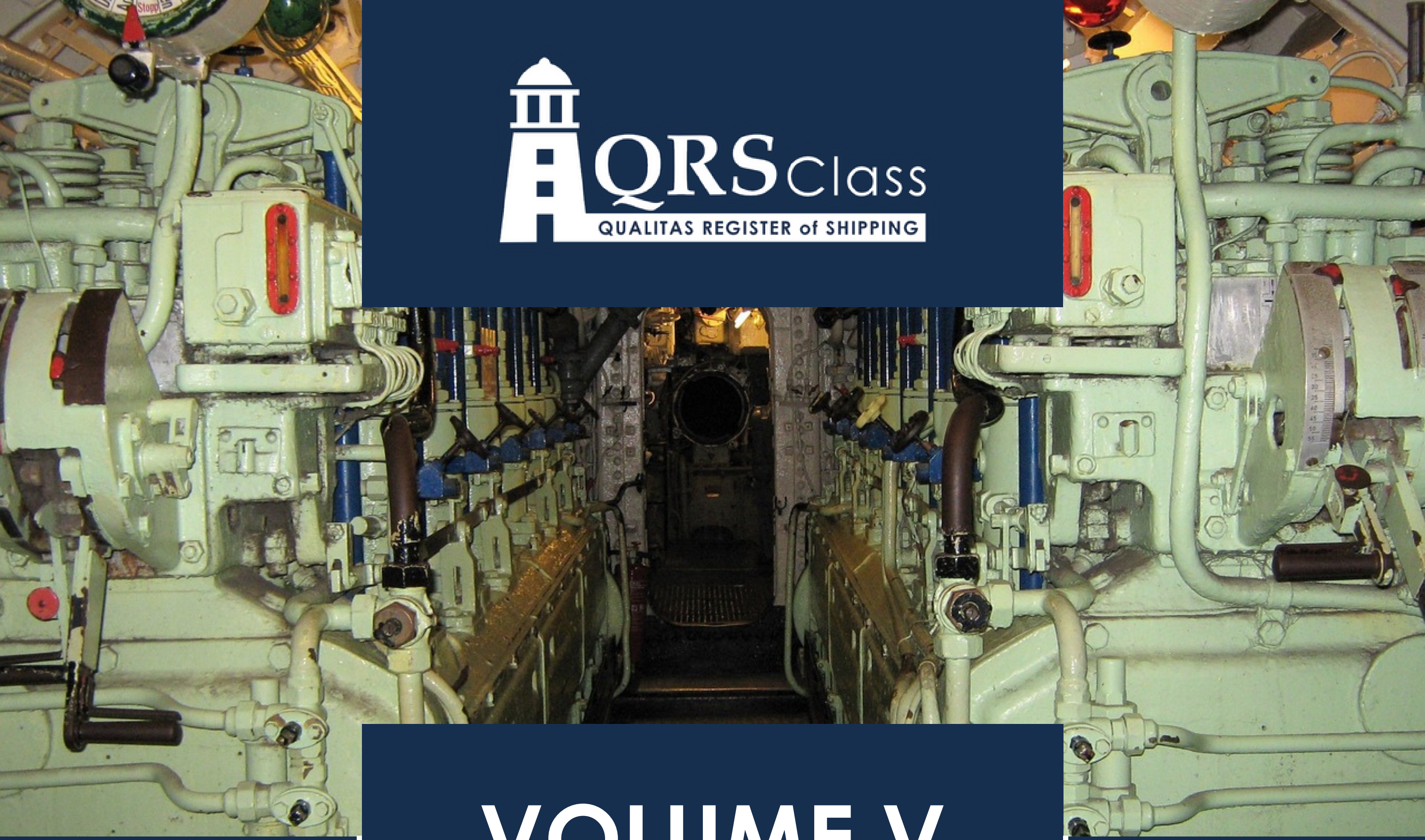

QRS Class
QUALITAS REGISTER of SHIPPING

VOLUME V

Machinery

The present edition of Machinery, Volume V has been approved by the General Manager and will enter in force on May 15, 2023.

The present edition of Machinery is based on the 2016 edition taking into account the amendments developed immediately before publication.

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General regulations for Classification

REVISION HISTORY

(Purely editorial amendments are not included in the Revision History)

Amended paras / chapters / sections	**Information on amendments**	**Entry-into force date**
Para 2.1	Application of technical solutions has been considered	15.05. 2023
Para 2.4	Requirements for crankshafts have been taken into account	15.05. 2023
Para 3.1	Distribution of dynamic balancing for parts has been included	15.05. 2023
Para 5.2	Requirement for steering gear as per IACS UR M42 (Rev.6 Mar 2022) has been introduced.	15.05. 2023

Machinery

Section 1 General

1.1. Application

1.1.1. The present Chapter is to be applied to the following engines and machinery:

.1. Oil engines.
.2. Steam turbines.
.3. Gears and couplings.
.4. Engines driving electric generators or auxiliary and deck machinery, units in assembly.
.5. Pumps
.6. Air compressors.
.7. Fans of main boilers and turbo blowers, or other types of fans of the oil engines.
.8. Fans
.9. Steering gear.
.10. Anchor machinery.
.11. Towing winches.
.12. Mooring machinery.
.13. Hydraulic drives.
.14. Centrifugal separators.
.15. Gas turbines.

1.2. Scope of supervision

1.2.1. The provisions specifying the procedure of survey conducted by *QRS Class* during the manufacture of the machinery and equipment, as well as the procedure of consideration and approval of technical documentation are contained in *General Regulations for the Supervision*.

1.2.2. *QRS Class* carries out the supervision during the manufacture of engines and machinery.

1.2.3. Prior to manufacturing of the machinery, the following documents are to be submitted to QRS Class approval:

.1. Oil engines.

.1.1. Engine particulars as per data sheet or specification.

.1.2. General view plans with engine longitudinal and transverse sections.

.1.3. Drawings of bedplate, columns, engine bed, crankcase, casing and other parts, cast or welded, with welding details and instructions.

.1.4. Assembly drawing of thrust bearing as well as thrust bearing casing, cast or welded, with welding details and instructions (if thrust bearing is integral with engine but not integrated in bedplate).

.1.5. Assembly drawing of cylinder cover.

.1.6. Drawing of the rods.

.1.7. Drawings of cylinder jacket or engine block as well as cylinder liner.

.1.8. Drawings of connecting rod, crosshead and rod.

.1.9. Drawings of crankshaft as an assembly and details.

.1.10. Drawings of counterweights (if not integral with crankshaft).

.1.11. Drawing of thrust shaft or intermediate shaft (is integral with engine).

.1.12. Drawing of piston as an assembly.

.1.13. Drawing of coupling bolts.

.1.14. Assembly drawing of camshaft and its drive.

.1.15. Specification of main details material with indication of test pressure values (where required).

.1.16. Drawings of securing engine structure to the foundation and arrangement of foundation bolts.

.1.17. Drawings of main piping and systems associated with engine:

- starting air;
- fuel oil;
- lubricating oil;
- cooling water;
- control, governing and protection;
- shielding and insulation of the gas exhaust pipes.

.1.18. Drawings of high pressure delivery fuel oil piping and their protection in case of damage.

.1.19. Drawings of the crankcase safety valves and scavenging air manifold and their arrangement.

.1.20. Strength calculations pertaining to machinery parts regulated by the rules.

.1.21. Test program for prototype and production models of engines

.1.22. Operating and maintenance instruction of the engine

.1.23. Drawing of the torsional vibration damper or antivibrator (if provided), description and operation manual.

.1.24. drawings of camshaft gear and chain drive.

.1.25. drawings of the hydraulic system for engine valve control.

.1.26. where engines incorporate the electronic control system, the failure mode and effects analysis shall be submitted to demonstrate that the failure of the electronic control system will not result in the loss of essential services for the operation of the engine and that operation of the engine will not be lost or degraded beyond the stated performance criteria of the engine.

.2. All other machinery regulated by the present Chapter except for oil engines:

.2.1. Machinery particulars as per data sheet or specification.

.2.2. General view plans with machinery longitudinal and transverse sections.

.2.3. Drawings of bedplates, crankcases, engine beds, casings, covers and other parts, cast or welded, with welding details and instructions.

.2.4. Drawings of crankshafts, thrust shafts, output and other shafts as well as their drives (gears)

.2.5. Drawings of connecting rods, piston rods and pistons.

.2.6. Drawings of cylinder covers and cylinder liners.

.2.7. Drawings of pinions, gear wheels and their shafts.

.2.8. Drawings of driving and driven parts of hydraulic gears, disengaging and flexible couplings.

.2.9. Drawing of thrust blocks built in the machinery.

.2.10. Drawings of rotors of steam and gas turbines and compressors as well as discs and impellers.

.2.11. Drawings of high pressure fuel oil piping and their protection in case of damage.

.2.12. Drawings of insulation and lining of gas exhaust piping associated with machinery

.2.13. Drawings of main pipings and fuel oil, lubricating oil, cooling, gas exhaust, scavenging, air control, governing, alarm, protection and other systems associated with machinery.

.2.14. Drawings of machinery hydraulic piping system with hydraulic drives.

.2.15. Drawings of securing machinery structure to bedplate and arrangement of foundation bolts (only for main machinery, electric generator drives, steering gears, anchor, mooring and towing machinery).

.2.16. Strength calculations of machinery parts, regulated by the Rules.

.2.17. List of main parts of machinery with material specification and all details for test pressure values (if required).

.2.18. Operation and service manuals.

.2.19. Test programs for prototype and production models of machinery.

1.2.4. Drawings of machinery parts listed in table 2000 are subject to agreement with *QRS Class*.

In the process of manufacture all these parts are subject to supervision of QRS Class.

1.2.5. Rotors, shafts and disks of steam turbines and gas turbine engines, as well as the bolts for joints of casings of high pressure turbines are subject to ultrasonic testing during manufacture. Shafts of main gears and tillers more than 100 kg in mass, pinions, tooth rims more than 250 kg in mass are subject to ultrasonic testing during manufacture. Parts of oil engines of steel are also subject to ultrasonic testing during manufacture in accordance with the requirements of *table 2100*.

Table 2000 Drawings of machinery parts

N°	Items	Materials
1	**Oil engines**	
1.1	Bedplate, crankcase, frames, thrust bearing casing integral with engine	cast iron, cast steel, forged steel, rolled steel, aluminium alloy
1.2	Cylinder block, cylinder covers, valve housing	cast iron cast steel

N°	Items	Materials
1.3	Cylinder liners and their parts	cast iron cast steel forged steel
1.4	Piston	cast iron cast steel forged steel aluminium alloy
1.5	Piston rod, crossheads, gudgeon pins and connecting rod	forged steel
1.6	Crankshaft, thrust shaft of the built-in thrust bearing	forged steel cast steel cast iron
1.7	Crankshaft detachable couplings	forged steel cast steel
1.8	Bolts and studs of the crossheads, crank and main bearings, cylinder covers	forged steel
1.9	Tie rods	forged steel
1.10	Connecting bolts of crankshaft sections	forged steel
1.11	Shaft and rotor of the turbocharger	forged steel
1.12	Camshaft, camshaft drive gears	forged steel
1.13	Speed governors and overspeed devices	
1.14	Safety valves of the crankcase (for engines having a bore exceeding 200 mm)	

N°	Items	Materials
1.15	Counterweights if they are not integral with the crankshaft	forged steel cast steel cast iron
2	**Steam turbines**	
2.1	Casings of turbines	cast iron cast steel rolled steel
2.2	Maneuvering gear casings, nozzle boxes	cast steel
2.3	Solid-forged rotors, shafts and disks	forged steel
2.4	Blades	forged steel cast steel
2.5	Shrouds and lashing wire	
2.6	Nozzles and diaphragms	cast iron forged steel cast steel
2.7	Gland seals	
2.8	Couplings	forged steel cast steel
2.9	Bolts for joints of rotor parts, split casings and couplings	forged steel
3	**Gears, elastic and disengaging couplings**	

N°	Items	Materials
3.1	Casing	cast iron forged steel rolled steel cast steel aluminum alloy
3.2	Shafts	forged steel
3.3	Pinions, wheel, wheel rims	forged steel cast steel
3.4	Coupling components transmitting the torque: 1. Rigid components 2. Elastic components	 rolled steel forged steel cast steel cast iron aluminium alloy rubber, synthetic material spring steel
3.5	Coupling bolts	forged steel
4	**Compressors and piston-type pumps**	
4.1	Crankshaft	forged steel cast steel cast iron
4.2	Piston rod	forged steel

N°	Items	Materials
4.3	Connecting rod	forged steel cast iron aluminium alloy
4.4	Piston	forged steel cast steel cast iron copper alloy aluminium alloy
4.5	Cylinder block, cylinder covers	cast steel cast iron
4.6	Cylinder liner	cast iron
5	**Centrifugal pumps, fans and air blowers**	
5.1	Shaft	rolled steel forged steel
5.2	Impeler, paddle wheel, lobes	cast steel copper alloy aluminium alloy
5.3	Casing	cast iron cast steel rolled steel copper alloy aluminium alloy
6	**Steering gear**	

N°	Items	Materials
6.1	Tiller of main and emergency gear	forged steel cast steel
6.2	Rudder quadrant	cast steel
6.3	Rudder stock yoke	forged steel
6.4	Pistons with rods	forged steel cast steel
6.5	Cylinders	steel tube cast steel cast iron
6.6	Drive shaft	forged steel
6.7	Pinions, gear wheels, tooth rims	forged steel cast steel cast iron
7	**Windlasses, capstans, mooring and towing winches**	
7.1	Drive, intermediate and output shafts	forged steel
7.2	Pinions, gear wheels and tooth rims	forged steel cast iron cast iron
7.3	Sprockets	cast steel cast iron
7.4	Claw clutches	forged steel cast steel
7.5	Brake band	rolled steel

N°	Items	Materials
8	**Hydraulic drives, screw, gear and rotary pumps**	
8.1	Shaft, screw, rotor	forged steel cast steel copper alloy
8.2	Piston rod	forged steel copper alloy
8.3	Piston	forged steel cast steel
8.4	Casing, cylinder and housing of screw pump	cast steel cast iron copper alloy
8.5	Pinions	forged steel cast steel cast iron copper alloy
9	**Centrifugal fuel and lubricating oil separators**	
9.1	Bowl shaft	forged steel
9.2	Bowl body, bowl discs	forged steel
9.3	Drive pinions	forged steel copper alloy
10	**Gas turbines**	
10.1	Casings of turbines and compressors, diaphragms and combustion chamber casings	rolled steel cast steel

N°	Items	Materials
10.2	Rotors and discs of turbines	forged steel
10.3	Rotors and discs of compressors	forged steel
10.4	Turbine blades	rolled steel forged steel cast steel
10.5	Compressor blades	forged steel cast steel
10.6	Shrouds and lashing wire	
10.7	Flame tubes of combustion chambers	rolled steel
10.8	Heat-exchanging surfaces of regenerators	rolled steel
10.9	Sealings	
10.10	Flanges of couplings	forged steel cast steel
10.11	Bolts for joints of rotor parts, turbines and compressor split casings	forged steel

Parts of internal combustion engines of steel are also subject to ultrasonic testing during manufacture in accordance with the requirements of Table 2100.

Table 2100 Requirements for the steel parts

N°	Cylinder bore, mm	Part N° according to table 2000
1	Up to 400 inclusive	*1.1, 1.2, 1.4, 1.6* and *1.7*
2	More than 400	*1.1, 1.2, 1.4-1.7*

1.2.6. For the oil engines the steel cast and forged parts listed in table 2200, their welded connections included, shall be tested during the manufacture for the absence of the surface defects by the magnetic particle or liquid penetrant method.

The rubber blades of main and auxiliary turbines, guide blades of main turbines and turbine blades of gas turbine engines are also to be subjected to the above testing.

Table 2200 Requirements for the steel cast and forged parts

N°	Cylinder bore, mm	Part N° according to table 2000
1	Up to 400 inclusive	*1.1, 1.5, 1.6*
2	More than 400	*all parts*

1.2.7. If there are doubts about the absence of defects in the part material, *QRS Class* may require to carry out a non-destructive testing of other machinery parts and their welded connections.

1.3. Hydraulic test

1.3.1. The machinery parts, with the exception of the oil engine parts, operating under excessive pressure are to be subjected to a hydraulic test by a pressure p_{test} after final machining and before protective coating is applied. The hydraulic test pressure p_{test}, in MPa, is found by the formula:

$$p_{test} = (1,5 + 0,1 \cdot k) \cdot p$$

where: p = working pressure, MPa;
k = factor taken from *table 2300*.

In all cases, the value of test pressure shall not be lower than the pressure setting with the safety valve fully open, but not less than 0,4 MPa for cooled spaces of parts and various seals and not less than 0,2 MPa in all other cases. If temperatures or working pressures exceed the ratings indicated in *table 2300*, the value of test pressure shall be approved by *QRS Class* in each case.

1.3.2. The machinery parts and assemblies may be tested separately along the spaces by test pressures prescribed in compliance with the working pressures and temperatures inside each space.

1.3.3. Parts of oil engines are to be tested according to the requirements specified in *table 2400*.

1.3.4. The machinery parts and assemblies are filled with petroleum products or their vapors (viz., reduction gear casings, sumps, and so on.) under hydrostatic or atmospheric pressure are to be tested for oil-tightness by the method approved by *QRS Class*. Oil-tightness tests of welded structures may be confined to welded seams only.

1.4. Operation test

On completion of assembly, adjustment and running-in, each piece of machinery shall be bench tested under the load conditions prior to installation aboard the ship. The test program shall be approved by *QRS Class*.

In particular cases, bench test may be substituted by test aboard the ship on agreement with *QRS Class*.

The pilot models of the machinery are to be tested under a program providing for checking reliability and long-term operational capacity of certain unit components and of the machinery as a whole.

1.5. General technical requirements

1.5.1. Machinery indicated in *1.1* is to remain operative under specified environmental conditions.

1.5.2. The design of the main engines intended for installation aboard single-shaft ships shall provide, as a rule, for a possibility of emergency operation at reduced power in case of a failure of parts, the replacement of which cannot be carried out aboard the ship or demands much time.

