Requirements concerning MACHINERY INSTALLATIONS

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M1 Cylinder overpressure monitoring of internal combustion engines

(1969)
(Rev. 1 1985)
(Rev.2 April 1999)

Deleted in Aug 2004

END

M2 Alarm devices of internal combustion engines

(1971)

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

END

M3 Speed governor and overspeed protective device

(1971)
(Rev. 1 1984)
(Rev. 2 1986)
(Rev. 3 1990)
(Rev. 4 June 2002)
(Corr. Aug 2003)
(Rev.5 Feb. 2006)

M3.1 Speed governor and overspeed protective device for main internal combustion engines

1. Each main engine is to be fitted with a speed governor so adjusted that the engine speed cannot exceed the rated speed by more than 15%.

2. In addition to this speed governor each main engine having a rated power of 220 kW and above, and which can be declutched or which drives a controllable pitch propeller, is to be fitted with a separate overspeed protective device so adjusted that the engine speed cannot exceed the rated speed by more than 20%. Equivalent arrangements may be accepted upon special consideration. The overspeed protective device, including its driving mechanism, has to be independent from the required governor.

3. When electronic speed governors of main internal combustion engines form part of a remote control system, they are to comply with UR M43.8 and M43.10 or M47 and namely with the following conditions:
   - if lack of power to the governor may cause major and sudden changes in the present speed and direction of thrust of the propeller, back up power supply is to be provided;
   - local control of the engines is always to be possible, as required by M43.10, and, to this purpose, from the local control position it is to be possible to disconnect the remote signal, bearing in mind that the speed control according to UR M3.1, subparagraph 1, is not available unless an additional separate governor is provided for such local mode of control.
   - In addition, electronic speed governors and their actuators are to be type tested according to UR E10.

M3
cont'd

NOTE:
The rated power and corresponding rated speed are those for which classification of the installation has been requested.

M3.2 Speed governor, overspeed protective and governing characteristics of generator prime movers

1. Prime movers for driving generators of the main and emergency sources of electrical power are to be fitted with a speed governor which will prevent transient frequency variations in the electrical network in excess of ±10% of the rated frequency with a recovery time to steady state conditions not exceeding 5 seconds, when the maximum electrical step load is switched on or off.

   In the case when a step load equivalent to the rated output of a generator is switched off, a transient speed variation in excess of 10% of the rated speed may be acceptable, provided this does not cause the intervention of the overspeed device as required by 3.1.1

2. At all loads between no load and rated power the permanent speed variation should not be more than ±5% of the rated speed.

3. Prime movers are to be selected in such a way that they will meet the load demand within the ship's mains.

   Application of electrical load should be possible with 2 load steps and must be such that prime movers – running at no load – can suddenly be 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. Steady state conditions should be achieved in not more than 5 seconds.

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

   Application of electrical load in more than 2 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 2 load steps (see Fig. 1) 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. In this case, due consideration is to be given to the power required for the electrical equipment to be automatically switched on after black-out and to the sequence in which it is connected. This applies analogously also for generators to be operated in parallel and where the power has to be transferred from one generator to another in the event of any one generator has to be switched off.

4. Emergency generator sets must satisfy the governor conditions as per items 1 and 2 even when**:**

   a) their total consumer load is applied suddenly, or

   b) their total consumer load is applied in steps, subject to:
      – the total load is supplied within 45 seconds since power failure on the main switchboard
      – the maximum step load is declared and demonstrated
      – the power distribution system is designed such that the declared maximum step loading is not exceeded
      – the compliance of time delays and loading sequence with the above is to be demonstrated at ship's trials.

5. In addition to the speed governor, each prime mover driving an electric generator and having a rated power of 220 kW and above must be fitted with a separate overspeed protective device so adjusted that the speed cannot exceed the rated speed by more than 15%.

6. For a.c. generating sets operating in parallel, the governing characteristics of the prime movers shall be such that within the limits of 20% and 100% total load the load on any generating set will not normally differ from its proportionate share of the total load by more than 15% of the rated power of the largest machine or 25% of the rated power of the individual machine in question, whichever is the less.
   For an a.c. generating set intended to operate in parallel, facilities are to be provided to adjust the governor sufficiently fine to permit an adjustment of load not exceeding 5% of the rated load at normal frequency.

NOTE:
For guidance, the loading for 4-stroke diesel engines may be limited as given by Figure 1. ►

M3
cont'd

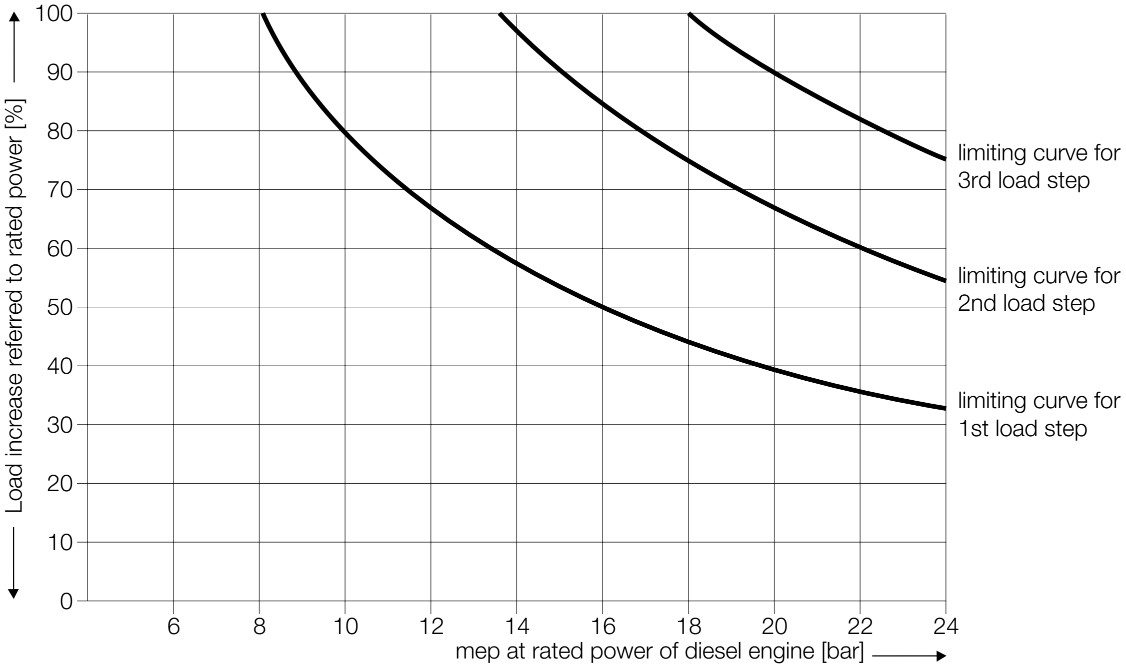

Fig. 1
Limiting curves for loading 4-stroke diesel engines step by step from no-load to rated power as function of the brake mean effective pressure

◄◄

M4 Deleted

Limits of flash point of oil fuel are covered by F35 as revised and should be referred to.

◄◄

M5 (1971) (Rev.1 1987) Mass production of internal combustion engines, procedure for inspection

M5.1 Field of application

The following procedure applies to the inspection of mass produced internal combustion engines having a bore not exceeding 300 mm.

◀

M5.2 Procedure for approval of mass production

M5.2.1 Request for approval - documents to be submitted

Upon requesting approval for mass production of a type of internal combustion engine, the Manufacturer must submit all the necessary data concerning this type of engine:

drawings
technical specification of the main parts
operation and maintenance manuals
list of subcontractors for the main parts.

M5.2.2 Examination of the manufacturing processes and quality control procedures

The Manufacturer will supply full information regarding the manufacturing processes and quality control procedures applied in the workshops. These processes and procedures will be thoroughly examined on the spot by the Surveyors.
The examination will specially concern the following points:

organisation of quality control systems
recording of quality control operations
qualification and independence of personnel in charge of quality control.

M5.2.3 Type test

A running test of at least 100 hours duration will be carried out on an engine chosen in the production line. The programme of this test is examined specially for each case.
At the end of the test, the main parts of the engine will be disassembled and examined.
Omission of the test for engines of well known type will be considered.

M5.2.4 Validity of approval

The Classification Society reserves the right to limit the duration of validity of the approval.
The Classification Society must be kept informed, without delay, of any change in the design of the engine, in the manufacturing or control processes or in the characteristics of the materials.

◀

M5.3 Continuous review of production

M5.l3.1 Access of Surveyors to the Workshops

The Classification Society Surveyors must have free access to the Workshops and to the Control Service premises and files.

M5.3.2 Survey of production

(a) Inspection and testing records are to be maintained to the satisfaction of the Surveyor.
(b) The system for identification of parts is to be approved.
(c) The Manufacturer must give full information about the quality control of the parts supplied by subcontractors, for which approval may be required.

▶

M5
cont'd

The Classification Society reserves the right to apply direct and individual inspection procedures for parts supplied by subcontractors when deemed necessary.

M5.3.2 Individual bench test

The Classification Society may require that a bench test be made under supervision of the Surveyor.

◀

M5.4 Compliance and inspection certificate

For every engine liable to be installed on a ship classed by the Classification Society, the Manufacturer is to supply a statement certifying that the engine is identical to the one which underwent the tests specified in 5.2.3 and give the inspection and test result.

This statement is to be made on a form agreed with the Classification Society. Each statement bears a number which is to appear on the engine.

A copy of this statement is to be sent to the Classification Society.

◀◀

M6 Test pressures for parts of internal combustion engines 1)

(1972)
(Rev. 1 1985)
(Rev. 2 1994)
(Rev.3 May, 1998)

No.	Item		Test pressure 2) [bar] 3)
1.	**Cylinder cover**, cooling space 4)		7
2.	**Cylinder liner**, over whole length of cooling space		7
3.	**Cylinder jacket**, cooling space		4 but not less than 1,5.P
4.	**Exhaust valve**, cooling space		4 but not less than 1,5.P
5.	**Piston crown**, cooling space (where the cooling space is sealed by piston rod or by piston rod and skirt, test after assembly) 4)		7
6.	**High pressure fuel injection system**	Fuel injection pump body, pressure side	1,5 . P or P + 300 whichever is the less
		Fuel injection valve	1,5 .P or P + 300 whichever is the less
		Fuel injection pipes	1,5 . P or P + 300 whichever is the less
7.	**Hydraulic System**	Piping, Pumps, actuators, etc. for hydraulic drive of valves	1,5 . P
8.	**Scavenge pump cylinder**		4
9.	**Turboblower**, cooling space		4 but not less than 1,5 . P
10.	**Exhaust pipe**, cooling space		4 but not less than 1,5 . P
11.	**Engine driven air compressor** (cylinders, covers, intercoolers and aftercoolers)	Air side	1,5 . P
		Water side	4 but not less than 1,5 . P
12.	**Coolers**, each side 5)		4 but not less than 1,5 . P
13.	**Engine driven pumps** (oil, water, fuel, bilge)		4 but not less than 1,5 . P

NOTES

1) In general, items are to be tested by hydraulic pressure as indicated in the Table. Where design or testing features may require modification of these test requirements, special consideration will be given.

2) P is the maximum working pressure in the part concerned.

3) 1 bar = 0,1 MPa = 0,1 N/mm^2.

4) For forged steel cylinder covers and forged steel piston crowns test methods other than pressure testing may be accepted. e.g. suitable non-destructive examination and dimensional control properly recorded.

5) Charge air coolers need only be tested on the water side.

◀◀

M7 Re-categorised as “recommendation” No.26

(1972)
(Rev.1
1987)

End of
Document

M8 Re-categorised as “recommendation” No.27

(1972)
(Rev.1
1989)

End of
Document

M9
(1972)
(Rev.1 1991)
(Corr. 1997)
(Rev.2 June 2000)
(Rev.3 Jan 2005)
(Corr.1 Nov 2005)
(Corr.2 Sept 2007)

Crankcase explosion relief valves for crankcases of internal combustion engines

M9.1 Internal combustion engines having a cylinder bore of 200 mm and above or a crankcase volume of 0.6 m^3 and above shall be provided with crankcase explosion relief valves in accordance with UR M9.2 to UR M9.13 as follows:

M9.1.1 Engines having a cylinder bore not exceeding 250 mm are to have at least one valve near each end, but, over eight crankthrows, an additional valve is to be fitted near the middle of the engine.

M9.1.2 Engines having a cylinder bore exceeding 250 mm but not exceeding 300 mm are to have at least one valve in way of each alternate crankthrow, with a minimum of two valves.

M9.1.3 Engines having a cylinder bore exceeding 300 mm are to have at least one valve in way of each main crankthrow.

M9.2 The free area of each relief valve is to be not less than 45 cm^2.

M9.3 The combined free area of the valves fitted on an engine must not be less than 115 cm^2 per cubic metre of the crankcase gross volume.

M9.4 Crankcase explosion relief valves are to 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.

M9.5 The valve discs in crankcase explosion relief valves are to be made of ductile material capable of withstanding the shock of contact with stoppers at the full open position.

NOTE

1. The total volume of the stationary parts within the crankcase may be discounted in estimating the crankcase gross volume (rotating and reciprocating components are to be included in the gross volume).

2. Engines are to be fitted with components and arrangements complying with Revision 3 of this UR, except for M9.8, M9.9 and the second bullet point in M9.10, when:

    1) an application for certification of an engine is dated on/after 1 January 2006; or

    2) installed in new ships for which the date of contract for construction is on or after 1 January 2006.

    The requirements of M9.8, M9.9 and the second bullet point in M9.10 apply, in both cases above, from 1 January 2008.

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

M9
(cont)

M9.6 Crankcase explosion relief valves are to be designed and constructed to open quickly and be fully open at a pressure not greater than 0.02 N/mm^2 (0.2bar).

M9.7 Crankcase explosion relief valves are to be provided with a flame arrester that permits flow for crankcase pressure relief and prevents passage of flame following a crankcase explosion.

M9.8 Crankcase explosion relief valves are to type tested in a configuration that represents the installation arrangements that will used on an engine in accordance with UR M66.

M9.9 Where crankcase relief valves are provided with arrangements for shielding emissions from the valve following an explosion, the valve is to be type tested to demonstrate that the shielding does not adversely affect the operational effectiveness of the valve.

M9.10 Crankcase explosion relief valves are to be provided with a copy manufacturer's installation and maintenance manual that is pertinent to the size and type of valve being supplied for installation on a particular engine. The manual is to contain the following information:

- Description of valve with details of function and design limits.
- Copy of type test certification.
- Installation instructions.
- Maintenance in service instructions to include testing and renewal of any sealing arrangements.
- Actions required after a crankcase explosion.

M9.11 A copy of the installation and maintenance manual required by UR M9.10 is to be provided on board ship.

M9.12 Plans of showing details and arrangements of crankcase explosion relief valves are to be submitted for approval in accordance with UR M44.

M9.13 Valves are to be provided with suitable markings that include the following information:

- Name and address of manufacturer
- Designation and size
- Month/Year of manufacture
- Approved installation orientation

End of Document

M10 Protection of internal combustion engines against crankcase explosions

(1972) (Rev.1 1991) (Corr. 1997) (Rev.2 Jan 2005) (Corr.1 Nov 2005) (Corr.2 Oct 2007) (Rev.3 Sept 2008)

M10.1 Crankcase construction and crankcase doors are to 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 by UR M9. Crankcase doors are to be fastened sufficiently securely for them not be readily displaced by a crankcase explosion.

M10.2 Additional relief valves are to 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.

M10.3 Scavenge spaces in open connection to the cylinders are to be fitted with explosion relief valves.

M10.4 Crankcase explosion relief valves are to comply with UR M9.

M10.5 Ventilation of crankcase, and any arrangement which could produce a flow of external air within the crankcase, is in principle not permitted except for dual fuel engines where crankcase ventilation is to be provided in accordance with UR M59.3.2.(1).

M10.5.1 Crankcase ventilation pipes, where provided, are to be as small as practicable to minimise the inrush of air after a crankcase explosion.

M10.5.2 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 is not to exceed 2.5 x 10^{-4} N/mm^2 (2.5 m bar).

M10.5.3 To avoid interconnection between crankcases and the possible spread of fire following an explosion, crankcase ventilation pipes and oil drain pipes for each engine are to be independent of any other engine.

Note:

1. The requirements of M10 Rev. 3 are to be uniformly implemented by IACS Societies for engines:

   i) when an application for certification of an engine is dated on or after 1 January 2010; or

   ii) which are installed in new ships for which the date of contract for construction is on or after 1 January 2010.

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

M10 (cont)

M10.6 Lubricating oil drain pipes from the engine sump to the drain tank are to be submerged at their outlet ends.

M10.7 A warning notice is to be fitted either on the control stand or, preferably, on a crankcase door on each side of the engine. This warning notice is to specify that, whenever overheating is suspected within the crankcase, the crankcase doors or sight holes are not to be opened before a reasonable time, sufficient to permit adequate cooling after stopping the engine.

M10.8 Oil mist detection arrangements (or engine bearing temperature monitors or equivalent devices) are required:

- 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
- 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 are to be of a type approved by classification societies and tested in accordance with UR M67 and comply with UR M10.9 to UR M10.20. Engine bearing temperature monitors or equivalent devices used as safety devices have to be of a type approved by classification societies for such purposes.

Note: For equivalent devices for high speed engines, refer to UI SC 133.

M10.9 The oil mist detection system and arrangements are to be installed in accordance with the engine designer's and oil mist manufacturer's instructions/recommendations. The following particulars are to be included in the instructions:

- 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.
- 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.
- The manufacturer's maintenance and test manual.
- 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.

M10.10 A copy of the oil mist detection equipment maintenance and test manual required by UR M10.9 is to be provided on board ship.

M10.11 Oil mist detection and alarm information is to be capable of being read from a safe location away from the engine.

M10.12 Each engine is to be provided with its own independent oil mist detection arrangement and a dedicated alarm.

