);

— σ_{db} is the design bearing strength;

— τ_d is the design shear strength;

— τ_u is the ultimate shear strength.

Additional requirements are given in Annex A (metals) and Annex B (composites)

For wood and composites, the strength values of the relevant annexes of ISO 12215-5 shall be used.

6 Rudder and steering arrangement, rudder types

6.1 General

6.1.1 General definition

The rudder and steering arrangement comprises all components necessary for manoeuvring the craft, from the rudder and the rudder operating gear to the steering position.

Rudder and steering equipment shall be arranged so as to permit inspection.

NOTE It is good practice that the rudder keeps the steering effect after grounding (for example, a spade rudder with the stock not going down to the bottom enables the rudder blade to break without bending the stock).

6.1.2 Multi-rudder arrangement

If the craft has several rudders, the following requirements apply to each one of the rudders.

NOTE On sailing craft, twin rudders, frequently canted outwards, are not usually protected from contact with floating objects by the keel, a skeg, the hull canoe body at centreline, etc. This is particularly the case for the windward rudder, close to the waterline, that can also be hit by breaking waves and can therefore support a part of the craft's weight. It is therefore current practice to have twin rudders installed on sailing craft that are significantly stronger than required in this part of ISO 12215, which only considers loads from normal lift forces. This enhanced strength is not quantified here.

6.1.3 Vertical support

The rudder stock or blade shall be supported vertically with limited axial upwards movement.

6.1.4 Hard over stops

Rudder stocks that are, or can be, actuated by a remote steering system (i.e. not directly by the tiller) shall be fitted with hard over stops, angled at 30° to 45° from zero lift position (usually at centreline). This also applies to rudders only actuated by a tiller of design category A and B.

Hard over stops can act on the rudder, the tiller, the quadrant, or any device directly connected to the rudder.

NOTE The need for stops is both to avoid excessive angle of attack and lift when running backwards and to avoid excessive range of movement of the steering system.

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6.1.5 Actuating system of the rudder

The following devices shall be able to transmit the rudder torque, T, defined in Clause 9, without exceeding their design stress, as defined in Clause 5:

— the actuating device that turns the rudder including the tiller, rudder arm and quadrant;

— the connection between the rudder stock and the actuating device (cone, square, key, etc.);

— the stops provided at either end of the tiller, rudder arm or quadrant stroke.

The connection between the rudder stock and the actuating device shall be designed to ensure alignment between the rudder blade and the tiller, actuating arm, etc. and allow a visual instant checking of this alignment.

6.1.6 Emergency tiller

Any component of the emergency tiller, where fitted, shall be able to transmit a rudder torque of 0,5 T, where T is defined in Clause 9, without exceeding its design stress defined in Clause 5.

6.2 Rudder types

This part of ISO 12215 is applicable to five types of rudder configuration: Type I to Type V, as shown in Figures 2 and 3. In all cases except case I c, the rudder blade is taken as rectangular or trapezoidal.

6.2.1 Type I (spade) rudders (see Figures 1 and 2)

The main variables are as follows:

— A is the rudder (spade) area;

— $\Lambda = \dfrac{h_r^2}{A}$ is the rudder geometric aspect ratio (1)

where h_r is the average height of the rudder;

— h_b is the height between rudder top and centre of hull bearing;

— c_1 and c_2 are, respectively, the top and bottom chords or their natural extension;

— co_1 and co_2 are the top and bottom compensation, respectively, i.e. the distance, measured from fore to aft, between the leading edge and the rotation axis;

— c is the chord length at the height of the centroid of rudder area;

— h_c is the height between rudder top and centroid of rudder area (this is the position where the rudder force is considered to act);

— h_{ou} and h_{in} are, respectively, any local height outside and inside the centre of hull bearing to be used in Figure 5;

— k_b is the rudder bending coefficient with $k_b = h_c/h_r$;

— r is the horizontal distance between the position of the resultant of the rudder force (taken at rudder centroid) and the rudder's rotational axis, as defined in Table 6, and shall not be taken less than r_{min};

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— u is, for Type I (spade) rudders, the horizontal distance from fore to aft, from the leading edge to the rudder rotational axis at the height of centroid of rudder area (i.e. the geometric centre of the profile area); u is positive if the leading edge is forward of the axis (see Figure 2 Types I a, I b, or I c) or negative in the opposite case (see Type I d).

6.2.2 Rudder spade with trapezoidal shape

For spade rudders with a trapezoidal (or close to) shape some values are easily calculated as follows:

$$A = h_r \frac{c_1 + c_2}{2} \quad \text{is the area of a trapezoidal spade;} \tag{2}$$

$$k_b = \frac{h_c}{h_r} = \frac{1 + 2\,\alpha}{3\,(1+\alpha)} \quad \text{for a trapezoidal spade;} \tag{3}$$

where $\alpha = \dfrac{c_2}{c_1}$ is the taper coefficient.

See Table 3.

Table 3 — Calculated values of k_b for a trapezoidal spade as a function of c_2/c_1

$c_2/c_1 = \alpha$	1,00	0,90	0,80	0,70	0,60	0,50	0,40	0,30	0,20
k_b	0,50	0,49	0,48	0,47	0,46	0,44	0,43	0,41	0,39

$$h_c = k_b \times h_r \tag{4}$$

$$c = c_1 - k_b(c_1 - c_2) \quad \text{for a trapezoidal spade} \tag{5}$$

$$u = co_1 - k_b(co_1 - co_2) \quad \text{for a trapezoidal spade} \tag{6}$$

The value of h_c can also be determined graphically, as shown in Figure 1.

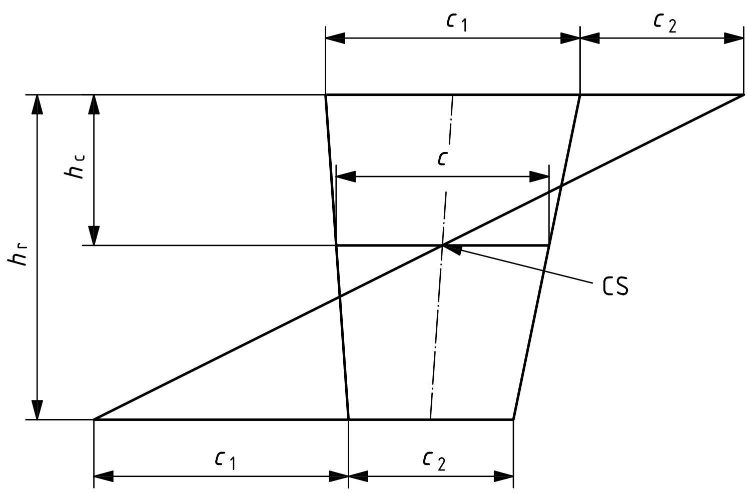

Figure 1 — Graphical determination of centroid, CS, of a trapeze

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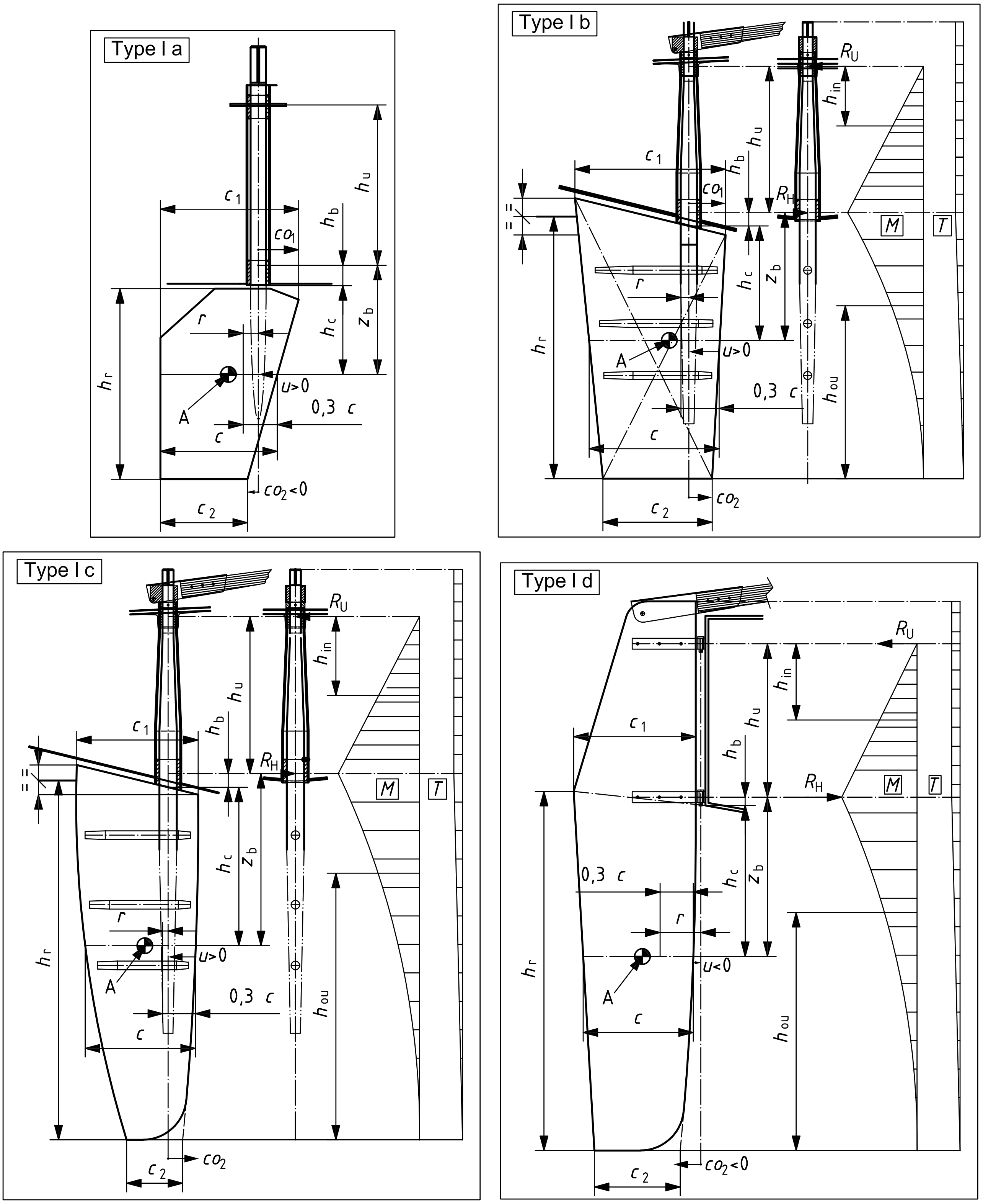

Type I a: Typical fast motor craft spade rudder with low aspect ratio and cut out top aft to avoid ventilation

Type I b: Near-rectangular shape

Type I c: Semi-elliptical shape typical on performance sailing craft

Type I d: Transom-hung spade rudder

NOTE The marking with a shaded circle shows the geometric centre of surface. The rudder force is located at the same height, but at a distance 0,3 c aft of the chord's leading edge.