1.5.3. The forged, cast and welded steel parts, as well as cast iron parts of the machinery are to be heat treated during manufacture.

1.5.4. The fasteners used in moving parts of machinery and gears, as well as fasteners difficult for access are to be properly designed or to have special arrangements aimed at preventing their self-loosening and self-releasing.

1.5.5. The heated surfaces of machinery and equipment are to be insulated.

1.5.6. The machinery parts that are in contact with a corrosive medium are to be made of an anticorrosive material or to have corrosion-resistant coatings.

1.5.7. Sea water cooling spaces of engines and coolers should be provided with protectors.

Table 2300 Temperatures and working pressures

Material	**Character-istic**	**Part N° according to table 2000**									
		120	**200**	**250**	**300**	**350**	**400**	**430**	**450**	**475**	**500**
Carbon steel	p, MPa	-	20	20	20	20	10	10	10	-	-
	k	0	0	1	3	5	8	11	14	-	-
Molybdenum and molybdenum-chrome steel with at least 0,4% molybdenum content	p, MPa	-	-	-	-	20	20	20	20	20	20
	k	0	0	0	0	0	1	2	3,5	6	11
Cast iron	p, MPa	6	6	6	6	-	-	-	-	-	-
	k	0	2	3	4	-	-	-	-	-	-

Material	Character-istic	Part N° according to table 2000									
		120	200	250	300	350	400	430	450	475	500
Bronze, brass and copper	p, Mpa	20	3	3	-	-	-	-	-	-	-
	k	0	3,5	7	-	-	-	-	-	-	-

Table 2400 Requirements for the parts of the Oil engines

Item	Test pressure[1]
Cylinder cover, cooling space Cylinder liner over the whole length of cooling space Piston crown, cooling space after assembly with the piston rod, if the latter forms a sealing	0,7 MPa
Cylinder block, cooling space Exhaust valve (body), cooling space Turbocharger, cooling space Exhaust piping, cooling space Coolers (from both sides)[2] Engine-driven pumps (lubricating oil, water, fuel booster, bilge) - working spaces	1,5p but not less than 0,4 MPa
Engine-driven compressors including cylinders, covers and air coolers: - air side - water side	1,5 p 1,5 p but no less than 0,4 MPa

Item	Test pressure[1]
Casings of the high pressure fuel pumps (pressure side), fuel valves and high pressure fuel pipes[3]	1,5p ó p+30 MPa whichever is the less
Scavenging pump cylinder	0,4 MPa
NOTES: 1. The above-stated norms may be changed for separate types of engine in agreement with QRS Class. 2. Turbocharger coolers are to be subjected to hydraulic tests only from the water side. 3. These pressure values for hydraulic tests do not apply to fuel pumps with the regulating edge of a plunger.	

1.6. Materials and welding

1.6.1. Materials of parts stated in items may be also selected according to the standards in force. In this case, the application of materials is to be subject to agreement with QRS Class during consideration of the technical documentation.

1.6.2. Materials of the parts of oil engines are to be subject to supervision of QRS Class in accordance with table 2500.

Table 2500 Parts of oil engines, according to table 2000, that are to be subject to supervision

N°	**Cylinder bore, mm**	**Part N° according to table 2000**
1	Up to 300 inclusive	*1.1, 1.5, 1.6, 1.9*
2	From 301 to 400 inclusive	*1.1, 1.2, 1.3, 1.5, 1.6, 1.8 1.9, 1.11*
3	More than 400	All parts from *1.1* to *1.12*

1.6.3. When the alloy steels, including heat resistant, high temperature oxidation resistant and high strength steels, or alloy cast iron is used for the machinery parts, the information on chemical composition, mechanical and special properties confirming suitability of the material for intended application shall be submitted to *QRS Class*.

1.6.4. The parts of steam turbines and gas turbine engines operating under the conditions of high temperatures (400 °C and above) are to be subjected to tensile test at the design temperature and, if necessary, QRS Class may require to submit the information on the average stress to produce rupture at the design temperature.

1.6.5. Graphite cast iron is allowed for use up to the temperature of 300 °C, and grey cast iron - up to 250 °C.

Machinery

Section 2 Internal combustion engines

2.1. General provisions

The requirements of the present Section are applicable to all oil engines of power output 55 kW and above.

Application of these requirements to the oil engines of power output less than 55 kW shall be subject to special consideration by *QRS Class* in each case.

2.2. General requirements

2.2.1. The engines are to be capable of working with an overload exceeding the rated power by at least 10 per cent for not less than one hour.

2.2.2. Irregularity of speed of a.c. diesel generating sets intended for parallel operation are such that the amplitude of angle oscillations of the generator shaft does not exceed 3,5°/p, where p is the number of pairs of generator poles.

2.2.3. The crosshead-type engines whose scavenge spaces are in open connection with the cylinders are to be provided with the fire extinguishing system approved by *QRS Class* which is entirely separate from the fire extinguishing system of the engine room.

The scavenge spaces of the main engines in ships with unattended machinery spaces of category *A* are to be equipped with a timely fire alarm and fire detection system.

2.2.4. The diesel generating sets intended as emergency sources are to be provided with self-contained fuel supply, cooling and lubricating systems.

2.2.5. Engines intended to drive emergency generators, which may be also used as sources of electrical power for non-emergency consumers shall be equipped with oil fuel and lubricating oil filters, as well as with monitoring equipment, alarm and protective devices as required for prime movers of the main sources of electrical power when in unattended operation.

Along with that, their oil fuel supply tanks shall be fitted with a low level alarm arranged at a level ensuring sufficient oil fuel capacity for the emergency services. Besides, such engines shall be designed for continuous operation and shall be subjected to a planned maintenance scheme ensuring that it is always available and capable of fulfilling its role in the event of an emergency at sea.

2.2.6. The rated power of the engines shall be determined under the following conditions:

atmospheric pressure, kPa —100;
air temperature, °C — 45;
relative humidity, % — 60;
sea water temperature, °C — 32.

2.2.7. In the crankshaft speed range (0 —1,2) nr, where nr is the rated speed, no restricted speed areas shall be permitted.

2.2.8. Fuel oil and lubricating oil pipes, valves, flanged connections, filters shall be screened or otherwise protected so that in case of their failure petroleum products fall onto hot surfaces.

2.2.9. Where special tools and gauges are required for maintenance purposes, these shall be supplied by the manufacturer. Engine servicing shall be performed in compliance with the manufacturer's recommendations.

2.2.10. For engines with electronic control system where the basic operation processes (fuel supply, gas exchange, starting and reversing, cylinder lubrication) are performed by means of hydraulic (pneumatic) systems controlled by programmable electronic devices upon a signal from the crankshaft-position sensor, a single failure of components of the electronic control system shall not result in the loss of manoeuvrability or in spontaneous stoppage of the engine.

2.3. Engine frame

2.3.1. The mating surfaces of the frame parts forming the engine crankcase are to be close-fitting and oil- and gas-tight as well as are to be fixed together by means of calibrating pieces.

2.3.2. The engine frame and conjugated parts shall be provided with draining arrangements (drain grooves, pipes, and so on.) and other facilities preventing penetration of fuel and water into the circulating oil. The cooling spaces of the cylinder blocks shall be fitted with drain arrangements providing complete drainage.

2.3.3. Engines with a cylinder bore in excess of 230 mm shall be fitted with alarm devices to give a signal indicating that the specified excess of the maximum combustion pressure in a cylinder has been reached.

The cooling spaces of the cylinder blocks are to be fitted with drain arrangements providing complete drainage.

2.3.4. Protection of internal combustion engines against crankcase explosions.

2.3.5. Crankcase construction and crankcase doors shall be of sufficient strength to withstand anticipated crankcase pressures that may arise during a crankcase explosion taking into account the installation of explosion relief valves required. Crankcase doors are to be fastened sufficiently securely for them not to be readily displaced by a crankcase explosion.

2.3.6. Additional relief valves shall be fitted on separate spaces of crankcase such as gear or chain cases for camshaft or similar drives, when the gross volume of such spaces exceeds 0,6 m^3.

2.3.7. Scavenge spaces in open connection to the cylinders shall be fitted with explosion relief valves.

2.3.8. Ventilation of crankcase, and other setups which could produce a flow of external air within the crankcase, is in principle not permitted except for dual fuel engines where crankcase ventilation shall be provided.

2.3.9. Crankcase ventilation pipes, where provided, shall be as small as practicable to minimise the inrush of air after a crankcase explosion. The ends of the ventilation pipes shall be fitted with flame arresting devices and arranged as to prevent water from getting into the engine.

Ventilation pipes shall be laid to the weather deck to locations preventing the suction of vapours into accommodation and service spaces.

For engines with power output up to 750 kW suction of gas from the crankcase by turbochargers or blowers may be admitted, provided reliable oil separators are fitted to prevent the oil from being carried into the engine with suction gas.

2.3.10. The oil mist detection system and arrangements shall be installed in accordance with the engine designer's and oil mist detection arrangements manufacturer's instructions / recommendations. The following particulars shall be included in the instructions:

.3.10.1. schematic layout of engine oil mist detection and alarm system showing location of engine crankcase sample points and piping or cable arrangements together with pipe dimensions to detector;

.3.10.2. evidence of study to justify the selected location of sample points and sample extraction rate (if applicable) in consideration of the crankcase arrangements and geometry and the predicted crankcase atmosphere where oil mist can accumulate;

.3.10.3. the manufacturer's maintenance and test manual;

.3.10.4. information relating to type or in-service testing of the engine with engine protection system test arrangements having approved types of oil mist detection equipment.

2.3.11. If a forced extraction of the oil mist atmosphere from the crankcase is provided (for mist detection purposes, for instance), the vacuum in the crankcase shall not exceed 250 P.

2.3.12. To avoid interconnection between crankcases and the possible spread of fire following an explosion, crankcase ventilation pipes and oil drain pipes for each engine shall be independent of other engines.

2.3.13. Lubricating oil drain pipes from the engine sump to the drain tank shall be submerged at their outlet ends.

Crankcase drain outlets shall be fitted with grates and grids preventing foreign objects from getting into the drain piping. The above requirement is also applied to engines with dry crankcase.

2.3.14. A warning notice shall be fitted either on the control stand or, preferably, on a crankcase door on each side of the engine. This warning notice shall specify that, whenever overheating is suspected within the crankcase, the crankcase doors or sight holes shall not be opened before a reasonable time, sufficient to permit adequate cooling after stopping the engine.

2.3.15. Oil mist detection arrangements (or engine bearing temperature monitors or equivalent devices) are required:

.3.15.1. for alarm and slow down purposes for low speed diesel engines of 2250 kW and above or having cylinders of more than 300 mm bore;

.3.15.2. for alarm and automatic shutoff purposes for medium and high speed diesel engines of 2250 kW and above or having cylinders of more than 300 mm bore.

Oil mist detection arrangements shall be of a type approved by the Register or any IACS member.

Engine bearing temperature monitors or equivalent devices used as safety devices shall be of a type approved by *Qualitas Register of Shipping* for such purposes.

2.3.16. An engine installed on board a ship shall be provided with a manufacturer's maintenance and test manual of oil mist detection arrangements.

2.3.17. Oil mist detection and alarm information shall be capable of being read from a safe location away from the engine.

2.3.18. Each engine shall be provided with its own independent oil mist detection arrangement and a dedicated alarm.

2.3.19. Oil mist detection and alarm systems shall be capable of being tested on the test bed and board under engine at standstill and engine running at normal operating conditions in accordance with test procedures approved by *Qualitas Register of Shipping*.

2.3.20. The oil mist detection arrangements shall provide an alarm indication in the event of a foreseeable functional failure in the equipment and installation arrangement.

2.3.21. The oil mist detection system shall provide an indication that lenses fitted in the equipment and used in determination of the oil mist level have been partially obscured to a degree that will affect the reliability of the information and alarm indication.

2.3.22. Where oil mist detection equipment includes the use of programmable electronic systems, the arrangements are the matter of special consideration by *QRS Class*.

2.3.23. Plans showing details and arrangements of oil mist detection and alarm arrangements shall be approved by *QRS Class*.

2.3.24. The equipment together with detectors shall be tested when installed on the test bed and on board ship to demonstrate that the detection and alarm system functionally operates. The testing arrangements shall be approved by *Qualitas Register of Shipping*.

2.3.25. Where sequential oil mist detection arrangements are provided the sampling frequency and time shall be as short as reasonably practicable.

2.3.26. Where alternative methods are provided for the prevention of the build-up of oil mist that may lead to a potentially explosive condition within the crankcase details shall be submitted for special consideration by *QRS Class*. the following information shall be included in the details to be submitted for consideration:

.3.26.1. engine particulars - type, power, speed, stroke, bore and crankcase volume;

.3.26.2. details of arrangements preventing the build up of potentially explosive conditions within the crankcase, e. g., bearing temperature monitoring, oil splash temperature, crankcase pressure monitoring and recirculation arrangements;

.3.26.3. evidence to demonstrate that the arrangements are effective in preventing the build up of potentially explosive conditions together with details of inservice experience;

.3.26.4. operating instructions and the maintenance and test instructions.

2.3.27. Where it is proposed to use the introduction of inert gas into the crankcase to minimise a potential crankcase explosion, details of the arrangements shall be submitted to QRS Class for consideration.

2.3.28. Engine crankcases explosion relief valves:

.3.28.1. Engines having a cylinder bore of 200 mm and above or a crankcase volume of 0,6 m3 and above shall be provided with crankcase explosion relief valves as follows.

.3.28.1.1. Engines having a bore exceeding 200 mm, but not exceeding 250 mm, shall have at least one valve near each end but, over 8 crankthrows, an additional valve shall be fitted near the middle of the engine.

.3.28.1.2. Engines having a bore exceeding 250 mm, but not exceeding 300 mm, shall have at least one valve in way of each alternate crankthrow with a minimum of two valves.

.3.28.1.3. Engines having a bore exceeding 300 mm shall have at least one valve in way of each main crankthrow.

.3.28.2. The free area of each relief valve shall be not less than 45 cm2.

.3.28.3. The combined free area of the valves fitted on an engine shall be not less than 115 cm2 per 1 m 3 of the crankcase gross volume. In estimating the crankcase gross volume the stationary parts may be discounted (however, the rotary and reciprocating components shall be included into the gross volume).

.3.28.4. Crankcase explosion relief valves shall be provided with lightweight spring-loaded valve discs or other quick-acting and self-closing devices to relieve a crankcase of pressure in the event of an internal explosion and to prevent the inrush of air thereafter.

.3.28.5. The valve discs in crankcase explosion relief valves shall be made of ductile material capable of withstanding the shock of contact with stoppers at the full open position.

.3.28.6. Crankcase explosion relief valves shall be designed to open quickly and be fully open at an overpressure in the crankcase of not greater than 0,02 MPa.

.3.28.7. Crankcase explosion relief valves shall be provided with flame arresters that permits flow for crankcase pressure relief and prevents passage of flame following a crankcase explosion.

.3.28.8. Crankcase explosion relief valves shall be of type approved by the Register and be tested in a configuration that represents the installation arrangements that will be used on an engine.

.3.28.9. Where crankcase explosion relief valves are provided with arrangements for shielding emissions from the valve following an explosion, the valve shall be type tested to demonstrate that the shielding does not adversely affect the operational effectiveness of the valve.

.3.28.10. In a delivery set of crankcase explosion relief valves a copy of the manufacturer's installation and maintenance manual shall be provided that is pertinent to the size and type of valve being supplied for installation on a particular engine. The manual shall contain the following information:

.3.28.10.1. description of the valve with details of functional and design limits;
.3.28.10.2. copy of Type Approval/Test Certificate;
.3.28.10.3. installation instruction;
.3.28.10.4. maintenance and in-service instructions including testing and replacement of sealing arrangements;
.3.28.10.5. actions required after a crankcase explosion.

.3.28.11. Details of crankcase explosion relief valves design and arrangement shall be submitted for QRS Class approval.

.3.28.12. Valves shall be provided with suitable marking including the following information:

.3.28.12.1. name and address of the manufacturer;
.3.28.12.2. designation and size;
.3.28.12.3. date of manufacture;
.3.28.12.4. approved installation orientation.

2.4. Crankshafts

2.4.1. The check calculation method as described below is applicable to solid-forged and semi-built crankshaft of forged or cast steel intended for marine diesel engines having the cylinders either in line or in V-arrangement, with one crankthrow between main bearings.

The crankshafts of engines having the cylinders arranged otherwise than stated in this paragraph, deviations from the steel crankshaft scantlings determined from the formula below as well as cast iron crankshafts may be approved on agreement with QRS Class provided calculations to substantiate such an arrangement or experimental data are submitted.

2.4.2. The outlets of oil bores into crankpins and journals shall be formed in such a way that the safety margin against fatigue at the oil bores is not less than that acceptable in the fillets. The engine manufacturer, if requested by QRS Class, shall submit documentation supporting his oil bore design.

2.4.3. For the calculation of crankshafts, the documents and particulars listed in the following are to be submitted:

- Crankshaft drawing which must contain all scantlings required by this Subsection.
- Type designation and kind of engine (in-line engine or V-type engine with adjacent connecting rods, forked connecting rod or articulated-type connecting rod.).
- Operating and combustion method (direct injection, precombustion chamber, and so on.).
- Number of cylinders.
- Rated power, kW.
- Rated engine speed, min^{-1}.
- Sense of rotation.

- Firing order with the respective ignition intervals and, where necessary, V-angle α_v, deg.
- Cylinder diameter, mm.
- Stroke, mm.
- Maximum cylinder pressure P_{max} , MPa.
- Charge air pressure, in MPa, before inlet valves or scavenge ports, whichever applies.
- Nominal compression ratio.
- Connecting rod length L_H; mm.
- Oscillating weight of one crank gear, in kg (in case of V-type engines, where necessary, also for the cylinder unit with master and articulated-type connecting rod or forked and inner connecting rod).
- Digitalized gas - pressure - versus - crank - angle curve presented at equidistant intervals and integrally divisible by the V-angle, but not more than 5°. For bending moments, shearing forces and torques.