M10.13 Oil mist detection and alarm systems are to 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 that are acceptable to the classification society.

M10
(cont)

M10.14 Alarms and shutdowns for the oil mist detection system are to be in accordance with UR M35 and UR M36 and the system arrangements are to comply with UR M29 and UR M30.

M10.15 The oil mist detection arrangements are to provide an alarm indication in the event of a foreseeable functional failure in the equipment and installation arrangements.

M10.16 The oil mist detection system is to provide an indication that any 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.

M10.17 Where oil mist detection equipment includes the use of programmable electronic systems, the arrangements are to be in accordance with individual classification society requirements for such systems.

M10.18 Plans of showing details and arrangements of oil mist detection and alarm arrangements are to be submitted for approval in accordance with UR M44 under item 28.

M10.19 The equipment together with detectors is to 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 are to be to the satisfaction of the classification society.

M10.20 Where sequential oil mist detection arrangements are provided the sampling frequency and time is to be as short as reasonably practicable.

M10.21 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 are to be submitted for consideration of individual classification societies. The following information is to be included in the details to be submitted for consideration:

- Engine particulars – type, power, speed, stroke, bore and crankcase volume.
- Details of arrangements prevent the build up of potentially explosive conditions within the crankcase, e.g., bearing temperature monitoring, oil splash temperature, crankcase pressure monitoring, recirculation arrangements.
- Evidence to demonstrate that the arrangements are effective in preventing the build up of potentially explosive conditions together with details of in-service experience.
- Operating instructions and the maintenance and test instructions.

M10.22 Where it is proposed to use the introduction of inert gas into the crankcase to minimise a potential crankcase explosion, details of the arrangements are to be submitted to the classification society for consideration.

End of Document

M11 (1972) Protective devices for starting air mains

In order to protect starting air mains against explosion arising from improper functioning of starting valves, the following devices must be fitted:

(i) an isolation non-return valve or equivalent at the starting air supply connection to each engine
(ii) a bursting disc or flame arrester
in way of the starting valve of each cylinder for direct reversing engines having a main starting manifold
at the supply inlet to the starting air manifold for non-reversing engines.

Devices under (ii) above may be omitted for engines having a bore not exceeding 230 mm.

END

M12 (1972) Fire extinguishing systems for scavenge manifolds

For crosshead type engines, scavenge spaces in open connection to the cylinder must be connected to an approved fire extinguishing system, which is to be entirely separate from the fire extinguishing system of the engine room.

END

M13 (1973) (Rev. 1 1989) Re-categorized as "recommendation" No. 28

END

M14 (1973) Mass production of internal combustion engines: definition of mass production

M14.1 Mass production may be defined, in relation to construction of marine engines for main and auxiliary purposes, as that machinery which is produced:

(i) in quantity under strict quality control of material and parts according to a programme agreed by the Classification Society;
(ii) by the use of jigs and automatic machines designed to machine parts to close tolerances for interchangeability, and which are to be verified on a regular inspection basis;
(iii) by assembly with parts taken from stock and requiring little or no fitting of the parts and which is subject to;
(iv) bench tests carried out on individual engines on a programme basis;
(v) appraisal by final testing of engines selected at random after bench testing.

M14.2 It should be noted that all castings, forgings and other parts for use in the forgegoing machinery are also to be produced by similar methods with appropriate inspection.

M14.3 The specification for machinery produced by the forgoing method must define the limits of manufacture of all component parts. The total production output is to be certified by the Manufacturer and verified as may be required, by the inspecting authority.

END

M15
(1974)
(Rev. 1
1989)

Re-categorized as "recommendation" No. 29

END

M16
(1974)
(Rev.1
Jan 2005)

Devices for emergency operation of propulsion steam turbines

In single screw ships fitted with cross compound steam turbines, the arrangements are to be such as to enable safe navigation when the steam supply to any one of the turbines is required to be isolated. For this emergency operation purpose the steam may be led directly to the L.P. turbine and either the H.P. or M.P. turbine can exhaust direct to the condenser. Adequate arrangements and controls are to be provided for these operating conditions so that the pressure and temperature of the steam will not exceed those which the turbines and condenser can safely withstand.

The necessary pipes and valves for these arrangements are to be readily available and properly marked. A fit up test of all combinations of pipes and valves is to be performed prior to the first sea trials.

The permissible power/speeds when operating without one of the turbines (all combinations) is to be specified and information provided on board.

The operation of the turbines under emergency conditions is to be assessed for the potential influence on shaft alignment and gear teeth loading conditions.

END

M17

Deleted 1 July 1998

END

M18 Parts of internal combustion engines for which material tests are required

(1972)
(Rev. 1 1986)
(Rev. 2 1996)
(Rev. 3 May, 1998)
(Rev.4 June 2000)

M18.1 The list given below applies to engines and superchargers not covered by M5 and M23.

M18.2 Parts for which material tests are required as given in M18.3 are the following ones:

(i) Crankshaft
(ii) Crankshaft coupling flange (non-integral) for main power transmissions
(iii) Coupling bolts for crankshaft
(iv) Steel piston crown
(v) Piston rod
(vi) Connecting rod together with connecting rod bearing caps
(vii) Crosshead
(viii) Cylinder liner, steel parts
(ix) Steel cylinder cover
(x) Bedplates of welded construction: plates and transverse bearing girders made of forged or cast steel
(xi) Frame and crankcase of welded construction
(xii) Entablatures of welded construction
(xiii) Tie rods
(xiv) Supercharger shaft and rotor, including blades. (Supercharger is understood as turbochargers and engine driven compressors (incl. “Root blowers”), but not auxiliary blowers.)
(xv) Bolts and studs for: cylinder covers, crossheads, main bearings, connecting rod bearings
(xvi) Steel gear wheels for camshaft drives.

M18.3 Material tests are required in accordance with the following:

Bore, b (mm)	Tests required for parts nos.
$b \leq 300$	1, 6, 10, 11, 12, 13
$300 < b \leq 400$	1, 6, 8, 9, 10, 11, 12, 13, 14, 15
$b > 400$	All parts

M18.4 This list does not deal with the following items for which material tests may also be required:

pipes and accessories of the air starting system and, possibly, other pressure systems, which are parts of engines.

M18.5 All required material tests are to be witnessed in the presence of the Society's representative.

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M19 (1974) Parts of internal combustion engines for which nondestructive tests are required

M19.1 The list given below covers only individually produced engines.

M19.2 Parts for which nondestructive tests are required as given in M19.3 and M19.4 are the following:

(i) Cast steel elements, including their welded connections, for bedplates (e.g. main bearing housings)
(ii) Solid forged crankshafts
(iii) Cast rolled or forged parts of fully built steel crankshafts
(iv) Cast or forged parts of semi-built steel crankshafts
(v) Connecting rods
(vi) Piston rods
(vii) Steel piston crowns
(viii) Tie rods*)
(ix) Bolts which receive a direct fluctuating load:
main bearing bolts, connecting rod bolts, crosshead bearing bolts, cylinder cover bolts
(x) Steel cylinder covers
(xi) Steel gear wheels for camshaft drives.

M19.3 Magnetic particle or liquid penetrant tests are required in accordance with the following and are to be at positions mutually agreed by the Surveyor and manufacturer, where experience shows defects are most likely to occur:

Bore, b (mm)	Test required for parts nos.
b ≤ 400	1, 2, 3, 4, 5
b > 400	All parts

M19.4 Ultrasonic testing is required, with Maker's signed certificate, in accordance with the following:

Bore, b (mm)	Tests required for parts nos.
b ≤ 400	1, 2, 3, 4, 7, 10
b > 400	1, 2, 3, 4, 5, 6, 7, 10

M19.5 For important structural parts of engines, examination of welded seams by approved methods of inspection may be required.

M19.6 In addition to tests mentioned above, where there is evidence to doubt the soundness of any engine component, non-destructive test by approved detecting methods may be required.

NOTE:

*) Magnetic particle test of tie rods be carried out at each threaded portion which is twice the length of the thread.

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M20 Periodical Survey of Machinery

(1974)
(Rev. 1 1977)
(Rev. 2 1983)
(Rev. 3 1992)
(Rev. 4 1995)

Deleted in November 2001. Requirements relocated to URs Z18 and Z21.

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M21 Mass production of internal combustion engines: type test conditions

(1974)
(Corr. Feb. 1999)
(Corr. Sept. 2003)

M21.1 Application

The following test conditions are to be applied to a type test of internal combustion engines for mass production of which the Maker has requested approval.

Omission or simplification of the type test may be considered for engines of well known type.

M21.2 Choice of engine tested

The choice of the engine to be tested, from the production line, is to be agreed with the classification Society.

M21.3 Duration and programme of tests

The duration and programme of tests is in principle as follows:

80 h at rated output
8 h at 110% overload
10 h at partial loads (1/4,2/4,3/4 and 9/10 of rated output)
2 h at intermittent loads
Starting tests
Reverse running of direct reversing engines
Testing of regulator – overspeed device – lubricating oil system failure alarm device.
Testing of the engine with turbocharger out of action when applicable.
Testing of minimum speed for main propulsion engines and the idling speed for auxiliary engines.

The tests at the above mentioned outputs are to be combined together in working cycles which are to be repeated subsequently with the whole duration within the limits indicated. The overload is to be alternately carried out with:

110% of rated output and 103% rpm
110 % of rated output and 100% rpm.

For prototype engines, the duration and programme of tests are to be specially agreed with the Classification Society.

M21.4 Condition of tests

The following particulars should be recorded:

ambient air temperature
ambient air pressure
atmospheric humidity
external cooling water temperature
fuel and lubrication oil characteristics.

M21.5 Measurements and recordings

In addition to those mentioned in M21.4, the following at least are to be measured or recorded:

engine r.p.m.
brake horsepower
torque
maximum combustion pressure
indicated pressure diagrams where practicable
exhaust smoke (with an approved smoke meter)
lubricating oil pressure and temperature

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M21
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exhaust gas temperature in exhaust manifold,
and, where facilities are available, from each cylinder,
and, *for supercharged engines*
r.p.m. of turbocharger
air temperature and pressures fore and after turboblower and charge cooler exhaust gas temperature and pressures fore and after turbine charge, charge air cooler, cooling water inlet temperature.

M21.6 Examination after test

After the type test, the main parts and especially those subject to wear, are to be disassembled and examined.

NOTES

1. For engines that are to be type approved for different purposes (multi-purpose engines), and that have different performances for each purpose, the programme and duration of test will be modified to cover the whole range of the engine performance taking into account the most severe values.

2. The rated output for which the engine is to be tested is the output corresponding to that declared by the manufacturer and agreed by the Classification Society, i.e. actual maximum power which the engine is capable of delivering continuously between the normal maintenance intervals stated by the manufacturer at the rated speed and under the stated ambient conditions.

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M23 Mass production of engines: mass produced exhaust driven turboblowers

(1975)
(Rev. 2 1981)
(Rev. 3 1991)

M23.1 Field of application

The following procedure applies to the inspection of exhaust driven turboblowers which are manufactured on the basis of mass production methods and for which the maker has requested the approval.

M23.2 Procedure of approval

M23.2.1 Request for approval: documents to be submitted

When the manufacturer of turboblowers built on the basis of mass production methods applies for a simplified method of inspection, the following documentation must be submitted in triplicate:

– cross-sectional drawings with main dimensions,
– drawings with necessary dimensions and material specifications as well as welding details of the rotating parts (shaft, wheels and blades),
– technical specifications including maximum operating conditions (maximum permissible r.p.m. and maximum permissible temperature),
– list of main current suppliers and subcontractors for rotating parts,
– operation and maintenance manuals.

M23.2.2 Material and quality control

The manufacturer will supply full information regarding the control organization as well as the inspection methods, the way of recording and proposed frequency, and the method of material testing of important parts. These processes and procedure will be throughly examined on the spot by the Surveyor.

M23.2.3 Type test

The type test is to be carried out on a standard unit taken from the assembly line and is to be witnessed by the Surveyor. Normally the type test is to consist of a hot running test of one hour's duration at maximum permissible speed and maximum permissible temperature. After the test the turboblower is to be opened up and examined.

Notes:
1. The performance data which may have to be verified are to be made available at the time of the type test.
2. For manufacturers who have facilities for testing the turboblower unit on an engine for which the turboblower is to be type approved, subsituition of the hot running test by a test run of one hour's duration at overload (110% of the rated output) may be considered.

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M23
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M23.2.4 Validity of approval

The Classification Society reserves the right to limit the duration of validity of approval. The approval will be invalid if there are any changes in the design, in the manufacturing or control processes or in the characteristics of the materials which haven not been approved in advance by the Classification Society.

M23.3 Continuous inspection of individual units

M23.3.1 Inspection by the Surveyor

The Surveyors must have the right to inspect at random the quality control measures and to witness the undermentioned tests as deemed necessary, as well as to have free access to all control records and subcontractors certificates.

M23.3.2 Testing of individual units

Each individual unit is to be tested in accordance with M23.3.4 – M23.3.7 by the maker who is to issue a final certificate.

M23.3.3 Identification of parts

Rotating parts of the turboblower are to be marked for easy identification with the appropriate certificate.

M23.3.4 Material tests

Material tests of the rotating parts are to be carried out by the maker or his subcontractor in accordance with the Classification Society's approval. The relevant certificate is to be produced and filed to the satisfaction of the Surveyor.

M23.3.5 Pressure tests

The cooling space of each gas inlet and outlet casings is to be hydraulically tested at pressure of either 0,4 N/mm^2 (4bar) or 1,5 times the maximum working pressure, whichever is the greater.

M23.3.6 Balancing and overspeed test

Each shaft and bladed wheel as well as the complete rotating assembly has to be individually dynamically balanced in accordance with the approved procedure for quality control.

All wheels (impellers and inducers) have to undergo an overspeed test for 3 minutes at 20% over the maximum speed at room temperature or 10% over the maximum speed at working temperature.

If each forged wheel is individually controlled by an approved nondestructive examination method no overspeed test may be required except for wheels of type test unit.

M23.3.7 Bench test

A mechanical running test of each unit for 20 minutes at maximum speed has to be carried out.

NOTE
Subject to the agreement of each individual Society, the duration of the running test may be reduced to 10 minutes provided that the manufacturer is able to verify the distribution of defects established during the running tests on the basis of a sufficient number of tested turbo-charges.

For manufacturers who have facilities in their Works for testing the turboblowers on an engine for which the turboblowers are intended, the bench test may be replaced by a test run of 20 minutes at oveload (110% of the rated output) on this engine.

Where the turboblowers are produced under a quality assurance system complying with recognised standards and subject to satisfactory findings of a historical audit, the Classification Society may accept that the bench test be carried out on a sample basis.

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M23
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M23.4 Compliance and certificate

For every turboblower unit liable to be installed on an engine intended for a ship classed by a Classification Society, the Manufacturer is to supply a statement certifying that the turboblower is identical with one that underwent the tests specified in M23.2.3 and that perscribed tests were carried out. Results of these tests are to be also stated. This statement is to be made on a form agreed with the Classification Society and copy is to be sent to the Classification Society.

Each satement bears a number which is to appear on the turboblower.

NOTE
1. In general, the pressure tests are to be carried out as indicated. Special consideration will be given where design or testing features may require modification of the test requirements.

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M24
(1975)
(Rev. 1 1976)

Requirements concerning use of crude oil or slops as fuel for tanker boilers

M24.1 In tankers crude oil or slops may be used as fuel for main or auxiliary boilers according to the following requirements. For this purpose all arrangement drawings of a crude oil installation with pipeline layout and safety equipment are to be submitted for approval in each case.

M24.2 Crude oil or slops may be taken directly from cargo tanks or flow slop tanks or from other suitable tanks. These tanks are to be fitted in the cargo tank area and are to be separated from non-gas-dangerous areas by means of cofferdams with gas-tight bulkheads.

M24.3 The construction and workmanship of the boilers and burners are to be proved to be satisfactory in operation with crude oil.

The whole surface of the boilers shall be gas-tight separated from the engine room. The boilers themselves are to be tested for gas-tightness before being used. The whole system of pumps, strainers, separators and heaters, if any, shall be fitted in the cargo pump room or in another room, to be considered as dangerous, and separated from engine and boiler room by gas-tight bulkheads. When crude oil is heated by steam or hot water the outlet of the heating coils should be led to a separate observation tank installed together with above mentioned components. This closed tank is to be fitted with a venting pipe led to the atmosphere in a safe position according to the rules for tankers and with the outlet fitted with a suitable flame proof wire gauze of corrosion resistant material which is to be easily removable for cleaning.

M24.4 Electric, internal combustion and steam (when the steam temperature is higher than 220°C) prime movers of pumps, of separators (if any), etc., shall be fitted in the engine room or in another non-dangerous room.

Where drive shafts pass through pump room bulkhead or deck plating, gas-tight glands are to be fitted.

The glands are to be efficiently lubricated from outside the pump room.

M24.5 Pumps shall be fitted with a pressure relief bypass from delivery to suction side and it shall be possible to stop them by a remote control placed in a position near the boiler fronts or machinery control room and from outside the engine room.

M24.6 When it is necessary to preheat crude oil or slops, their temperature is to be automatically controlled and a high temperature alarm is to be fitted.

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M24
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M24.7 The piping for crude oil or slops and the draining pipes for the tray defined in M24.9 are to have a thickness as follows:

External diameter of pipes, d_e	thickness, t
$d_e \leq 82,5$ mm	$t \geq 6,3$ mm
88,9 mm $< d_e \leq$ 108 mm	$t \geq 7,1$ mm
114,3 mm $< d_e \leq$ 139,7 mm	$t \geq 8$ mm
152,4 mm $\leq d_e$	$t \geq 8,8$ mm

Their connections (to be reduced to a minimum) are to be of the heavy flange type. Within the engine room and boiler room these pipes are to be fitted within a metal duct, which is to be gas-tight and tightly connected to the fore bulkhead separating the pump room and to the tray. This duct (and the enclosed piping) is to be fitted at a distance from the ship's side of at least 20% of the vessel's beam amidships and be at an inclination rising towards the boiler so that the oil naturally returns towards the pump room in the case of leakage or failure in delivery pressure. It is to be fitted with inspection openings with gas-tight doors in way of connections of pipes within it, with an automatic closing drain-trap placed on the pump room side, set in such a way as to discharge leakage of crude oil into the pump room.