Figure 2 — Spade rudders: Type I

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6.2.3 Rudder types II to V (see Figure 3)

The dimensions are the same as for spade rudders, except that:

— A is the total area of the moving part of the rudder, divided into A_1 and A_2 in Type V;

— A_3 is the skeg area (only used to determine the type in Figure 3);

— h_r is the average height of the rudder;

— $\Lambda = \frac{h_r^2}{A_0}$ is the effective rudder geometric aspect ratio (7)

where A_0 is the rudder effective area (moving part plus effective part of the skeg, see Table 4);

— $c = A_0/h_r$ is the mean chord;

— h_s is the height of the skeg/horn between hull and mid-skeg bearing for Type V and the lower bearing for Types III and IV.

Table 4 gives values of A and A_0 according to rudder type.

Table 4 — Rudder types and effective areas

Type	Value	
	A	A_0
II	A	
III	A_1	$A_1 + A_3$
IV	A_1	A_1
V	$A_1 + A_2$	$A_1 + A_2 + A_3$

For Type V, h_d and h_e are the portions of h_r above and below the skeg bearing, respectively.

For Types II to V:

— u is, for rudder Types II and IV, the horizontal distance, fore to aft, from the leading edge of the rudder to the stock vertical axis at the height of centroid of rudder area. For rudder Types III and V, u is measured aft of the leading edge of the partial or full narrow skeg (see Figure 3);

— r is the horizontal distance between the position of the centroid of rudder area and the rudder's rotational axis, as defined in Table 6, and shall not be taken less than r_{min}.

The rudders of Types II to V are considered to be held by three bearings (two bearings inside the hull and one skeg bearing, see 8.3.1)

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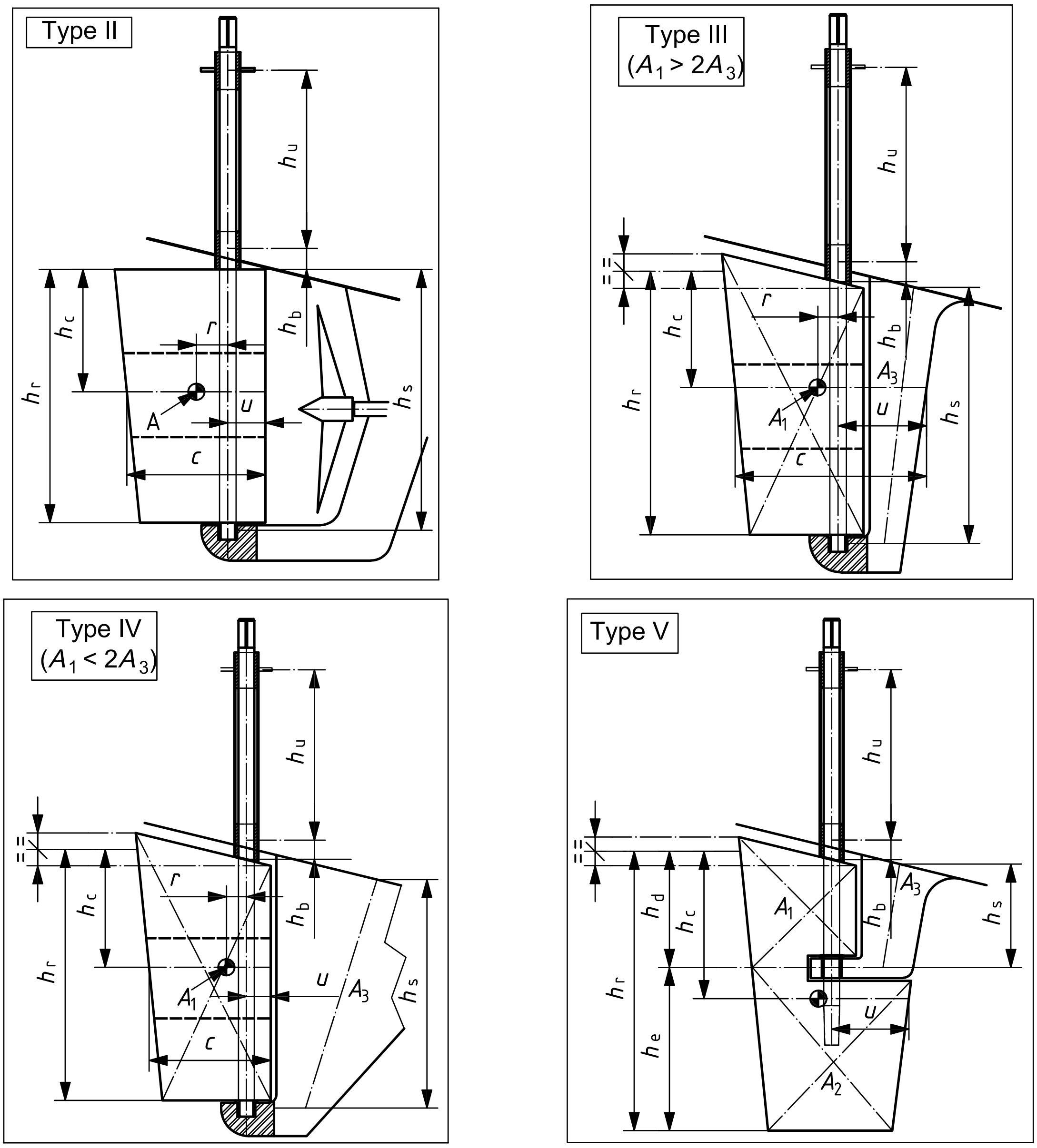

Type II Supported by skeg (solepiece) and skeg bearing

Type III Narrow full skeg

Type IV Wide full skeg

Type V Partial skeg

Figure 3 — Other rudder types: Types II to V

7 Design rudder force calculation

7.1 General

The design rudder force, F, shall be taken as follows:

— for motor craft, the greater of F_1 and F_2, defined in 7.2 and 7.3, respectively;

— for sailing craft, the force F_1, defined in 7.2.

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7.2 Force F_1 and corresponding load case

This case corresponds to loads associated with boat handling in the design category sea state.

$$F_1 = 23 \times L_{WL} \times k_{SEA} \times k_{LD}^2 \times k_{GAP} \times k_{USE} \times A \quad (8)$$

where

k_{SEA} =

- 1,4 for sailing craft of design categories A and B and motor craft of design category A,
- 1,2 for motor craft of design category B,
- 1,0 for craft of design categories C and D;

NOTE 1 k_{SEA} recognizes that in higher design categories the sea and related waves can induce higher lateral loads than in smooth water.

k_{LD} = 6,15 for motor craft of all design categories and sailing craft of design categories C and D;

for sailing craft of design categories A and B,

$$k_{LD} = \frac{L_{WL}}{\left(\dfrac{m_{LDC}}{1\,025}\right)^{1/3}} \quad (9)$$

but shall not be taken less than 6,15;

NOTE 2 k_{LD} recognizes that slender sailing craft can experience additional speeds due to surfing. It is derived from an established yacht rudder scantling guide that has been in use for many years.

k_{GAP} =

- 1,0 for rudders where the root gap (average clearance between the hull and the rudder root plane) is less than 5 % of the mean rudder chord. This gap shall not be exceeded at any rudder angle,
- 0,85 for rudders which are surface piercing (e.g. transom held) or exceed the gap limitation or can otherwise exhibit significant 3-D flow over the root;

NOTE 3 k_{GAP} recognizes that in general, 3-D flow over the root reduces rudder forces. Where there is doubt regarding the configuration under consideration, the conservative approach is to use k_{GAP} = 1.

k_{USE} = 1 for all craft but may be taken as 0,9 for category C and D sailing craft which are essentially used for close inshore racing with suitable safety procedures in place and for which the rudder can be easily inspected on a regular basis. If k_{USE} is taken as 0,9, a warning requiring regular inspection of rudder(s) should be included in the owner's manual.

NOTE 4 Rudder aspect ratio does not feature in the above formula since experimental evidence suggests that maximal rudder force is fairly insensitive to aspect ratio. The lift slope increases with increasing aspect ratio but the angle of maximal force reduces to maintain a sensibly constant rudder force coefficient.

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7.3 Force F_2 and corresponding load case

This case corresponds to loads connected with motor craft handling during a turn at speed in slight seas. It is therefore only applicable to motor craft.

$$F_2 = 370 \times \Lambda^{0,43} \times V_{MAX}^{1,3} \times k_{GAP} \times k_{SERV} \times k_{FLAT} \times k_{SIG} \times A \qquad (10)$$

where

Λ is the geometric aspect ratio defined in Equation (1) or (7);

V_{MAX} is the craft maximum speed in calm water and m_{LDC} conditions;

k_{GAP} is as given in 7.2;

k_{SERV} =

- 1,0 for design category A and B craft,
- 0,8 for design category C and D craft (may also be taken as 1);

NOTE 1 k_{SERV} recognizes that design category C and D craft generally operate in circumstances where the consequences of rudder problems are less severe than for ocean-going craft (i.e. proximity of other craft, shallow water and ability to anchor). The use of this factor is optional.

If $k_{SERV} = 0,8$ is used, a note to this effect should be placed in the owner's manual.

$$k_{FLAT} = 1,08 - 0,008 \times V_{MAX} \text{ with } 0,75 \leq k_{FLAT} < 1 \qquad (11)$$

NOTE 2 k_{FLAT} considers that a flat plate or wedge generates less lift at the same angle of attack than a typical NACA section used to develop the above equations.

NOTE 3 Aspect ratio is included in Equation (11) since the rudder dimensioning force is based on a rudder angle that is lower than the stall angle. The dimensioning speed is lower than V_{MAX} since high speed craft cannot execute practical manoeuvres at this speed. The dimensioning speed and practical rudder angle are derived by realistically achievable steady turning radius to craft length ratio values based on craft test data.

A note should be entered into the user's manual to the effect that owners are expected to execute responsible craft handling and helm actuation rates (degrees/second) should reflect the prevailing craft speed.

$k_{SIG} = 1,25$

NOTE 4 The coefficient of 370 in Equation (10) corresponds to the expected force when executing a reasonably tight high-speed turn. As it is not an extreme load case, it is necessary to use a lower design stress than is used for F_1 to cover the expected large number of times that F_2 will be experienced during the life of the boat. The enhanced design stress factor is $k_{SIG} = 1,25$.

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8 Rudder bending moment and reactions at bearings

8.1 General

Knowledge of the bending moment, reaction at bearings and torque is necessary to calculate the resistant part of the rudder blade, whether the rudder stock, the blade fin, or a combination of both.

The analysis of the bending moment and the reaction at bearings varies with rudder type:

— 8.2 analyses spade rudders;

— 8.3 analyses skeg rudders.

8.2 Analysis of spade rudder (Type I)

8.2.1 Values of k_b, bending moment M and reactions at bearings for spade rudders (Type I)

$$M_H = F \times z_b \tag{12}$$

is the design rudder bending moment (at hull bearing) for spade rudders, where

— F is determined according to 7.1;

— z_b is the bending moment lever for spade rudders (see 6.2.1):

$$z_b = (k_b \times h_r) + h_b = h_c + h_b \tag{13}$$

where k_b is the rudder bending coefficient, determined according to rudder type, as follows.

To calculate z_b, one shall first determine the value of h_c:

— for a trapezoidal or near trapezoidal shape, either

a) use the value of k_b given by Equation (3) or Table 3, or

b) apply the graphical method shown in Figure 1;

— for other shapes, find $h_c = k_b \times h_r$ by any geometrical method

and then calculate $z_b = h_c + h_b$.