Details of crankshaft material:

- Material designation (according to standards, and so on.).
- Chemical composition.
- Tensile strength, R_m; MPa.
- Yield strength, R_e; MPa.
- Reduction in area at break, Z; %.
- Elongation, A_5; %.

- Impact energy, k_V; J
- Method of material melting process (basic oxygen furnace, open-hearth furnace, electric furnace, and so on.).
- Heat treatment applied.
- Type of forging (free form forged, continuous grain flow forged, and so on.; with description of the forging process).
- Surface treatment of fillets, journals and pins (induction hardened, flame hardened, nitrided, rolled, shot peened).
- Hardness at surface, H_V
- Hardness as a function of depth; mm
- Extension of surface hardening.

For engines with articulated-type connecting rod the following details should be submitted additionally:

L_A :	Distance to link point; mm;
α_N :	link angle, deg;
L_H :	connecting rod length; mm;
L_N :	articulated-type connecting rod length; mm.

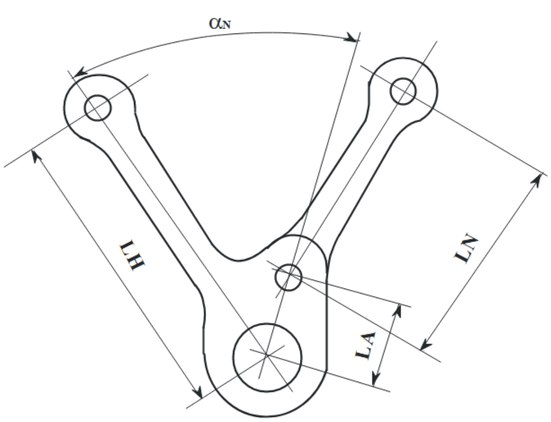


Figure 100 Articulated-type connecting rod

2.5. Calculation of alternating stresses due to bending moments and shearing forces

1. Assumptions

The calculation is based on a statically determined system, so that only one single crankthrow is considered of which the journals are supported in the centre of adjacent bearings and which is subject to gas and inertia forces (see figs 200 and 300).

The nominal bending moment is taken as a moment with the bending lever (distance L_1), due to the radial components of the connecting rod force. For crankthrows with two connecting rods acting upon one crankpin the nominal bending moment is taken as a bending moment obtained by vector sum of the moments that are applied by such connecting rods.

The nominal alternating stresses due to bending moments and shearing forces are to be related to the cross-sectional area of the crank web in the center of the overlap of the pins or at the center of the adjacent generating lines of the two pins if they do not overlap (see *fig 400*).

2. Calculation of nominal alternating bending and shearing stresses. The maximum and minimum bending moment values M_{Bmax} y M_{Bmin} as well as the maximum and minimum shearing force values Q_{max} y Q_{min} should be submitted to *QRS Class*, determined by calculating the radial forces acting upon the crankpin owing to gas and inertia forces.

In agreement with *QRS Class*, a simplified calculation of the radial forces may be submitted. The nominal alternating bending moment M_{BN} (in $N \cdot m$), will be determined as:

$$M_{BN} = \pm \frac{1}{2} \cdot (M_{Bmax} - M_{Bmin})$$

The nominal alternating bending stress σ_{BN} (in MPa), will be determined from the formula:

$$\sigma_{BN} = \pm \frac{M_{BN}}{W_{eq}} \cdot 10^3 \cdot K_e$$

where:

W_{eq} = moment of resistance related to cross-sectional area of web, mm³;

$$W_{eq} = \frac{B \cdot W^2}{6};$$

K_e = factor equal to 0,8 for 2-stroke engines and 1,0 for 4-stroke engines.

The nominal alternating shearing stress σ_{QN} (in MPa), will be determined from the formula:

$$\sigma_{QN} = \pm \frac{Q_N}{F} \cdot K_e$$

where:

Q_N = nominal alternating shearing force, in N;

$$Q_N = \pm \frac{1}{2} \cdot (Q_{max} - Q_{min});$$

F = area related to cross-section of web, in mm²;

$$F = B \cdot W$$

3. Calculation of alternating bending stresses in fillets. The alternating bending stress in a crankpin fillet σ_{BH}, in MPa, will be determined from the formula:

$$\sigma_{BH} = \pm (\alpha_B \cdot \sigma_{BN})$$

where:
α_B = stress concentration factor for bending in crankpin fillet.

The alternating bending stress in a journal fillet σ_{BG}, in MPa, will be determined from the formula:

$$\sigma_{BG} = \pm (\beta_B \cdot \sigma_{BN} + \beta_Q \cdot \sigma_{QN})$$

where:
β_B = stress concentration factor for bending in journal fillet;
β_Q = stress concentration factor for shearing.

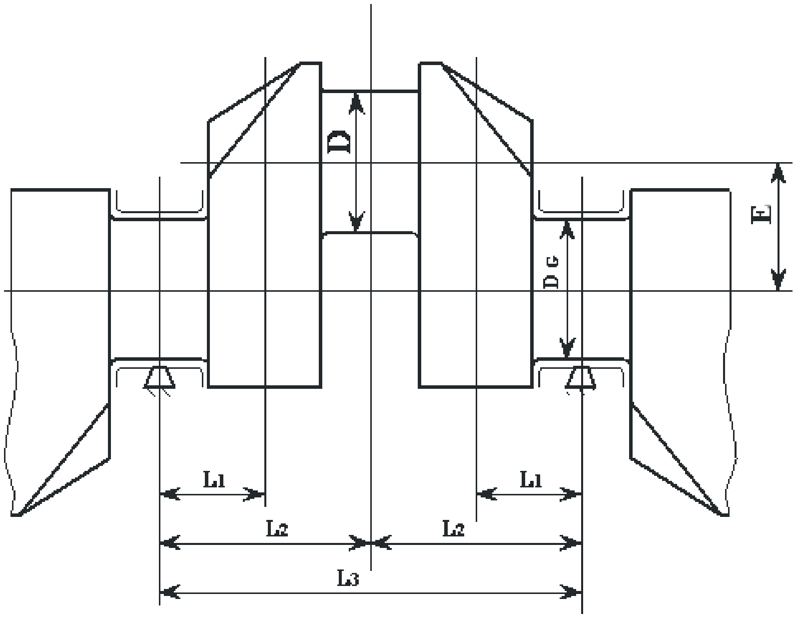


Figure 200 Crankthrow for in-line engine

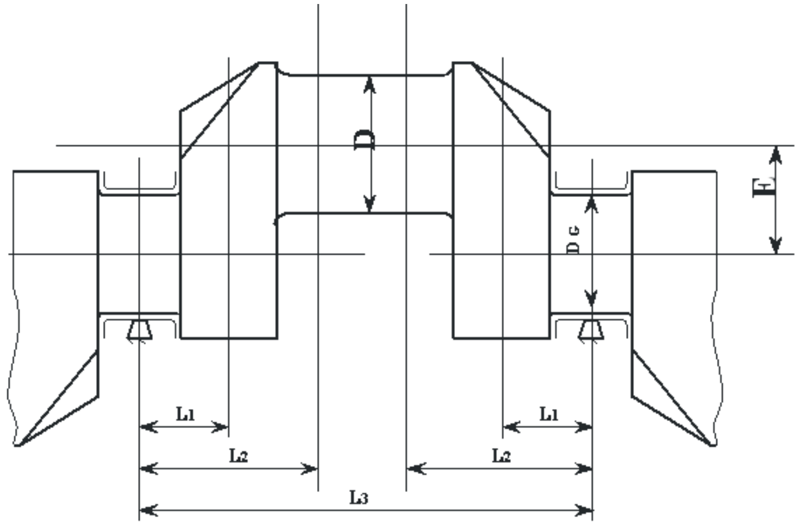


Figure 300 Crankthrow for V-type engine where in a common crankpin are installed two connecting rods

2.6. Calculation of alternating torsional stresses

2.6.1. Calculation of nominal alternating torsional stresses

The calculation for nominal alternating torsional stresses is to be undertaken by the engine Manufacturer according to the information below. The maximum values obtained from such calculations should be submitted to *QRS Class*.

The maximum and minimum alternating torques are to be ascertained for each crankthrow and for the entire speed range by means of a harmonic synthesis of the forced vibrations from the 1st order up to and including the 16th order for 2-stroke cycle engines and from 0,5th order up to and including the 12th order for 4-stroke cycle engines.

Allowance must be made for the dampings that exist in the system and unfavourable conditions (misfiring in one of the cylinders). The speed ranges should be selected in such a way that the transient response can be recorded with sufficient accuracy.

The nominal alternating torsional stress τ_N, in MPa, referred to crankpin or journal should be determined from the formula:

$$\tau_N = \pm \frac{M_T}{M_P} \cdot 10^3$$

where:

M_T = nominal alternating torque, in $N \cdot m$, to be determined from the formula:

$$M_T = \pm \frac{1}{2} (M_{Tmax} - M_{Tmin});$$

M_{Tmax} = maximum value of the torque with consideration of the mean torque, in $N \cdot m$;
M_{Tmin} = minimum value of the torque with consideration of the mean torque, in $N \cdot m$;
W_P = polar moment of resistance related to cross-sectional area of bored crankpin or bored journal (in mm³), and determined from the formula:

$$W_P = \frac{P}{16} \cdot \left(\frac{D^4 - D^4_{BH}}{D}\right) \text{ ó}$$

$$W_P = \frac{P}{16} \cdot \left(\frac{D^4_G - D^4_{BG}}{D_G}\right);$$

For D, D_{BH} and D_{BG}.

2.7. Calculation of alternating torsional stresses in fillets

In the crankpin fillet, the alternating torsional stress τ_H, (in MPa), should be determined from the formula:

$$\tau_H = \pm\left(\alpha_T \cdot \tau_N\right)$$

where:
α_T = stress concentration factor for torsion in crankpin fillet.

In the journal fillet, the alternating torsional stress, τ_G, (in MPa), should be determined from the formula:

$$\tau_G = \pm\left(\beta_T \cdot \tau_N\right)$$

where:

β_T = stress concentration factor for torsion in journal fillet.

2.8. Calculation of stress concentration factors

2.8.1. Where the stress concentration factor can not be furnished by reliable measurements the values may be evaluated by means of the formula applicable to the fillets of solid-forged web-type crankshafts and to the crankpin fillets of semi-built crankshafts.

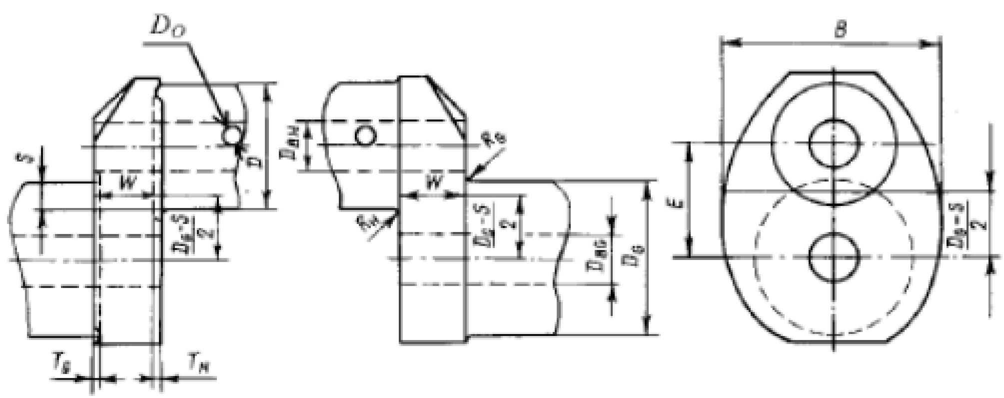


Figure 400 Crank dimensions necessary for the calculation of stress concentration factors

D = crankpin diameter, mm; D_{BH} = diameter of bore in crankpin, mm; R_H = fillet radius of crankpin, mm; T_H = recess of crankpin, mm; D_G = journal diameter, mm; D_{BG} = diameter of bore in journal, mm; D_O = diameter of oil bore in crankpin, mm; R_G = fillet radius of journal, mm; T_G = recess of journal, mm; E = pin eccentricity, mm; S = pin overlap, mm; *S-(D - DG)/2 - E*; *W*, *B* = web thickness and width, mm.

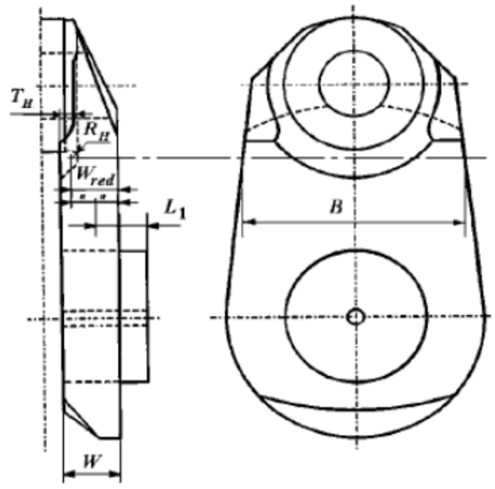


Fig. 500 Crank dimensions without web overlap necessary for calculation of stress concentration factors at TH > RH

Wred = design thickness of web, mm; Wred = W - TH + RH

For the calculation of stress concentration factors in crankpin and journal fillets, the following expressions shall be applied:

$s = \frac{S}{D}$ with $0,5 \leq S \leq 0,7$

$w = \frac{W}{D}$ with $0,2 \leq w \leq 0,8$

$b = \frac{B}{D}$ with $1,2 \leq b \leq 2,2$

$d_G = \frac{D_{BG}}{D}$ with $0 \leq d_G \leq 0,8$

$d_H = \frac{D_{BH}}{D}$ with $0 \leq d_H \leq 0,8$

$t_H = \frac{T_H}{D}$

$t_G = \frac{T_G}{D}$

For crankpin fillets:

$r = \frac{R_H}{D}$ with $0,03 \leq r \leq 0,13$

For journal fillets:

$r = \frac{R_G}{D}$ with $0,03 \leq r \leq 0,13$

The factor $f_{t'}$ which accounts for the influence of a recess in the fillets is valid if:

$t_H \leq \frac{R_H}{D}$ and $t_G \leq \frac{R_G}{D}$ and is to be applied within the range $-0,3 \leq s \leq 0,5$

2.8.2. Crankpin fillet

The stress concentration factor for bending α_B shall be determined from the formula:

$$\alpha_B = 2,6914 \cdot \left[f(s,w) \cdot f(w) \cdot f(b) \cdot f\left(d_G\right) \cdot f(r) \cdot f\left(d_H\right) \cdot f(t)\right]$$

where:

$$f(s,w) = -4,1883 + 29,2004 \cdot w - 77,5925 \cdot w^2 + 91,9454 \cdot w^3 - 40,0416 \cdot w^4 + (1-s) \cdot (9,5440 - 58,3480 \cdot w + 159,3415 \cdot w^2 - 192,5846 \cdot w^3 + 85,2916 \cdot w^4) + (1-s)^2 \cdot (-3,8399 + 25,0444 \cdot w - 70,5571 \cdot w^2 + 87,0328 \cdot w^3 - 39,1832 \cdot w^4);$$

$$f(w) = 2,1790 \cdot w^{(0,7171)};$$

$$f(b) = 0,6840 - 0,0077 \cdot b + 0,1473 \cdot b^2;$$

$$f(r) = 0,2081 \cdot r^{(-0,5231)};$$

$$f\left(d_G\right) = 0,9993 + 0,27 \cdot d_G - 1,0211 \cdot d_G^{\ 2} + 0,5306 \cdot d_G^{\ 3};$$
$$f\left(d_H\right) = 0,9978 + 0,3145 \cdot d_H - 1,5241 \cdot d_H^{\ 2} + 2,4147 \cdot d_H^{\ 3};$$
$$f(t) = 1 + \left(t_H + t_G\right) \cdot (1,8 + 3,2 \cdot s);$$

The stress concentration factor for torsion α_t shall be determined from the formula:

$$\alpha_t = 0,8 \cdot f(r,s) \cdot f(b) \cdot f(w)$$

where:

$$f(r,s) = r^{[-0,322+0,1015\cdot(1-s)]};$$
$$f(b) = 7,8955 - 10,654 \cdot b + 5,3482 \cdot b^2 - 0,8$$
$$f(w) = w^{(-0,145)}$$

2.8.3. Journal fillet

The stress concentration factor for bending β_B shall be determined from the formula:

$$\beta_B = 2,7146 \cdot \left[f_B(s,w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B\left(d_G\right) \cdot f_B\left(d_H\right) \cdot f(t)\ \right]$$

where:

$$f_B(s,w) = -\ 1,7625 + 2,9821 \cdot w - 1,5276 \cdot w^2 + (1 - s) \cdot \left(5,1169 - 5,8089 \cdot w + 3,1391 \cdot w^2\right) + (1 - s)^2 \cdot \left(-\ 2,1567 + 2,3297 \cdot w - 1,2952 \cdot w^2\right);$$
$$f_B(w) = 2,2422 \cdot w^{(0,7548)};$$

$$f_B(b) = 0,5616 + 0,1197 \cdot b + 0,1176 \cdot b^2;$$

$$f_B(r) = 0,1908 \cdot r^{(-0,5568)};$$

$$f_B(d_G) = 1,0012 - 0,6441 \cdot d_G + 1,2265 \cdot {d_G}^2;$$

$$f_B(d_H) = 1,0012 - 0,1903 \cdot d_H + 0,0073 \cdot {d_H}^2;$$

$$f(t) = 1 + (t_H + t_G) \cdot (1,8 + 3,2 \cdot s);$$

The stress concentration factor for shearing β_Q shall be determined from the formula:

$$\beta_Q = 3,0128 \cdot \left[f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(t)\right]$$

where:

$$f_Q(s) = 0,4368 + 2,1630 \cdot (1 - s) - 1,5212 \cdot (1 - s)^2;$$

$$f_Q(w) = \frac{w}{0,0637 + 0,9369 \cdot w};$$

$$f_Q(b) = - 0,5 + b;$$

$$f_Q(r) = 0,5331 \cdot r^{(-0,2038)};$$

$$f_Q(d_H) = 0,9937 - 1,1949 \cdot d_H + 1,7373 \cdot {d_H}^2;$$

$$f(t) = 1 + (t_H + t_G) \cdot (1,8 + 3,2 \cdot s);$$

The stress concentration factor for torsion β_T shall be:

$$\beta_T = \alpha_T$$

if the diameters and fillet radii of crankpin and journal are the same and:

$$\beta_T = 0,8 \cdot f(r,s) \cdot f(b) \cdot f(w)$$

if crankpin and journal diameters and/or radii are of different sizes $f(r, s)$, $f(b)$ and $f(w)$ are to be determined from the formula above; in this case r is the ratio of the journal fillet radius to the journal diameter.

$$r = \frac{R_G}{D_G}$$

2.8.4. Outlet of oil bore

The stress concentration factor for bending γ_B shall be determined by the formula:

$$\gamma_B = 3 - 5,88 \cdot d_0 + 34,6 \cdot d_0^2$$

The stress concentration factor for torsion γ_T shall be determined by the formula:

$$\gamma_T = 4 - 6 \cdot d_0 + 30 \cdot d_0^2$$

2.9. Additional bending stresses

2.9.1. In addition to the alternating bending stresses in fillets further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying as given in table 2600.