In order to detect leakages, level position indicators with relevant alarms are to be fitted on the drainage tank defined in M24.9. Also a vent pipe is to be fitted at the highest part of the duct and is to be led to the open in a safe position. The outlet is to be fitted with a suitable flame proof wire gauze of corrosion-resistant material which is to be easily removable for cleaning.

The duct is to be permanently connected to an approved inert gas system or steam supply in order to make possible:

injection of inert gas or steam in the duct in case of fire or leakage
purging of the duct before carrying out work on the piping in case of leakage.

M24.8 In way of the bulkhead to which the duct defined in M24.7 is connected, delivery and return oil pipes are to be fitted on the pump room side, with shut-off valves remotely controlled from a position near the boiler fronts or from the machinery control room. The remote control valves should be interlocked with the hood exhaust fans (defined in M24.10) to ensure that whenever crude oil is circulating the fans are running.

M24.9 Boilers shall be fitted with a tray or gutterway of a height to the satisfaction of the Classification Society and be placed in such a way as to collect any possible oil leakage from burners, valves and connections.

Such a tray or gutterway shall be fitted with a suitable flame proof wire gauze, made of corrosion resistant material and easily dismountable for cleaning. Delivery and return oil pipes shall pass through the tray or gutterway by means of a tight penetration and shall then be connected to the oil supply manifolds.

A quick closing master valve is to be fitted on the oil supply to each boiler manifold.
The tray or gutterway shall be fitted with a draining pipe discharging into a collecting tank in pump room. This tank is to be fitted with a venting pipe led to the open in a safe position and with the outlet fitted with wire gauze made of corrosion resistant material and easily dismountable for cleaning.
The draining pipe is to be fitted with arrangements to prevent the return of gas to the boiler or engine room.

M24.10 Boilers shall be fitted with a suitable hood placed in such a way as to enclose as much as possible of the burners, valves and oil pipes, without preventing, on the other side, air inlet to burner register.

The hood, if necessary, is to be fitted with suitable doors placed in such a way as to enable inspection of and access to oil pipes and valves placed behind it. It is to be fitted with a duct leading to the open in a safe position, the outlet of which is to be fitted with a suitable flame wire gauze, easily dismountable for cleaning. At least two mechanically driven exhaust fans having spark proof impellers are to be fitted so that the pressure inside the hood is less than that in the boiler room. The exhaust fans are to be connected with automatic change over in case of stoppage or failure of the one in operation.

The exhaust fan prime movers shall be placed outside the duct and a gas-tight bulkhead penetration shall be arranged for the shaft.

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M24
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Electrical equipment installed in gas dangerous areas or in areas which may become dangerous (i.e. in the hood or duct in which crude-oil piping is placed) is to be of certified safe type as required by Classification Societies.

M24.11 When using fuel oil for delivery to and return from boilers fuel oil burning units in accordance with Classification Societies' Rules shall be fitted in the boiler room. Fuel oil delivery to, and returns from, burners shall be effected by means of a suitable mechanical interlocking device so that running on fuel oil automatically excludes running on crude oil or vice versa.

M24.12 The boiler compartments are to be fitted with a mechanical ventilation plant and shall be designed in such a way as to avoid the formation of gas pockets.
Ventilation is to be particularly efficient in way of electrical plants and machinery and other plants which may generate sparks. These plants shall be separated from those for service of other compartments and shall be in accordance with Classification Societies' requirements.

M24.13 A gas detector plant shall be fitted with intakes in the duct defined in M24.7, in the hood duct (downstream of the exhaust fans in way of the boilers) and in all zones where ventilation may be reduced. An optical warning device is to be installed near the boiler fronts and in the machinery control room. An acoustical alarm, audible in the machinery space and control room, is to be provided.

M24.14 Means are to be provided for the boiler to be automatically purged before firing.

M24.15 Independent of the fire extinguishing plant as required by Classification Societies' Rules, an additional fire extinguishing plant is to be fitted in the engine and boiler rooms in such a way that it is possible for an approved fire extinguishing medium to be directed on to the boiler fronts and on to the tray defined in M24.9. The emission of extinguishing medium should automatically stop the exhaust fan of the boiler hood (see M24.8).

M24.16 A warning notice must be fitted in an easily visible position near the boiler front. This notice must specify that when an explosive mixture is signalled by the gas detector plant defined in M24.13 the watchkeepers are to immediately shut off the remote controlled valves on the crude oil delivery and return pipes in the pump room, stop the relative pumps, inject inert gas into the duct defined in M24.7 and turn the boilers to normal running on fuel oil.

M24.17 One pilot burner in addition to the normal burning control is required.

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M25 Astern power for main propulsion

(1975)
(Rev. 1 1984)
(Rev. 2 1997)
(Rev.3 July 2003)

M25.1 In order to maintain sufficient manoeuvrability and secure control of the ship in all normal circumstances, the main propulsion machinery is to be capable of reversing the direction of thrust so as to bring the ship to rest from the maximum service speed. The main propulsion machinery is to be capable of maintaining in free route astern at least 70% of the ahead revolutions.

M25.2 Where steam turbines are used for main propulsion, they are to be capable of maintaining in free route astern at least 70% of the ahead revolutions for a period of at least 15 minutes. The astern trial is to be limited to 30 minutes or in accordance with manufacturer’s recommendation to avoid overheating of the turbine due to the effects of “windage” and friction.

M25.3 For the main propulsion systems with reversing gears, controllable pitch propellers or electric propeller drive, running astern should not lead to the overload of propulsion machinery.

NOTES:
1. The head revolutions as mentioned above are understood as those corresponding to the maximum continuous ahead power for which the vessel is classed.
2. The reversing characteristics of the propulsion plant are to be demonstrated and recorded during trials.

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M26 Safety devices of steam turbines

(1976)
(Corr.1
Feb 2005)

M26.1 Governors and speed control

M26.1.1 All main and auxiliary turbines are to be provided with overspeed protective devices to prevent the design speed from being exceeded by more than 15%.

Where two or more turbines are coupled to the same gear wheel set, the Classification Society may agree that only one overspeed protective device be provided for all the turbines.

M26.1.2 Arrangement is to be provided for shutting off the steam to the main turbines by suitable hand trip gear situated at the manoeuvring stand and at the turbine itself.

Hand tripping for auxiliary turbines is to be arranged in the vicinity of the turbine overspeed protective device.

M26.1.3 Where the main turbine installation incorporates a reverse gear, electric transmission, controllable pitch propeller or other free-coupling arrangement, a separate speed governor in addition to the overspeed protective device is to be fitted and is to be capable of controlling the speed of the unloaded turbine without bringing the overspeed protective device into action.

M26.1.4 Where exhaust steam from auxiliary systems is led to the main turbine it is to be cut off at activation of the overspeed protective device.

M26.1.5 Auxiliary turbines driving electric generators are to have both:

a speed governor which, with fixed setting, is to control the speed within the limit of 10% for momentary variation and 5% permanent variation when the full load is suddenly taken off, and
an overspeed protective device which is to be independent of speed governor, and is to prevent the design speed from being exceeded by more than 15% when the full load is suddenly taken off (see M26.1.1).

M26.2 Miscellaneous safety arrangements

M26.2.1 Main ahead turbines are to be provided with a quick acting device which will automatically shut off the steam supply in the case of dangerous lowering of oil pressure in the bearing lubricating system. This device is to be so arranged as not to prevent the admission of steam to the astern turbine for braking purposes.

Where deemed necessary by the Classification Society appropriate means are to be provided to protect the turbines in case of:

abnormal axial rotor displacement,
excessive condenser pressure,
high condensate level.

M26.2.2 Auxiliary turbines having governors operated other than hydraulically in which the lubricating oil is inherent in the system, are to be provided with an alarm device and a means of shutting off the steam supply in the case of lowering of oil pressure in the bearing lubricating oil system.

M26.2.3 Main turbines are to be provided with a satisfactory emergency supply of lubricating oil which will come into use automatically when the pressure drops below a predetermined value.

The emergency supply may be obtained from a gravity tank containing sufficient oil to maintain adequate lubrication until the turbine is brought to rest or by equivalent means. If emergency pumps are used these are to be so arranged that their operation is not affected by failure of the power supply. Suitable arrangement for cooling the bearings after stopping may also be required.

M26
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M26.2.4 To provide a warning to personnel in the vicinity of the exhaust end steam turbines of excessive pressure, a sentinel valve or equivalent is to be provided at the exhaust end of all turbines. The valve discharge outlets are to be visible and suitably guarded if necessary. When, for auxiliary turbines, the inlet steam pressure exceeds the pressure for which the exhaust casing and associated piping up to exhaust valve are designed, means to relieve the excess pressure are to be provided.

M26.2.5 Non-return valves, or other approved means which will prevent steam and water returning to the turbines, are to be fitted in bled steam connections.

M26.2.6 Efficient steam strainers are to be provided close to the inlets to ahead and astern high pressure turbines or alternatively at the inlets to manoeuvring valves.

NOTE
The hand trip gear is understood as any device which is operated manually irrespective of the way the action is performed, i.e. mechanically or by means of external power.

END

M27 (1976) Bilge level alarms for unattended machinery spaces

M27.1 All vessels are to be fitted with means for detecting a rise of water in the machinery space bilges or bilge wells. Bilge wells are to be large enough to accommodate normal drainage during the unattended period. The number and location of wells and detectors is to be such that accumulation of liquids may be detected at all normal angles of heel and trim.

M27.2 Where the bilge pumps start automatically, means shall be provided to indicate if the influx of liquid is greater than the pump capacity or if the pump is operating more frequently than would normally be expected. In this case, smaller bilge wells to cover a reasonable period of time may be permitted. Where automatically controlled bilge pumps are provided special attention shall be given to oil pollution prevention requirements.

M27.3 Alarms are to be given at the main control station, engineers' accommodation area and at the bridge.

END

M28 (1978) Ambient reference conditions

For the purpose of determining the power of main and auxiliary reciprocating internal combustion engines, the following ambient reference conditions apply for ships of unrestricted service:

Total barometric pressure	1000 mbar
Air temperature	+45°C
Relative humidity	60%
Sea water temperature (charge air coolant-inlet)	32°C

NOTE
The engine manufacturer shall not be expected to provide simulated ambient reference conditions at a test bed.

END

M29 Alarm systems for vessels with periodically unattended machinery spaces

(1978)
(Rev. 1 1979)
(Rev.2 1995)
(Rev.3 1997)

M29.1 Definition

The alarm system is intended to give warning of a condition in which deviation occurs outside the preset limits on selected variables. The arrangement of the alarm display should assist in identifying the particular fault condition and its location within the machinery space. Alarm systems, including those incorporating programmable electronic systems, are to satisfy the environmental requirements of IACS UR E10.

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M29.2 General requirements

Where an alarm system is required by the Rules, the system is to comply with the conditions given in M29.2.1 - M29.2.10.

M29.2.1 The system is to be designed to function independently of control and safety systems so that a failure or malfunction in these systems will not prevent the alarm system from operating. Common sensors for alarms and automatic slowdown functions are acceptable as specified in M35 Table 1 and 2 as Gr 1.

M29.2.2 Machinery faults are to be indicated at the control locations for machinery.

M29.2.3 The system is to be so designed that the engineering personnel on duty are made aware that a machinery fault has occurred.

M29.2.4 If the bridge navigating officer of the watch is the sole watchkeeper then, in the event of a machinery fault being monitored at the control location for machinery, the alarm system is to be such that this watchkeeper is made aware when:

(i) a machinery fault has occurred,
(ii) the machinery fault is being attended to,
(iii) the machinery fault has been rectified. Alternative means of communication between the bridge area, the accommodation for engineering personnel and the machinery spaces may be used for this function.

M29.2.5 Group alarms may be arranged on the bridge to indicate machinery faults. Alarms associated with faults requiring speed reduction or the automatic shut down of propulsion machinery are to be separately identified.

M29.2.6 The alarm system should be designed with self monitoring properties. In so far as practicable, any fault in the alarm system should cause it to fail to the alarm condition.

M29.2.7 The alarm system should be capable of being tested during normal machinery operation. Where practicable means are to be provided at convenient and accessible positions, to permit the sensors to be tested without affecting the operation of the machinery.

M29.2.8 Upon failure of normal power supply, the alarm system is to be powered by an independent standby power supply, e.g. a battery. Failure of either power supply to the alarm system is to be indicated as a separate alarm fault. Where an alarm system could be adversely affected by an interruption in power supply, change-over to the stand by power supply is to be achieved without a break.

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M29
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M29.2.9

(a) Alarms are to be both audible and visual. If arrangements are fitted to silence audible alarms they are not to extinguish visible alarms.
(b) The local silencing of bridge or accommodation alarms is not to stop the audible machinery space alarm.
(c) Machinery alarms should be distinguishable from other audible alarms, i.e. fire, CO_2 flooding.
(d) The alarm system is to be so arranged that acknowledgement of visual alarms is clearly noticeable.

M29.2.10 If an alarm has been acknowledged and a second fault occurs before the first is rectified, then audible and visual alarms are to operate again.

Alarms due to temporary failures are to remain activated until acknowledged.

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M30
(1978)
(Rev. 1
1997)

Safety systems for vessels with periodically unattended machinery spaces

M30.1 Definition

The safety system is intended to operate automatically in case of faults endangering the plant so that:
(i) normal operating conditions are restored (by starting of standby units), or
(ii) the operation of the machinery is temporarily adjusted to the prevailing conditions (by reducing the output of machinery), or
(iii) machinery and boilers are protected from critical conditions by stopping the machinery and shutting off the fuel to the boilers respectively (shutdown).

M30.2 General requirements

M30.2.1 Where a safety system is required by the Rules, the system is to comply with M30.2.2 - M30.2.8.

M30.2.2 Operation of the safety system shall cause an alarm.

M30.2.3 The safety system intended for the functions listed under M30.1 (iii) is to be independent of all other control and alarm systems so that failure or malfunction in these systems will not prevent the safety system from operating. For the safety systems intended for functions listed under M30.1(i) and (ii), complete independence of other control and alarm systems is not required.

M30.2.4 In order to avoid undesirable interruption in the operation of machinery, the system is to intervene sequentially after the operation of alarm system by:

Starting of standby units,
load reduction or shutdown, such that the least drastic action is taken first.

M30.2.5 The system should be designed to 'fail safe'. The characteristics of 'fail safe' of a system is to be evaluated on the basis not only of the safety system itself and its associated machinery, but also on the inclusion of the whole machinery installation as well as the ship.

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M30
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M30.2.6 Safety systems of different units of the machinery plant are to be independent. Failure in the safety system of one part of the plant is not to interfere with the operation of the safety system in another part of the plant.

M30.2.7 When the system has been activated, means are to be provided to trace the cause of the safety action.

M30.2.8 When the system has stopped a unit, the unit is not to be restarted automatically before a manual reset has been carried out.

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M31
(1978)

Continuity of electrical power supply for vessels with periodically unattended machinery spaces

M31.1 The continuity of electrical power on vessels with periodically unattended machinery spaces is to be assured in accordance with M31.2 and M31.3.

M31.2 For vessels having the electrical power requirements normally supplied by one ship's service generator in case of loss of the generator in operation, there shall be adequate provisions for automatic starting and connecting to the main switchboard of a standby generator of sufficient capacity to permit propulsion and steering and to ensure the safety of the ship with automatic re-starting of the essential auxiliaries including, where necessary, sequential operations. This standby electric power is to be available automatically in not more than 45 seconds.

M31.3 For vessels having the electrical power requirements normally supplied by two or more ship's service generating sets operating in parallel, arrangements are to be provided (by load shedding, for instance) to ensure that in case of loss of one of these generating sets, the remaining ones are kept in operation without overload to permit propulsion and steering and to ensure the safety of the ship.

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M32 (1979) Definition of diesel engine type

M32.1 General

Engines are of the same type if they do not vary in any detail included in the definition in M32.2. When two engines are to be considered of the same type it is assumed that they do not substantially differ in design and their design details, crankshaft, etc., and the materials used meet Rule requirements and are approved by the Classification Society.

M32.2 Definition

The type of internal combustion engine expressed by the Engine Builder's designation is defined by:
- the bore,
- the stroke,
- the method of injection (direct or indirect injection),
- the kind of fuel (liquid, dual-fuel, gaseous),
- the working cycle (4-stroke, 2-stroke),
- the gas exchange (naturally aspirated or supercharged),
- the maximum continuous power per cylinder at maximum continuous speed and/or maximum continuous brake mean effective pressure,[1]
- the method of pressure charging (pulsating system, constant pressure system),
- the charging air cooling system (with or without intercooler, number of stages),
- cylinder arrangement (in-line, vee).[2]

NOTES
1. After a large number of engines has been proved successfully by service experience, an increase in power up to maximum 10% may be permitted, without any further type test, provided approval for such power is given.
2. One type test suffices for the whole range of engines having different numbers of cylinders.

END

M33 (1981) (Rev. 1 (1981) Scantlings of intermediate shafts

UR M33 was replaced by UR M68 in February 2005.

END

M34 Scantlings of coupling flanges

(1980)

M34.1 For intermediate, thrust and propeller shaft couplings having all fitted coupling bolts, the coupling bolt diameter is not less than that given by the following formula:

$$d_b = 0.65 \sqrt{\frac{d^3 (T + 160)}{iDT_b}}$$

where

d_b = diameter (mm) of fitted coupling bolt
d = Rule diameter (mm), i.e., minimum required diameter of intermediate shaft made of material with tensile strength T, taking into account ice strengthening requirements where applicable
i = number of fitted coupling bolts
D = pitch circle diameter (mm) of coupling bolts
T = tensile strength (N/mm^2) of the intermediate shaft material taken for calculation
T_b = tensile strength (N/mm^2) of the fitted coupling bolts material taken for calculation

while: $T \leq T_b \leq 1{,}7T$, but not higher than 1000 N/mm^2.