The reactions at bearings for spade rudders are as follows:

$$R_U = F \frac{z_b}{h_u} \tag{14}$$

is the reaction at the upper bearing (at deck or intermediate level), where h_u is the vertical distance between the centres of the upper and lower bearings (see Figure 2);

$$R_H = R_U + F \tag{15}$$

is the reaction at the hull bearing.

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8.3 Analysis of skeg rudders (Types II to V)

8.3.1 General

Rudders supported by a skeg or horn are considered to be held, from bottom to top, by three bearings (see Figure 3):

— a skeg bearing, with reaction R_S;

— a hull bearing located close to the hull bottom at the rudder level, with reaction R_H;

— an upper bearing located at deck level at the rudder level or an intermediate level between hull and deck, with reaction R_U.

Rudders with only two bearings (at hull and skeg level) are not recommended and are outside the scope of this part of ISO 12215. They rely entirely on skeg strength and stiffness, and they are not current practice. The loads and design stresses given by this part of ISO 12215 may be used to analyse their strength.

The rudderstock is considered to be continuous outside or inside of the hull. It may be made with two or several elements provided the connection between these elements is able to transmit the shear force and bending moment.

8.3.2 Methods of calculation

Rudders of Types II to V may be analysed by one of the following methods:

— continuous beam theory (also known as the three-moment equation) or the method in Annex C;

— the simplified method of 8.3.4.

8.3.3 Continuous beam theory

Continuous beam theory treats the upper rudder stock and lower rudder stock (including the blade) as being simply supported at each bearing. The lateral displacement shall be assumed to be zero at the upper and hull bearings but to deflect at the skeg bearing. The skeg is represented by a spring of stiffness k_S. This is defined as the force required to cause a unit lateral displacement at the skeg bearing and is expressed in meganewtons per metre.

The rudder force may be distributed to correspond with the blade area centroid, i.e. a trapezoidal load rate distribution. If using a matrix-displacement solution, any stiffness variations along the length of the blade/rudder may be represented by as many beam elements as are required. The skeg may also be modelled by beam elements in place of a single spring at the skeg bearing position providing care is taken to model the junction using a roller bearing condition. This is the recommended procedure which will also yield bearing forces. Annex C offers an alternative approach to analysing a Type II-V rudder as a continuous beam.

8.3.4 Simplified method

The simplified method (see Figure 4) allows the bending moment to be estimated at the hull bearing and at the skeg bearing only.

This method assumes that

— the distance between the hull and top bearing is sufficiently short that the stock may be considered as fully fixed at the hull bearing,

— the rudderstock has a near constant flexural rigidity, EI, between the three bearings,

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— the rudder force is uniformly distributed over the average rudder height, h_r, and

— the gap between the point where the stock emerges from the hull bearing and the top of the rudder blade is small.

This assumption is generally conservative (i.e. bending moments will be overestimated). The method should be regarded as a compromise which takes some account of skeg flexibility but does not claim to provide the complete solution.

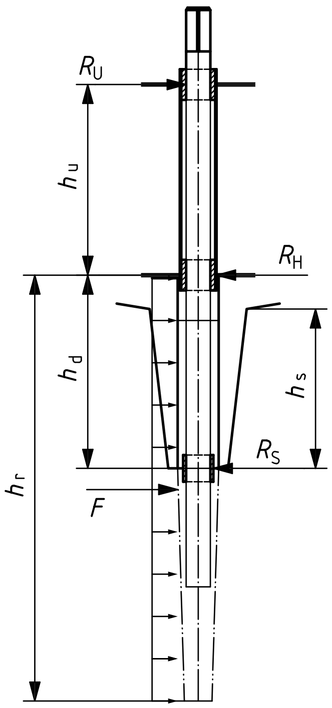

Figure 4 — Idealization for simplified method

NOTE 1 For simplicity of calculation, the simple method uses distances shown as h_d and h_r that are in fact $h_d + h_b$ and $h_r + h_b$, respectively; but as their ratio is used, the difference is not significant.

The following design moments at skeg bearing and hull bearing, respectively, shall be used in Clause 10 for stress checking:

$M_S = \frac{F \times (h_r - h_d)^2}{2h_r}$ is the design bending moment on the rudderstock at skeg bearing (16)

$M_H = F \times h_r \times \left(0{,}5 - \chi \cdot \frac{h_d}{h_r}\right)$ is the design bending moment on the rudderstock at hull bearing (17)

where

F is determined according to 7.1;

$$\chi = \frac{0{,}75 \times \left(\frac{h_r}{h_d}\right) + 0{,}125 \times \left(\frac{h_d}{h_r}\right) - 0.5}{1 + \frac{3\,EI_R}{h_d^3 \times k_S}} = \frac{R_S}{F} \quad (18)$$

where

χ is the ratio of the reaction force generated at the skeg, R_S, to the rudder force, F. For calculation of χ, EI_R is the average flexural rigidity of the rudder stock and blade. For strength analysis at the skeg or hull bearing, the calculations shall be based on the actual section at this point (i.e. the stock alone in the case of the hull bearing).

k_S is the stiffness coefficient of the skeg, which, for the case where the skeg can be idealized as a cantilever, may be estimated from

$$k_S = \frac{3 \times EI_S}{L_S^3} \tag{19}$$

where

EI_S is the average flexural rigidity of the skeg in MN·m²;

L_S is the effective length of the skeg from its top to skeg bearing.

NOTE 2 L_S can be equal to h_s as shown in Figure 3 but can also be entirely unrelated as in the case of a Type II rudder supported at the base by a horizontal bar which is attached to the keel.

The design bending moment of the skeg, at the point where the skeg emerges from the hull, to be analysed in accordance with Clause 14, is

$$M_S = \chi \times F \times L_S \tag{20}$$

The reactions are:

$R_S = \chi \times F$ is the reaction at skeg bearing; (21)

$R_H = F - R_S$ is the reaction at hull bearing. (22)

In the simplified method, the rudder is treated as being fully fixed at hull bearing level, with no rotation. If one wishes to have an order of magnitude of the reaction at upper bearing (to check the bushing pressure or the strength of the bearing support and connection), the following equation may be used:

$R_U = \frac{M_H}{h_u}$ is the order of magnitude (± 30 %) of the reaction at upper bearing. (23)

9 Rudder design torque, T

$T = F \times r$ is the rudder design torque (24)

where

F is as defined in 7.1;

r is the rudder torque arm but shall not be taken less than r_{min}, as defined in Table 5.

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Table 5 — Values of r and r_{min} according to rudder type

Type	r	r_{min}
I	0,3 $c - u$	0,1 c
II	0,3 $c - u$	0,1 c
III	0,5 $c - u$	0,05 c
IV	0,25 $c - u$	0,05 c
V	$\left(0{,}2\dfrac{h_d}{h_r}+0{,}3\right)c-u$	$\left(0{,}1-0{,}05\dfrac{h_d}{h_r}\right)c$
NOTE c and u are defined in 6.2.3.		

10 Rudder and rudder stock design

10.1 Load bearing parts of the rudder

The design bending moment, M, and the torque, T, to be resisted by the rudder are defined in Clauses 8 and 9, respectively.

The load bearing parts of the rudder may be:

- the rudder stock, usually where the rudder stock is metallic and where the rudder skins are less stiff than the stock;
- the rudder blade or spade itself (FRP, wood, plywood, or metal) as often in Type I d, or for Types II to IV where the stock is only there at the level of the bearings and acts as a pintle;
- the rudder stock and the blade where they constitute a single metal casting or welding, as in Type I a, or where the rudder and stock are both made of structural FRP;
- a combination of the above configuration, etc.

Where the stock is not continuous in the rudder, the continuity of the load path shall be ensured. In the Type V rudder shown in Figure 3, the stock stops at a certain distance below the skeg bearing. Where there is no more stock, the rudder blade alone has to support the bending moment.

Structural rudder stocks are the most common solution, so will be discussed first, in 10.2 to 10.6

The rudder stock diameter can normally be determined with only the rudder bending moment, M, and the rudder torque, T, both previously defined, and design stress, defined in Clause 5.

To help the user, formulae are given below to determine the required diameter for metallic solid and tubular stocks. Other information is also given for non-metallic stocks.

10.2 Metal rudder stock material

Considerations on metal choice for rudder stock are given in Annex A.

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10.3 Design stress for metal rudder stock

The design stresses are, in general, given in Clause 5 and Table 2. For metal, the pre-computed values of design stresses shall be taken from Table A.1 for the metals shown, but specific data derived from tests may also be used.

If derived from tests, the mechanical properties σ_u and σ_y for the determination of σ_d shall be 90 % of the mean relevant tested value or the mean value minus two standard deviations, whichever is the lesser.

10.4 Required diameter for solid circular metal rudder stocks

$z_{eq} = \sqrt{z_b{}^2 + 0{,}75\,r^2}$ is the equivalent lever in pure bending (for bending plus torsion) (25)

or

$M_{eq} = \sqrt{M^2 + 0{,}75\,T^2}$ is the equivalent bending moment (for bending plus torsion) (26)

where M and T are the bending moment and torque, respectively, at the critical section, i.e. hull bearing for Type I rudders (see 8.2.1) and the greater of hull bearing and skeg bearing for other types of rudder (see 8.3).

$$d = 21{,}68 \times \left(\frac{F \times z_{eq}}{\sigma_d}\right)^{1/3} = 21{,}68 \times \left(\frac{M_{eq}}{\sigma_d}\right)^{1/3} \quad (27)$$

is the required diameter for solid circular metal rudder stocks.

10.5 Vertical variation of the diameter of a Type I rudder (spade)

The bending moment, M, and torque, T, vary vertically, as does the demand for the local stock diameter.

Annex E gives details on how this variation can be computed.

Figure 5 give computed values, on the left side for h_{in}/h_u (above hull bearing) and on the right side h_{ou}/h_r (below hull bearing, i.e. outside the hull):

— the solid line represents the values for a rectangular spade with $\alpha = 1$;

— the dotted line represents the values for a trapezoidal spade with $\alpha = 0{,}5$.

Results for intermediate values of α may be obtained by interpolation.

To simplify the manufacture and avoid a curved profile of the shape of the stock, which is complicated to machine, the following simple manufacturing approximations may be used.

— Above the hull bearing, the diameter is constant at d_{max} [i.e. d given by Equation (27) until $h_{in}/h_u = 0{,}85$, and is then tapered above reducing to 0,53 d_{max} at the top (see small dotted line on the left side of Figure 5)].

— Below the hull bearing, the rudder stock is tapered from $h_{ou}/h_r = 0{,}95$, reducing to $d/d_{max} = 0{,}5$ for $h_{ou}/h_r = 0{,}3$ (see the dash-dotted line on the right side of Figure 5).

Similar approximation can be applied for $\alpha \neq 1$.

CAUTION — These values do not consider the eventual equivalent diameter reduction due to key notches or square sections as shown in Figure 6.