Table 2600 Stress component σ_{add}

Type of engine	σ_{add}; MPa
Crosshead engines Trunk piston engines	± 30 ± 10

For crosshead type engines the additional stress (30 MPa) includes stress due to axial vibrations (20 MPa) and stress due to misalignment and bedplate deformation (10 MPa).

It is recommended that a value of 20 MPa be used where axial vibration calculation results of the complete dynamic system (engine / shafting / gearing / propeller) are not available. Where axial vibration calculation results of the complete dynamic system are available, the calculated figures may be used instead.

2.10. Calculation of equivalent alternating stresses

2.10.1. For the crankpin fillet, the equivalent alternating stresses σ_{VH}, in MPa, is to be determined from the formula:

$$s_{VH}=\pm\sqrt{\left(s_{BH}+s_{add}\right)^2+3\cdot t_H^2}$$

For the journal fillet, the equivalent alternating stress s_{VG}, in MPa, is to be determined from the formula:

$$s_{VG}=\pm\sqrt{\left(s_{BG}+s_{add}\right)^2+3\cdot t_G^2}$$

For the outlet of crankpin oil bore, the equivalent alternating stress σ_{VO} in MPa shall be determined by the formula:

$$\sigma_{VO} = +\frac{1}{3}\sigma_{BO}\left[1+2\sqrt{1+2,25(\sigma_{TO}/\sigma_{BO})^2}\right]$$

2.11. Calculation of fatigue strength

2.11.1. Where the fatigue strength for a crankshaft cannot be furnished by reliable measurements, the fatigue strength σ_{DWH} and σ_{DWG}, in Mpa, may be evaluated by means of the following formula:

- Related to the crankpin diameter:

$$s_{DWH} = \pm K\cdot(0,42\cdot s_B+39,3)\cdot(0,264+1,073\cdot D^{-0,2}+\frac{785-s_B}{4900}+\frac{196}{s_B}\cdot\sqrt{\frac{1}{R_H}})$$

- Related to the journal diameter:

$$s_{DWG} = \pm K\cdot(0,42\cdot s_B+39,3)\cdot(0,264+1,073\cdot D_{G^{-0,2}}+\frac{785-s_B}{4900}+\frac{196}{s_B}\cdot\sqrt{\frac{1}{R_G}})$$

- Related to the crankpin in outlet of crankpin oil bore:

$$\sigma_{DWO} = K(0,42R_m+39,3)(0,264+1,073D^{-0,2}+\frac{785-\sigma_B}{4900}+\frac{196}{\sigma_B}\sqrt{\frac{2}{D_O}}$$

where:

σ_B = minimum tensile strength of crankshaft material, MPa;

K = factor for different types of forged and cast crankshafts without surface treatment equal to:

1,05 for continuous grain flow forged or drop-forged crankshafts;

1,0 for free form forged crankshafts;

0,93 for cast steel crankshafts.

2.11.2. For calculation purposes R_H and R_G are not to be taken less than 2 mm.

2.11.3. Where no results of the fatigue test conducted on full size crankthrows or crankshafts which have been subjected to surface treatment are available, the K factors for crankshafts without surface treatment are to be used.

2.11.4. In each case the experimental values of fatigue strength carried out with full size crankthrows or crankshafts are subject to special consideration of *QRS Class*. The survival probability for fatigue strength values derived from testing is not to be less than 80% of the average value.

2.12. Calculation of shrink-fits of semi-built crankshafts

2.12.1. General provisions

All crank dimensions necessary for the calculation of the shrink-fit are shown in *fig 600*.

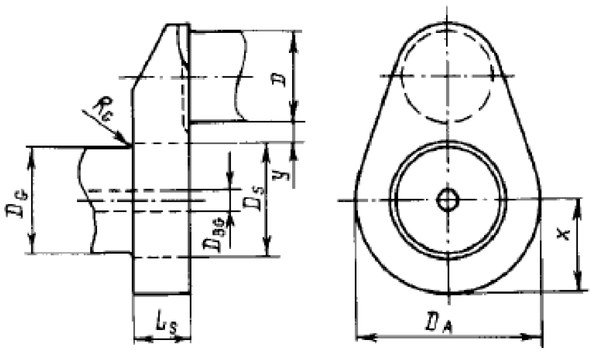


Figure 600 Crankthrow of semi-built crankshaft

D_S = shrink diameter (mm);

L_S = length of shrink-fit (mm);

D_A = outside diameter of web or twice the minimum distance x between centre-lines of journals and outer contours of web, whichever is less (mm);

Y= distance between the adjacent generating lines of journal and pin (mm) y>0,05DS :
Where y is less than $0,1D_S$, special consideration shall be given to the effect of stress due to the shrink on the fatigue strength at the crankpin fillet.

The radius of the transition from the journal to the shrink diameter should not be less than the greater of the two values:

$R_G \geq 0,015 \cdot D_G$ and $R_G{}^{3}0,5\cdot\left(D_S-D_G\right)$

2.12.2. The calculation of the minimum oversize Z_{min} is to be carried out for the crankthrow with the maximum torque M_{Tmax} using the formula:

$$Z_{min}\ ^{3}\ \frac{4\cdot 10^3}{p\cdot m}\cdot\frac{S_R\cdot M_{Tmax}}{E_M\cdot D_S\cdot L_S}\cdot\frac{1-Q_A^2\cdot Q_S^2}{\left(1-Q_A^2\right)\cdot\left(1-Q_S^2\right)}$$

where:

Z_{min} = minimum oversize, mm;

S_r = safety factor against slipping to be taken not less than 2;

μ = coefficient for static friction equal to 0,20 where $\frac{L_S}{D_S}\geq 0,4$;

E_M = Young's modulus, MPa;

$Q_A = \frac{D_S}{D_A}$;

$Q_S = \frac{D_{BG}}{D_S}$.

where: $Q_O = \frac{\sigma_{DWO}}{\sigma_{VO}} \geq 1,15 \quad D_{BG} \leq D_S\sqrt{\frac{1-4000 S_R M_{T\max}}{\mu\pi D_S^2 L_S \mathrm{R}_{eH}}}$

2.12.3. The minimum oversize , in mm, is also to be calculated according to the following formula:

$$Z_{min} \geq \frac{\sigma_S \cdot D_S}{E_M}$$

where:

σ_S = minimum yield strength of material for crank web, in MPa.

2.12.4. The maximum permissible oversize , in mm, is calculated in accordance with the following formula:

$$Z_{max} \leq \frac{\sigma_S \cdot D_S}{E_M} + \frac{0,8 \cdot D_S}{1000}$$

2.4.11 Acceptability factor.

1. Adequate dimensioning of a crankshaft is ensured if the acceptability factors (the ratio of the equivalent alternating stress to the fatigue strength) for both the crankpin and journal fillets satisfy the criteria:

$$Q_H = \frac{\sigma_{DWH}}{\sigma_{VH}} \geq 1,15$$

$$Q_G = \frac{\sigma_{DWG}}{\sigma_{VG}} \geq 1,15$$

$$Q_O = \frac{\sigma_{DWO}}{\sigma_{VO}} \geq 1,15$$

2. At the junction of the web with the journal or pin, the radius of the fillet is not to be less than *0,05 D*.

   Where crankshafts have flanges, the radius of the fillet at the junction of the flange with the journal is not to be less than *0,08 D*.

3. The edges of the oil holes are to be rounded to a radius of not less than 0,25 of the diameter of the hole with a smooth finish. In built and semi-built crankshafts, no keys or pins are permitted for joining a crankpin or journal to the web. On the outer sides of the junction of webs to pins or

journals, reference marks should be provided.

2.5 Scavenging and supercharging

1. In the event of failure of one of the turbochargers, the operation and manoeuvrability of the main engine are to be guaranteed (emergency operation).

2. In the case of main engines which, when started or operating at low rotational speed, do not receive enough air from the turbochargers, an auxiliary supercharging system should be provided to ensure engine operation, at which supercharging from the turbochargers would be sufficient.

   For the case of failure of this system, an additional means to duplicate its functions should be provided.

3. Where supercharging air is cooled, the scavenge manifolds are to be fitted with thermometers and condensate drain cocks after each air cooler.

4. The scavenge manifolds of two-stroke engines with positive displacement-type scavenging pumps, as well as the scavenge manifolds having direct connection with the cylinders, are to be provided with the relief valves set for a pressure exceeding that of the scavenging air by not more than 50%.

The free area of the relief valves is not to be less than 30 cm^2 per cubic meter of the manifold volume including the volume of the underpiston spaces in crosshead engines fitted with diaphragms if these spaces are not used as scavenging pumps.

5. Scavenge manifolds and underpiston spaces are to be provided with draining arrangements for removing accumulations of sludge and water.

6. The air intake pipes of engines and scavenging-supercharging units are to be fitted with safety gauzes.

2.6 Fuel system

1. The high-pressure oil fuel pumps are to be equipped with a device for quick shutting off the fuel supply to cylinders of the engine. Exemption from this requirement is allowed for engines with cylinders not over 180 mm in bore having grouped fuel pumps.

2. The high-pressure oil fuel injection pipes are to be made from thick-walled seamless steel pipes without welded or soldered intermediate joints.

3. External high-pressure fuel delivery lines between the high-pressure fuel pumps and fuel injectors shall be protected with a jacketed piping system capable of containing fuel from a high pressure line failure. A jacketed pipe incorporates an outer pipe, into which the high-pressure fuel pipe is placed, forming a permanent assembly. The jacketed piping system shall include a means for collection of leakages and arrangements and shall

be provided with an alarm in case of a fuel line failure.

When in return piping the propulsion of pressure with peak to peak values exceeds 1,6 MPa, shielding of this piping is also required.

4. The fuel injection pumps and fuel delivery piping shall be designed so that they can withstand the pressure fluctuation or special means shall be provided to reduce it even to the point of disappearance.

5. For the main engine provision shall be made for an arrangement to limit the fuel supply by the rated power mode.

2.7 Lubrication

1. The lubricators supplying oil for lubricating the cylinders are to be fitted with an arrangement enabling control of the amount of oil delivered to each point. To supervise the oil supply to all points to be lubricated, flow indicators are to be provided in a position convenient for observation.

2. Every union supplying lubricating oil to the two-stroke engine cylinders, as well as the unions arranged in the upper part of the cylinder liner are to be provided with a non-return valve.

3. The turbochargers and governors with ball or roller bearings shall have independent lubricating oil systems. Departure from this requirement may be allowed only on special agreement with *QRS Class*.

4. Provision shall be made to prevent penetration of water and fuel oil into the circulating oil and the entry of oil into the cooling water.

2.8 Cooling

Where telescopic devices are employed for cooling pistons or for supplying lubricating oil to moving parts, protection from hydraulic shocks is to be provided.

2.9 Starting arrangements

1. The manifold supplying starting air from the master starting air valve to the cylinder starting valves is to be fitted with one or more relief valves and with a device relieving the manifold of pressure after the engine has been started.

   The relief valve is to be loaded to a pressure not more than 1,2 times that in the starting air manifold. The relieving device and the relief valve may be fitted directly on the master starting air valve.

   An alternative device designed to protect the starting air manifold from the effects of inner explosions is also admitted.

2. Flame arresters or bursting discs are to be fitted on each branch pipe for air supply to the starting valves of the reversing engine cylinder covers.

   In case of non-reversing engines at least one flame arrester or bursting disc is to be fitted on the manifold supplying starting air from the main starting air valve to the manifold.

Flame arresters or bursting discs may be omitted for the engines having a bore not exceeding 230 mm.

3. It is recommended to equip electrically-started engines with engine-driven generators for the automatic charging of the starting storage batteries.

4. Emergency diesel generators shall be capable of being readily started in their cold condition at the ambient temperature of 0 °C. Where such starting is impractical or at lower temperatures at the space, provision shall be made for heating devices to ensure safe starting and taking up the load by the diesel generators.

   When necessary, provision is to be made for the heating devices to ensure safe starting and taking up the load according to the requirements stated above.

2.10 Exhaust arrangements

In two-stroke engines fitted with the exhaust gas turbo-blowers which operate on the impulse systems, provisions are made to prevent broken piston rings from entering the turbine casing.

2.11 Controls and governors

1. The starting and reversing arrangements shall eliminate the possibility of:

   .1 Running the engine in the direction opposite to the required one.

   .2 Reversing the engine when the fuel supply is cut.

   .3 Starting the engine before reversal is completed.

   .4 Starting the engine with the power-driven turning gear engaged.

2. Each main engine shall have a speed governor adjusted so that the engine speed cannot exceed the rated (nominal) speed by more than 15%.

   In addition to the governor, each main engine of power output 220 kW and upwards which may have a disengaging clutch or which is driving a controllable-pitch propeller, is to be provided with a separate overspeed device adjusted so that the engine speed cannot exceed the rated speed by more than 20%.

   The overspeed device shall be activated after the speed governor.

3. Each prime mover for driving a generator shall be fitted with a speed governor which shall meet the following requirements:

.1 When 100% of the generator rated power is suddenly thrown off, the transient speed variations shall not exceed 10% of the rated speed.

.2 When a prime mover running at no-load is suddenly loaded to 50% of the rated power of the generator followed by the remaining 50% after an interval sufficient to restore the speed to steady state, the transient speed variations shall not exceed 10% of the rated speed.

Application of electrical load in more than two load steps can only be permitted, if the conditions within the ship's mains permit the use of such prime movers which can only be loaded in more than two load steps (*fig 700*) and provided that this is already allowed for in the designing stage. This is to be verified in the form of system specifications to be approved and to be demonstrated at ship's trials.

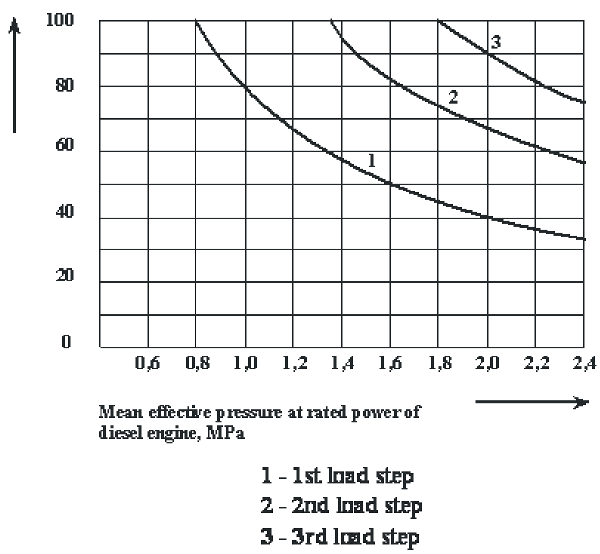


Figure 700 Limiting curves for loading 4-stroke diesel engines step by step from no load to rated power as function of the brake mean effective pressure.

In this case the power required for the electrical equipment to be automatically switched on after black-out as well as the sequence in which it is connected shall fit the load steps. This applies analogously also to generators to be operated in parallel and where the power has to be transferred from one generator to another in the event one generator has to be switched off.

.3 For the operation of the a.c. generators in parallel, from 20 to 100% of its load, this latter shall be distributed for each generator in a way proportional to their power and shall not be different in more than 15% of the rated power of the most powerful generator or in more than 25% of the rated power of the aforementioned generator, depending on which is lesser.

.4 At all loads between no-load and rated power the permanent speed variation shall not exceed the rated speed by more than 5% of the rated speed.

.5 Steady state conditions are those at which the speed variation does not exceed ± 1% of the declared speed at the new power.

.6 for main engines driving shaft-generators, the values of load-relief and load-on shall comply with the load of the engines.

.7 when 100 per cent of the generator rated power is thrown off, a transient speed variation in excess of 10 percent of the rated speed may be acceptable, provided this does not cause the intervention of the overspeed device.

4. The characteristics of the speed governor for the emergency generator driving engine shall meet the requirements of *2* when a 100% load is taken off and put on. At stepwise loading the full (100 percent) load shall be provided in 45 s after power loss on the main switchboard busbars.

   The time delay and successive stepwise loading shall be demonstrated during sea trials of the ship.

5. Provision shall be made for local and remote control of speed variation within 10%. of the nominal value.

   Remote control of speed variation for generators to be operated in parallel shall be arranged to provide the possibility of controlling them by one operator.

6. In addition to the speed governor each driving engine stated in *2.11.3* having a power 220 kW and above is to be fitted with a separate overspeed protective device adjusted so that the speed cannot exceed the rated speed by more than 15%.

7. The overspeed protective device, including its driving mechanism, shall be independent from the speed governor.

8. Protection system of main and auxiliary engines, apart from the overspeed protective device, shall provide complete cut-off of the fuel when the pressure of lubricating oil in the system drops below the allowable value.

2.12 Instruments and alarm devices

1. The control stations of the main engines are to be fitted with instruments for measuring:

   .1 Crankshaft speed and, where disengaging couplings are provided, also the propeller shaft speed.

   .2 Speed of turbo-blowers (for engines with more than 1500 kW of power).

   .3 Lubricating oil pressure before the engine.