M34.2 The design of coupling bolts in the shaftline other than that covered by M34.1 are to be considered and approved by the Classification Society individually.

M34.3 For intermediate shafts, thrust shafts and inboard end of propeller shafts the flange is to have a minimum thickness of 0,20 times the Rule diameter d of the intermediate shaft or the thickness of the coupling bolt diameter calculated for the material having the same tensile strength as the corresponding shaft, whichever is greater.

Special consideration will be given by the Classification Societies for flanges having non-parallel faces, but in no case is the thickness of the flange to be less than the coupling bolt diameter.

M34.4 Fillet radii at the base of the flange should in each case be not less than 0,08 times the actual shaft diameter.

Fillets are to have a smooth finish and should not be recessed in way of nuts and bolt heads.

The fillet may be formed of multiradii in such a way that the stress concentration factor will not be greater than that for a circular fillet with radius 0,08 times the actual shaft diameter.

◄◄

M35 Alarms, remote indications and safeguards for main reciprocating I.C. engines installed in unattended machinery spaces

(1980)
(Rev.1 1993)
(Rev.2 1996)
(Rev.3 1997)
Rev.4 1999)
(Rev.5 Aug 2008)

35.1 General

Alarms, remote indications and safeguards listed in Table 1 and 2 are respectively referred to slow speed (crosshead) and medium/high speed (trunk piston) reciprocating i.c. engines.

35.2 Alarms

A system of alarm displays and controls is to be provided which readily ensures identification of faults in the machinery and satisfactory supervision of related equipment. This may be provided at a main control station or, alternatively, at subsidiary control stations. In the latter case, a master alarm display is to be provided at the main control station showing which of the subsidiary control stations is indicating a fault condition.

The detailed requirements covering communications of alarms from machinery spaces to the bridge area and accommodation for engineering personnel, are contained in M29.

35.3 Remote indications

Remote indications are required only for ships which are operated with machinery space unattended but under a continuous supervision from a position where control and monitoring devices are centralized, without the traditional watch service being done by personnel in machinery space.

35.4 Safeguards

35.4.1 Automatic start of standby pumps – slow down

A suitable alarm is to be activated at the starting of those pumps for which the automatic starting is required.

Note:

1. The requirements of M35 Rev.5 are to be uniformly implemented by IACS Societies for engines:
    i) when an application for certification of an engine is dated on or after 1 January 2010; or
    ii) which are installed in new ships for which the date of contract for construction is on or after 1 January 2010.

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

M35
(cont)

35.4.2 Automatic reduction of power

If overriding devices of the required automatic reduction of power are provided, they are to be so arranged as to preclude their inadvertent operation, and a suitable alarm is to be activated by their operation.

35.4.3 Automatic stop – shut down

If overriding devices of the required automatic stops are provided, they are to be so arranged as to preclude their inadvertent operation, and a suitable alarm is to be operated by their activation. When the engine is stopped automatically, restarting after restoration of normal operating conditions is to be possible only after manual reset, e.g. by-passing the control lever through the 'stop' position.
Automatic restarting is not permissible (see M30.2.8).

M35
(cont)

Table 1

	Gr 1			Gr 2	Gr 3
Monitored parameters for slow speed diesel engines	**Remote Indication**	**Alarm activation**	**Slow down with alarm**	**Automatic start of standby pump with alarm**	**Shut down with alarm**
1.0 Fuel oil system					
Fuel oil pressure after filter (engine inlet)	x	low		x	
Fuel oil viscosity before injection pumps or Fuel oil temp before injection pumps		high low			
Leakage from high pressure pipes		x			
Level of fuel oil in daily service tank[1]		low			
Common rail fuel oil pressure		low			
2.0 Lubricating oil system					
Lub oil to main bearing and thrust bearing, pressure	x	low	x	x	x
Lub oil to crosshead bearing pressure[2]	x	low	x	x	x
Lub oil to camshaft pressure[2]		low		x	x
Lub oil to camshaft temp[2]		high			
Lub oil inlet temp		high			
Thrust bearing pads temp or bearing outlet temp		high	x		x
Main, crank, crosshead bearing, oil outlet temp or Oil mist concentration in crankcase[3]		high	x		
Flow rate cylinder lubricator. Each apparatus		low	x		
Level in lubricating oil tanks[4]		low			
Common rail servo oil pressure		low			
3.0 Turbocharger system					
Turbocharger lub oil inlet pressure[9]		low			
Turbocharger lub oil outlet temp each bearing[10]		high			
Speed of turbocharger	x				
4.0 Piston cooling system					
Piston coolant inlet pressure[5]		low	x	x	
Piston coolant outlet temp each cylinder		high	x		
Piston coolant outlet flow each cylinder[8]		low	x		
Level of piston coolant in expansion tank		low			
5.0 Sea water cooling system					
Sea water pressure		low		x	

Gr 1 Common sensor for indication, alarm, slow down
Gr 2 Sensor for automatic start of standby pump with alarm
Gr 3 Sensor for shut down

M35
(cont)

Table 1 (continued)

	Gr 1			Gr 2	Gr 3
Monitored parameters for slow speed diesel engines	**Remote Indication**	**Alarm activation**	**Slow down with alarm**	**Automatic start of standby pump with alarm**	**Shut down with alarm**
6.0 Cylinder fresh cooling water system					
Cylinder water inlet pressure		low	x	x	
Cylinder water outlet temp (from each cylinder) or Cylinder water outlet temp (general)[6]		high	x		
Oily contamination of engine cooling water system[7]		x			
Level of cylinder cooling water in expansion tank		low			
7.0 Starting and control air systems					
Starting air pressure before main shut-off valve	x	low			
Control air pressure		low			
Safety air pressure		low			
8.0 Scavenge air system					
Scavenge air receiver pressure	x				
Scavenge air box temp (fire)		high	x		
Scavenge air receiver water level		high			
9.0 Exhaust gas system					
Exhaust gas temp after each cylinder	x	high	x		
Exhaust gas temp after each cylinder. Deviation from average.		high			
Exhaust gas temp before each T/C	x	high			
Exhaust gas temp after each T/C	x	high			
10.0 Fuel valve coolant					
Pressure of fuel valve coolant		low		x	
Temperature of fuel valve coolant		high			
Level of fuel valve coolant in expansion tank		low			
11.0 Engine speed/direction of rotation.	x				
Wrong way		x			
12.0 Engine overspeed					x
13.0 Control-Safety-Alarm system power supply failure		x			

M35
(cont)

1 High-level alarm is also required if no suitable overflow arrangement is provided.

2 If separate lub oil systems are installed.

3 When required by UR M10.8 or by SOLAS Reg. II-1/47.2.

4 Where separate lubricating oil systems are installed (e.g. camshaft, rocker arms, etc.), individual level alarms are required for the tanks.

5 The slow down is not required if the coolant is oil taken from the main cooling system of the engine.

6 Where one common cooling space without individual stop valves is employed for all cylinder jackets.

7 Where main engine cooling water is used in fuel and lubricating oil heat exchangers.

8 Where outlet flow cannot be monitored due to engine design, alternative arrangement may be accepted.

9 Unless provided with a self-contained lubricating oil system integrated with the turbocharger.

10 Where outlet temperature from each bearing cannot be monitored due to the engine/turbocharger design alternative arrangements may be accepted. Continuous monitoring of inlet pressure and inlet temperature in combination with specific intervals for bearing inspection in accordance with the turbocharger manufacturer's instructions may be accepted as an alternative.

M35
(cont)

Table 2

	Gr 1			Gr 2	Gr 3
Monitored parameters for medium and high speed diesel engines	**Remote Indication**	**Alarm activation**	**Slow down with alarm**	**Automatic start of standby pump with alarm**	**Shut down with alarm**
1.0 Fuel oil system					
Fuel oil pressure after filter (engine inlet)	x	low		x	
Fuel oil viscosity before injection pumps or Fuel oil temp before injection pumps[1]		high low			
Leakage from high pressure pipes		x			
Level of fuel oil in daily service tank[2]		low			
Common rail fuel oil pressure		low			
2.0 Lubrication oil system					
Lub oil to main bearing and thrust bearing, pressure	x	low		x	x
Lub oil filter differential pressure	x	high			
Lub oil inlet temp	x	high			
Oil mist concentration in crankcase[3]		high			x
Flow rate cylinder lubricator. Each apparatus		low	x		
Common rail servo oil pressure		low			
3.0 Turbocharger system					
Turbocharger lub oil inlet pressure[5]	x	low			
Turbocharger lub oil temperature each bearing[8]		high			
4.0 Sea Water cooling system					
Sea Water pressure	x	low		x	
5.0 Cylinder fresh cooling water system					
Cylinder water inlet pressure or flow	x	low	x	x	
Cylinder water outlet temp (general)[6]	x	high	x		
Level of cylinder cooling water in expansion tank		low			
6.0 Starting and control air systems					
Starting air pressure before main shut-off valve	x	low			
Control air pressure	x	low			

Gr 1 **Common sensor for indication, alarm, slow down**
Gr 2 **Sensor for automatic start of standby pump with alarm**
Gr 3 **Sensor for shut down**

M35
(cont)

Table 2 (continued)

	Gr 1			Gr 2	Gr 3
Monitored parameters for medium and high speed diesel engines	**Remote Indication**	**Alarm activation**	**Slow down with alarm**	**Automatic start of standby pump with alarm**	**Shut down with alarm**
7.0 Scavenge air system					
Scavenge air receiver temp		high			
8.0 Exhaust Gas system					
Exhaust gas temp after each cylinder[7]	x	high	x		
Exhaust gas temp after each cylinder. Deviation from average[7]		high			
9.0 Engine speed	x				
10.0 Engine overspeed					x
11.0 Control-Safety-Alarm system power supply failure		x			

1 For heavy fuel oil burning engines only.

2 High-level alarm is also required if no suitable overflow arrangement is provided.

3 When required by UR M10.8 or by SOLAS Reg. II-1/47.2. One oil mist detector for each engine having two independent outputs for initiating the alarm and shut-down would satisfy the requirement for independence between alarm and shut-down system.

4 If necessary for the safe operation of the engine.

5 Unless provided with a self-contained lubricating oil system integrated with the turbocharger.

6 Two separate sensors are required for alarm and slow down.

7 For engine power > 500 kW/cyl.

8 Where outlet temperature from each bearing cannot be monitored due to the engine/ turbocharger design alternative arrangements may be accepted. Continuous monitoring of inlet pressure and inlet temperature in combination with specific intervals for bearing inspection in accordance with the turbocharger manufacturer's instructions may be accepted as an alternative.

End of Document

M36 Alarms and safeguards for auxiliary reciprocating internal combustion engines driving generators in unattended machinery spaces

(1980)
(Rev.1 1993)
(Rev.2 June 2000)
(Rev.3 Sep 2008)

M36.1 General

This UR refers to medium/high speed (trunk piston) reciprocating i. c. engines on fuel oil.

M36.2 Alarms

All monitored parameters for which alarms are required to identify machinery faults and associated safeguards are listed in Table 1.

All these alarms are to be indicated at the control location for machinery as individual alarms; where the alarm panel with individual alarms is installed on the engine or in the vicinity, common alarm in the control location for machinery is required.

For communication of alarms from machinery space to bridge area and accommodation for engineering personnel detailed requirements are contained in M29.

Note:

1. The requirements of M36 Rev.3 are to be uniformly implemented by IACS Societies for engines:
   i) when an application for certification of an engine is dated on or after 1 January 2010; or
   ii) which are installed in new ships for which the date of contract for construction is on or after 1 January 2010.

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

M36
(cont)

Table 1

Monitored parameters	Alarm	Shut down
Fuel oil leakage from high pressure pipes	x	
Lubricating oil temperature	high	
Lubricating oil pressure	low	x
Oil mist concentration in crankcase[3]	high	x
Pressure or flow of cooling water	low	
Temperature of cooling water or cooling air	high	
Level in cooling water expansion tank, if not connected to main system	low	
Level in fuel oil daily service tank	low	
Starting air pressure	low	
Overspeed activated		x
Fuel oil viscosity before injection pumps or fuel oil temp before injection pumps[1]	low high	
Exhaust gas temperature after each cylinder[2]	high	
Common rail fuel oil pressure	low	
Common rail servo oil pressure	low	

Notes:

1. For heavy fuel oil burning engines only.
2. For engine power above 500 kW/cyl.
3. When required by UR M10.8 or by SOLAS Reg. II-1/47.2. one oil mist detector for each engine having two independent outputs for initiating the alarm and shut-down would satisfy the requirement for independence between alarm and shut-down system.

End of Document

M37 Scantlings of propeller shafts
(1981)

UR M37 was replaced by UR M68 in February 2005.

END

M38 (1981) k-factors for different shaft design features (intermediate shafts) - see M33

UR M38 was replaced by UR M68 in February 2005.

END

M39 (1981) k-factors for different shaft design features (propeller shafts) - see M37

UR M39 was replaced by UR M68 in February 2005.

M40 Ambient conditions – Temperatures
(1981)

M40.1 The ambient conditions specified under M40.2 are to be applied to the layout, selection and arrangement of all shipboard machinery, equipment and appliances as to ensure proper operation.

M40.2 Temperatures

Air

<table>
<tr><th>Installations,
components</th><th>Location,
arrangement</th><th>Temperature range (°C)</th></tr>
<tr><td rowspan="3">Machinery and electrical installations [1]</td><td>In enclosed spaces</td><td>0 to +45[2]</td></tr>
<tr><td>On machinery components, boilers,
In spaces subject to higher and lower temperatures</td><td>According to specific local conditions</td></tr>
<tr><td>On the open deck</td><td>–25 to +45[2]</td></tr>
</table>

Water

Coolant	Temperature (°C)
Seawater Charge air coolant inlet to charge air cooler	+32[2] see UR M28

NOTES
1. Electronic appliances are to be suitable for proper operation even with an air temperature of +55°C.
2. The Classification Society may approve other temperatures in the case of ships not intended for unrestricted service.

M41 Automation - type testing conditions for control and instrumentation equipment

UR E10 superseded UR M41 (1991)

M42 Steering gear

(1981)
(Rev. 1 1986)
(Rev.2 1995)
(Rev.3 1997)

Preamble

In addition to the requirements contained in the Amendments to the 1974 SOLAS Convention, Chapter II – I Reg. 29 and 30, and related Guidelines (see Annex 2 of IMCO document MSC XLV/4) the following requirements apply to new ocean-going vessels of 500 GRT and upwards. These requirements may be applied to other vessels at the discretion of the Classification Society.

1. Plans and specifications

Before starting construction, all relevant plans and specifications are to be submitted to the Classification Society for approval.

2. Definitions

The definitions relating to steering gear are given in Appendix 1.

3. Power piping arrangements

3.1 The power piping for hydraulic steering gears is to be arranged so that transfer between units can be readily effected.

3.2 Where the steering gear is so arranged that more than one system (either power or control) can be simultaneously operated, the risk of hydraulic locking caused by single failure is to be considered.

3.3 For all vessels with non-duplicated actuators, isolating valves are to be fitted at the connection of pipes to the actuator, and are to be directly fitted on the actuator.

3.4 Arrangements for bleeding air from the hydraulic system are to be provided where necessary.

3.5 Piping, joints, valves, flanges and other fittings are to comply with Classification Society requirements for Class 1 components. The design pressure is to be in accordance with paragraph M42.6.8.

4. Rudder Angle Limiters

Power-operated steering gears are to be provided with positive arrangements, such as limit switches, for stopping the gear before the rudder stops are reached. These arrangements are to be synchronized with the gear itself and not with the steering gear control.

5. Materials

Ram cylinders; pressure housings of rotary vane type actuators; hydraulic power piping valves, flanges and fittings; and all steering gear components transmitting mechanical forces to the rudder stock (such as tillers, quadrants, or similar components) should be of steel or other approved ductile material, duly tested in accordance with the requirements of the Classification Society. In general, such material should not have an elongation of less than 12 per cent nor a tensile strength in excess of 650 N/mm^2.
Grey cast iron may be accepted for redundant parts with low stress level, excluding cylinders, upon special consideration.

▶

M42
cont'd

6. Design

6.1 The construction should be such as to minimize local concentrations of stress.

6.2 Welds
a) The welding details and welding procedures should be approved.
b) All welded joints within the pressure boundary of a rudder actuator or connecting parts transmitting mechanical loads should be full penetration type or of equivalent strength.

6.3 Oil seals
a) Oil seals between non-moving parts, forming part of the external pressure boundary, should be of the metal upon metal type or of an equivalent type.
b) Oil seals between moving parts, forming part of the external pressure boundary, should be duplicated, so that the failure of one seal does not render the actuator inoperative. Alternative arrangements providing equivalent protection against leakage may be accepted at the discretion of the Administration.

6.4 All steering gear components transmitting mechanical forces to the rudder stock, which are not protected against overload by structural rudder stops or mechanical buffers, are to have a strength at least equivalent to that of the rudder stock in way of the tiller.

6.5 For piping, joints, valves, flanges and other fittings see paragraph M42.3.4.

6.6 Rudder actuators other than those covered by Regulation 29.17 and relating Guidelines should be designed in accordance with Class 1 pressure vessels (notwithstanding any exemptions for hydraulic cylinders).

6.7 In application of such rules the permissible primary general membrance stress is not to exceed the lower of the following values:

$$\frac{\sigma B}{A} \text{ or } \frac{\sigma y}{B}$$

where:

σB = specified minimum tensile strength of material at ambient temperature
σy = specified minimum yield stress or 2 per cent proof stress of the material, at ambient temperature

A and B are given by the Table 1.