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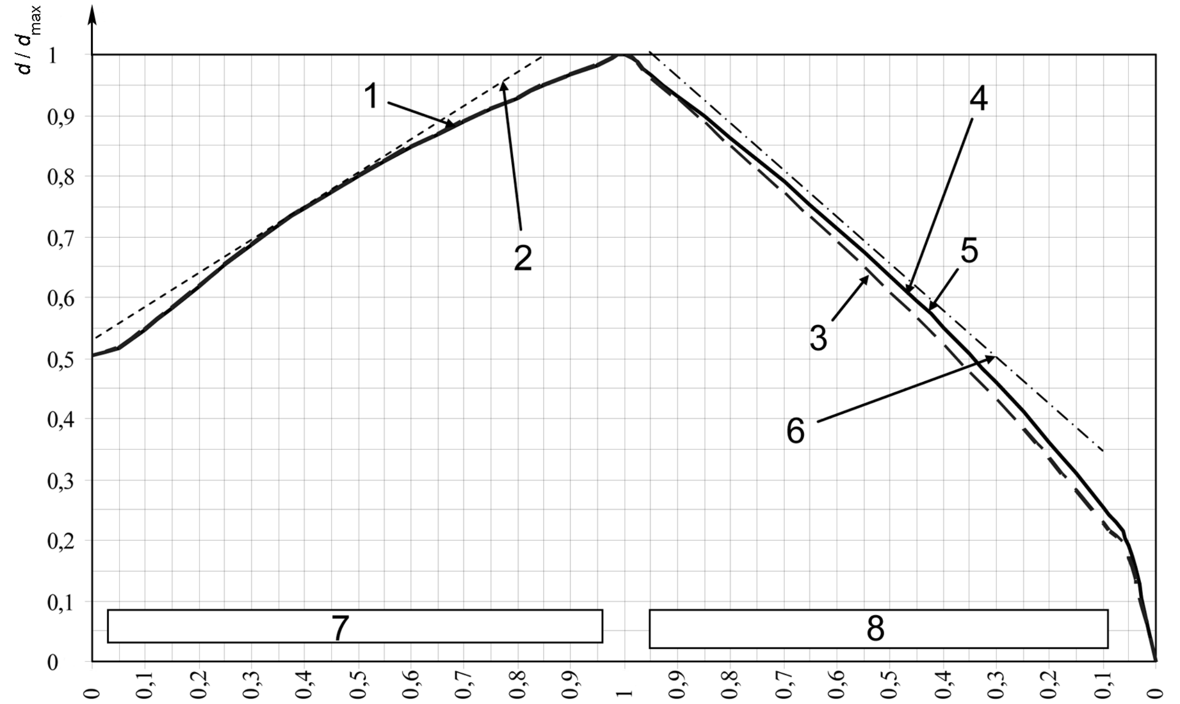

Key

1 d/d_{max} between bearings

2 tapered stock approximation from $d = d_{max}$ at h_{in}/h_u = 0,85 to d = 0,53 d_{max} at h_{in} = 0

3 α = 0,5 for a trapezoidal spade

4 d/d_{max} below lower bearing

5 α = 1 for a rectangular spade

6 tapered stock approximation from $d = d_{max}$ at h_{ou}/h_u = 0,95 and passing through d/d_{max} = 0,5 for h_{ou}/h_r = 0,3

7 values of h_{in}/h_u (above hull bearing)

8 values of h_{ou}/h_r (below hull bearing)

Figure 5 — Variation of d/d_{max} as a function of h_{in}/h_u or h_{ou}/h_r

NOTE 1 For ease of understanding, the diameter required by Equation (27) is called d_{max} whereas the required diameter at height h_{in} or h_{ou} is called d.

NOTE 2 The shape of the curve on the extreme left is twisted because T remains constant while M is close to zero. The extreme right of the curve is also kinked because the differences of variation of T and M are more significant for small values.

NOTE 3 In the lower part of the spade, there is no longer need for stock if the blade itself is able to support the bending moment and torque, and is also able to transmit them to the stock above.

10.6 Round tubular stocks

Where round tubular rudder stocks are used, the outer and inner diameters shall comply with

$$d = \sqrt[3]{\frac{d_o^4 - d_i^4}{d_o}} = \left(\frac{d_o^4 - d_i^4}{d_o}\right)^{1/3} \tag{28}$$

where

d is the required diameter of solid round rudder stock as defined in 10.4;

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d_o is the required outer diameter of round tubular rudder stock;

d_i is the required inner diameter of round tubular rudder stock.

In order to prevent the risk of local buckling and to provide adequate local strength at the level of the bearings or any load inducing device (keys, tiller arms, etc.), the wall thickness shall not be less than 0,1 d_o.

Tabulated values of Equation (28) are given in Table 6.

Table 6 — Equivalent diameter d

Equivalent diameter d for a tube with outer diameter d_o and wall thickness $t = (d_i - d_o)/2$											
d_o mm	$t = (d_i - d_o)/2$ mm										
	3	**4**	**5**	**6**	**7**	**8**	**10**	**12**	**14**	**16**	**18**
30	25,2	26,8	27,9	28,6	29,2	29,5	29,9	30,0	30,0	30,0	30,0
40		33,6	35,2	36,5	37,5	38,2	39,1	39,7	39,9	40,0	40,0
50			41,9	43,7	45,0	46,1	47,7	48,8	49,4	49,7	49,9
60				50,3	52,1	53,5	55,8	57,3	58,3	59,0	59,5
70					58,7	60,5	63,3	65,3	66,8	67,9	68,7
80						67,1	70,5	73,0	74,9	76,4	77,5
90							77,3	80,3	82,7	84,5	85,9
100							83,9	87,3	90,1	92,3	94,1
110								94,1	97,3	99,8	101,9
120								100,7	104,2	107,1	109,5
130									110,9	114,2	116,9
140									117,4	121,0	124,0
150										127,7	131,0
160										134,2	137,8
170											144,5

10.7 Non-circular metal rudder stocks

For non-circular rudder stocks,

$$\sigma_d \geqslant \sqrt{\sigma^2 + 3\tau^2} \text{ is the design stress for combined bending and twisting} \quad (29)$$

where

$$\sigma = \frac{M}{SM_B} \text{ is the bending stress (} M_H \text{ for Type I and } M_H \text{ or } M_S \text{ for Types II to V)} \quad (30)$$

where

SM_B, in cm^3, is the minimum section modulus under bending;

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$\tau = \dfrac{T}{SM_{\mathrm{To}}}$ is the shear stress (31)

where

M and T are determined in Clauses 8 and 9, respectively,

SM_{To}, in cm^3, is the minimum section modulus under torsion (see Annex D).

Annex D gives the means of determining SM_{B} and SM_{To}.

10.8 Simple non-isotropic rudder stocks (e.g. wood or FRP)

10.8.1 Blade acting both as rudder and rudder stock

A traditional rudder arrangement is a rudder blade directly connected to the transom, and acting both as a rudder and a rudder stock (see Figure 2, Type I d). This rudder blade is usually a faired plank underwater sometimes transformed into an unfaired plank above water. Annex D gives the section modulus of some typical shapes.

10.8.2 Stress requirements for wood or plywood rudders

For wooden rudder stocks, the following inequality shall be fulfilled:

$$\left(\frac{\sigma}{\sigma_{\mathrm{u}}}\right)^2 + \left(\frac{\tau}{\tau_{\mathrm{u}}}\right)^2 < 0{,}25 \qquad (32)$$

where

σ_{u} is the ultimate flexural strength (modulus of rupture) parallel to the stock/blade axis;

τ_{u} is the minimum shear strength parallel to the stock/blade axis.

10.9 Complex structural rudders and rudder stocks in composite

Complex composite rudder stocks, including those where the stock blends into the blade, shall be analysed according to Annex B.

10.10 Check of deflection of Type I rudder stocks between bearings

On spade rudders, the rudder stock deflects between the hull and upper bearings due to the bending moment induced by the side force, F, with two effects:

— difficulty of rotation of the stock, or even risk of “seizing” of the stock in cylindrical bearings. Self-aligning bearings are therefore recommended;

— risk of the rudder stock touching or damaging its tube.

In order to prevent the latter drawback, one of the following conditions shall be fulfilled:

— the deflection of the rudder stock between hull bearing and upper bearing is not greater than 15 % of the outer diameter of the stock; or

— $h_{\mathrm{u}}/d = 1{,}08 \times \left(\dfrac{E}{\sigma_{\mathrm{d}}}\right)^{0{,}5}$ is the maximum h_{u}/d ratio not to surpass (33)

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where

h_u and d are both expressed with the same unit, millimetres or metres;

CAUTION — In the rest of this part of ISO 12215, h_u is in metres and d in millimetres.

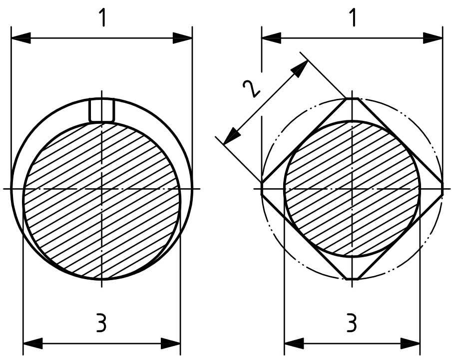
E is the elastic modulus of the stock, in newtons per square millimetre (see Annex D for approximate E for metals);

d is the outer diameter of a circular stock (whether solid or hollow).

Annex F gives the background of this requirement and some pre-calculated values.

11 Equivalent diameter at the level of notches

At the level of key notches or square section, the actual diameter shall be replaced by an equivalent diameter defined geometrically, as explained in Figure 6. This equivalent diameter shall be used for comparison with the required diameter, d, in Clause 10.



Key

1 actual diameter

2 square side

3 equivalent diameter

Figure 6 — Determination of equivalent diameter at the level of key notches or square section

12 Rudder bearings, pintles and gudgeons

12.1 Bearing arrangement

Bearings shall be designed to support and transmit to the skeg or hull structure the reaction loads given in Clause 8.

Bearings may be plain bearings (cylindrical, conical, spherical, etc.), roller bearings, or any arrangement of the two kinds.

The design bearing pressure on plain bushings shall not be greater than the allowable pressure given by the bushing provider or manufacturer (static pressure).

NOTE The bearing area is the product (bearing bushing length multiplied by outer stock diameter), in square millimetres. The bearing pressure is the bearing reaction divided by the bearing area, in newtons per square millimetre.

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Bearings usually need lubrication (water, oil, grease, etc.) and the manufacturer's recommendation in this field shall be fulfilled. If relevant, information for lubrication may be given in the owner's manual.

If bearings are of roller type, they shall be designed to operate in sea water or be adequately protected by a device allowing them to operate in water (stuffing box, lip gasket, etc.).

Unless specifically engineered, plain bearings shall have a length of 1,2 × to 1,5 × the diameter of the rudder stock or pintle.

Plain bearing bushings shall be secured in their support against rotation and axial movement.

As any material flexes under bending, self-aligning or spherical bearings that can follow this bending will induce a better holding and a rotation with less friction. This is particularly true for material with a low E/σ_d ratio, like titanium, which bends significantly under load, as emphasized in 10.10.

12.2 Clearance between stock and bearings

The geometry between the rudder stock and bearings shall allow the proper functioning clearance, i.e.:

- a clearance large enough (see below) to allow a free rotation of the stock around its axle. Too small a clearance can cause difficult rotation or even seizing of the stock. This is particularly true with plastic bushings that expand from water absorption;
- a clearance not so large as to permit vibration at speed.

Bearings made with hygroscopic material expand with time when they absorb water or when their temperature rises because of weather or friction. As the hull bearing is usually in water and the top bearing out of water, they will usually expand differently.