   .4 Fresh water pressure in the cooling system.

   .5 Sea water pressure in the cooling system.

   .6 Starting air pressure before the master starting valve.

   .7 Fuel pressure before high-pressure pumps (where a fuel booster pump is provided).

   .8 Pressure in the cooling systems of the fuel valves and pistons.

   .9 Pressure in the reverse gear system.

   .10 Pressure in the supercharging air manifolds.

   .11 Temperature of exhaust gases at each cylinder (for engines with cylinder bore of 180 mm and over).

   .12 Exhaust gas temperature before and after the turbo-blowers.

.13 Cooling water and oil temperature at the engine inlet and at the outlet from each cylinder and piston for engines with cylinder bore of more than 180 mm; for engines with cylinder bore of 180 mm and less, as well as the cylinder block covers only at the engine inlet and outlet.

.14 Lubricating oil temperature before the engine.

.15 Air temperature after air coolers.

.16 Current and voltage in the charging circuit and voltage in the discharging circuit of the starting storage batteries (for electrically-started engines).

.17 Oil fuel temperature before high-pressure pumps (for oil fuel requiring preheating).

Where gearing with the circulating lubricating oil system is provided, the main engine control station shall be fitted with instruments for measuring the pressure of the lubricating oil at the inlet and the temperature of oil at the outlet. Devices for visual checking of the oil flow to each bearing are to be fitted directly on the gearing, except for planet pinions of the planetary gear. Where hydraulic, pneumatic or electrical couplings are provided, the main engine control station is to be fitted with the instruments for measuring the electrical power supply and pressure in the hydro pneumo systems.

.18 The oil pressure in the control system of the clutches and main engines with the reducing gears.

2. Main and auxiliary engines above 37 kW must be fitted with an alarm device with audible and visual signals for failure of the lubricating oil system.

   The following warning alarms are recommended to be provided:

   .1 Pressure or flow drop in the fresh water cooling systems.

   .2 Drop of lubricating oil level in the gravity tank of the turbo-blowers.

   .3 Rise of the lubricating oil temperature at the engine inlet.

   .4 Rise of the cooling water or oil temperature at the engine outlet.

   .5 Drop of the preheating temperature of heavy fuel oil.

   .6 Decrease of the pressure in the exhaust valves drive hydraulic system.

2.13 Torsional vibration damper. Antivibrator

1. The damper structure should make air removal possible when filling the damper with oil or silicone liquid.

2. Lubrication of a spring damper should, as a rule, be effected from the lubricating oil circulation system of the engine.

3. The design of the damper fitted at the free end of the crankshaft shall make it possible to connect devices for measuring torsional vibration to the crankshaft.

Machinery

Section 3 Gears, disengaging and elastic couplings

3.1 General requirements

1. Parts rotating at speeds 5 to 20 m/s should be statically balanced while those rotating at speeds over 20 m/s should be dynamically balanced. The accuracy of dynamic balancing should be determined on the basis of the formula:

$$\gamma = \frac{24000}{n} \quad \text{with } v > 300 \text{ m/s}$$

$$\gamma = \frac{63000}{n} \quad \text{with } v = 20 \text{ m/s}$$

where:

γ = distance between the center of gravity and the geometrical axis of rotation of the part concerned, mm;

n = rotational speed, s-1;

v = peripheral velocity, in m/s.

For peripheral velocities between 20 and 300 m/s, it is to be determined by interpolation. The rigid elements of couplings should be balanced together with the parts they rigidly adjoin.

2. The design of the main gearing shall provide an access to all bearings.

   The gear cases shall have a sufficient number of sight openings with easily detachable covers for carrying out internal inspection.

   The sight openings are to be arranged so as to allow an inspection of the teeth over their full length and of the bearings inside the gearing.

   The application of this requirement to the planetary gear is subject to special consideration by *QRS Class*.

3. The gear cases are to be provided with venting arrangements.

   The vent pipes are to be led to the upper weather deck or other positions where uptake is provided.

   The ends of the vent pipes are to be fitted with flame-arresting devices and arranged as to prevent water from getting into the gearing.

4. Where the main thrust bearing is housed in the gearing case, the lower part of the case shall have proper strengthening.

5. Each sleeve and thrust bearing is to be provided with a temperature measuring device.

   When required by *QRS Class* , the temperature measuring device may be also provided for rolling bearings.

3.2 Gearing

General provisions

1. The requirements of the present Subsection are applicable to main and auxiliary gearing with cylindrical wheels, external and internal toothing, having spur or helical teeth with involute profile.

   Main gearing as well as auxiliary turbo-gearing refer to Group A, the remaining auxiliary gearing refer to Group B.

2. Planetary gear shall be fitted with equalisers. The rim of the epicyclic wheel with more than 3 planetary pinions shall be flexible in radial direction.

Gear wheels

1. The pinions of main gearing shall be manufactured from alloy steel with the ultimate tensile strength of 620 MPa and above. For auxiliary gears, both constructional steels with lower physical and chemical properties

and cast iron, bronze and non-metallic materials may be used.

2. The hardness of the pinion teeth material is to be at least by 15% higher than that of the wheel material. This requirement does not apply to machinery with strengthened gear wheels surface (carburization, nitriding, surface hardening, and so on.).

3. The roots of the teeth are to be formed with smooth fillets of radius not less than 0,3 m_n.

4. The strength of teeth and other pinion and wheel elements shall be proved by calculations. These calculations of steel gear teeth for the basic criteria of durability (contact surface endurance and bending endurance) and for depth strength (for gears with chemically and thermally hardened teeth and with a large module) shall be based on the requirements of the Chapter. In some cases, for high loads and speeds a calculation of the scuffing load capacity may be required.

5. Technical documentation on gearing to be submitted to *QRS Class* approval is to cover the following data:

M_1 : Torques transmitted from each pinion at maximum continuous load, Nm.

d_1 : Pitch-circle diameter of each pinion or gear wheel, cm.

B : Working width of the gear rim, cm.

Z1 and ***Z2***	:	Numbers of teeth in pinion and in gear wheel.
$i = \frac{Z_2}{Z_1}$	:	Gear ratio of each stage.
mn and ***mt***	:	Moduli in normal and transverse sections, cm.
t_a	:	Axial pitch, mm.
β_d, β_b	:	Helical angle on pitch and base cylinders, deg.
α_d	:	Profile angle of initial contour of the teeth, deg.
ε_s	:	Profile recovering coefficient for helical gearing.
1 and 2	:	Addendum modification coefficients.
α_n	:	Pressure angle in normal section, deg.
v	:	Circumferential speed in meshing, m/s.
R_m	:	Tensile strength of tooth core material, MPa.
$HB(HRC)$	:	Material hardness of operating surfaces of teeth.

Type of surface hardening of teeth:

δ : Depth of core hardness, cm.

HB_c : Hardness of teeth core material.

Δt_o : Pitch error, μm.

In addition, for planetary gear:

υ : Poisson's constant.

a_p : Number of planet pinions.

i_p : Gear ratio between driving and driven shafts of machinery.

h_o : Minimum thickness of sun wheel rim, cm.

d_m : Diameter of sun wheel rim, cm.

A : Distance between shafts, cm.

N_{ne} : Equivalent number of stress alternations for each gear wheel of gearing B.

6. Gearing is to meet the following conditions:

Surface strength of operating surfaces of teeth:

$\sigma_k \leq [\sigma_k]$

and bending strength:

$\sigma \leq [\sigma]$

Calculated value of surface stresses, in MPa, is determined from the formula:

$= \frac{}{} \sqrt{\frac{(\quad)}{}}$

where:

k = sign (+) for external gearing;

k_d = sign (-) internal gearing;

C = factor taken from *table 2700*;

ϑ_k = factor taken from the formula:

φ_k = $$k_d = 1 + \frac{B \cdot d_1 \cdot g}{M_1} \cdot C$$

λ_{min} = coefficient equal to:

$[\sigma_k]$ = **1** for spur gearing;

$[\sigma_k]$ = **0,5** for helical gearing;

$[\sigma_k]$ = *g* is taken from *table 2800*;

$[\sigma_k]$ = $k_d = 1$ at < 1,5 m/s;

$[\sigma_k]$ = **1,1** for spur gearing;

$[\sigma_k]$ = **1,45** for group A helical and herringbone gearing;

$[\sigma_k]$ = **1,25** for group B helical and herringbone gearing;

1 for helical gearing and spur gearing with altitude correction and uncorrected;

for spur gearing with angle correction φ_k ;

1 for spur and helical gearing with ratio $1 \geq B / t_a \geq 3$;

for spur gearing and helical gearing with ratio $1 \leq B / t_a < 3$ permissible surface stresses, MPa, are determined as follows:

At HB 270:

2,2 HB

At 270 < HB 350:

+ ( )

At through and surface hardening up to 40 HRC 270:

+ ( )

For alloy carburized steel at 54 HRC 64:

+ ( )

For high alloy carburized steel at 54 HRC 64:

+ ( )

For nitrite and cyanide gear wheels:

+ ( )

Table 2700 Coefficient *k*

Machine	Method of connection	*k* value	
		A gearing	B gearing
Steam turbine	All methods	1	1
Electromotor	Ditto	1	1
Internal combustion engine	Hydrodynamic or electromagnetic clutch	1	1
	Elastic coupling	1,2	1,1
	Rigid coupling	1,4	1,3

Table 2800 *g* values

Pitch	Module m_n, in mm						
error	Over	Over	Over	Over	Over	*U*	*g*
(Δ_{to})	1	2,5	6	10	16		
in	up to	up to	up to	up to	up to		
µm	2,5	6	10	16	30		
	$\Delta_{to} <$ 6	$\Delta_{to} <$ 7	$\Delta_{to} <$ 9	$\Delta_{to} <$ 11,5	-	0,32	25
	6	7	9	11,5	-	0,46	45

	10	11	14	19	-	0,65	80
	16	18	22	30	Δ_{to} 45	0,9	150
	25	28	36	48	70	1,15	300
	-	45	55	75	110	1,7	500

For **spur gearing** the lesser of two values $[\sigma_k]_1$ and $[\sigma_k]_2$ is determined for pinion 1 and wheel 2.

For **helical gearing**:

$$[s_k]=0,5\cdot([s_k]_1+[s_k]_2)$$

where:

$[s_k]_1$ and $[s_k]_2$ = are the permissible surface stresses, respectively, for pinion and wheel determined according to the above-stated relation, in MPa.

In this case, if $[\sigma_k]$ value of helical gearing exceeds the lesser of two values $[\sigma_k]_1$ and $[\sigma_k]_2$ by more than 20%, then it is to be taken equal to:

1,1$[\sigma_k]$ of the lesser value at > 20 m/s;

1,2$[\sigma_k]$ of the lesser value at 20 m/s.

For **turbine gearing**:

$$[\sigma_k] = 0{,}9 \cdot [\sigma_k]_2$$

For **group B gearings** the allowable stresses may be increased by 40% at *HB 350* and by 25% at *HB > 350* (*HRC > 38*)

NOTES:

1. If the value of α_d of the gearings differs from 20º, $[\sigma_k]$ values determined according to recommended relations are to be multiplied by $\sqrt{sen\,2 \cdot \alpha_d \;/\; sen\,40^o}$.

2. It is not recommended to use nitriding for cylindrical gearings with overhanging pinion; this is also true for gearings with non-overhanging pinion but with insufficiently rigid construction.

For **planetary gear** the calculated value of the surface stresses is also determined from *formula:*

where:

M_1 = torque moment, in N• m;

for gearing I (*fig 800*):

$M_1 = \frac{|M_a|}{a_p} \cdot \Omega_a$ at $Z_a \leq Z_g$

$M_1 = M_g \cdot \Omega_a$ at $Z_a > Z_g$

For gearing II (*fig 800*):

$M_1 = M_g \cdot \Omega_b$

For gearing III: (*fig. 800*):

$M_1 = M_f \cdot \Omega_e$;

Ω_a = coefficient of irregular load distribution between planetary pinions taken equal to:

1,1 for diagrams A and B (*fig 800*), with a floating suspension of the sun wheels;

2 for diagram C (*fig 800*) at a_p > 1 and non-floating sun wheel *a*;

1,25 for diagram C (*fig 800*) at a_p > 1 and floating sun wheel *a*;

Ω_b = coefficient of irregular load distribution between planetary pinions taken equal to:

1,1 for diagrams A y B (*fig 800*), with a floating suspension of the sun wheels;

For diagrams A and B (*fig 800*) without a floating suspension, according to *fig 900*;
For diagram C (*fig 800*) at floating wheel *e*

$$1+\frac{Z_b}{Z_a\cdot|i_p|}\cdot(\Omega_a-1)$$

1 for diagram C (*fig 800*) at floating wheel *e*;

Ω_e = coefficient of irregular load distribution between planetary pinions taken equal to:
1 for diagram C (*fig 800*) at floating wheel *b*;
For diagram C (*fig 800*) at floating wheel *e*

$$1+\frac{Z_b}{Z_a\,|i_p|}\,(:_a-1)$$
;

k_{nr} = 1;

k_d = shall be determined according to *table 2900*.

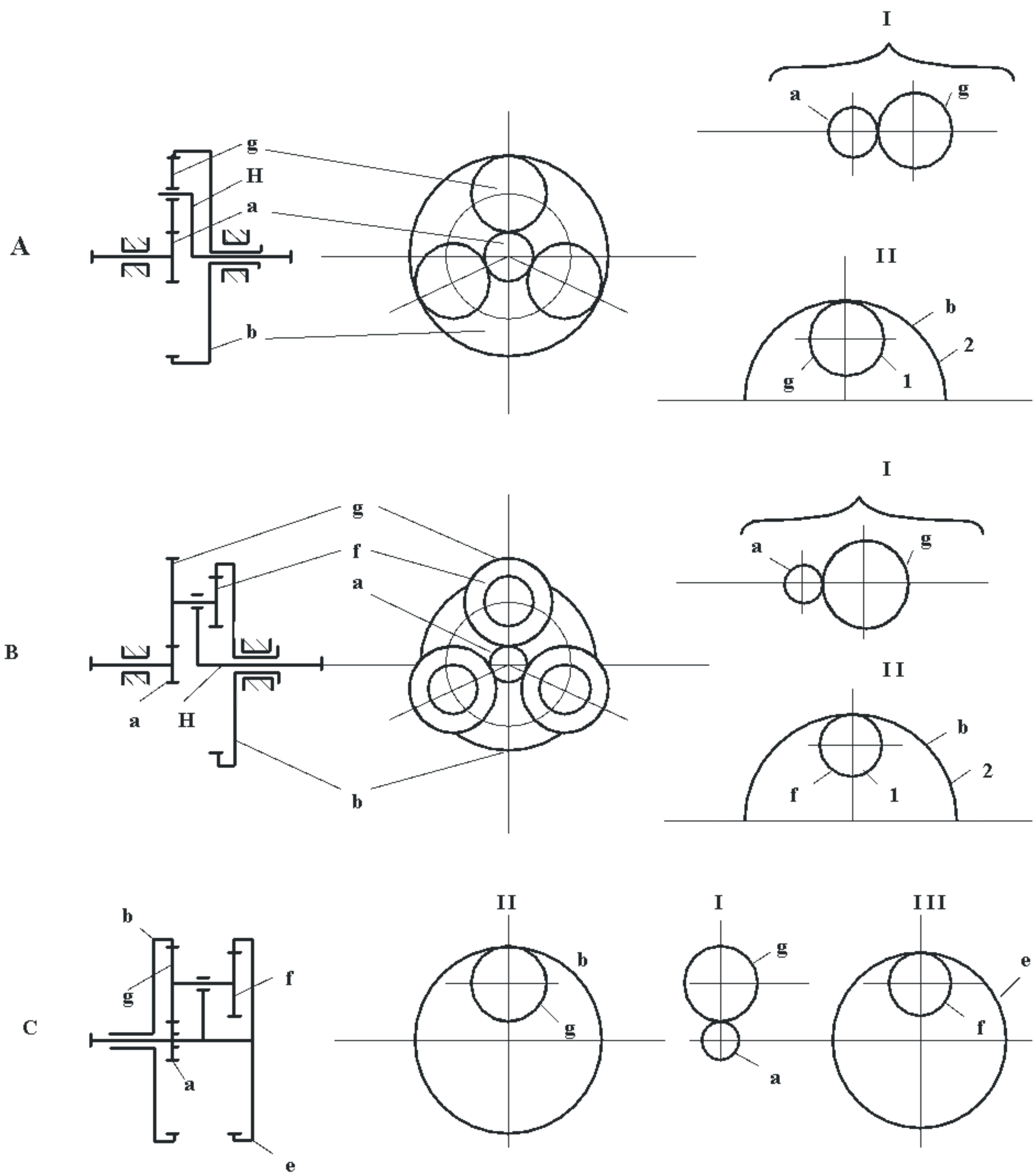


Figure 800 Diagrams of planetary gearings.