Table 1

	Steel	Cast Steel	Nodular Cast Iron
A	3.5	4	5
B	1.7	2	3

6.8 The design pressure is to be at least equal to the greater of the following:

(i) 1.25 times the maximum working pressure,
(ii) the relief valve setting.

6.9 Accumulators, if any are to comply with Classification Society requirements for pressure vessels.

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M42 cont'd

7. Dynamic loads for fatigue and fracture mechanic analysis

The dynamic loading to be assumed in the fatigue and fracture mechanics analysis considering Regulation 29.2.2 and 29.17.1 and relating Guidelines, will be established at the discretion of the Classification Society.
Both the case of high cycle and cummulative fatigue are to be considered.

8. Hoses

8.1 Hose assemblies of type approved by the Classification Society may be installed between two points where flexibility is required but should not be subjected to torsional deflection (twisting) under normal operating conditions. In general, the hose should be limited to the length necessary to provide for flexibility and for proper operation of machinery.

8.2 Hoses should be high pressure hydraulic hoses according to recognized standards and suitable for the fluids, pressures, temperatures and ambient conditions in question.

8.3 Burst pressure of hoses should not be less than four times the design pressure.

9. Relief valves

Relief valves for protecting any part of the hydraulic system which can be isolated, as required by Regulation 29.2.3 should comply with the following:

(1) The setting pressure should not be less than 1.25 times the maximum working pressure.
(2) The minimum discharge capacity of the relief valve(s) should not be less than the total capacity of the pumps, which can deliver through it (them), increased by 10 per cent.

Under such conditions the rise in pressure should not exceed 10 per cent of the setting pressure. In this regard, due consideration should be given to extreme foreseen ambient conditions in respect of oil viscosity.

The Classification Society may require, for the relief valves, discharge capacity tests and/or shock tests.

10. Electrical installations

Electrical Installations should comply with the requirements of the Classification Society.

11. Alternative source of power

Where the alternative power source required by Regulation 29.14 is a generator, or an engine driven pump, automatic starting arrangements are to comply with the requirements relating to the automatic starting arrangements of emergency generators.

12. Monitoring and alarm systems

12.1 Monitoring and alarm systems, including the rudder angle indicators, should be designed, built and tested to the satisfaction of the Classification Society.

12.2 Where hydraulic locking, caused by a single failure, may lead to loss of steering, an audible and visual alarm, which identifies the failed system, shall be provided on the navigating bridge.

NOTE:
This alarm should be activated whenever.g:

– position of the variable displacement pump control system does not correspond with given order; or

– incorrect position of 3-way full flow valve or similar in constant delivery pump system is detected.

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M42
cont'd

13. Operating instructions

Where applicable, following standard signboard should be fitted at a suitable place on steering control post on the bridge or incorporated into operating instruction on board:

CAUTION

IN SOME CIRCUMSTANCES WHEN 2 POWER UNITS ARE RUNNING SIMULTANEOUSLY THE RUDDER MAY NOT RESPOND TO HELM. IF THIS HAPPENS STOP EACH PUMP IN TURN UNTIL CONTROL IS REGAINED.

The above signboard is related to steering gears provided with 2 identical power units intended for simultaneous operation, and normally provided with either their own control systems or two separate (partly or mutually) control systems which are/may be operated simultaneously.

Note: Existing vessels according to SOLAS 1986 shall have minimum the above signboard, when applicable.

14. Testing

14.1 The requirements of the Classification Society relating to the testing of Class 1 pressure vessels, piping, and relating fittings including hydraulic testing apply.

14.2 A power unit pump is to be subjected to a type test. The type test shall be for a duration of not less than 100 hours, the test arrangements are to be such that the pump may run in idling conditions, and at maximum delivery capacity at maximum working pressure. During the test, idling periods are to be alternated with periods at maximum delivery capacity at maximum working pressure. The passage from one condition to another should occur at least as quickly as on board. During the whole test no abnormal heating, excessive vibration or other irregularities are permitted. After the test, the pump should be disassembled and inspected. Type tests may be waived for a power unit which has been proven to be reliable in marine service.

14.3 All components transmitting mechanical forces to the rudder stock should be tested according to the requirements of the Classification Society.

14.4 After installation on board the vessel the steering gear is to be subjected to the required hydrostatic and running tests.

15. Trials

The steering gear should be tried out on the trial trip in order to demonstrate to the Surveyor's satisfaction that the requirements of the Rules have been met. The trial is to include the operation of the following:

(i) the steering gear, including demonstration of the performances required by Regulation 29.3.2 and 29.4.2. For controllable pitch propellers, the propeller pitch is to be at the maximum design pitch approved for the maximum continuous ahead R.P.M. at the main steering gear trial.
If the vessel cannot be tested at the deepest draught, alternative trial conditions may be specially considered.
In this case for the main steering gear trial, the speed of ship corresponding to the number of maximum continuous revolution of main engine could apply.
(ii) the steering gear power units, including transfer between steering gear power units.
(iii) the isolation of one power actuating system, checking the time for regaining steering capability.
(iv) the hydraulic fluid recharging system.
(v) the emergency power supply required by Regulation 29.14.
(vi) the steering gear controls, including transfer of control and local control.
(vii) the means of communication between the wheelhouse, engine room, and the steering gear compartment.
(viii) the alarms and indicators required by regulations 29, 30 and M42.12, these tests may be effected at dockside.
(ix) where steering gear is designed to avoid hydraulic locking this feature shall be demonstrated.

◀▶

M42
App.1

Appendix 1

Definitions relating to steering gear

1. Steering gear control system means the equipment by which orders are transmitted from the navigating bridge to the steering gear power units. Steering gear control systems comprise transmitters, receivers, hydraulic control pumps and their associated motors, motor controllers, piping and cables.

2. Main steering gear means the machinery, rudder actuator(s), the steering gear power units, if any, and ancillary equipment and the means of applying torque to the rudder stock (e.g. tiller or quadrant) necessary for effecting movement of the rudder for the purpose of steering the ship under normal service conditions.

3. Steering gear power unit means:

   (a) in the case of electric steering gear, and electric motor and its associated electrical equipment,
   (b) in the case of electrohydraulic steering gear, an electric motor and its associated electrical equipment and connected pump,
   (c) in the case of other hydraulic steering gear, a driving engine and connected pump.

4. Auxiliary steering gear means the equipment other than any part of the main steering gear necessary to steer the ship in the event of failure of the main steering gear but not including the tiller, quadrant or components serving the same purpose.

5. Power actuating system means the hydraulic equipment provided for supplying power to turn the rudder stock, comprising a steering gear power unit or units, together with the associated pipes and fittings, and a rudder actuator. The power actuating systems may share common mechanical components, i.e. tiller, quadrant and rudder stock, or components serving the same purpose.

6. Maximum ahead service speed means the greatest speed which the ship is designed to maintain in service at sea at her deepest sea going draught at maximum propeller RPM and corresponding engine MCR.

7. Rudder actuator means the component which converts directly hydraulic pressure into mechanical action to move the rudder.

8. Maximum working pressure means the maximum expected pressure in the system when the steering gear is operated to comply with 29.3.2.

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M43 (1982) Bridge control of propulsion machinery for unattended machinery spaces

M43.1 Under all sailing conditions, including manoeuvring, the speed, direction of thrust and, if applicable, pitch of the propeller shall be fully controllable from the navigating bridge.

M43.2 In principle the remote control mentioned under M43.1 is to be performed by a single control device for each independent propeller, with automatic performance of all associated services including, where necessary, means of preventing overload and prolonged running in critical speed ranges of the propelling machinery.

M43.3 The bridge control system is to be independent from the other transmission system; however, one control lever for both system may be accepted.

M43.4 Operations following any setting of the bridge control device including reversing from the maximum ahead service speed in case of emergency are to take place in an automatic sequence and with time intervals acceptable to the machinery.

M43.5 The main propulsion machinery shall be provided with an emergency stopping device on the navigating bridge and independent from the bridge control system.

M43.6 Remote starting of the propulsion machinery is to be automatically inhibited if conditions exist which may hazard the machinery, e.g. shaft turning gear engaged, drop of lubricating oil pressure.

M43.7 For steam turbines a slow-turning device should be provided which operates automatically if the turbine is stopped longer than admissible. Discontinuation of this automatic turning from the bridge must be possible.

M43.8 The design of the bridge control system shall be such that in case of its failure an alarm is given. In this case the speed and direction of the propeller thrust is to be maintained until local control is in operation, unless this is considered impracticable. In particular, lack of power (electric, pneumatic, hydraulic) should not lead to major and sudden change in propulsion power or direction of propeller rotation.

M43.9 The number of automatic consecutive attempts which fail to produce a start shall be limited to maintain sufficient starting air pressure. An alarm shall be provided at an air pressure level, which still permits main engine starting operation.

M43.10 It shall be possible for the propulsion machinery to be controlled from a local position even in the case of failure in any part of the automatic or remote control systems.

M43.11 Remote control of the propulsion machinery shall be possible only from one control location at one time; at such locations interconnected control positions are permitted.

M43.12 The control system shall include means to prevent the propelling thrust from altering significantly when transferring control from one control to another.

M43.13 Each control location is to be provided with means to indicate which of them is in control. Propolusion machinery orders from the navigating bridge shall be indicated in the engine control room or at the manoeuvring platform, as appropriate.

M43.14 The transfer of control between the navigating bridge and machinery spaces shall be possible only in the main machinery space or the main machinery control room.

◄◄

M44 Documents for the approval of diesel engines

(1982)
(Rev. 1 1983)
(Rev. 2 1984)
(Rev. 3 1986
(Rev. 4 1989)
(Rev. 5 1992)
Rev.6 (Nov 2003)
(Rev.7 May 2004)

For each type of engine that is required to be approved the documents listed in the following table and as far as applicable to the type of engine are to be submitted to the Classification Society for approval (A), approval of materials and weld procedure specifications (A*), or for information (R) by each engine manufacturer (see Note 4). After the approval of an engine type has been given by the Classification Society for the first time, only those documents as listed in the table which have undergone substantive changes will have to be submitted again for consideration by the Classification Society. In cases where 2 identifications (R/A*) are given, the first refers to cast design and the second to welded design. The assignment of the letter R does not preclude possible comments by the individual Classification Society.

No.	A/R	Item
1	R	Engine particulars as per attached sheet
2	R	Engine transverse cross-section
3	R	Engine longitudinal section
4	R/A*	Bedplate and crankcase, cast or welded with welding details and instructions[9]
5	R	Thrust bearing assembly[3]
6	R/A*	Thrust bearing bedplate, cast or welded with welding details and instructions[9]
7	R/A*	Frame/framebox, cast or welded with welding details and instructions[1,9]
8	R	Tie rod
9	R	Cylinder head, assembly
10	R	Cylinder liner
11	A	Crankshaft, details, each cylinder No.
12	A	Crankshaft, assembly, each cylinder No.
13	A	Thrust shaft or intermediate shaft (if integral with engine)
14	A	Shaft coupling bolts
15	R	Counterweights (if not integral with crankshaft), including fastening
16	R	Connecting rod
17	R	Connecting rod, assembly
18	R	Crosshead, assembly[2]
19	R	Piston rod, assembly[2]
20	R	Piston, assembly
21	R	Camshaft drive, assembly

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M44
cont'd

No.	A/R	Item
22	A	Material specifications of main parts with information on non-destructive material tests and pressure tests[8]
23	R	Arrangement of foundation (for main engines only)
24	A	Schematic layout or other equivalent documents of starting air system on the engine[6]
25	A	Schematic layout or other equivalent documents of fuel oil system on the engine[6]
26	A	Schematic layout or other equivalent documents of lubricating oil system on the engine[6]
27	A	Schematic layout or other equivalent documents of cooling water system on the engine[6]
28	A	Schematic diagram of engine control and safety system on the engine[6]
29	R	Shielding and insulation of exhaust pipes, assembly
30	A	Shielding of high pressure fuel pipes, assembly[4]
31	A	Arrangement of crankcase explosion relief valve[5]
32	R	Operation and service manuals[7]
33	A	Schematic layout or other equivalent documents of hydraulic system (for valve lift) on the engine
34	A	Type test program and type test report
35	A	High pressure parts for fuel oil injection system[10]

FOOTNOTES:
1. only for one cylinder.
2. only necessary if sufficient details are not shown on the transverse cross section and longitudinal section.
3. if integral with engine and not integrated in the bedplate.
4. all engines.
5. only for engines of a cylinder diameter of 200 mm or more or a crankcase volume of 0.6 m^3 or more.
6. and the system so far as supplied by the engine manufacturer. Where engines incorporate electronic control systems a failure mode and effects analysis (FMEA) is to be submitted to demonstrate that failure of an 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 an acceptable performance criteria of the engine.
7. operation and service manuals are to contain maintenance requirements (servicing and repair) including details of any special tools and gauges that are to be used with their fitting/settings together with any test requirements on completion of maintenance.
8. for comparison with Society requirements for material, NDT and pressure testing as applicable.
9. The weld procedure specification is to include details of pre and post weld heat treatment, weld consumables and fit-up conditions.
10. The documentation to contain specification of pressures, pipe dimensions and materials.

NOTES:
1. The approval of exhaust gas turbochargers, charge air coolers, etc. is to be obtained by the respective manufacturer.
2. Where considered necessary, the Society may request further documents to be submitted. This may include details of evidence of existing type approval or proposals for a type testing programme in accordance with M50.
3. The number of copies to be submitted is left to each Society.
4. A Licensee is to submit, for each engine type manufactured, a list of all documents required by the Classification Society with the relevant drawing numbers and revision status from both Licensor and Licensee.
   Where the Licensee proposes design modifications to components, the associated documents are to be submitted by the Licensee for approval or for information. In case of significant modifications a statement is to be made confirming the Licensor's acceptance of the changes.
   In all cases a complete set of documents will be required by the surveyor(s) attending the Licensee's work.
5. Where the operation and service manuals identify special tools and gauges for maintenance purposes (see footnote 7) refer to UR P2.7.4.14.
6. The FMEA reports required by FOOTNOTE 6 will not be explicitly approved by the Classification Society

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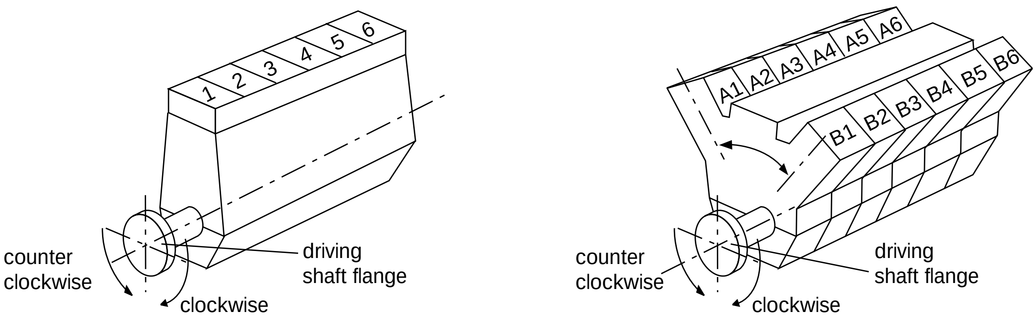
M44
cont'd

DATA SHEET

for calculation of Crankshafts for I.C. Engines

– based on IACS UR M 53 –

1	Engine Builder
2	Engine Type Designation
3	Stroke-Cycle ☐ 2 SCSA ☐ 4 SCSA
4	Kind of engine ☐ In-line engine ☐ V-type engine with adjacent connecting rods ☐ V-type engine with articulated-type connecting rods ☐ V-type engine with forked/inner connecting rods ☐ Crosshead engine ☐ Trunk piston engine
5	Combustion Method ☐ Direct injection ☐ Precombustion chamber ☐ Others: ______________________
6	 Fig. 1 Designation of the cylinders
7	Sense of Rotation (corresponding to Item 6) ☐ Clockwise ☐ Counter clockwise

▶

M44
cont'd

8	Firing Order (corresponding to Item 6 and 7)		
9	Firing Intervals [deg] (corresponding to Item 8)		
10	Rated Power		kW
11	Rated Engine Speed		1/min
12	Mean Effective Pressure		bar
13	Mean Indicated Pressure		bar
14	Maximum Cylinder Pressure (Gauge)		bar
15	Charge Air Pressure (Gauge) (before inlet valves or scavenge ports)		bar
16	Nominal Compression Ratio		–
17	Number of Cylinders		–
18	Diameter of Cylinders		mm
19	Length of Piston Stroke		mm
20	Length of Connecting Rod (between bearing centers)		mm
21	Oscillating Mass of one cylinder (mass of piston, rings, pin, piston rod, crosshead, oscillating part of connecting rod)		kg
22	Digitalized Gas Pressure Curve (Gauge) – presented at equidistant intervals [bar versus crank angle] – (intervals not more than 5° CA) ☐ given in the appendix		

	Additional Data of V-type Engines		
23	V-Angle α_V (corresponding to Item 6)		deg
	For the Cylinder Bank with Articulated-type Connecting Rod (Dimensions corresponding to Item 27)		
24	Maximum Cylinder Pressure (Gauge)		bar
25	Charge Air Pressure (Gauge) (before inlet valves or scavenge ports)		bar
26	Nominal Compression Ration		–

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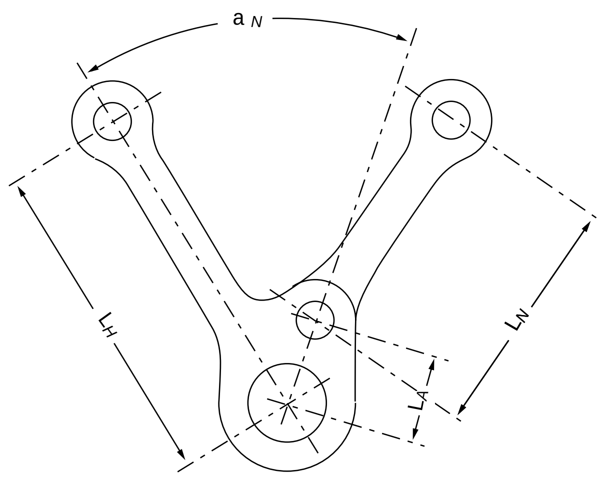
M44
cont'd

27	 Articulated-type connecting rod		
28	Distance to Link Point L_A		mm
29	Link Angle α_N		deg
30	Length of Connecting Rod L_N		mm
31	Oscillating Mass of one cylinder (mass of piston, rings, pin, piston rod, crosshead, oscillating part of connecting rod)		kg
32	Digitalized Gas Pressure Curve (Gauge) – presented at equidistant intervals [bar versus crank angle] – (intervals not more than 5° CA) ☐ given in the appendix		
	For the Cylinder Bank with Inner Connecting Rod		
33	Oscillating Mass of one cylinder (mass of piston, rings, pin, piston rod, crosshead, oscillating part of connecting rod)		kg

Data of Crankshaft (Dimensions corresponding to Item 39) Note: For asymmetric cranks the dimensions are to be entered both for the left and right part of crank throw.	
34	Drawing No.
35	Kind of crankshaft (e.g. solid-forged crankshaft, semi-built crankshaft, etc.)