Where the bushing manufacturer specifies the necessary clearance between stock and bushing, this shall be adopted. In the absence of such information, Equations (34) and (35) may be used. Table 7 gives pre-calculated values for these equations.

$$D - d = \frac{1{,}5 \times d}{1000} + 0{,}1 + \text{water soaking expansion, in millimetres, is the minimum recommended value.} \quad (34)$$

$$D - d = \frac{3 \times d}{1000} + 0{,}2 + \text{water soaking expansion, in millimetres, is the maximum recommended value.} \quad (35)$$

Table 7 — Computed recommended values of diametric clearance $D - d$ between stock and bushing

Dimensions in millimetres

Stock outer diameter d	**Diametric clearance** $D - d$	
	min.	max.
40	0,16	0,32
60	0,19	0,38
80	0,22	0,44
100	0,25	0,50
120	0,28	0,56
140	0,31	0,62
160	0,34	0,68
180	0,37	0,74
200	0,40	0,80

NOTE These values correspond roughly to values between tolerances H11-c11 and D11-d11 of ISO 286-2.

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13 Rudder stock structure and rudder construction

13.1 Rudder stock structure

The rudder shall be designed to transmit, where necessary, the bending moment, M, and torque, T, from the blade to the stock.

This is usually done by a set of arms, rods, connecting plates, FRP lamination, etc. These shall be designed not to exceed σ_d in any of their elements, under M and T.

If the connection between the rudder stock and its structure is made by welding, the mechanical properties shall be the "as welded" ones in the areas affected by welding heat.

13.2 Rudder construction

The rudder construction and structure shall be able to withstand the hydrodynamic pressure loads and transfer them to the rudder stock or pintles.

If the rudder stock does not extend to the bottom of the rudder, the rudder shall have, below the end of the rudder stock and at the level of the rudder stock axis, at least the mechanical properties enabling the bending and torsional resistance defined above, according to the rudder type.

13.3 FRP rudder blades

13.3.1 Blades with a core

13.3.1.1 Requirements on core

Where the rudder blade is made of an FRP skin on a foam core, this foam shall be made of PVC with a density of at least 65 kg/m^3, or a core material of at least an equivalent compressive and shear strength (see ISO 12215-5:2008, 10.5). This core material shall be able to transfer the loads, usually by bearing load from the blade to the rudder stock structure, or be properly reinforced.

13.3.1.2 Requirements on the skins

Where the rudder blade is made of an FRP skin on a foam core, the minimum required mass of fibre per square metre, w_r, shall be as follows:

$$w_r = 1{,}1 \times k_5 \times (0{,}115 \times L_{WL} + 0{,}15) \text{ kg/m}^2 \tag{36}$$

where

k_5 is the fibre type factor:

— $k_5 = 1{,}0$ for E-glass reinforcement containing up to 50 % of chopped strand mat by mass;

— $k_5 = 0{,}9$ for continuous glass reinforcement (i.e. bi-axials, woven roving, unidirectionals);

— $k_5 = 0{,}7$ for continuous reinforcement using aramid or carbon or hybrids thereof.

13.3.1.3 Skins on a wood or plywood core

The requirements in 13.1 apply.

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13.3.2 Blade skins without internal core

The blades shall be strong enough or properly stiffened, in order to transfer the loads to the rudder stock.

It is considered reasonable that the skin fibre mass be $2 \times w_r$.

13.4 Non-FRP rudder blades

The pressure and shear loads on the skins shall be transferred to the rudder stock with an appropriate device, with normal and shear stresses not greater than the values required or implied in Clauses 7, 8 and 9.

14 Skeg structure

14.1 General

The skeg shall be able to support and transmit to the craft's structure the skeg reactions defined in Clause 8 without exceeding the design stresses defined above. The transmission of the reaction and its corresponding cantilever bending moment and torque shall be carefully considered. In some cases the skeg is as highly loaded as a ballast keel and therefore its structural arrangements might need the same structural arrangement as for keels, i.e. floors, etc.

14.2 Design stress

For metal, FRP or wooded skegs, the design stresses are defined in Clause 5. ISO 12215-5:2008, Annex C might also be useful.

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Annex A
(normative)

Metal for rudder stock

A.1 Metal rudder stock material

Metal rudder stocks may be made with one of the metals displayed in Table A.1, with the following recommendations and restrictions. Other metals or alloys may also be used, provided they are suited to usage in a marine environment.

The standards cited below are listed with full reference in the bibliography.

A.1.1 General

Welding generally reduces corrosion resistance and mechanical properties, and impairs heat treating. Therefore,

— at the level of significantly stressed welds, the “as welded” properties given in Table A.1 shall be used;

— there shall be no “end to end” joint rudder stocks using welding within 0,15 h_r of the point of maximal mechanical stress, unless specifically engineered.

A.1.2 Stainless steel

For stainless steel, there seems to be at least three common denominations, in addition to the chemical contents.

— The US AISI denomination (close to ASTM/UNS) is the most popular worldwide.

— EN 10088-3:1995 gives all stainless steel bars and wires a number 1.xxxx. It seems to be the standard most used in the European Union.

— ISO 16143-2:2004 classes austenitic steels from 0 to 49, austeno-ferritic from 50 to 59, ferritic from 60 to 79, martensitic from 80 to 99 and precipitation hardened above 100. ISO 16143-2:2004 seems clearer than EN 10088-3:1995, but is not yet widely used.

Table A.2 gives AISI, ISO and EN denominations for some steels typically used for rudder stocks.

AISI 316L, with low carbon content, has better corrosion resistance than AISI 316, but they are both recommended.

AISI 304 or 304L are not fully recommended for use under the waterline as there is a risk of crevice corrosion.

These steels can be cold drawn, stamped or forged, and reach high mechanical values (bolts, wires, etc.), but are then so hard that they are difficult to machine. They are usually found in the “as produced” state.

Precipitation hardened (PH) stainless steels (and some other steels) need to be heat treated to gain their strength, but are brittle if not softened, a heat treatment which lowers their mechanical properties.

On precipitation hardened (PH) steel F16 PH, 17-4 PH or similar, welds shall only be used as point or tack welding, to lock elements in place, not in elements subject to significant mechanical stress, and away from areas with high stresses. These steels have lower corrosion resistance than AISI 316L, but have been applied satisfactorily on rudder stocks for many years.

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DX 45 (AISI F51) may be structurally welded, provided it is done in a neutral atmosphere, preferably argon.

Of course stainless steels do not remain stainless in all conditions, particularly when there is a lack of oxygen and in high temperatures. Check with the supplier that the steel is suitable for use as a rudder stock for marine use, and if and how it may be welded.

A.1.3 Aluminium alloys

For aluminium alloys, EN 13195-1 follows the US denomination, and is the most popular.

Series 5 000 corresponds to Al-Mg alloys that can be cold drawn but not heat treated. They are mainly found in plates and sheets.

Series 6 000 corresponds to Al-Mg-Si alloys that can be heat treated. They are mainly found in bars and extrusions.

Series 7 000 corresponds to Al-Zn-Mg alloys that can be heat treated. They are mainly found in bars and extrusions. Their marine resistance to corrosion is poor, so they shall be avoided except in specific cases. In any case, the owner's manual shall include information on the risks and on the required inspection intervals and maintenance. The grades containing copper shall be avoided.

Series 2 000 aeronautic alloys containing copper are not suitable for marine use.

A.1.4 Titanium alloys

As Ta 6 V has high mechanical properties, it allows small diameter rudder stocks. As its elastic modulus is moderate (45 000 N/mm^2), this might induce large deflections. One shall therefore check that the deflection of the stock under the nominal force, F, does not impair the functioning of the rudder stock, nor the resistance and watertightness of the rudder tube. See 10.10.

A.2 Design stress for metallic rudder stock

The various metals displayed in Table A.1 can be found in a huge variety of brands, of cold working or of heat treatment, and their mechanical properties can change significantly for the same base material.

Standards usually give minimum values of mechanical properties, often significantly lower than the ones of the actual piece used. The values of Table A.1 are therefore indicative; they can be used for the cited state as default value, but values derived from tests or from a certificate obtained from the metal manufacturer or supplier are recommended.

For precipitation hardened (PH) stainless steels, and due to the wide variety of heat treatments, the values given are indicative, and precise data shall be taken from a certificate obtained from the steel manufacturer or supplier.

If derived from tests, the values of σ_u and σ_y used in Table 2 and in the rest of this part of ISO 12215 shall be 90 % of the mean relevant tested value or the mean value minus two standard deviations, whichever is the lesser.

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Table A.1 — Values of σ_d for metallic rudder stocks

Material	Chemical composition	σ_y [a] unwelded	σ_u [a] unwelded	σ_{yw} welded	σ_{uw} welded	σ_d [a] unwelded	σ_{dw} welded	E Approx
		N/mm^2						
Stainless steels								
AISI 304	X5 Cr Ni 18.9	195	500	195	195	195	98	$2,05 \times 10^5$
AISI 316, 316L	X5 Cr Ni Mo 17.2.2	195	500	195	195	195	98	
AISI 329 not cold worked	X3 Cr Ni Mo N 27-5-2	500	650	500	500	325	250	
AISI 329 cold worked	X3 Cr Ni Mo N 27-5-2	780	900	780	780	450	390	
17-4 PH, F16 PH	X5 Cr Ni Cu Nb 16.4	720	1 000		NR[c]	500		
DX45, Uranus[b]	X2 Cr Ni No N 22.5.3	450	660	450	450	330	225	
Mild steel								
E24/A		235	400	235	400	200	200	$2,1 \times 10^5$
E32-AH 32		315	470	315	470	235	235	
E36-AH 36		355	490	355	490	245	245	
Aluminium alloys, series 5000								
5086 0/H111	Al,Mg,4	100	240	100	240	100	100	$7,0 \times 10^4$
5086 H32	Al,Mg,5	185	275	100	240	138	100	
5083 0/H111	Al,Mg,4,5 Mn 07	125	275	125	125	125	63	
5083 H32	Al,Mg,4,5 Mn 07	125	275	125	125	125	63	
5059 Alustar 0/H111	Al, Mg, 5-6	145	290	145	290	145	145	
5059 Alustar H32	Al, Mg, 5-6	220	305	145	290	153	145	
Aluminium alloys, series 6000								
6005 A T6	Al,Si,Mg (A)	215	260	115	165	130	83	$7,0 \times 10^4$
6061 T6	Al,Mg1,Si,Cu	240	260	115	165	130	83	
6082 T6	Al,Mg1,Si,Mn	260	310	115	170	155	85	
Titanium alloys								
UTA 6V		820	900			450		$1,1 \times 10^5$
Copper alloys								
Bronze-Mn		245	510			245		$1,1 \times 10^5$
Bronze-Ni-Al		390	740			370		
Monel 400		350	550			275		$1,8 \times 10^5$
Monel 500		690	960			480		

[a] Properties in unwelded state, $\sigma_d = \min(\sigma_y; 0,5\ \sigma_u)$.

[b] To be welded in neutral atmosphere (argon).

[c] NR = not recommended.