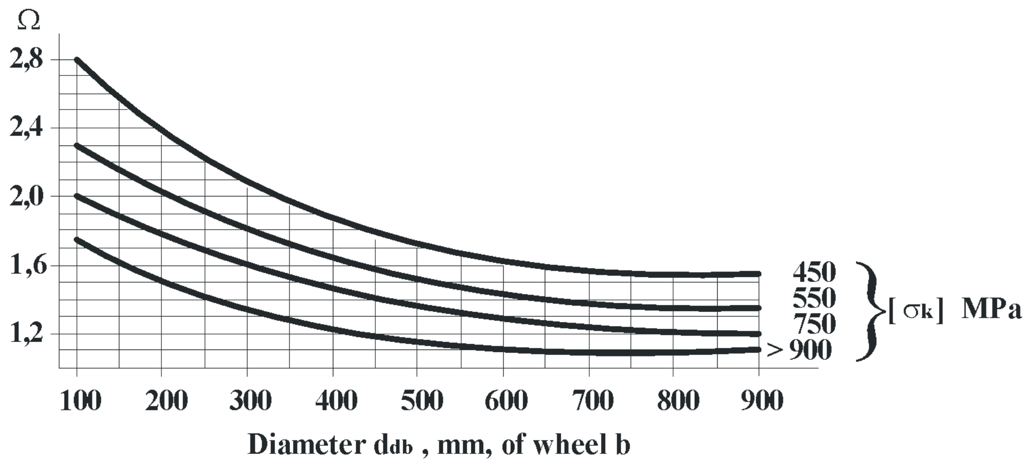


Figure 900 Approximate values of coefficient of irregularity Ω

Table 2900 Value k_d

Application	Type of teeth	Large components firmly fixed with pinions	
		absent	present
Group A gearings	spur	$k_d = 1 + 1,3 \cdot k_p$	$k_d = 1 + 2,6 \cdot k_p$
Group B gearings at $HB_{1,2} > 350$	helical	$k_d = 1 + 1,4 \cdot k_p$	$k_d = 1 + 0,8 \cdot k_p$
Group B gearings at $HB_{1,2} < 350$	spur helical	$k_d = 1 + 0,6 \cdot k_p$ $k_d = 1 + 0,18 \cdot k_p$	$k_d = 1 + 1,2 \cdot k_p$ $k_d = 1 + 0,36 \cdot k_p$

NOTES:

1. The value k_p shall be determined from the formula:

$$k_p = \frac{0,1 \cdot U \cdot v \cdot d_1 \cdot B}{M_1 \cdot k_{nr}} \cdot \sqrt{\frac{A}{i}}$$

2. If at $\beta_d = 0$, the value determined from the *table 2900* exceeds value $k_{max} = 1 + \frac{0,1 \cdot U \cdot v \cdot d_1 \cdot B}{M_1 \cdot k_{nr}} \cdot \sqrt{\frac{A}{i}}$ then it should be taken $k_d = k_{max}$.

At $\beta_d \neq 0$, the second addendum shall not exceed the value k_{max}.

The values *g* and *U*, are to be taken from the *table 2800*.

The permissible surface stresses for planetary gearings are to be determined as follows:

1. For group A gearings from the formula for normal cylindrical gearings.

2. For group B spur gearings is to be taken the lesser value between:
   $[\sigma_k]_1 \cdot k_{c1}$ and $[\sigma_k]_2 \cdot k_{c2}$
   determined for pinion (1) and wheel (2).

3. For group B helical gearing the value shall be:
   $0,5 \cdot \left([s_k]_1 \cdot k_{c1} + [s_k]_2 \cdot k_{c2} \right)$

   In this case, it shall not exceed by more than 20% to the lesser of the two values between:
   $[\sigma_k]_1 \cdot k_{c1}$ and $[\sigma_k]_2 \cdot k_{c2}$

   The coefficient k_c shall be determined from the formula:

$k_c = \sqrt[6]{10^7 \;/\; N_{ne}}$ at HB ≤ 350

$k_c \quad \sqrt[6]{5 \;\; 10^6 \;\; (HRC - 32)/N_{ne}}$ at HB > 350

If it happens that k_c is lesser than 1, then it should be taken $k_c = 1$.

The calculated value of tooth bending stresses, in MPa, is to be determined from the formula:

$$\sigma = \frac{M_1 \cdot k_1 \cdot k_{nr} \cdot k_d \cdot \cos\beta_d}{d_1 \cdot m_n \cdot B \cdot y_b \cdot \xi_s}$$

where:

k_1	=	2	for spur gearings;
		1,65	for helical and herringbone gearings;
y_b	=	tooth shape factor;	
$[\sigma]$	=	permissible bending stresses determined from the *table 3000*.	

4. For planetary gearing the calculated bending stresses value, in MPa, is to be determined from the formula:

$$\sigma = \frac{M_1 \cdot k \cdot k_1 \cdot k_{nr} \cdot k_u \cdot \cos\beta_d}{d_1 \cdot m_n \cdot B \cdot y_b \cdot \xi_s}$$

where:

y_b = tooth shape factor for internal gearing.
For planetary pinion with a thin wall rim, the value shall be determined from *fig 1000*;

k_u = coefficient determined from the *table 3100*.

Table 3000 Permissible stresses $[\sigma]$

Hardness of operating surfaces of teeth	Type of heat treatment	Permissible stresses	
		reversible	non-reversible
HB 350	Tempering $R_m < 1180$ MPa	$[s]=(0,24\cdot R_m+59)/[n_u]$	$[s]=(0,35\cdot R_m+88)/[n_u]$
HRC 38	Carburization, surface hardening over the whole root	$[s]=(0,29\cdot R_m+78)/[n_u]$	$[s]=(0,5\cdot R_m+118)/[n_u]$
-	Nitriding	$[s]=(0,29\cdot R_m+69)/[n_u]$	$[s]=(0,43\cdot R_m+103)/[n_u]$

where:

$[n_u]$ = safety factor.

For gear wheels without surface hardening of teeth:

- 1,9 for propulsion gearing;
- 1,6 for auxiliary gearing.

For gear wheels with surface hardening of teeth:

- 2 for propulsion gearing;
- 1,8 for auxiliary gearing.

Notes:

1. Bending strength calculation shall be carried out for the gear wheel with the lesser value of the product $y_b \cdot [R_m]$.
2. If the maximum moment of the astern running does not exceed 70% of the nominal moment of the ahead running, the permissible value of the teeth stresses for the reversible gear may be assumed equal to 85% of the permissible bending stress for the non-reversible gear.

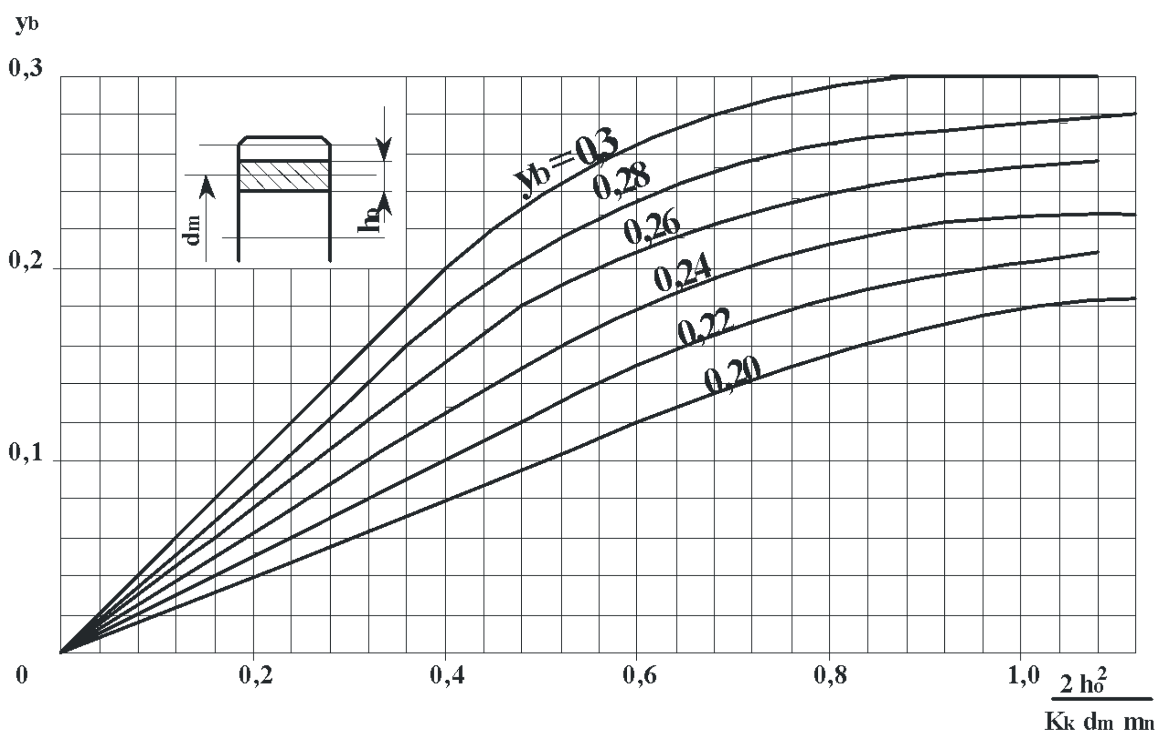
Table 3100 Coefficient k_u

Application	Type of teeth	Large components firmly fixed with pinion	
		absent	present
Gearings of both groups	spur helical	$k_u = 1+1,5 \cdot k_p$ $k_u = 1+0,6 \cdot k_p$	$k_u = 1+3,0 \cdot k_p$ $k_u = 1+1,2 \cdot k_p$
NOTE: For k_p see notes to *table 2900*.			

Table 3200 Bending permissible stresses [σ]

Hardness of operating surface of teeth	Type of heat treatment	Permissible stresses	
		reversible	non-reversible
HB 350	Tempering $R_m < 1180$ MPa	$[s]=\frac{0,24 \cdot R_m+59(k_{cu})}{[n_u]}$	$[s]=\frac{0,35 \cdot R_m+88(k_{cu})}{[n_u]}$
HRC 38	Carburization, surface hardening over the whole root	$[s]=(0,34 \cdot R_m+78(k_{cu}))/[n_u]$	$[s]=(0,5 \cdot R_m+118(k_{cu}))/[n_u]$
-	Nitriding, Cyaniding	$[s]=(0,29 \cdot R_m+69(k_{cu}))/[n_u]$	$[s]=(0,43 \cdot R_m+103(k_{cu}))/[n_u]$

where:		
k_{cu}	=	coefficient which shall be determined as follows: 1 for group A bearings; for group B bearings: $k_{cu} = \sqrt[6]{4 \cdot 10^6 / N_{ne}}$ at HB 350; $k_{cu} = \sqrt{4 \cdot 10^6 / N_{ne}}$ at HB> 350; If k_{cu} is lesser than 1, then it should be taken $k_{cu} = 1$.



NOTE: At $\frac{h_0^2}{K_k \cdot d_m \cdot m_n} > 1,14$ then y_b should not be specified.

Figure 1000 Diagram for determining tooth shape factor y_b of planetary pinions with pliable rim (with sliding bearings $K_k = 1$)

The bending permissible stresses $[\sigma]$ shall be determined by the *table 3200*

NOTES:

If the maximum moment of the astern running does not exceed 70% of the nominal moment of the ahead running, permissible teeth bending stresses of pinions and wheels for the reversible gear may be assumed equal to 85% of the permissible teeth stresses for the non-reversible gear.

Gear wheels with chemical and thermal hardening of the teeth surface are to be additionally examined for the depth strength according to the following condition:

$$\sqrt{\chi_{gl}} \cdot \sigma_k \leq \left[\sigma_{kgl}\right]$$

where:

χ_{gl} = coefficient determined from *fig. 1100*, depending on the value of:

$$k_{gl} = \frac{10^6 \cdot \delta}{1,7 \cdot \sigma_k \cdot \rho_r}$$

ρ_r = reduced radius of curvature in the meshing pole determined, mm, as follows:

$$Y_r \quad \frac{m_s \;\; Z_1 \;\; i \;\; \mathrm{sen}\,\Delta_n}{2 \;\; (i \; \rho \; 1) \;\; \cos^2 E_b}$$

sign (+) - for external meshing

sign (-) - for internal meshing

M_s = face modulus, cm;

$[\sigma_{kgl}]$ = permissible stresses on the depth strength, in MPa, determined as:

$$[\sigma_{kgl}] = HB_c$$

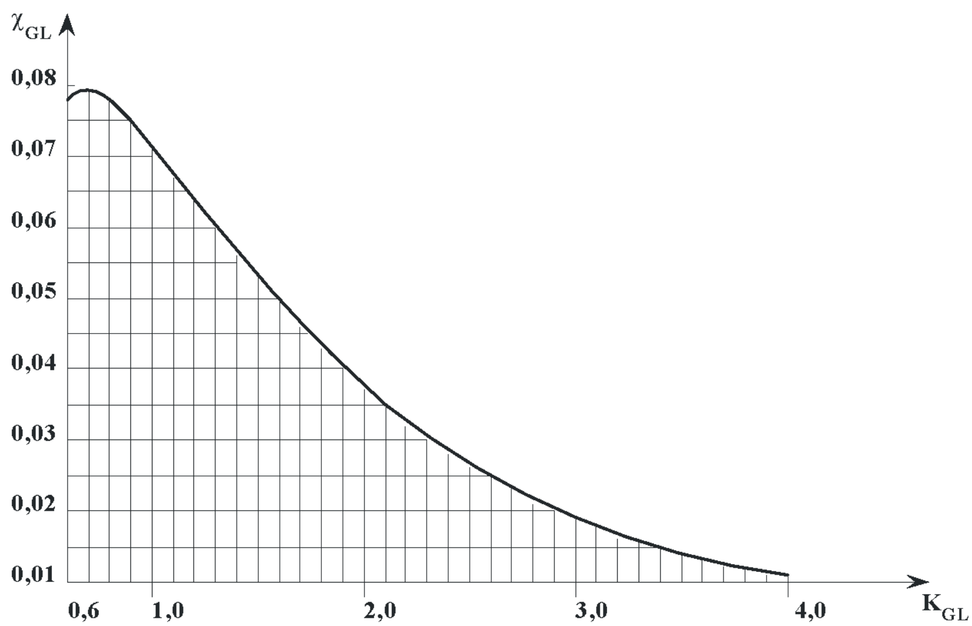


Figure 1100 Diagram for determining χ_{gl} depending on k_{gl}

Shafts

1. The shaft diameter of a larger wheel is not to be less than 1,10 times the diameter of the intermediate shaft when the driving pinions are set at an angle of 120° and more and not less than 1,15 times the diameter of the intermediate shaft in all other cases, the mechanical properties of material of the wheel shaft and intermediate shaft being taken into consideration.

2. For ships with ice strengthening category ICE2, the shafts, pinions and gear wheels of main gears are to be designed for a torque exceeding the main engine designed torque by 15%.
   For ships with ice strengthening category ICE1 the shafts, pinions and gear wheels of main gears are to be designed for a torque exceeding the main engine designed torque by 25%.

For ships with ice strengthening category RICE the shafts, pinions and gear wheels of main gears are to be designed for a torque exceeding the main engine designed torque by 50%.

Pinions and their shafts and gear wheels of the main gears of ships with ice strengthening category RICEI, shall be designed for a torque exceeding the design torque of the main machinery by at least 100%.

The provisions stated in this rule shall not be applicable to the engines protected against overloads of the torque.

Lubrication

1. The toothing and sleeve bearings of the main gears are to be provided with forced lubrication. The possibility of oil pressure regulation is to be provided.

   Provision is to be made for a safety device excluding the oil pressure rise above the permissible value.

2. Lubricating oil is to be delivered to the toothing through sprayers.

   The sprayers shall provide the oil feed in form of fanned-out compact jet with the adjacent jets being overlapped.

   The sprayers are to be arranged so that, while running ahead or astern, the oil is captured in the toothing.

Oil supply to the bearings and sprayers, as well as the oil removal from them, shall be carried out in such a way that oil foaming and emulsification do not take place.

Control, protection and regulation.

Provision shall be made for pressure meters at the inlet to the gearing lubrication systems and for temperature meters at inlet and outlet, as well as for a meter of oil level within the reduction gear casing.

Each sleeve and thrust bearing shall be provided with a temperature measuring device. For transferring power of less than 2250 kW, oil temperature measurement at outlet may be permitted for journal bearings. When required by *QRS Class*, the temperature measuring device may also be provided for rolling bearings.

To prevent an inadmissible rise of lubricating oil temperature in bearings or drop of the lubricating oil pressure, provision shall be made for a warning alarm system.

3.3 Elastic and disengaging couplings

The requirements of this Chapter apply to the elastic and disengaging couplings of main and auxiliary machinery. As far as practical, these requirements apply to electromagnetic and hydraulic disengaging couplings as well.

In ships with one main engine, the shaft coupling design should ensure, in case of coupling failure, the ship running at a speed sufficient for easy steering.

Elastic couplings

1. The ultimate static moment of the elastic component material, i.e. rubber or similar synthetic material, being in shear or tension should be at least eight times the torque transmitted by the coupling.

2. For the purpose of main machinery and diesel generator sets analysis, additional loads due to torsional vibrations should be considered.

3. The elastic couplings of diesel generator sets should withstand moments arising as a result of short-circuit. Where no such information is available, the maximum torque should be at least 4,5 times the nominal torque transmitted by the coupling.

4. The possibility should be provided of fully loading the elastic components, made of rubber or another similar synthetic material, of main machinery plant and diesel generator sets couplings within the temperature range 5 to 60 °C.

Disengaging couplings

1. The disengaging couplings of main machinery should be provided with devices to prevent slipping during appreciable periods of time.

2. It should be possible to control the disengaging couplings of main machinery from the stations from which the main machinery is controlled. Directly at the disengaging couplings, local emergency control arrangements should be provided.

3. Where two or more engines devoted to a common propeller shaft are driving it through disengaging couplings, their control arrangement should make a simultaneous engagement of the engines impossible when running in opposite directions.

Turning gear

A power-driven turning gear is to be provided with an interlocking to preclude the possibility of the drives and couplings engagement when the turning gear is engaged.

Machinery

Section 4 Auxiliary machinery

4.1 Power-driven air compressors

General requirements

1. The air inlets of compressors are to be fitted with strainers.

2. The compressors are to be designed so that the air temperature at the air cooler outlet is not in excess of 90 °C.

3. The compressor cooling water spaces are to be fitted with drain arrangements.

Safety devices

1. Each compressor stage or directly after it is to be fitted with a safety valve preventing the pressure rise in the stage above 1,1 of the rated pressure when the delivery pipe valve is closed. The safety valve design shall prevent the possibility of its adjustment or disconnection after being fitted on the compressor.

2. The compressor crankcases of more than 0,5 m^3 in volume are to be fitted with safety valves.

3. On the delivery pipe immediately after the compressor a safety fuse or a signal device is to be fitted and it shall operate at the air temperature in excess of 120 °C.

4. The casings of the coolers are to be fitted with safety devices providing for a free escape of air in case the pipes are broken.

Crankshaft

1. Cast iron crankshafts, as well as departures from the dimensions of steel crankshafts may be allowed in agreement with *QRS Class* , provided the confirming calculations or test data are submitted.