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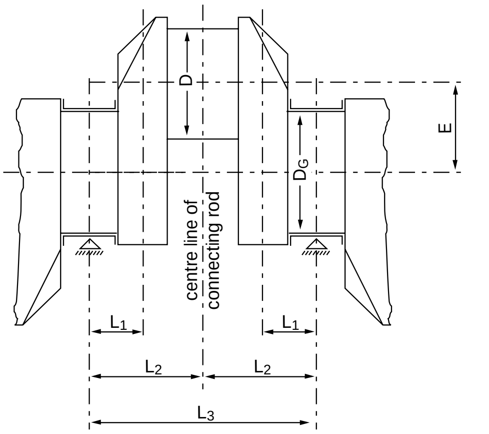
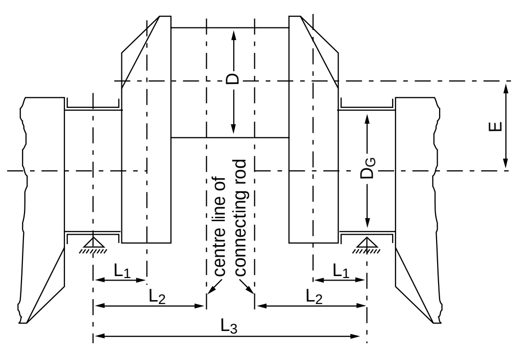
M44
cont'd

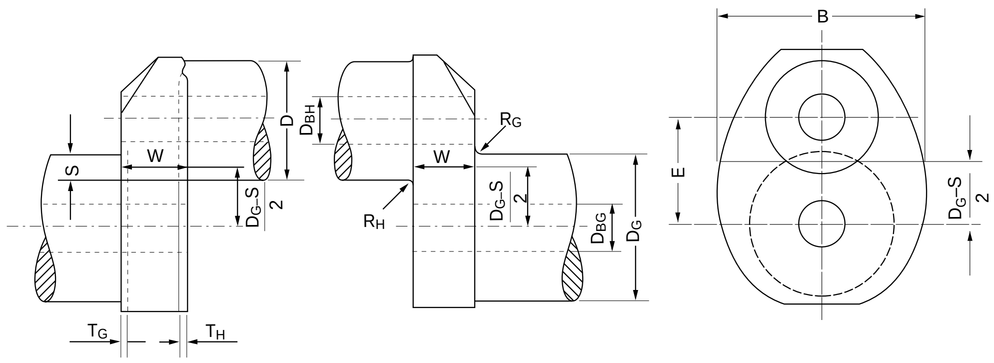
36	Method of Manufacture (e.g. free form forged, cast steel, etc.) ☐ Description of the forging process – if c.g.f forged or drop-forged – given in the appendix	
37	Heat treatment (e.g. tempered)	
38	Surface Treatment of Fillets, Journals and Pins (e.g. induction hardened, nitrided, rolled, etc.) ☐ Full details given in the appendix	



Crank throw for in-line engine



Crank throw for engine with 2 adjacent connecting rods



Crank dimensions necessary for the calculation of stress concentration factors

40	Crankpin Diameter D	mm
41	Diameter of Bore in Crankpin D_{BH}	mm
42	Fillet Radius of Crankpin R_H	mm
43	Recess of Crankpin T_H	mm

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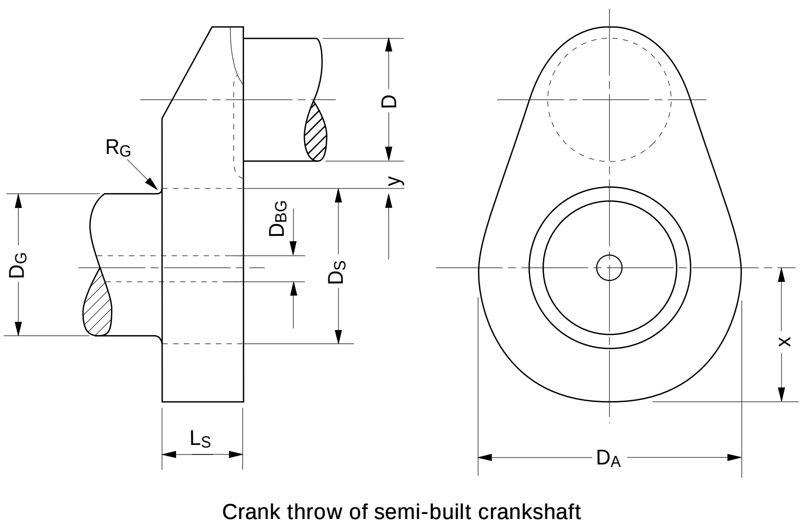
M44
cont'd

44	Journal Diameter D_G		mm
45	Diameter of Bore in Journal D_{BG}		mm
46	Fillet Radius of Journal R_G		mm
47	Recess of Journal T_G		mm
48	Web Thickness W		mm
49	Web Width B		mm
50	Bending Length L_1		mm
51	Bending Length L_2		mm
52	Bending Length L_3		mm
53	Oil Bore Design ☐ Safety margin against fatigue at the oil bores is not less than than acceptable in the fillets		
54	Diameter of Oil Bore		mm
55	Smallest Edge Radius of Oil Bore		mm
56	Surface Roughness of Oil Bore Fillet		μm
57	Inclination of Oil Bore Axis related to Shaft Axis		deg

Additional Data for Shrink-Fits of Semi-Built Crankshafts
(dimensions corresponding to Item 58)

58



Crank throw of semi-built crankshaft

M44
cont'd

59	Shrink Diameter D_S		mm
60	Length of Shrink-Fit L_S		mm
61	Outside Diameter of Web D_A or Twice the Minimum Distance x (the small er value is to be entered)		mm
62	Amount of Shrink-Fit (upper and lower tolerances)		mm
			%o
63	Maximum Torque (ascertained according to M 53.2.2.2 with consideration of the mean torque)		Nm

	Data of Crankshaft Material Note: Minimum values of mechanical properties of material obtained from longitudinal test specimens		
64	Material Designation (according to DIN, AISI, etc.)		
65	Method of Material Melting Process (e.g. open-hearth furnace, electric furnace, etc.)		
66	Tensile Strength		N/mm^2
67	Yield Strength		N/mm^2
68	Reduction in Area at Break		%
69	Elongation A_5		%
70	Impact Energy – KV		J
71	Young's Modulus		N/mm^2
	Additional Data for Journals of Semi-Built Crankshafts		
72	Material Designation (according to DIN, AISI, etc.)		
73	Tensile Strength		N/mm^2
74	Yield Strength		N/mm^2

▶

M44
cont'd

Data According to Calculation of Torsional Stresses Note: In case the Society is entrusted with carrying out a forced vibration calculation to determine the alternating torsional stresses to be expected in the engine and possibly in its shafting, the data according to M53.2.2.1 are to be submitted.			
75	Max. Nominal Alternating Torsional Stress (ascertained by means of a harmonic synthesis according to M53.2.2.2 and related to cross-sectional area of bored crank pin)		N/mm^2
76	Engine Speed (at which the max. nominal alternating torsional stress occurs)		1/min
77	Minimum Engine Speed (for which the harmonic synthesis was carried out)		1/min

Data of Stress Concentration Factors (S.C.F.) and/or Fatigue Strength Furnished by Reliable Measurements Note: To be filled in only when data for stress concentration factors and/or fatigue are furnished by the engine manufacturer on the basis of measurements. Full supporting details are to be enclosed.			
78	S.C.F. for Bending in Crankpin Fillet α_B		–
79	S.C.F. for Torsion in Crankpin Fillet α_T		–
80	S.C.F. for Bending in Journal Fillet β_B		–
81	S.C.F. for Shearing in Journal Fillet β_Q		–
82	S.C.F. for Torsion in Journal Fillet β_T		–
83	Allowable Fatigue Strength of Crankshaft σ_{DW}		N/mm^2

Remarks	
84	

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M44
cont'd

Remarks (continued)	

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M45 Ventilation of engine rooms

(1982)
(Rev. 1
1987)

Machinery spaces shall be sufficiently ventilated so as to ensure that when machinery or boilers therein are operating at full power in all weather conditions, including heavy weather, a sufficient supply of air is maintained to the spaces for the operation of the machinery.

Interpretation:
The sufficient amount of air is to be supplied through suitably protected openings arranged in such a way that they can be used in all weather conditions, taking into account Reg. 19 of the 1966 Load Line Convention.

◀◀

M46 Ambient conditions – Inclinations

(1982
Rev. 1
June
2002)

M46.1 The ambient conditions specified under M46.2 are to be applied to the layout, selection and arrangement of all shipboard machinery, equipment and appliances to ensure proper operation.

M46.2 Inclinations

Installations, components	Angle of inclination [°][2]			
	Athwartships		Fore-and-aft	
	static	dynamic	static	dynamic
Main and auxiliary machinery	15	22,5	5[4]	7,5
Safety equipment, e.g. emergency power installations, emergency fire pumps and their devices Switch gear, electrical and electronic appliances[1] and remote control systems	22,5[3]	22,5[3]	10	10

NOTES:
1. Up to an angle of inclination of 45° no undesired switching operations or operational changes may occur.
2. Athwartships and fore-end-aft inclinations may occur simultaneously.
3. In ships for the carriage of liquefied gases and of chemicals the emergency power supply must also remain operable with the ship flooded to a final athwartships inclination up to maximum of 30°.
4. Where the length of the ship exceeds 100m, the fore-and-aft static angle of inclination may be taken as 500/L degrees where L = length of the ship, in metres, as defined in UR S2.

The Society may consider deviations from these angles of inclination taking into consideration the type, size and service conditions of the ship.

◀◀

M47 (1983) Bridge control of propulsion machinery for attended machinery spaces

Installations shall comply with the requirements of M43. If the slow-turning device referred to in M43.7 is arranged to be operated manually, automatic operation will noMt be required.

END

M48 (1983) Permissible limits of stresses due to torsional vibrations for intermediate, thrust and propeller shafts

UR M48 was replaced by UR M68 in February 2005.

END

▶

M49 Availability of Machinery

(1984)
(Rev 1
1996)
(Rev. 2
1998)

Deleted in Dec 2003

(E8 has been merged with UR M49 to form a new UR M61 (Dec 2003))

◄◄

M50
(1986)
(Rev.1 1987)
(Rev.2 1991)
(Corr. Feb 1999)
(Rev.3 Jan 2008)

Programme for type testing of non-mass produced I.C. engines

M50.1 General

Upon finalization of the engine design for production of every new engine type intended for the installation on board ships, one engine shall be presented for type testing as required by the Rules of Classification Societies.

A type test carried out for a particular type of engine at any place at any manufacturer will be accepted for all engines of the same type built by licensees and licensors.

Engines which are subjected to type testing are to be tested in accordance to the scope as specified below.

It is taken for granted that:

1.1 this engine is optimised as required for the condition of the type,

1.2 the investigations and measurements required for reliable engine operation have been carried out during internal tests by the engine manufacturer and

1.3 the design approval has been obtained for the engine type in question on the basis of documentation requested (UR M44) and the Classification Societies have been informed about the nature and extent of investigations carried out during the pre-production stages.

The type test is subdivided into three stages, namely:

- Stage A - Internal tests

Functional tests and collection of operating values including test hours during the internal tests, the relevant results of which are to be presented to the Classification Societies during the type test.

Testing hours of components which are inspected according to M50.4 shall be stated.

- Stage B - Type approval test

Test approval test in the presence of the Classification Societies' representatives.

- Stage C - Component inspection

Component inspections by the Classification Societies after completion of the test programme.

The engine manufacturer will have to compile all results and measurements for the engine tested during the type test in a type test report, which will have to be handed over to the Classification Society in question.

Note:
1. The requirements of M50 Rev.3 are to be uniformly implemented by IACS Societies for engines which are presented for type testing on or after 1 January 2009.

M50 (cont)

M50.2 Stage A - Internal tests

Function tests and collection of operating data during the internal tests.

During the internal tests the engine is to be operated at the load points important for the engine manufacturer and the pertaining operating values are to be recorded. The load points may be selected according to the range of application.

If an engine can be satisfactorily operated at all load points without using mechanically driven cylinder lubricators this is to be verified.

For engines which may operate of heavy fuel oil, the suitability for this will have to be proved in an appropriate form, at Manufacturer's (Licensor or Licensee) testbed in general, but, where not possible, latest on board for the first engine to be put into service.

2.1 Normal case

The normal case includes:

2.1.1 The load points 25%, 50%, 75%, 100% and 110% of the maximum rated power

- along the nominal (theoretical) propeller curve and at constant speed for propulsion engines

- at constant speed for engines intended for generating sets.

2.1.2 The limit points of the permissible operating range. These limit points are to be defined by the engine manufacturer.

2.2 Emergency operation situations

For turbocharged engines the achievable continuous output is to be determined in the case of turbocharger damage.

- engines with one turbocharger, when rotor is blocked or removed

- engines with two or more turbochargers, when damaged turbochargers are shut off.

M50.3 Stage B - Type approval test

During the type test the tests listed under 3.1 to 3.3 are to be carried out in the presence of the Classification Societies and the results achieved are to be recorded and signed by the attending representatives. Deviations from this programme, if any, are to be agreed between the engine manufacturer and the Classification Societies.

3.1 Load points

Load points at which the engine is to be operated according to the power/speed diagram (figure 1).

The data to be measured and recorded when testing the engine at various load points are to include all necessary parameters for the engine operation.

The operating time per load point depends on the engine size (achievement of steady-state condition) and on the time for collection of the operating values.

M50
(cont)

Normally, an operating time of 0.5 hour can be assumed per load point.

At the rated power as per 3.1.1 and operating time of two hours is required. Two sets of readings are to be taken at a minimum interval of one hour.

3.1.1 Rated power, i.e. 100% output at 100% torque and 100% speed corresponding to load point 1.

3.1.2 100% power at maximum permissible speed corresponding to load point 2.

3.1.3 Maximum permissible torque (normally 110%) at 100% speed corresponding to load point 3.

or maximum permissible power (normally 110%) and speed according to nominal propeller curve corresponding to load point 3a.

3.1.4 Minimum permissible speed at 100% torque corresponding to load point 4.

3.1.5 Minimum permissible speed at 90% torque corresponding to load point 5.

3.1.6 Part loads, e.g. 75%, 50%, 25% of rated power and speed according to nominal propeller curve corresponding to points 6, 7 and 8.

and at rated speed with constant governor setting corresponding to points 9, 10 and 11.

3.2 Emergency operation

Maximum achievable power when operating along the nominal propeller curve and when operating with constant governor setting for rated speed as per 2.2.

3.3 Functional tests

3.3.1 Lowest engine speed according to nominal propeller curve.

3.3.2 Starting tests, for non-reversible engines and/or starting and reversing tests, for reversible engines.

3.3.3 Governor test.

3.3.4 Testing the safety system, particularly for overspeed and low lub. oil pressure.

3.3.5 Integration Test: For electronically controlled diesel engines integration tests shall verify that the response of the complete mechanical, hydraulic and electronic system is as predicted for all intended operational modes. The scope of these tests shall be agreed with the Society for selected cases based on the FMEA required in UR M44.

NOTE:
For engines, intended to be used for emergency services supplementary tests according to the regulations of administration may be required.

M50
(cont)

M50.4 Stage C - Component inspection

Immediately after the test run the components of one cylinder for in-line engines and two cylinders for V-engines are presented for inspections.

The following components are concerned:

- Piston removed and dismantled
- Crosshead bearing, dismantled
- Crank bearing and main bearing, dismantled
- Cylinder liner in the installed condition
- Cylinder head, valves disassembled
- Control gear, camshaft and crankcase with opened covers.

NOTE:
If deemed necessary by the representative of Classification Society further dismantling of the engine may be required.

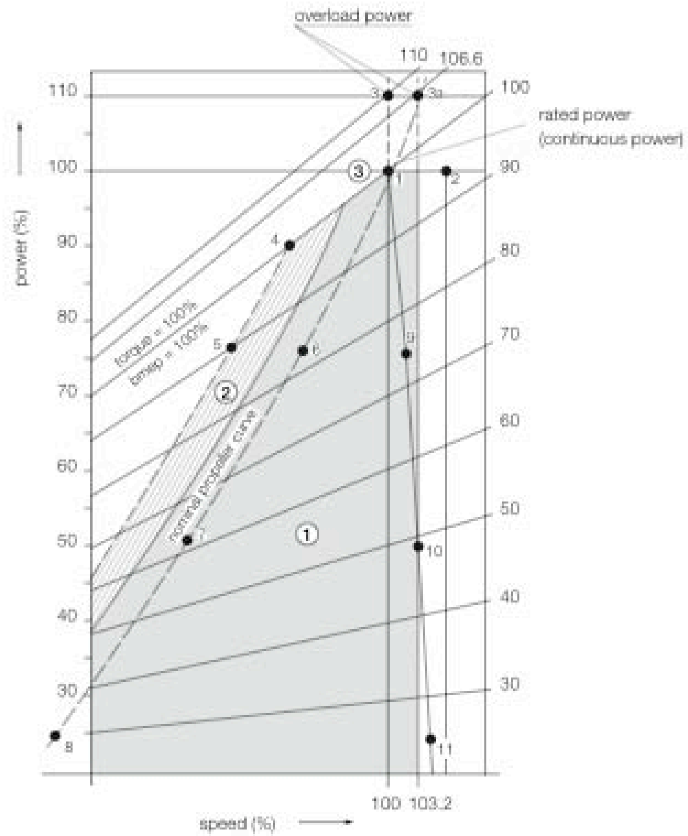
M50.5 Notes

5.1 If a type tested engine which has proven reliability in service is increased in output by not more than 10%, new type approval test is not necessary as laid down in UR M32. The agreement for granting an increased output will be subject to prior plan approval.