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Table A.2 — Equivalence between stainless steel standard denominations

Stainless Steel AISI, common name brand	Chemical composition	AISI	EN 10088-3	ISO 16143-2
AISI 304, 304L	Cr Ni 18.9	304L	1,4307	1
AISI 316, 316L	Cr Ni Mo 17.12.2 -17.12.3	316L	1,4404-1,4432	21-22
AISI 329 not cold worked	Cr Ni Mo N 27-5-2	329	1,4460	55
AISI 329 cold worked	Cr Ni Mo N 27-5-2	329	1,4460	55
17-4 PH, F16 PH	Cr Ni Cu Nb 16.4	630	1,4545	101
AISI F51, DX45, Uranus	Cr Ni Mo N 22.5.3	UNS 31803	1,4462	52

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Annex B
(normative)

Complex composite rudder stock design

B.1 Scope

Annex B is intended to apply to composite stock and blade arrangements where the laminate schedule is likely to involve complex unidirectional and bi-axial fibres (including double-bias ± 45° fibres) with respect to the stock axis, and these fibres can be tapered out as the stock passes down the rudder blade. The rudder blade is assumed to be effectively attached to the stock and can be considered to be capable of making a contribution to the strength and stiffness of the component.

B.2 Design loads

The design side force, F, maximal bending moment, M, and torque, T, shall be taken directly from Clauses 7, 8 and 9, respectively.

For spade rudders, the bending moment and torque at any vertical position below hull bearing may be calculated by considering only the blade area below the point under consideration (see also 10.5 and Annex E)

The bending moment shall be determined by assuming that the side force given in Clause 7 is uniformly distributed over the blade area in a vertical sense. In the case of an unsupported blade below the point under consideration, the vertical lever may be based on values given in Clause 7.

The torque lever, r, shall be calculated from Table 5 on the mean chord of the area below the point under consideration.

B.3 Design criteria

The strength of the stock and/or blade shall be assessed under local bending moment and torque, as defined in Clause 10, at bearing level and at several vertical locations, including at least midway between bearings and in the middle of the unsupported span.

B.3.1 Strength criteria

For FRP rudder stocks, the following equation shall be fulfilled:

$$\left(\frac{\sigma}{\sigma_{\mathrm{u}}}\right)^2 + \left(\frac{\tau}{\tau_{\mathrm{u}}}\right)^2 < 0{,}25 \quad \text{(B.1)}$$

where

σ_{u} is the smaller of the ultimate tensile and compressive strength parallel to the stock axis;

τ_{u} is the minimum shear strength parallel to the stock axis.

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B.4 Methods of analysis

The strength and stiffness of the stock and blade may be assessed using one of the following two methods:

a) only the unidirectional fibres are considered as effective in carrying the bending moment, with only the bi-axial or diagonal fibres being considered effective in carrying the torque;

b) all fibres are effective.

If the first method is used, the component will be considered satisfactory if

1) the bending stress in the unidirectional fibres is less than 50 % of the minimum of the ultimate compressive (or tensile) stress, and

2) the shear stress in the other fibres is less than 50 % of the ultimate shear strength.

In the second method, a laminate analysis using classical lamination theory shall be performed and the component under combined bending and torsion shall satisfy Inequality (B.1).

B.5 Other considerations

B.5.1 Skin wrinkling for sandwich construction

In the case of a sandwich blade face in compression, the compressive stress shall not exceed 50 % of the face wrinkling strength (see ISO 12215-5 for explanations).

$$\sigma_{\mathrm{dc}} = 0{,}3\ \sqrt[3]{E_{\mathrm{s}} \times E_{\mathrm{c}} \times G_{\mathrm{c}}} \tag{B.2}$$

where

σ_{dc} is the design stress in compression of the outer and inner skin;

E_{s} is the compressive elastic modulus of skins, in 0°/90° in-plane axis of panel;

E_{c} is the compressive elastic modulus of core, perpendicular to skins;

G_{c} is the core shear modulus, in the direction parallel to load.

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Annex C
(normative)

Complete calculation for rudders with skeg

C.1 General

The skeg provides an extra support to the rudder stock. As the rudder stock has three supports, it is a statically indeterminate structure. Compared to a spade rudder, the maximum bending moment on the rudder stock is dependent on the relative stiffness of the skeg compared to the stiffness of the stock.

Annex B is intended to be used when it is required to determine the actual support provided by the skeg (full or partial). Annex B need not be used if the skeg is treated as flexible and the stock scantling is obtained considering the rudder as a spade rudder.

C.2 Rudder with partial skeg

The rudder area, A, is divided into 3 parts (See Figure C.1):

$$A = A_d + A_f + A_e \tag{C.1}$$

is the sum of the three sub-areas of the blade.

NOTE Some dimensions differ slightly from the main core of this part of ISO 12215-8. For example, the blade area of the rudder is divided here into 3 parts A_d, A_f and A_e in contrast with the simple 2 parts in Figure 3.

$$M_H = \frac{C_1 \times (h_d^2 - h_b^2) \times (0{,}5 + 0{,}125\ w) + C_2 \times h_e^2 \times (0{,}5 + \frac{h_d}{h_e} - 0{,}25\ w)}{1 + w \times (1 + \frac{h_u}{h_d} \times \frac{I_d}{I_u} \times \frac{E_d}{E_u})} \tag{C.2}$$

is the bending moment at the hull bearing;

$$M_S = \frac{C_2 \times h_e^2}{2} \tag{C.3}$$

is the bending moment on the rudder stock at the skeg bearing;

$$T_H = \frac{F}{A} \left(A_d \times r_d + A_f \times r_f + A_e \times r_e\right) \tag{C.4}$$

is the torque at the top and hull bearing;

$$T_S = \frac{F}{A} \left(A_f \times r_f + A_e \times r_e\right) \tag{C.5}$$

is the torque at the skeg bearing;

$$C_1 = \frac{F \times (A_d + 0{,}5 \times A_f)}{A\ (h_r - h_e)} \tag{C.6}$$

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$$C_2 = \frac{F \times (A_e + 0{,}5 \times A_f)}{A \times h_e}$$, and (C.7)

$$w = \frac{h_d^3}{3 \times I_d * E_d \times h_S \left(\frac{r_s^2}{G_S J_S} + \frac{h_s^2}{3\ I_S\ E_S} \right)} \quad \text{(C.8)}$$

are non-dimensional coefficients;

$r_d = 0{,}2 c_d - x_d$ is the lever radius of area A_d, but shall not be taken less than 0,125 c_d; (C.9)

$r_f = 0{,}2 c_f - x_f$ is the lever radius of area A_f, but shall not be taken less than 0,125 c_f; (C.10)

$r_e = 0{,}33 c_e - x_e$ is the lever radius of area A_e, but shall not be taken less than 0,125 c_e; (C.11)

r_h is the mean lever radius for twisting of the skeg;

C_i are the horizontal lengths of the rudder sub-areas at the centre of area;

A_i and x_i are the distance from the leading edges to the centreline of the skeg bearing, with i = d, f and e;

F is the force on the rudder, defined in Clause 7;

I_u and E_u are, respectively, the mean second moment (cm^4) and elastic modulus of the upper rudder stock;

I_d and E_d are, respectively, the mean second moment (cm^4) and elastic modulus of the rudder (blade and stock) between hull bearing and skeg bearing;

I_S and E_S are, respectively, the mean second moment (cm^4) and elastic modulus of the rudder skeg;

J_S and G_S are, respectively, the mean polar moment (cm^4) and shear modulus of the rudder skeg.

$J_S = \frac{4 \times a^2 \times t}{s}$ is an approximation of the skeg polar moment, in centimetres to the power 4, (C.12)

where

- a is the mean horizontal area enclosed by the outer perimeter of the skeg plating, in centimetres;
- t is the mean thickness of the skeg plating, in centimetres;
- s is the mean median perimeter of the skeg plating, in centimetres.

Figure C.1 schematizes the dimensions and the vertical variations of the bending and twisting moments.

From the values of M and T, Clause 7 shall be used to determine the rudder stock's required dimensions.

The reaction on the upper bearing is $R_U = \frac{M_H}{h_u}$ (C.13)

The reaction on the hull bearing is $R_H = R_U \left(1 + \frac{h_u}{h_d} \right) + \frac{C_1}{2\ h_d} \left(h_d - h_s \right)^2 - \frac{M_H}{h_d}$ (C.14)

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The reaction on the skeg bearing is $R_S = F + R_U - R_H$ (C.15)

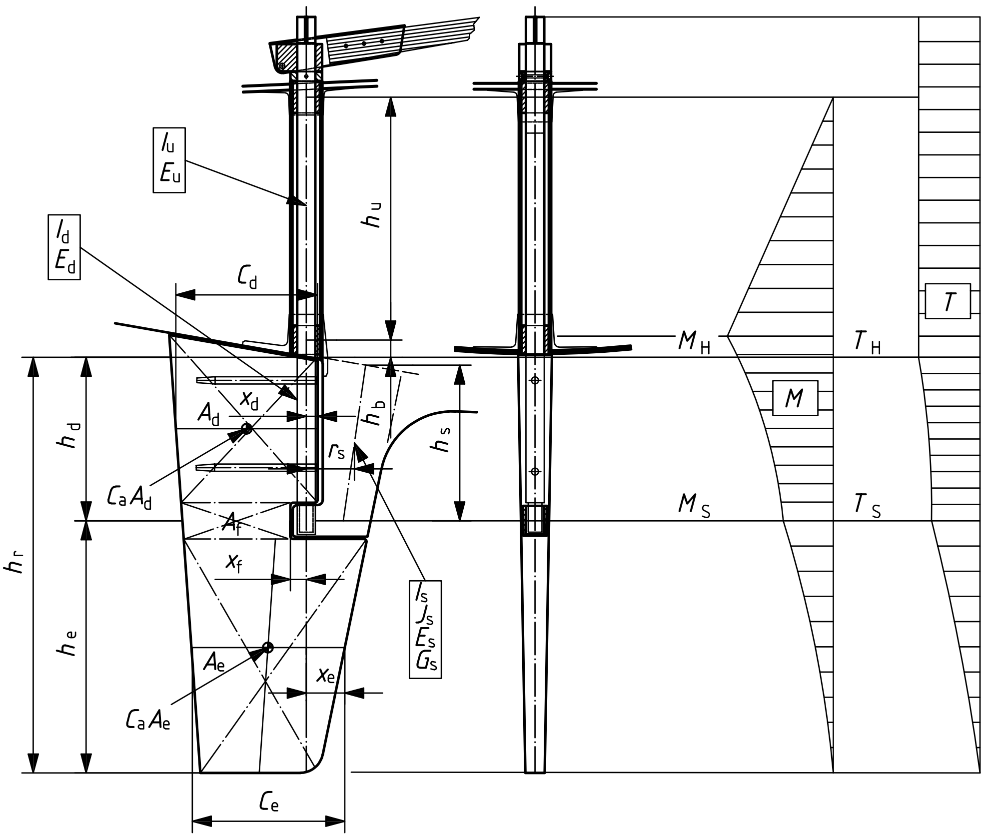

Figure C.1 — Rudder with a partial skeg

C.3 Rudder with full skeg

This is a simplified case compared to the previous one, with $A_d = A$, $A_f = A_e = 0$, $h_e = 0$ and $h_d \approx h_s \approx h_r$.

$$M_H = \frac{\frac{F}{h_d}(h_d^2 - h_b^2)(0{,}5 + 0{,}125\ w)}{1 + w\left(1 + \frac{h_u}{h_d} \times \frac{I_d}{I_u} \times \frac{E_d}{E_u}\right)}$$ is the bending moment at the hull bearing; (C.16)

$M_S = 0$ is the bending moment on the rudder stock at the skeg bearing (i.e. simply supported);

$T_H = F \times r$ is the torque at the top of the rudder stock and hull bearing; (C.17)