2. The crankshafts are to be made of steel having a tensile strength of 410 to 780 MPa.

   The use of steel having a tensile strength over 780 MPa, is subject to special consideration by *QRS Class* in each case.

   Cast iron crankshaft shall be manufactured of the spheroidal graphite cast iron or ferrite-pearlite structure.

3. The crankpin diameter d_c, in mm, of the compressor is not to be less than that determined by the formula:

$$d_c = 0,25 \cdot k' \cdot \sqrt[3]{D_{cal}{}^2 \cdot p_c \cdot \sqrt{0,3 \cdot L_{cal}^2 \cdot f + \left(S \cdot j_1\right)^2}}$$

where:

D_{cal} = calculated diameter of the cylinder, mm;
for single-stage compression:
$D_{cal} = D$

for two and multi-stage compression in separate cylinders:
$D_{cal} = D_{hp}$

for two-stage compression by a tandem piston:
$D_{cal} = 1{,}4 \cdot D_{hp}$

for two-stage compression by a differential piston:
$D_{cal} = \sqrt{D_{lp}^2 - D_{hp}^2}$;

D = diameter of the cylinder, in mm;

D_{hp} = diameter of high-pressure cylinder, in mm;

D_{lp} = diameter of low-pressure cylinder, in mm;

P_c = delivery pressure of high-pressure cylinder for air compressor, MPa, for refrigerant compressors.

L_{cal} = calculated span between main bearings, mm, equal to:

when one crank is arranged between two main bearings:
$L_{cal} = L'$;

when to cranks are arranged between two main bearings:
$L_{cal} = 1{,}1 \cdot L'$;

L' = actual span between centres of the main bearings, mm;

S = piston stroke, mm;
k' = coefficient taken in accordance with *table 3300*;
f = coefficient taken in accordance with *table 3400*;
φ_1 = coefficient taken in accordance with *table 3500*.

Table 3300 Values of coefficient k'

Tensile strength, R_m, MPa	390	20	590	690	780	900
k'	1,43	1,35	1,28	1,23	1,2	1,18

Table 3400 Values of coefficient f

Angle between the cylinder axes	0° (in line)	45°	60°	90°
f	1,0	2,9	1,96	1,21

Table 3500 Values of coefficient φ_1

Number of cylinders	1	2	4	6	8
φ_1	1,0	1,1	1,2	1,3	1,4

4. The thickness of crank web h_c, in mm, is to be not less than that determined by the formula:

$$h_c = 0,115 \cdot k \cdot D_{cal} \cdot \sqrt{\frac{(y_1 \cdot y_2 + 0,4) \cdot p_c \cdot c_1 \cdot f_1}{b}}$$

where:

$$k = a \cdot \sqrt[3]{\frac{R_m}{2 \cdot R_m - 430}};$$

R_m = tensile strength of material, MPa.

Where the tensile strength exceeds 780 MPa, R_m equal to 780 Mpa should be adopted for calculation purposes;

a = **0,9** in the case of shafts the surface of which is nitrided as a whole or hardened by another method approved by *QRS Class* .

0,95 in the case of shafts forged by closed-die or continuous grain-flow methods.

1 in the case of shafts not subjected to hardening;

y_1 = coefficient taken in accordance with *table 3600*;

y_2 = coefficient taken in accordance with *table 3700*;

Table 3600 Values of coefficient y_1

r/h	ε/h						
	0	**0,2**	**0,4**	**0,6**	**0,8**	**1,0**	**1,2**
0,07	4,5	4,5	4,28	4,1	3,7	3,3	2,75
0,10	3,5	3,5	3,34	3,18	2,88	2,57	2,18
0,15	2,9	2,9	2,82	2,65	2,4	2,07	1,83
0,20	2,5	2,5	2,41	2,32	2,06	1,79	1,61
0,25	2,3	2,3	2,2	2,1	1,6	1,7	1,4

NOTE:

1. *r* = fillet radius, mm.
2. ε = absolute amount of overlapping, mm (see *fig 1200*).
3. For crankshafts having the distance x between journals and pins the values of coefficient ψ_1 are to be taken valid for ratio $\varepsilon/h = 0$.

Table 3700 Values of coefficient y_2

b/d	**1,2**	**1,4**	**1,5**	**1,8**	**2,0**	**2,2**
ψ_2	0,92	0,95	1,0	1,08	1,15	1,27

where:

p_c = delivery pressure;

$c1$ = distance from the centre of the main bearing to mid-plane of the web; for cranks arranged between two main bearings, the distance is taken to the midplane of the web remotest from the support, mm;

b = web thickness, mm;

f_1 = coefficient taken in accordance with *table 3800*.

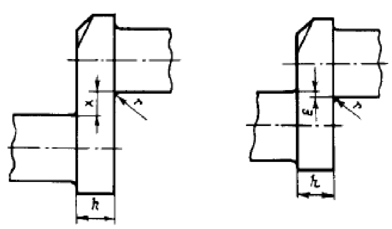


Figure 1200 Distance from the centre of the main bearing to the mid-plane.

Intermediate values of coefficients given in the tables are determined by linear interpolation.

Table 3800 Values of coefficient f_1

Angle between the cylinder axes	0° (in line)	45°	60°	90°
f_1	1,0	1,7	1,4	1,1

Instruments

1. A pressure gauge is to be fitted after each stage of the compressor.

2. Provision shall be made to measure the air temperature at the delivery pipe immediately after the compressor.

3. The instrumentation of the attached compressors is to be subjected to special consideration by *QRS Class* in each case.

4.2 Pumps

Provision is to be made to prevent the pumped fluid from penetration to the bearings. However, this does not apply to the pumps where the pumped fluid is employed for lubrication of bearings.

The pump glands arranged on the suction side are recommended to be fitted with hydraulic seals.

Safety devices

1. If the design of the pump does not preclude the possibility of pressure rising above the rated value, a safety valve is to be fitted on the pump

casing or on the pipe before the first stop valve.

2. In pumps intended for transferring flammable liquids, the by-pass from safety valves is to be effected to the suction side of the pump.

3. Provision is to be made to prevent hydraulic impacts; use of the by-pass valves for this purpose is not recommended.

Strength testing

The critical speed of the pump rotor is not to be less than **1,3** of the rated speed.

Self-priming pumps

The pumps provided with self-priming devices shall ensure operation under *dry suction* conditions and are to be fitted generally with arrangements preventing the self-priming device from operating with contaminated water.

The self-priming pumps shall have the place for connecting a vacuum pressure gauge.

Additional requirements for the pumps transferring flammable liquids

1. Sealing of the shaft is to be such that the leakages occurring will not cause the formation of vapours and gases in the amount sufficient to

produce the flammable air/gas mixture.

2. The possibility of excessive heating and ignition in sealing of the rotating parts due to friction energy should be excluded.

3. When the materials are of low electrical conductivity (plastics, rubber, and so on.) are used in the pump structure, provision is to be made for removal of the electrostatic charges by insertion of the conductive additives into them or use of the devices for removal of the charges and for their transfer to the body.

4.3 Fans, blowers and turbochargers

1. The requirements of the present Subs. shall be complied with when designing and manufacturing fans as well as boiler fans and internal combustion engine turbo blowers.

2. The rotors of fans and air blowers with couplings as well as turbocharger rotor assemblies are to be dynamically balanced in accordance with *3.1*.

3. The suction pipes of fans, blowers and turbochargers are to be protected against entry of foreign materials.

4. The lubricating oil system of the turbocharger bearings is to be arranged so as to prevent the oil from getting into the supercharging air.

Strength testing

1. The impeller is to be dimensioned such that at a speed equal to 1,3 of the rated speed the reference stresses at all sections are not in excess of *0,95* R_{eH} of the element material.

   The impellers of the turbines and blowers shall be also tested for strength during at least 3 min at a speed equal to 1,2 times of the designed speed.

   Series specimens may not be subjected to such testing, provided each impeller forging shall be tested by one of the approved non-destructive testing methods.

   Such testing of the prototype of the turbine and blower impellers is mandatory.

2. In agreement with *QRS Class* other safety factors for the turbo-blowers may also be permitted if the calculation procedures are used taking account of the stress concentrations and plasticity (finite element method).

Additional requirements for the pump room fans in tankers, compartments for the carriage of dangerous goods and load compartments in which are carried motor vehicles with fuel in their tanks.

1. The air gap between the impeller and the casing shall be not less than 0,1 of the impeller shaft bearing diameter, but not less than 2 mm (it is

permitted to be not more than 13 mm).

2. Protection screens of not more than 13 mm square mesh are to be fitted in the inlet and outlet of ventilation ducts to prevent the entrance of objects into the fan housing.

3. To prevent electrostatic charges both in the rotating body and casing, they are to be made of antistatic materials. Furthermore, the installation on board of the ventilation units is to ensure their safe bonding to the ship's hull.

4. The impeller and the housing (in the way of the impeller) are to be made of materials which are recognized as being spark proof.

   The following combinations of materials of impeller and housing are considered spark proof:

   .1 Non-metallic antistatic materials.

   .2 Non-ferrous-based alloys.

   .3 Austenitic stainless steel.

   .4 Impeller is made of aluminium alloy or magnesium alloy and housing is made of cast iron steel (austenitic stainless steel included), if a ring of suitable thickness of non-ferrous materials is fitted inside the housing in the way of impeller.

   .5 Combinations of cast iron and steel impellers and housings (including the case when impeller or housing is made of austenitic stainless steel), provided the tip clearance is not less than 13 mm.

5. Other combinations of materials of impellers and housings may also be permitted if they are recognized as non-sparking by appropriate tests.

6. The following combinations of materials of impeller and housing are not permitted:

   .1 Impellers are made of aluminium alloy or magnesium alloy and housings are made of ferrous-based alloys.

   .2 Impellers are made of ferrous-based alloys and housings are made of aluminium or magnesium alloys.

   .3 Impellers and housings are made of ferrous-based alloys with less than 13 mm tip clearance.

4.4 Centrifugal separators

1. The separator design shall preclude the leakage of oil products and vapours under all conditions of separation.

2. The separator bowls shall be dynamically balanced. The position of the removable parts shall be marked. The design of the disc holder and bowl shall preclude the possibility of misassembly.

3. Rotor-stator systems shall be designed so that the critical speed exceeds the operating speed both in empty and in filled condition.

The critical speed less than the rated speed may be allowed only provided that proofs of continuous safe operation of the separator are submitted.

4. The design of coupling shall preclude the possibility of sparking and impermissible heating under all conditions of the separator operation and besides, shall make possible to dissipate the heat on the working surfaces.

Strength testing

Besides, the strength of rotating separator parts should be checked under stresses arising at rotational speeds exceeding the design speed at least by 30%; in this case, the total stresses in the parts should not exceed $0,95\ R_{eH}$ of the material of which they are made.

At the Manufacturer's test bench, the strength of the rotating parts of the prototype separator should be tested by a rotation speed exceeding the design speed by 30% at least.

Instrumentation, control and protection

1. A device for the control over the separation process shall be provided.

2. It is advisable that the separators be provided with a device automatically shutting off the drive and stopping the separator when inadmissible vibration occurs.

Machinery

Section 5 Deck machinery

5.1 General requirements

1. The brake straps and their fastenings are to be resistant to sea water and petroleum products. The brake straps are to be heat-resistant at temperatures up to 250 °C.

   The permissible heat resistance of connections between the brake strap and the frame is to be above the temperature of heating of the connections for all possible operating conditions of the machinery.

2. The machinery having both manual and power drives is to be provided with interlocking arrangements preventing their simultaneous operation.

3. The deck machinery control arrangements are to be made so that heaving-in is performed when the handwheel is turned to the right or when the lever is shifted backwards while veering out is carried on when the handwheel is turned to the left or the lever is shifted forwards. Locking of brakes is to be carried out turning the handwheels to the right while releasing is affected by turning to the left.

4. The control devices, as well as the instrumentation shall be arranged so as to provide the observation of them from the control place.

5. Winch drums having the multilayer rope winding with the ropes that can be subjected to the load in several layers are to have flanges protruding above the upper layer of winding by not less than 2,5 times the rope diameter.

6. If used for oil-recovery operations, cargo winches and topping of derricks, cargo handling machinery, luffing gear, slewing and travelling machinery of cranes and hoists, and other deck machinery installed in danger zones 0,1 and 2 should be issued for it by a competent body.

5.2 Steering gear

1. Main and auxiliary steering gear are to be arranged so that a single failure in one of them will not render the other one inoperative.

2. Main steering gear, comprising two or more identical power units, is to be arranged so that a single failure in its piping or in one of the power units will not impair the integrity of the remaining part of the steering gear.

3. In oil tankers, oil tankers (60 °C), or chemical carriers of 10 000 tons gross tonnage and upwards, hydraulic steering gear is to be provided with audible and visual alarms to give the indication of hydraulic fluid leakage in all parts of the hydraulic system as well as with the arrangements of automatic isolation of the defective part of the system so that the steering capability shall regained in not more than 45 s after the loss of the

defective port of the hydraulic system.

4. The design of the gears is to provide an emergency changeover from the main steering gear to the auxiliary one during not more than 2 min.

5. Steering gears are to provide for a continuous operation under the most severe service conditions. The design of the steering gear is to exclude the possibility of its failure with a ship running astern at maximum speed.

6. As a rated torque of the steering gear M_r the torque is taken corresponding to the rudder (steering nozzle) angle equal to 35° for the main steering gear and 15° for the auxiliary steering gear when operating under the nominal parameters (nominal pressure in the inner spaces of hydraulic and electro hydraulic gears, nominal current and voltage in the electric steering gear motors, and so on.). In this case, the torque corresponding to the rudder angle 0° is not to be less than 0,82 M_r.

7. In case of the hydraulic steering gear provision is to be made for the fixed storage tank for hydraulic fluid with the capacity sufficient to fill at least one power actuating system, the equalising tank included. This fixed tank is to be provided with a water level indicator connected to the hydraulic gear by the piping so that its hydraulic systems can be filled directly from the tiller room. Each equalising tank is to be provided with a minimum water level alarm.

8. Every oil tanker, oil tanker (60 °C), or chemical carrier of 10 000 tons gross tonnage and upwards is to comply with the following requirements:

.1 the main steering gear shall be arranged so that in the event of loss of steering capability due to a single failure in a part of one of the power actuating systems of the main steering gear excluding the tiller, quadrant or components serving the same purpose as well as seizure of the rudder actuators, steering capability could be regained in not more than 45 s after the loss of one power actuating system;

.2 the main steering gear shall comprise either:

.2.1 two independent and separate power actuating systems; or

.2.2 In this case the interconnection of hydraulic systems shall be provided. Loss of hydraulic fluid from power actuating systems shall be capable of being detected and the defective system automatically isolated so that the other actuating system (systems) is (are) to remain fully operative;

.3 steering gears other than of the hydraulic type shall achieve equivalent standards.

9. For oil tankers, oil tankers (≥60 °C), or chemical carriers of 10000 gross tonnage and upwards but of less than 100000 tons deadweight, at *QRS Class* discretion, solutions other than those set out in 6.2.1.8, which need not apply the single failure criterion to the rudder actuator or actuators, may be permitted provided, that an equivalent safety standard is achieved and that:

.1 following loss of steering capability due to a single failure of a part of the piping system or of one of the power units, steering capability shall be regained within 45 s; and

.2 where a steering gear includes only a single rudder actuator, special consideration is given to stress analysis for the design including fatigue and fracture mechanics analysis, as appropriate, to the material used, the installation of sealing arrangements and testing and inspection as well as to the provision of effective maintenance.

10. Hydraulic system components in the power actuating or hydraulic servo systems controlling the power systems of the steering gear (e.g. solenoid valves, magnetic valves) shall be considered as part of the steering gear control system and shall be duplicated and separated.

    Hydraulic system components in the steering gear control system that are part of a power unit may be regarded as being duplicated and separated when there are two or more separate power units provided and the piping to each power unit can be isolated.

Power of steering gear

1. The main steering gear is to be capable of putting the rudder (steering nozzle) over from 35°on one side to 30° on the other side in not more than 28 s when the rudder stock is affected by a rated torque of the steering engine.

2. The auxiliary steering gear is to be capable of putting the rudder (steering nozzle) over from 15° on one side to 15° on the other side in not more than 60 s.

3. The steering gear power units are to permit a torque overload of at least 1,5 times the rated torque for a period of 1 min.

Hand-operated steering gear

1. The main hand-operated steering gear is to be of self-braking design.

   The auxiliary hand-operated steering gear is to be either of self-braking design or to have a locking device provided that it is reliably controlled from the control station.

Protection against overload and reverse rotation

1. The main and auxiliary steering gears are to have protection against overloads of the gear elements and assemblies when a rudder stock torque equal to 1,5 times the corresponding rated value arises. In case of hydraulic steering gear the safety valves may be used set to a pressure meeting the above-mentioned requirements, but not in excess of **1,25** times the maximum working pressure in the inner spaces of the hydraulic steering gear.

   The design of the safety device shall permit its sealing.

   The minimum capacity of the relief valves is to exceed the total pump capacity by 10%. In this case, the rated pressure of the hydraulic steering gear cavities is not to exceed the pressure to which the relief valves are adjusted.

2. For the main hand-operated steering gear it is sufficient to provide the gear with buffer springs instead of the protection against overload.

   For the auxiliary hand-operated steering gear the fulfilment of the requirement for protection against overload is not compulsory.

3. The pumps of hydraulic steering engines are to be provided with protective devices preventing rotation of the inoperative pump in the opposite direction or with an automatic arrangement shutting out the flow of liquid through the inoperative pump.

Braking device

The steering gear is to be fitted with a brake or some other device which provides keeping the rudder (the steering nozzle) steady at specific positions when the latter exerts a rated torque without allowing for the efficiency of the rudder stock bearings.

Where the pistons or blades of the hydraulic steering gear can be locked by closing the oil pipeline valves, a special braking device may be omitted.

Limit switches

Each power-operated steering gear is to be provided with a device discontinuing its operation before the rudder (the steering nozzle) reaches the rudder (the steering nozzle) stops.