5.2 Each engine type has to be type tested as per definition of engine type given in UR M32.

5.3 If an electronically controlled diesel engine has been type tested as a conventional engine the Society may waive tests required by this UR provided the results of the individual tests would be similar.

M50
(cont)



① = range of continuous operation

② = range of intermitted operation

③ = range of short-time overload operation

Figure 1 Power/Speed Diagram

End of Document

M51
(1987)
(Rev.1 1990)
(Corr. 1997)
(Rev.2 July 2003)
(Rev.3 Jan 2008)

Programme for trials of i.c. engines to assess operational capability

1. Works trials (acceptance test)

The Programme for trials has been written on the assumption that a Classification Society may require that after the tests the fuel delivery system will be blocked so as to limit the engines to run at not more than 100% power.

Engines, which are to be subjected to trials on the test bed at the manufacturer's works and under the Society's supervision according to the Rules Classification Societies, are to be tested in accordance with the scope as specified below. Exceptions to this require the agreement of the Society.

1.1 Scope of works trials

For all stages, the engine is going to be tested, the pertaining operation values are to be measured and recorded by the engine manufacturer. All results are to be compiled in an acceptance protocol to be issued by the engine manufacturer.

In each case all measurements conducted at the various load points shall be carried out at steady operating conditions. The readings for 100% power (rated power at rated speed) are to be taken twice at an interval of at least 30 minutes.

1.1.1 Main engines driving propellers

a) 100% power (rated power) at rated engine speed n_o:

at least 60 min – after having reached steady conditions.

b) 110% power at engine speed n – 1,032 n_o:

30-45 min – after having reached steady conditions.

NOTE:
After running on the test bed, the fuel delivery system of main engines is normally to be so adjusted that overload power cannot be given in service.

c) 90% (or normal continuous cruise power), 75%, 50% and 25% power in accordance with the nominal propeller curve.

d) Starting and reversing manoeuvres.

e) Testing of governor and independent overspeed protective device.

f) Shut down device.

Note:
1. The requirements in M51 Rev.3 are to be uniformly implemented by IACS Societies for engines; when an application for certification for an engine is dated on or after 1 January 2009.

2. The "date of application for certification of the engine" is the date of whatever document the Classification Society requires/accepts as an application or request for certification of an individual engine.

M51
(cont)

1.1.2 Main engines driving generators for propulsion

The test is to be performed at rated speed with a constant governor setting under conditions of:

a) 100% power (rated power) at rated engine speed:

at least 50 min – after having reached steady conditions.

b) 110% power:

30 min – after having reached steady conditions.

NOTE:
After running on the test bed, the fuel delivery system of diesel engines driving generators must be adjusted such that overload (110%) power can be given in service after installation on board, so that the governing characteristics including the activation of generator protective devices can be fulfilled at all times.

c) 75%, 50% and 25% power and idle run.

d) Start-up tests.

e) Testing of governor and independent overspeed protective device.

f) Shut-down device.

1.1.3 Engines driving auxiliaries

Test to be performed in accordance with 1.1.2.

NOTE:
After running on the test bed, the fuel delivery system of diesel engines driving generators must be adjusted such that overload (110%) power can be given in service after installation on board, so that the governing characteristics including the activation of generator protective devices can be fulfilled at all times.

1.2 Inspection of components

Random checks of components to be presented for inspection after the works trials are left to the discretion of each Society.

1.3 Parameters to be measured

The data to be measured and recorded, when testing the engine at various load points are to include all necessary parameters for the engine operation. The crankshaft deflection is to be checked when this check is required by the manufacturer during the operating life of the engine.

1.4 In addition the scope of the trials may be expanded depending on the engine application.

M51 (cont)

1.5 Integration tests: For electronically controlled diesel engines integration tests shall verify that the response of the complete mechanical, hydraulic and electronic system is as predicted for all intended operational modes. The scope of these tests shall be agreed with the Society for selected cases based on the FMEA required in UR M44.

2. Shipboard trials

After the conclusion of the running-in programme, prescribed by the engine manufacturer, engines are to undergo the trials as specified below:

2.1 Scope of trials

2.1.1 Main propulsion engines driving fixed propellers

a) At rated engine speed n_o: at least 4 hours
and
at engine speed corresponding
to normal continuous cruise power: at least 2 hours

b) At engine speed $n = 1{,}032\ n_o$: 30 minutes
(where the engine adjustment permits, see 1.1.1 b)

c) At minimum on-load speed

d) Starting and reversing manoeuvres

e) In reverse direction of propeller
rotation during the dock or sea trials
at a minimum engine speed of $n = 0{,}7\ n_o$: 10 minutes

f) Monitoring, alarm and safety systems.

2.1.2 Main propulsion engines driving controllable pitch propellers or reversing gears 2.1.1 applies as appropriate.

Controllable pitch propellers are to be tested with various propeller pitches.

2.1.3 Single main engines driving generators for propulsion

The tests to be performed at rated speed with a constant governor setting under conditions of:

a) 100% power (rated propulsion power): at least 4 hours
and
at normal continuous cruise propulsion power: at least 2 hours

b) 110% power (rated propulsion power): 30 minutes

c) In reverse direction of propeller rotation
at a minimum speed of 70% of the nominal propeller speed: 10 minutes

d) Starting manoeuvres

e) Monitoring, alarm and safety systems

M51
(cont)

NOTE:
Tests are to be based on the rated electrical powers of the electric propulsion motors.

2.1.4 Engines driving auxiliaries

Engines driving generators or important auxiliaries are to be subjected to an operational test for at least 4 hours. During the test, the set concerned is required to operate at its rated power for an extended period.

It is to be demonstrated that the engine is capable of supplying 100% of its rated power, and in the case of shipboard generating sets account shall be taken of the times needed to actuate the generator's overload protection system.

2.1.5 The suitability of engine burn residual or other special fuels is to be demonstrated, if machinery installation is arranged to burn such fuels.

2.2 In addition the scope of the trials may be expanded in consideration of the special operating conditions, such as towing, trawling etc.

End of Document

M52 Length of aft stern bush bearing

(1986)

M52.1 Oil lubricated bearings of white metal

1.1 The length of white metal lined bearings is to be not less than 2,0 times the rule diameter of the shaft in way of the bearing.

1.2 The length of the bearing may be less provided the normal bearing pressure is not more than 8 bar as determined by static bearing reaction calculation taking into account shaft and propeller weight which is deemed to be exerted solely on the aft bearing divided by the projected area of the shaft. However, the minimum length is to be not less than 1,5 times the actual diameter.

M52.2 Oil lubricated bearings of synthetic rubber, reinforced resin or plastic materials

2.1 For bearings of synthetic rubber, reinforced resin or plastics materials which are approved for use as oil lubricated stern bush bearings, the length of the bearing is to be not less than 2,0 times the rule diameter of the shaft in way of the bearing.

2.2 The length of bearing may be less provided the nominal bearing pressure is not more than 6 bar as determined by static bearing reaction calculation taking into account shaft and propeller weight which is deemed to be exerted solely on the aft bearing divided by the projected area of the shaft. However, the minimum length is to be not less than 1,5 times the actual diameter.

Where the material has proven satisfactory testing and operating experience, consideration may be given to an increased bearing pressure.

M52.3 Water lubricated bearings of lignum vitae

Where the bearing comprises staves of wood (known as lignum vitae), the length of the bearing is to be not less than 4,0 times the rule diameter of the shaft in way of the bearing.

NOTE:
Lignum vitae is the generic name for several dense, resinous hardwoods with good lubricating properties. The original high quality Lignum Vitae is almost unobtainable and other types of wood such as Bulnesia Sarmiento (or Palo Santo or Bulnesia Arabia) are commonly used now.

M52.4 Water lubricated bearings of synthetic material

4.1 Where the bearing is constructed of synthetic materials which are approved for use as water lubricated stern bush bearings such as rubber or plastics the length of the bearing is to be not less than 4,0 times the rule diameter of the shaft in way of the bearing.

4.2 For a bearing design substantiated by experiments to the satisfaction of the Society consideration may be given to a bearing length not less than 2,0 times the rule diameter of the shaft in way of the bearing.

End of Document

UR M 53 Calculation of Crankshafts for I.C. Engines

(1986)
(Rev.1, Dec 2004)

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M 53.1 GENERAL

1.1. Scope

These Rules for the design of crankshafts are to be applied to I.C. engines for propulsion and auxiliary purposes, where the engines are capable of continuous operation at their rated power when running at rated speed.

Where a crankshaft design involves the use of surface treated fillets, or when fatigue parameter influences are tested, or when working stresses are measured, the relevant documents with calculations/analysis are to be submitted to Classification Societies in order to demonstrate equivalence to the Rules.

1.2. Field of application

These Rules apply only to solid-forged and semi-built crankshafts of forged or cast steel, with one crankthrow between main bearings.

1.3. Principles of calculation

The design of crankshafts is based on an evaluation of safety against fatigue in the highly stressed areas.

The calculation is also based on the assumption that the areas exposed to highest stresses are :

- fillet transitions between the crankpin and web as well as between the journal and web,
- outlets of crankpin oil bores.

When journal diameter is equal or larger than the crankpin one, the outlets of main journal oil bores are to be formed in a similar way to the crankpin oil bores, otherwise separate documentation of fatigue safety may be required.

Calculation of crankshaft strength consists initially in determining the nominal alternating bending (see § M53.2.1) and nominal alternating torsional stresses (see § M53.2.2) which, multiplied by the appropriate stress concentration factors (see § M53.3), result in an equivalent alternating stress (uni-axial stress) (see § M53.5). This equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material (see § M53.6). This comparison will show whether or not the crankshaft concerned is dimensioned adequately (see § M53.7).

1.4. Drawings and particulars to be submitted

For the calculation of crankshafts, the documents and particulars listed below are to be submitted :

- crankshaft drawing
  (which must contain all data in respect of the geometrical configurations of the crankshaft)
- 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
  (2-stroke or 4-stroke cycle/direct injection, precombustion chamber, etc.)
- number of cylinders
- rated power [kW]
- rated engine speed [r/min]
- direction of rotation (see. fig. 1)
- firing order with the respective ignition intervals and, where necessary,
  V-angle α_V [°] (see fig. 1)

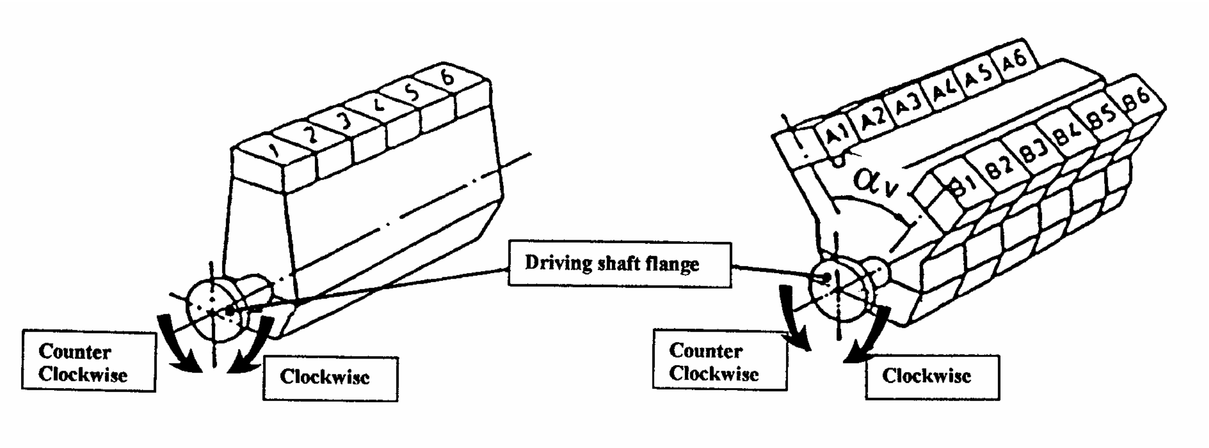


Fig. 1 – Designation of the cylinders

- cylinder diameter [mm]
- stroke [mm]
- maximum net cylinder pressure Pmax [bar]
- charge air pressure [bar]
  (before inlet valves or scavenge ports, whichever applies)
- connecting-rod length L_H [mm]
- all individual reciprocating masses acting on one crank [kg]
- digitized gas pressure curve presented at equidistant intervals [bar versus Crank Angle] (at least every 5° CA)
- for engines with articulated-type connecting-rod (see fig. 2)

  * distance to link point L_A [mm]
  * link angle α_N [°]
  - connecting-rod length L_N [mm]

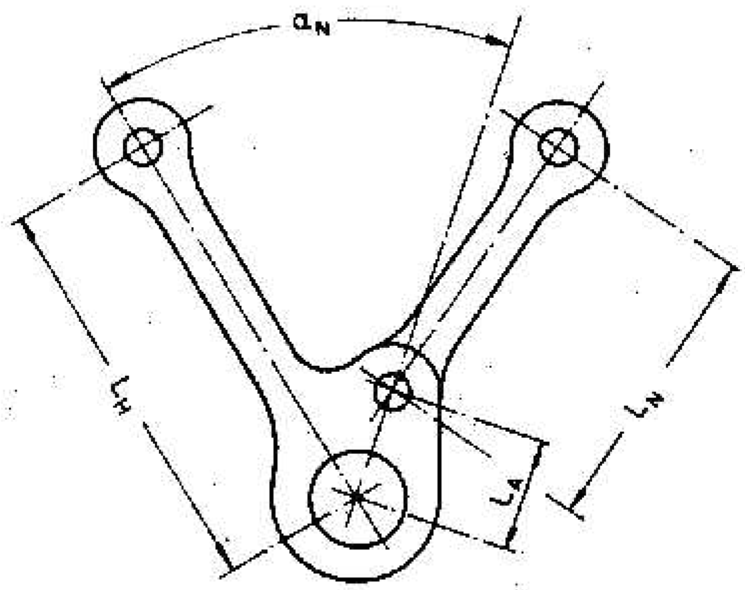


Fig. 2 – articulated-type connecting-rod

- details of crankshaft material

  * material designation
    (according to ISO,EN,DIN, AISI, etc..)

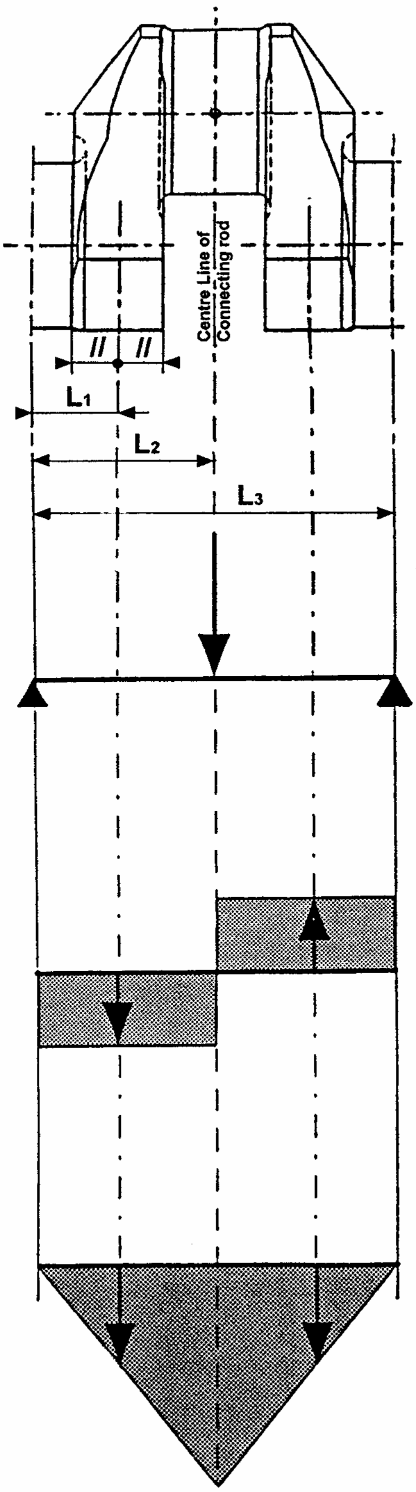
  * mechanical properties of material
    (minimum values obtained from longitudinal test specimens)

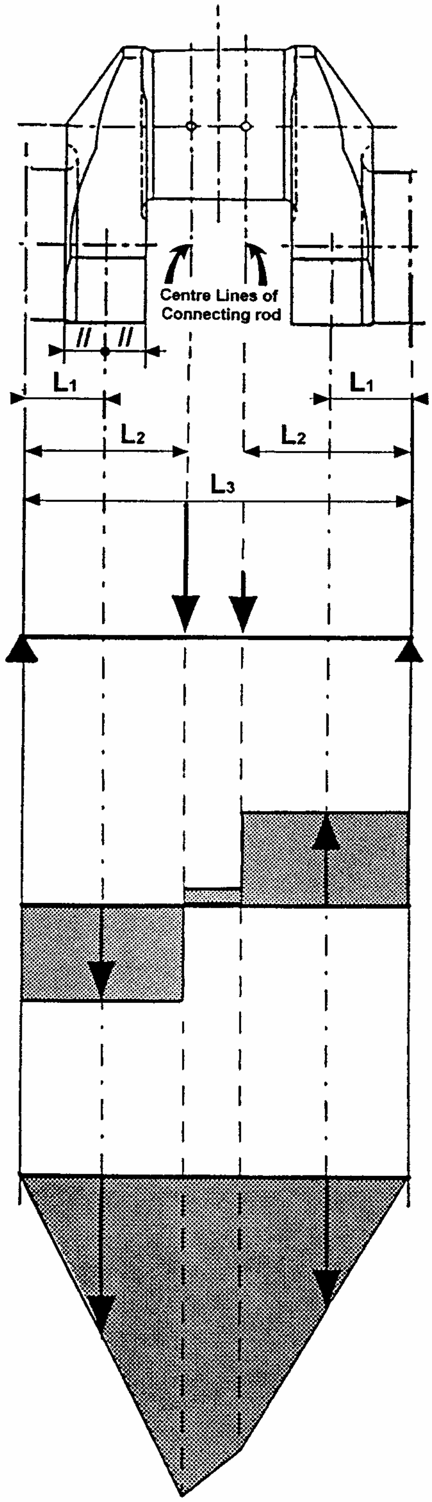
    - tensile strength [N/mm²]
    - yield strength [N/mm²]
    - reduction in area at break [%]
    - elongation A_5 [%]
    - impact energy – KV [J]

  * type of forging
    (free form forged, continuous grain flow forged, drop-forged, etc… ; with description of the forging process)

- Every surface treatment affecting fillets or oil holes shall be subject to special consideration

- Particulars of alternating torsional stress calculations, see item M 53.2.2.