$T_S = 0$ is the torque on the rudder stock at the skeg bearing level (i.e. simply supported);

$$w = \frac{h_d^3}{3 \times I_b \times E_b \left(\frac{r_s^2}{G_S \times J_S} + \frac{h_s^2}{3 \times I_S \times E_S} \right)}.$$ (C.18)

The reaction on the upper bearing is $R_U = \frac{M_H}{h_u}$ (C.19)

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The reaction on the hull bearing is $R_H = R_U(1+\frac{h_u}{h_d}) + \frac{F}{2 \times h_d^2}(h_d - h_b)^2$ (C.20)

The reaction on the skeg bearing is $R_S = F + R_U - R_H$ (C.21)

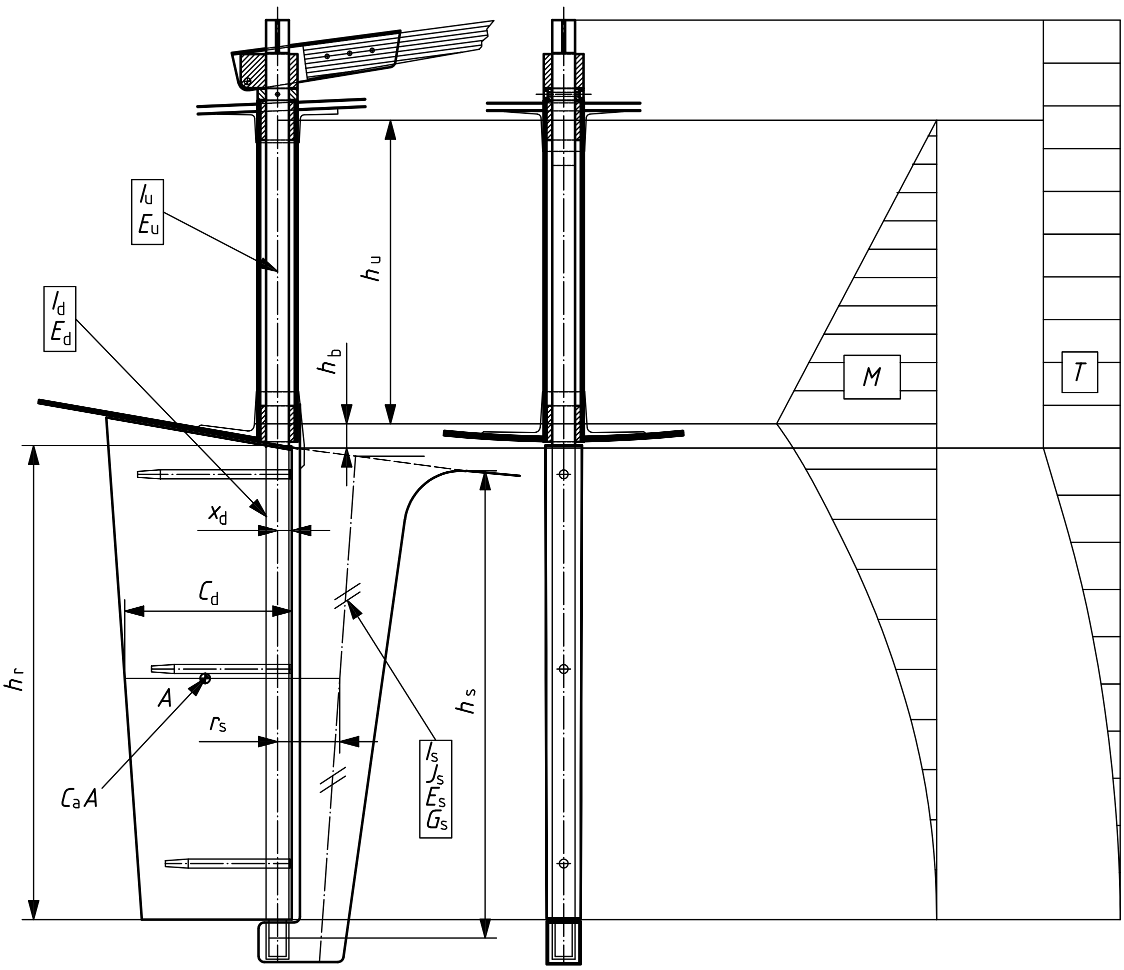

Figure C.2 — Rudder with a full skeg

C.4 Skeg load and scantlings

The load on the skeg, R_S, is defined above in Equation (C.15) or (C.21). The skeg strength and structure shall be analysed according to Clause 14.

NOTE Although the hydrodynamic load on the skeg is generally not considered in this part of ISO 12215, it might need consideration if the skeg has an area greater than 25 % of that of the rudder.

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Annex D
(informative)

Geometrical properties of some typical rudder blade shapes

D.1 Dimensions

The dimensions of rectangular or streamlined sections are as follows (see Figure D.1):

— L_f is the overall longitudinal dimension of the foil section, in millimetres;

— b_f is the overall transversal dimension of the foil section, in millimetres;

— t_f is the wall thickness of a hollowed section, in millimetres.

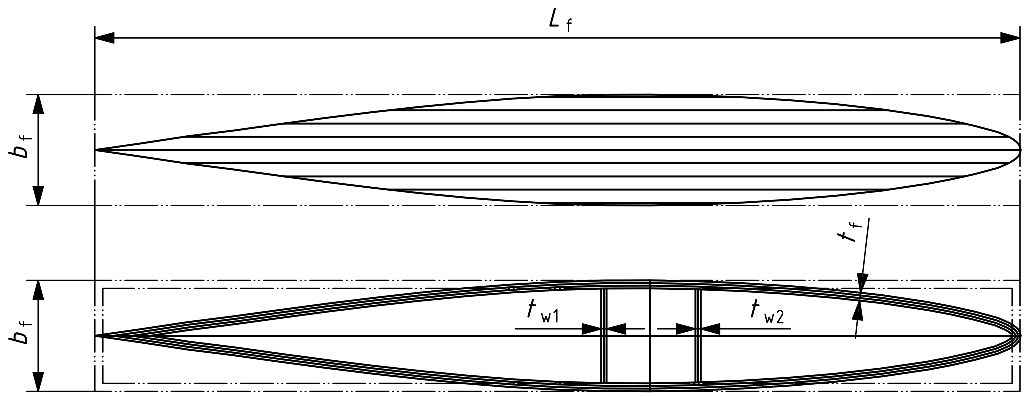

Figure D.1 — Sketch of solid and hollowed rectangular or streamlined sections

D.2 Geometric properties (bending)

D.2.1 Transverse section modulus and second moment

The minimum transverse section modulus, in cubic centimetres, is

$$SM_T = \frac{20 \times I_T}{b_f} \quad \text{(D.1)}$$

Where I_T is the transverse second moment of area in centimetres to the power 4 and can be obtained with reasonable accuracy from

$$I_T = \frac{{k_f}^2}{1{,}2 \times 10^5} \times L_f \times {b_f}^3 \quad \text{(D.2)}$$

which is the transverse section second moment for solid sections, in centimetres to the power 4,

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where

$$k_{\text{f}} = \frac{\text{fin area}}{L_{\text{f}} \times b_{\text{f}}} \tag{D.3}$$

is the foil shape coefficient whose values are given in Table D.1 for typical shapes.

$$I_{\text{T}} = \frac{k_{\text{f}}^2}{1{,}2 \times 10^5} \times \left[L_{\text{f}} \times b_{\text{f}}^3 - (L_{\text{f}} - 2t_{\text{f}}) \times (b_{\text{f}} - 2t_{\text{f}})^3 \right] + \sum \frac{t_{\text{wi}} \times d_{\text{wi}}^3}{1{,}2 \times 10^5} \tag{D.4}$$

is the transverse second moment for hollow sections, in centimetres to the power 4, where t_{wi} is the thickness of a transverse mounted internal web and d_{wi} is the depth, both in millimetres.

Table D.1 — Values of k_{f} and k_{f1} for typical planform shapes

Fin shape/planform	Value of k_{f}	Value of k_{f1}
Solid rectangle	1,000	1,00
Elliptical	0,786	1,00
Diamond	0,500	1,00
Parabolic	0,667	1,00
NACA 00XX	0,684	1,16
NACA 65aXX	0,670	1,16

D.2.2 Longitudinal section modulus and second moment

The minimum longitudinal section modulus, in cubic centimetres, is

$$SM_{\text{L}} = \frac{20 \times I_{\text{L}}}{k_{\text{f1}} \times L_{\text{f}}} \tag{D.5}$$

where k_{f1} is a factor which corrects the maximum lever as given in Table D.1. It is equal to 1 for all sections which are symmetrical about the midplane between leading and trailing edges, and 1,16 for NACA 00XX sections.

The longitudinal second moment of area, I_{L}, can be obtained with reasonable accuracy from

$$I_{\text{L}} = \frac{k_{\text{f}}^2}{1{,}2 \times 10^5} \times b_{\text{f}} \times L_{\text{f}}^3 \tag{D.6}$$

which is the longitudinal second moment for solid sections, in centimetres to the power 4,

$$I_{\text{L}} = \frac{k_{\text{f}}^2}{1{,}2 \times 10^5} \times \left[b_{\text{f}} \times L_{\text{f}}^3 - (b_{\text{f}} - 2t_{\text{f}}) \cdot (L_{\text{f}} - 2t_{\text{f}})^3 \right] \tag{D.7}$$

which is the longitudinal second moment for hollow sections, in centimetres to the power 4.

The additional effect of transverse webs may be added using $\sum t_{\text{wi}} \times d_{\text{wi}} \times x_{\text{i}}^2$, where x_{i} is the distance from the longitudinal centroid of the fin and webs and the web centres. Unless the webs are located towards the leading or trailing edges, the effect of neglecting the webs will be small.

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D.3 Geometric properties (torsion)

The torsional properties are required to calculate the shear stress induced in the fin by the torque where there is no rudder stock (or below its lower limit).

The maximum shear stress generally occurs parallel to the fin centreline (i.e. in the fore and aft direction). It is given by

$$\tau = \frac{T}{SM_{\mathrm{To}}}, \text{ which is the maximum shear stress,} \tag{D.8}$$

where

T is the maximum torque defined in Clause 9;

SM_{To} is the torsional section modulus and can be estimated with reasonable accuracy as follows.

NOTE 1 The index $_{\mathrm{To}}$, for torsion, is used to avoid confusion with transversal section modulus.

$$SM_{\mathrm{To}} = \frac{L_{\mathrm{f}} \times {b_{\mathrm{f}}}^2 \times k_{\mathrm{f}}^2}{3\,000} \text{ for a solid section;} \tag{D.9}$$

$$SM_{\mathrm{To}} = \frac{2 \times k_{\mathrm{f}} \times t_{\mathrm{f}} \times (L_{\mathrm{f}} - t_{\mathrm{f}}) \times (b_{\mathrm{f}} - t_{\mathrm{f}})}{1\,000} \text{ for a hollow non-circular section.} \tag{D.10}$$

NOTE 2 For circular sections, whether solid or hollow, the torsion modulus is J/r, therefore Equations (25) and (26) consider correctly both σ and τ.