Rudder (steering nozzle) indicators

The steering gear segment rack or the hydraulic steering engine crosshead guide, or the element rigidly coupled with the rudder stock is to be fitted with a dial calibrated in not more than 1° to indicate the actual position of the rudder (the steering nozzle).

Strength testing

1. The main and auxiliary steering gear components to be used in flux of force lines are to be checked for strength under the stress corresponding to the rated torque, and the piping and other steering gear components subjected to internal hydraulic pressure-to the rated pressure.

   The design pressure for calculations to determine the scantlings of piping and other steering gear components subjected to internal hydraulic pressure is to be at least 1,25 times the maximum working pressure to be expected under the operational conditions. In this case, at the discretion of *QRS Class* , fatigue criterion is to be applied for the design of piping and components, taking account of pulsating pressures due to dynamic loads.

   In all above cases the reference stresses in the components are not to exceed $0,4\ R_{eH}$ for the steel components and $0,18\ R_m$ for the components of spheroidal cast iron.

2. The stresses in the elements are common for both the main and auxiliary steering gears (viz., tiller, segment, reduction gear, and so on.) shall not exceed 80% of the stresses tolerable.

3. The steering gear elements unprotected from overloads by safety devices, shall have strength corresponding to the rudder stock strength.

Connection with rudder stock

1. The connection of the steering engine or gear with the elements rigidly coupled with the rudder stock shall eliminate the possibility of breakdown on the steering gear when the rudder stock is shifted in the axial direction.

2. Connecting of the tiller hub or segment rack with the rudder stock is to be designed to transmit no less than double rated torque M_r. The height of the hubs of loose segment racks and auxiliary tillers is not to be less than 0,8 of the diameter of the rudder stock head. In case of press keyless fitted solid hubs on the rudder stock the friction coefficient is to be taken no more than **0,13**.

3. The split hubs are to be fastened with at least two bolts on each side and have two keys.

   The keys are to be arranged at an angle of 90° to the split joints plane.

5.3 Anchor machinery

Drive

1. The drive engine power of the anchor machinery is to provide for an uninterrupted heaving-in of one anchor chain together with the anchor of the normal holding power at a speed not less than 0,15 m/s for a period of 30 min with the pull on the sprocket P_1, in N, not less than determined from the formula:

$$P_1 = 9{,}8 \cdot a \cdot d^2$$

where:

a = coefficient (in kg m/ mm² s²) equal to:

3,75 for category 1 anchor chain;

4,25 for category 2 anchor chain;

4,75 for category 3 anchor chain;

d = anchor chain diameter, mm.

In agreement with the Register, reduction of coefficient *a* is permitted for the chain diameters of 28 mm and less.

For supply vessels the pull on the sprocket P_2, in N, is to be not less than:

$$P_2 = 11{,}1 \cdot (g \cdot h + G)$$

where:

g = mass of anchor chain linear metre, kg/m;

h = specified depth of anchorage, m, but not less than:
200 m - for ships with equipment number 720 or less;
250 m - for ships with equipment number over 720;

G = anchor mass, in kg.

Heaving-in speed of the anchor chain is to be measured on the length of two shackles beginning from the moment when three shackles are in suspended condition.

2. As the anchor approaches the hawse, the drive shall provide for heaving-in speed not over 0,17 m/s.

   It is recommended that the speed during the pulling of the anchor into the hawse should be not more than 0,12 m/s.

3. To break the anchor out, the anchor machinery drive shall build up a pull on the sprocket of at least **1,5 times** the rated value in 2 min without a speed requirement.

Brakes and clutches

1. The anchor machinery is to be fitted with clutches arranged between the sprocket and its drive shaft.

   The anchor machinery with non-self braking gear shall be provided with automatic brakes switched in when the driving energy disappears or the driving engine fails.

2. The automatic brake is to ensure a braking torque corresponding to a force in the chain on the sprocket not less than $1{,}3 \cdot P$ or $1{,}3 \cdot P_2$.

3. Each chain sprocket is to be fitted with a brake, the braking torque of which with the sprocket disconnected from the drive shall provide for holding of the anchor chain without slipping of the brake on exposure to the force in the chain:

.1 Equal to **0,45** of the breaking load in the chain, where the anchor gear is provided with the anchor chain stopped intended for anchorage.

.2 Equal to **0,8** of the breaking load in the chain without the above-mentioned stopper.

The force applied to the brake drive handle is not to exceed 740 N.

Chain sprockets

1. The chain sprockets shall have not less than five cams. For horizontal shaft sprockets the wrapping angle is to be not less than 115°, while for vertical shaft sprockets, not less than 150°.

2. The chain sprockets shall ensure passing the joining links in both horizontal and vertical positions.

3. The construction of sprocket shall not permit skipping of the links over the cams:

under all conditions of operation of the machinery from the main drive;
when ship is lying at anchorage;
when paying out the anchor with the chain cable through free dropping

with periodical braking by the band brake and when the speed of paying out is approximately 4 m/s.

Overload protection

If the machinery drive is capable of developing a torque building up an effort on the sprocket exceeding 0,5 of the anchor chain test load, provision shall be made for a safety arrangement installed between the drive and the machinery to prevent exceeding the above-mentioned load.

Strength calculation

The machinery elements shall be checked for strength when the sprocket is affected by efforts corresponding to the maximum torque of the drive or to the moment of the extreme protection setting and also by the chain breaking load acting after the hose, as well as by the wave forces. These requirements do not cover oil tankers and bulk carriers to suit the requirements of Part X V I I I "Common Structural Rules for Double Hull Oil Tankers" and Part X IX "Common Structural Rules for Bulk Carriers".). The reference stresses in the elements, which may arise from the influence of the above-mentioned loads, shall not exceed **0,95 R_{eH}** of the element material. For the purpose of complying with this requirement the use is allowed of the protecting devices (e.g., extreme moment clutch) fitted between the drive and the machinery.

1. The anchor machinery elements situated in lines of force flow shall be checked for strength when affected by stresses corresponding to the rated pull on the sprocket P1 or P2 . In this case, the reference stresses in the elements shall not exceed $0,4R_{eH}$ of the element material.

2. The following pressures and associated areas shall be applied:

   200 kN/m2 normal to the shaft axis and away from the forward perpendicular, over the projected area in this direction;

   150 kN/m2 parallel to the shaft axis and acting both inboard and outboard separately, over the multiple of /times the projected area where / is determined by the formula:
   f=1+B/H

   where
   B = width of machinery measured parallel to the shaft axis;

   H = overall height of machinery but not more than 2,5.

3. Forces in bolts, chocks and stoppers securing the machinery to the deck shall be calculated. The machinery is supported by *N* bolt groups, each containing one or more bolts *(refer to Fig. 1400)*.

4. The axial force *Ri* in the bolt group or one bolt, positive in tension, may be determined by the formula:
   $R_i = R_{xi} + R_{yi} - R_{si}$

Where: $R_{xi} = P_x h_{xi} A_i / I_x$

$$R_{yi} = P_y h_{yi} A_i / I_y$$

P_x=force acting normal to the shaft axis, kN;

P_y=force acting parallel to the shaft axis, either inboard or outboard, whichever gives the greater force in *i* bolt group, kN ;

h =shaft height above the windlass mounting, cm;

x_h, y_i = x and *y* coordinates of *i* bolt group from the centroid of all *N* bolt groups, positive in the direction opposite to that of the applied force, cm;

A_i = cross sectional area of all bolts in *i* group, cm^2;

$I_x = \Sigma A_f x_i^2$ for *N* bolt groups;

$I_y = \Sigma A_i y_i^2$ for *N* bolt groups;

R_{si} = static reaction at *i* bolt group, due to weight of windlass.

5. Shear forces *Fxi* and *Fyi* applied to *i* bolt group, and the resultant combined force *Fi* may be determined by the formula:

$$F_{xi} = (P_x - \alpha g M) / N$$

$$F_{yi} = (P_y - \alpha g M) / N$$

$$F_i = (F_{xi}^2 + F_{yi}^2)^{0,5}$$

6. Axial tensile and compressive forces and lateral forces are considered in the design of supporting structures.

7. Tensile axial stresses in the individual bolts in each *i* bolt group shall be calculated. The horizontal forces *Fxi* and *Fyi* shall normally be reacted by shear chocks. Where "fitted" bolts are designed to support these shear forces in one or both directions, equivalent stresses in the individual bolts shall be calculated, and compared to the allowable stresses. Where synthetic compounds are incorporated in the holding down arrangements, due account shall be taken in the calculations. The safety factor against bolt proof strength shall not be less than 2,0.

Additional requirements

1. The requirements of this Subsection apply to the remote-controlled anchor machinery.

2. If the provision is made for remote control of paying out the chain cable with the sprocket disconnected from the anchor machinery drive, a device is to be fitted ensuring an automatic braking by the band brake in order that the maximum speed of paying out will not exceed 3 m/s and the minimum speed will not be less than 1,4 m/s without regard to the initial acceleration. In ships with equipment number of 400 and less it is permissible not to install a device for an automatic braking by the band brake.

3. The chain sprocket brake is to provide for smooth stopping of the chain cable when paying it out for a period of not more than 5 s and not less than 2 s from the moment of initiation of the signal from the control station.

4. Provision is to be made at the remote control station for an indicator of the length of the chain cable paid out and the indicator of the paying out speed of the cable with the mark of the 3m/s of the maximum permissible speed.

5. Machinery and machinery elements for which the remote control is provided are to be manually operated from the local position. The failure of an element or the whole remote control system is not to adversely affect the normal operation of the anchor machinery and equipment manually operated from the local position.

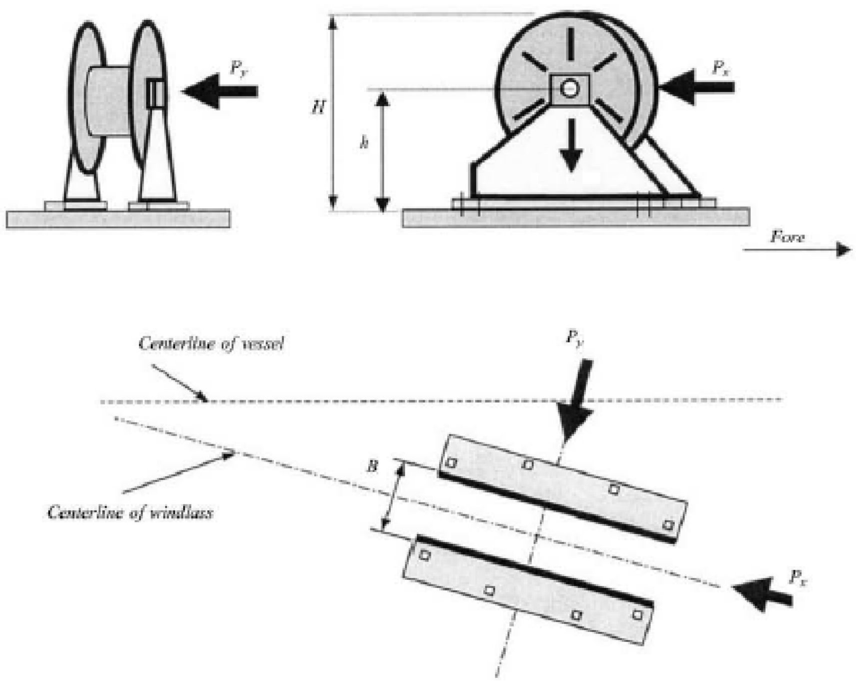


Figure 1300 Direction of forces

Note. P_y shall be examined from both inboard and outboard directions separately.

The sign convention for y_i is reversed when P_y is from the opposite direction as shown

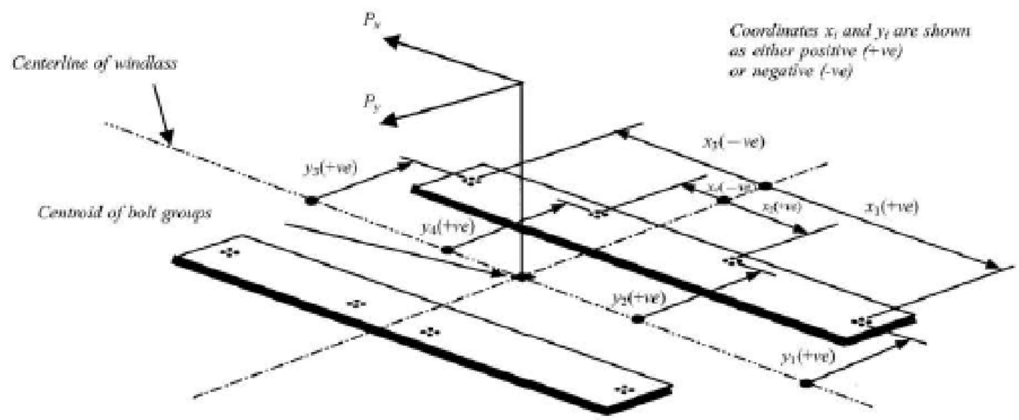


Figure. 1400 Sign convention

where

a = coefficient of friction equal to 0,5;

M = mass of windlass, t;

g = gravity acceleration, m/s^2;

N = number of bolt groups.

5.4 Mooring machinery

Drive.

The mooring machinery drive shall provide for an uninterrupted heaving-in of a mooring line at a rated pull with the rated speed for a period of not less than 30 min.

The speed, v, of heaving-in of a mooring line on the first rope winding layer on the drum with the nominal pulling force *F* shall not be less than stated in *Table 3900*.

The speed of heaving-in of a mooring line by the use of a warping drum at the rated pull shall not be over 0,3 m/s.

Under the rated conditions of the mooring machinery its drive shall develop the pull on the first rope winding layer on the drum equal at least to 1,5 times the rated value in 2 min.

Overload protection.

If the maximum torque of the drive may bring about a larger load on the mooring machinery elements, an overload protection shall be provided.

Brake.

1. The mooring machinery shall be provided with an automatic brake ensuring a hold, without a slip on the mooring line at a pull equal to 1,5 times the rated one when the driving energy disappears or the driving engine fails.

2. The mooring machinery drum shall be provided with a brake, a braking torque of which shall ensure keeping the mooring line from unreeling at a pull in the line equal to 0,8 times the breaking load of the line on the first rope winding layer on the drum.

   The force applied to the brake drive handle shall not exceed 740 N.

If the drum is fitted with an arresting or other safety device, the possibility shall be provided for disengaging the drum by an approved means when the mooring cable is under the load.

Strength calculation.

1. The mooring machinery elements situated in lines of force flow shall be checked for strength under the rated pull on the mooring drum. In this case, the reference stresses in the elements shall not exceed $0{,}4R_{eH}$ of the element material.

Table 3900

v, in m/s	0,25	0,20	0,16	0,13
F, in kN	Up to 80	81 — 160	161 — 250	Above 250

2. The elements of the mooring machinery and the elements of its fastening to foundation shall be checked for strength under the effect of the maximum torque of the drive and when the drum is affected by an effort equal to breaking force of the mooring cable.

   Besides, the strength of the warping drum shaft under the load applied in the middle of its length, equal to the breaking force of the mooring cable shall be checked.

In all above-mentioned cases, the stress in the elements shall not exceed 0,95*Re H* of the element material.

The strength of the mooring machinery elements shall allow for all possible kinds and geometrical directions of the loads that may arise during operation.

The strength of the mooring rope intended for operation with the mooring machinery shall be indicated on the machinery.

Automatic mooring winches.

1. The performance characteristics and durability of the automatic mooring winches shall not be inferior to the similar-purpose non-automatic machinery.

2. Automatic winches shall be equipped with the manual control to provide the possibility of nonautomatic operation.

3. The following shall be provided:

   sound warning alarm operating with the maximum permissible length of the mooring rope veered out;

   an indicator of the actual pull in the mooring rope under the automatic operation.

   For pull measuring it is recommended to install sensors with electric output signals.

5.5 Towing winches

1. Where automatic devices are used for governing the tension of the towline, provision shall be made to enable checking the value of tension at every moment. The tension 1. Connecting of hydraulic steering gear pipelines and those of the hydraulic power systems of CPP to other hydraulic systems is not permitted.

   Connecting of pipelines of the engine-room trunk closures hydraulic drive systems to other hydraulic systems is not permitted.

Machinery

Section 6 Hydraulic Drives

Hydraulic Drives

1. Where the pipeline servicing hydraulic anchor machinery is connected to other hydraulic system pipelines, the latter shall be serviced by two separate pump units, each of which shall ensure the anchor gear operation with nominal pull and at nominal heaving-in speed.

2. The hydraulic system failure shall not cause the failure of machinery or arrangement.

3. Fluids to be used in the hydraulic systems shall be selected with regard to temperature conditions that may occur during operation.

Strength calculation

The hydraulic machinery elements situated in lines of force flow shall be checked under the stresses corresponding to the working pressure. In this case, the reference stresses in elements shall not exceed $0,4R_{eH}$ of the element material.

In cases specified the elements shall be checked for strength under the stresses corresponding to the opening pressure of the safety valves. In this case, the reference stresses in elements shall not exceed $0,95R_{eH}$ of element material.

Safety and other arrangements

1. The hydraulic machinery shall be protected by safety valves, which operating pressure shall not exceed 1,1 times the maximum rated pressure.

2. The working fluid from the safety valve shall be led to the drain pipeline or to the oil tank.

3. Arrangements for complete air expulsion when filling the machinery and the pipeline with the working fluid, as well as for leakage replenishment and drainage shall be provided.

4. The hydraulic systems shall be provided with the filters of appropriate capacity and filtration purity of the working fluid.

   For continuously operating hydraulic systems (hydraulic steering gear, hydraulic couplings, and so on.) provision shall be made for filter cleaning without interruption of the system operation.

5. Oil seals between fixed parts forming a part of the external pressure limit shall be of "metal on metal" type .

   Oil seals between moving parts forming a part of external pressure limit shall be doubled in such a way that the failure of one seal would not disable the executive actuator.

The alternative arrangements providing the equivalent leakage protection may be accepted upon the special agreement with the *QRS Class*.

6. Hydraulic working cylinder rods that are heavily affected by dust and subject to icing shall be protected against such effects.

7. The hydraulic machinery shall be provided with a sufficient amount of the instruments to monitor its operation.

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