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Annex E
(informative)

Vertical variation of diameter for Type I rudders

E.1 General

The bending moment, M, and torque, T, vary vertically, as does the local demand for stock diameter. Equation (27) can be written, using Equation (26), as

$$d_{\text{max}} = 21{,}68 \times \left(\frac{M_{\text{eq max}}}{\sigma_{\text{d}}} \right)^{1/3} = 21{,}68 \times \left[\frac{\left(M_{\text{max}}^2 + 0{,}75 T_{\text{max}}^2 \right)^{1/2}}{\sigma_{\text{d}}} \right]^{1/3} \tag{E.1}$$

This diameter d_{max} is the required diameter at maximum bending moment (at hull bearing) (see NOTE 2 in 10.5). At any position above or below the hull bearing, the diameter, d, necessary to follow the variations of M and T can be found as follows

$$\frac{d}{d_{\text{max}}} = \left(\frac{M_{\text{eq}}}{M_{\text{eq max}}} \right)^{1/3} = \left(\frac{M^2 + 0{,}75 T^2}{M_{\text{max}}^2 + 0{,}75 T_{\text{max}}^2} \right)^{1/6} \tag{E.2}$$

In general, with properly balanced spade rudders $T_{\text{max}} \approx 0{,}15\ M_{\text{max}}$, and therefore

$$\frac{d}{d_{\text{max}}} = \left[\frac{M^2 + 0{,}75 T^2}{M_{\text{max}}^2 + 0{,}75 x \left(0{,}15^2 \times M_{\text{max}}^2 \right)} \right]^{1/6} = \left(\frac{M^2 + 0{,}75 T^2}{1{,}017\ M_{\text{max}}^2} \right)^{1/6} = \left\{ \frac{M^2 + 0{,}75 T^2}{\left[\left(\frac{1}{0{,}15} \right)^2 + 0{,}75 \right] \times T_{\text{max}}^2} \right\}^{1/6} \tag{E.3}$$

and finally
$$\frac{d}{d_{\text{max}}} = \left[\frac{1}{1{,}017} \left(\frac{M}{M_{\text{max}}} \right)^2 + \frac{0{,}75}{45{,}19} \left(\frac{T}{T_{\text{max}}} \right)^2 \right]^{1/6} = \left[0{,}983 \left(\frac{M}{M_{\text{max}}} \right)^2 + 0{,}0166 \left(\frac{T}{T_{\text{max}}} \right)^2 \right]^{1/6} \tag{E.4}$$

E.2 In a trapezoidal spade (outside of the hull)

For a trapezoidal spade, if we call h the height of any section from the bottom of the blade, with $0 < h < h_{\text{r}}$, for clarity the value of h outside the hull is called h_{ou} in Figure 2 and Figure 5.

$$\frac{M}{M_{\text{max}}} = \left(\frac{h}{h_{\text{r}}} \right)^2 \cdot \left[\frac{3\alpha + (1-\alpha) \frac{h}{h_{\text{r}}}}{2\alpha + 1} \right] \tag{E.5}$$

$$\frac{T}{T_{\text{max}}} = \frac{1}{(1+\alpha)} \left[2\alpha \frac{h}{h_{\text{r}}} + (1-\alpha) \cdot \left(\frac{h}{h_{\text{r}}} \right)^2 \right] \tag{E.6}$$

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where α is the taper ratio defined in 6.2.2. For a rectangular spade, the formulae simplify as $\alpha = 1$.

The values of ratio d/d_{max} in Equation (E.4) are computed in Table E.1 from Equations (E.5) and (E.6).

E.3 Above hull bearing (inside of the hull)

If we consider that the torque is counterbalanced at the top bearing, with the tiller or quadrant, T is constant above hull bearing, and M decreases linearly (see Figure 2, Type I b).

$$\frac{M}{M_{max}} = \left(\frac{h}{h_u}\right) \text{ is the value inside the hull} \tag{E.7}$$

with $0 < h < h_u$, for clarity the value of h inside the hull is called h_{in} in Figure 2 and Figure 5.

E.4 Computed values

Computed values for $\alpha = 1$, $\alpha = 0{,}75$ and $\alpha = 0{,}5$ are given in Table E.1. Intermediate values can be found by interpolation. This table served as a basis for Figure 5.

Table E.1 — Computed values for $\alpha = 1$, $\alpha = 0{,}75$ and $\alpha = 0{,}5$

Position	h/h_{max}	$\alpha = 1$			$\alpha = 0{,}75$			$\alpha = 0{,}5$		
		RECTANGULAR SPADE			**TRAPEZE SPADE**			**TRAPEZE SPADE**		
		M/M_{max}	T/T_{max}	d/d_{req}	M/M_{max}	T/T_{max}	d/d_{req}	M/M_{max}	T/T_{max}	d/d_{req}
Top bearing	0,00	0,00	1,00	0,51	0,00	1,00	0,51	0,00	1,00	0,51
	0,10	0,10	1,00	0,55	0,10	1,00	0,55	0,10	1,00	0,55
	0,20	0,20	1,00	0,62	0,20	1,00	0,62	0,20	1,00	0,62
	0,30	0,30	1,00	0,69	0,30	1,00	0,69	0,30	1,00	0,69
	0,40	0,40	1,00	0,75	0,40	1,00	0,75	0,40	1,00	0,75
h_{in}/h_u	0,50	0,50	1,00	0,80	0,50	1,00	0,80	0,50	1,00	0,80
	0,60	0,60	1,00	0,85	0,60	1,00	0,85	0,60	1,00	0,85
	0,70	0,70	1,00	0,89	0,70	1,00	0,89	0,70	1,00	0,89
	0,80	0,80	1,00	0,93	0,80	1,00	0,93	0,80	1,00	0,93
	0,90	0,90	1,00	0,97	0,90	1,00	0,97	0,90	1,00	0,97
Hull bearing	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
	0,90	0,81	0,90	0,93	0,80	0,89	0,93	0,79	0,87	0,92
	0,80	0,64	0,80	0,86	0,63	0,78	0,86	0,61	0,75	0,85
	0,70	0,49	0,70	0,79	0,48	0,67	0,78	0,45	0,63	0,77
	0,60	0,36	0,60	0,71	0,35	0,57	0,70	0,32	0,52	0,69
h_{ou}/h_r	0,50	0,25	0,50	0,64	0,24	0,46	0,62	0,22	0,42	0,61
	0,40	0,16	0,40	0,55	0,15	0,37	0,54	0,14	0,32	0,52
	0,30	0,09	0,30	0,46	0,08	0,27	0,45	0,07	0,23	0,43
	0,20	0,04	0,20	0,36	0,04	0,18	0,35	0,03	0,15	0,33
	0,10	0,01	0,10	0,25	0,01	0,09	0,24	0,01	0,07	0,23
Rudder bottom	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00

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Annex F
(informative)

Type I rudders — Deflection of stock between bearings

F.1 General

CAUTION — The following calculation corresponds to an idealization considering that the rudder stock is free to rotate at both bearings. This is not far from reality for self-aligning bearings with low friction, but it is not true for non-self-aligning bearings, for which the actual deflection can be less than calculated below.

NOTE To simplify the calculation, x will be taken as 0 at upper bearing and h_u at hull bearing. The calculation is only valid if EI is constant between hull bearing and upper bearing.

$$M = R_U \times x \tag{F.1}$$

is the bending moment, where R_U is the reaction at upper bearing (see Figure 2, Type I b).

$$EI \frac{d^2_y}{d^2_x} = M = R_U x \text{ is the differential equation of deflection,} \tag{F.2}$$

and after first integration,

$$EI \frac{dy}{dx} = R_U \frac{x^2}{2} + A,$$

and with proper value of A,

$$EI \frac{dy}{dx} = R_U \frac{x^2}{2} + R_U \frac{h_u^2}{6} \tag{F.3}$$

and after second integration,

$$EIy = R_U \frac{x^3}{6} + Ax + B, \tag{F.4}$$

where A and B are integration constants.

At $x = 0$, $y = 0$ so $B = 0$; and at $x = h_u$, $y = 0$, so

$$A = -\frac{R_U \times h_u^2}{6}, \text{ and}$$

$$y = \frac{1}{6EI}\left(R_U \times x^3 - R_U \times h_u^2 \times x\right). \tag{F.5}$$

The maximal deflection occurs where $dy/dx = 0$ for $x = (1/3)^{0,5} \times h_u = 0,577\, h_u$.

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Therefore,

$$y = \frac{R_U}{6EI}\left((0,577\,h_u)^3 - h_u^2 \times 0,577\,h_u\right) = \frac{M_H \times h_u^2}{6EI}\left(0,577^3 - 0,577\right) = -0,064\,2\,\frac{M_H \times h_u^2}{EI} \tag{F.6}$$

If the outer diameter is such that σ_d is reached at the outer fibre of the stock,

$$\sigma_d = \frac{M_H}{SM} = \frac{M_H \times d}{2 \times I} \tag{F.7}$$

And therefore,

$$\frac{M_H}{I} = \frac{2 \times \sigma_d}{d}$$

where d is the outer diameter of the stock (solid or hollow), hence

$$y_{max} = 0,128\,\frac{h_u^2 \times \sigma_d}{d \times E} \tag{F.8}$$

is the absolute value of the maximum deflection of the stock between hull bearing and upper bearing.

F.2 Limitation of deflection

On plain plastic bearings, the thickness of the bushing is usually 15 % of the outer diameter of the stock. If the deflected stock does not touch the inner face of the rudderport (stock tube), the deflection shall not be greater than 0,15 × d, and:

$$y_{max} = 0,128\,\frac{h_u^2 \times \sigma_d}{d \times E} = 0,15 \times d \tag{F.9}$$

Therefore

$$\frac{E}{\sigma_d} = \frac{0,128}{0,15} \times \frac{h_u^2}{d^2} \tag{F.10}$$

and, by inverting, the limit not to surpass is

$$\frac{h_u}{d} = 1,08 \times \left(\frac{E}{\sigma_d}\right)^{0,5} \tag{F.11}$$

This value is considering that the bearings are self-aligning (free rotation), and with cylindrical bearings the deflexion will be less.

F.3 Computed values

Table F.1 gives the result of the calculation.

It is obvious that the ratios for FRP rudder stocks are low, but composite stocks are usually tubular, which means an outer diameter greater than for a solid stock.

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Table F.1 — Computed maximum values of h_u/d ratio and of h_u for $30 < d < 120$ mm

Material	Type	E N/mm^2	σ_d N/mm^2	h_u/d max.	d mm 30	60	90	120
					h_u max. mm			
Stainless steel	316L	210 000	195	34	1 030	2 070	3 100	4 130
Stainless steel	F16 PH	210 000	500	22	650	1 290	1 940	2 580
Aluminium	5086 H111	70 000	100	28	830	1 670	2 500	3 330
Aluminium	6061 T6	70 000	130	24	730	1 460	2 190	2 920
Titanium	UTA6V	110 000	450	16	490	980	1 480	1 970
Glass RP	70 % UD $\Psi = 50$ %	19 950	166	12	350	690	1 040	1 380
Carbon RP	70 % UD $\Psi = 50$ %	68 300	255	17	520	1 030	1 550	2 060
Hard wood	$\rho = 600$ kg/m^3	10 500	45	16	480	960	1 440	1 920

From this table, one can see that a hollow stock with a greater outside diameter than for a solid stock is a favourable feature for stiffness, particularly for FRP which is a lighter material than metal. On the other hand, a greater diameter means a greater friction moment for the same friction coefficient.

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