Connecting-rod acting component forces (F_R or F_T)

Radial shear force diagrams (Q_R)

Bending moment diagrams (M_{BR} or M_{BT})

Fig. 3 Crankthrow for in line engine

Fig. 4 Crankthrow for Vee engine with 2 adjacent connecting-rods

L1 = Distance between main journal centre line and crankweb centre
(see also Fig 5 for crankshaft without overlap)
L2 = Distance between main journal centre line and connecting-rod centre
L3 = Distance between two adjacent main journal centre lines

M 53.2 CALCULATION OF STRESSES

2.1. Calculation of alternating stresses due to bending moments and radial forces

2.1.1 Assumptions

The calculation is based on a statically determined system, composed of a single crankthrow supported in the centre of adjacent main journals and subject to gas and inertia forces. The bending length is taken as the length between the two main bearing midpoints (distance L_3, see fig. 3 and 4).

The bending moments M_{BR}, M_{BT} are calculated in the relevant section based on triangular bending moment diagrams due to the radial component F_R and tangential component F_T of the connecting-rod force, respectively (see fig.3).

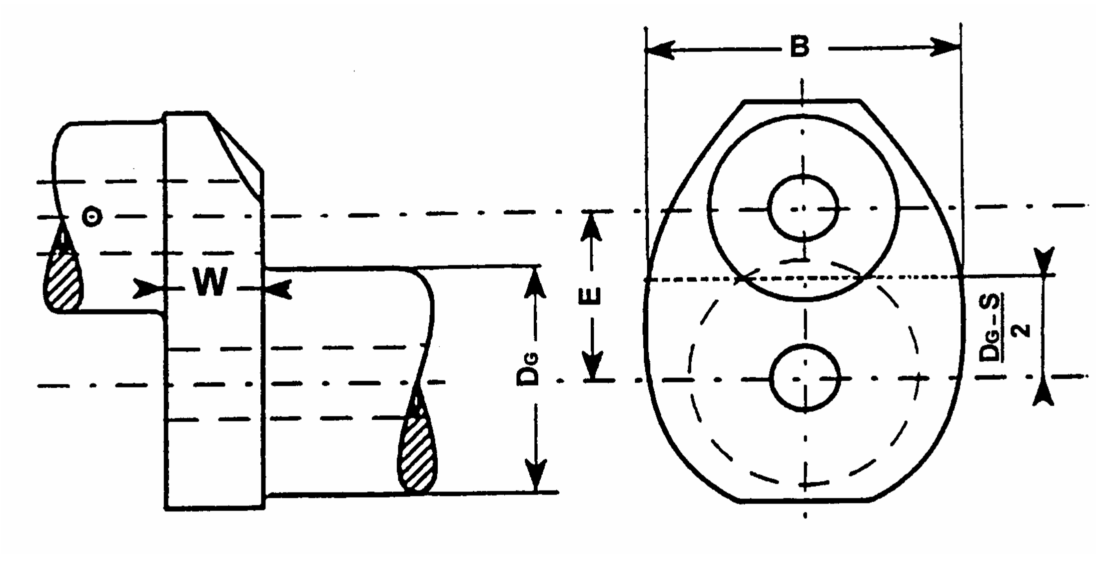
For crankthrows with two connecting-rods acting upon one crankpin the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams according to phase (see fig.4).

2.1.1.1 Bending moments and radial forces acting in web

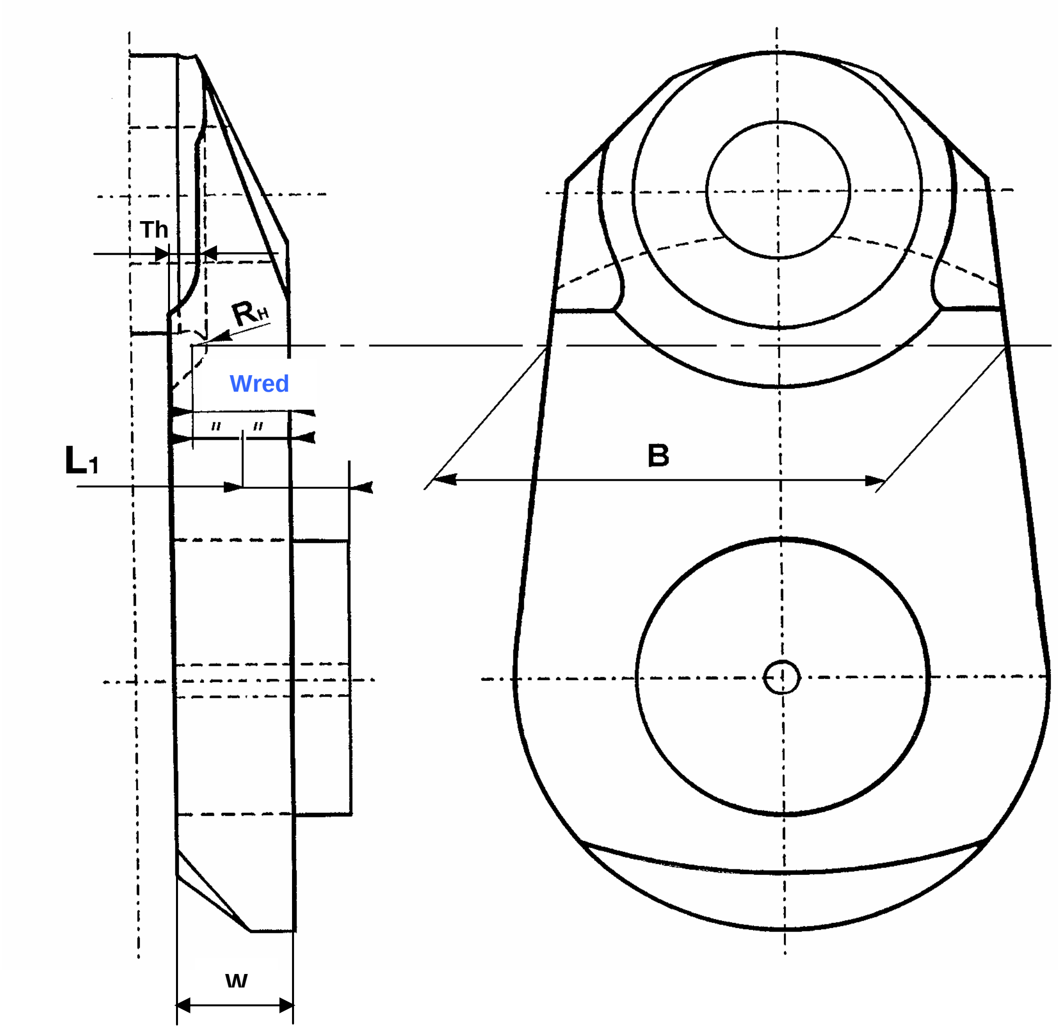
The bending moment M_{BRF} and the radial force Q_{RF} are taken as acting in the centre of the solid web (distance L_1) and are derived from the radial component of the connecting-rod force.

The alternating bending and compressive stresses due to bending moments and radial forces are to be related to the cross-section of the crank web. This reference section results from the web thickness W and the web width B (see fig. 5).

Mean stresses are neglected.



Overlapped crankshaft

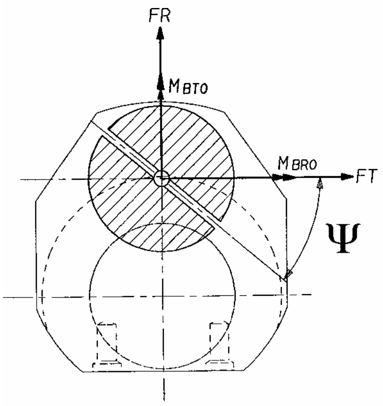


Crankshaft without overlap

Fig: 5 – Reference area of crankweb cross section

2.1.1.2 Bending acting in outlet of crankpin oil bore

The two relevant bending moments are taken in the crankpin cross-section through the oil bore.



M_{BRO} is the bending moment of the radial component of the connecting-rod force

M_{BTO} is the bending moment of the tangential component of the connecting-rod force

Fig. 6 : Crankpin section through the oil bore

The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin.

Mean bending stresses are neglected.

2.1.2 Calculation of nominal alternating bending and compressive stresses in web

The radial and tangential forces due to gas and inertia loads acting upon the crankpin at each connecting-rod position will be calculated over one working cycle.

Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments M_{BRF}, M_{BRO} , M_{BTO} and radial forces Q_{RF} - as defined in M53 2.1.1.1 and 2.1.1.2. - will then be calculated.

In case of V-type engines, the bending moments - progressively calculated from the gas and inertia forces - of the two cylinders acting on one crankthrow are superposed according to phase. Different designs (forked connecting-rod, articulated-type connecting-rod or adjacent connecting-rods) shall be taken into account.

Where there are cranks of different geometrical configurations in one crankshaft, the calculation is to cover all crank variants.
The decisive alternating values will then be calculated according to :

$$X_N = \pm\frac{1}{2}\left[X_{max} - X_{min}\right]$$

where :

X_N is considered as alternating force, moment or stress
X_{max} is maximum value within one working cycle
X_{min} is minimum value within one working cycle

2.1.2.1 Nominal alternating bending and compressive stresses in web cross section

The calculation of the nominal alternating bending and compressive stresses is as follows :

$$\sigma_{BFN} = \pm \frac{M_{BRFN}}{W_{eqw}} \bullet 10^3 \bullet Ke$$

$$\sigma_{QFN} = \pm \frac{Q_{RFN}}{F} \bullet Ke$$

Where :

σ_{BFN} [N/mm²] nominal alternating bending stress related to the web

M_{BRFN} [Nm] alternating bending moment related to the center of the web (see fig. 3 and 4)

$$M_{BRFN} = \pm \frac{1}{2}\left[M_{BRF\,max} - M_{BRF\,min}\right]$$

W_{eqw} [mm³] section modulus related to cross-section of web

$$W_{eqw} = \frac{B \bullet W^2}{6}$$

Ke empirical factor considering to some extent the influence of adjacent crank and bearing restraint
with : Ke = 0.8 for 2-stroke engines
Ke = 1.0 for 4-stroke engines

σ_{QFN} [N/mm²] nominal alternating compressive stress due to radial force related to the web

Q_{RFN} [N] alternating radial force related to the web (see fig. 3 and 4)

$$Q_{RFN} = \pm \frac{1}{2}\left[Q_{RF_{max}} - Q_{RF_{min}}\right]$$

F [mm²] area related to cross-section of web

$F = B \bullet W$

2.1.2.2 Nominal alternating bending stress in outlet of crankpin oil bore

The calculation of nominal alternating bending stress is as follows :

$$\sigma_{BON} = \pm \frac{M_{BON}}{W_e} \bullet 10^3$$

where :

σ_{BON}	[N/mm²]	nominal alternating bending stress related to the crank pin diameter
M_{BON}	[Nm]	alternating bending moment calculated at the outlet of crankpin oil bore

$$M_{BON} = \pm \frac{1}{2}\left[M_{BO_{max}} - M_{BO_{min}}\right]$$

with $M_{BO} = \left(M_{BTO} \bullet \cos\psi + M_{BRO} \bullet \sin\psi\right)$

and ψ $[°]$ angular position (see fig. 6)

We	[mm³]	section modulus related to cross-section of axially bored crankpin

$$W_e = \frac{\pi}{32}\left[\frac{D^4 - D_{BH}^4}{D}\right]$$

2.1.3. Calculation of alternating bending stresses in fillets

The calculation of stresses is to be carried out for the crankpin fillet as well as for the journal fillet.

For the crankpin fillet :

$$\sigma_{BH} = \pm\left(\alpha_B \bullet \sigma_{BFN}\right)$$

where :

σ_{BH}	$\left[N/mm^2\right]$	alternating bending stress in crankpin fillet
α_B	$[-]$	stress concentration factor for bending in crankpin fillet (determination - see item M53.3)

For the journal fillet (not applicable to semi-built crankshaft) :

$$\sigma_{BG} = \pm(\beta_B \bullet \sigma_{BFN} + \beta_Q \bullet \sigma_{QFN})$$

where :

σ_{BG}	$[N/mm^2]$	alternating bending stress in journal fillet
β_B	$[-]$	stress concentration factor for bending in journal fillet (determination - see item M53.3)
β_Q	$[-]$	stress concentration factor for compression due to radial force in journal fillet (determination - see item M53.3)

2.1.4 Calculation of alternating bending stresses in outlet of crankpin oil bore

$$\sigma_{BO} = \pm(\gamma_B \bullet \sigma_{BON})$$

where :

σ_{BO}	$[N/mm^2]$	alternating bending stress in outlet of crankpin oil bore
γ_B	$[-]$	stress concentration factor for bending in crankpin oil bore (determination - see item M53.3)

2.2. Calculation of alternating torsional stresses

2.2.1 General

The calculation for nominal alternating torsional stresses is to be undertaken by the engine manufacturer according to the information contained in item M 53.2.2.2.

The manufacturer shall specify the maximum nominal alternating torsional stress.

2.2.2 Calculation of nominal alternating torsional stresses

The maximum and minimum torques are to be ascertained for every mass point of the complete dynamic system 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 15th order for 2-stroke cycle engines and from the 0.5th order up to and including the 12th order for 4-stroke cycle engines. Whilst doing so, allowance must be made for the damping that exists in the system and for unfavourable conditions (misfiring [*] in one of the cylinders). The speed step calculation shall be selected in such a way that any resonance found in the operational speed range of the engine shall be detected.

Where barred speed ranges are necessary, they shall be arranged so that satisfactory operation is possible despite their existence. There are to be no barred speed ranges above a speed ratio of $\lambda \geq 0.8$ for normal firing conditions.

The values received from such calculation are to be submitted to Classification Society.

The nominal alternating torsional stress in every mass point, which is essential to the assessment, results from the following equation :

$$\tau_N = \pm \frac{M_{TN}}{W_P} \bullet 10^3$$

$$M_{TN} = \pm \frac{1}{2}\left[M_{T\,max} - M_{T\,min}\right]$$

$$W_p = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}{}^4}{D}\right) \quad \text{or} \quad W_p = \frac{\pi}{16}\left(\frac{D_G{}^4 - D_{BG}{}^4}{D_G}\right)$$

where :

τ_N	[N/mm²]	nominal alternating torsional stress referred to crankpin or journal
M_{TN}	[Nm]	maximum alternating torque
W_P	[mm^3]	polar section modulus related to cross-section of axially bored crankpin or bored journal
M_{Tmax}	[Nm]	maximum value of the torque
M_{Tmin}	[Nm]	minimum value of the torque

*) Misfiring is defined as cylinder condition when no combustion occurs but only compression cycle.

For the purpose of the crankshaft assessment, the nominal alternating torsional stress considered in further calculations is the highest calculated value, according to above method, occurring at the most torsionally loaded mass point of the crankshaft system.
Where barred speed ranges exist, the torsional stresses within these ranges are not to be considered for assessment calculations.

The approval of crankshaft will be based on the installation having the largest nominal alternating torsional stress (but not exceeding the maximum figure specified by engine manufacturer).

Thus, for each installation, it is to be ensured by suitable calculation that this approved nominal alternating torsional stress is not exceeded. This calculation is to be submitted for assessment.

2.2.3. Calculation of alternating torsional stresses in fillets and outlet of crankpin oil bore

The calculation of stresses is to be carried out for the crankpin fillet, the journal fillet and the outlet of the crankpin oil bore.

For the crankpin fillet :

$$\tau_H = \pm(\alpha_T \bullet \tau_N)$$

where :

τ_H	$[N/mm^2]$	alternating torsional stress in crankpin fillet
α_T	$[-]$	stress concentration factor for torsion in crankpin fillet (determination - see item M53.3)
τ_N	$[N/mm^2]$	nominal alternating torsional stress related to crankpin diameter

For the journal fillet (not applicable to semi-built crankshafts)

$$\tau_{G} = \pm(\beta_{T} \bullet \tau_{N})$$

where :

τ_G	$[N / mm^2]$	alternating torsional stress in journal fillet
β_T	$[-]$	stress concentration factor for torsion in journal fillet (determination - see item M53.3)
τ_N	$[N / mm^2]$	nominal alternating torsional stress related to journal diameter

For the outlet of crankpin oil bore

$$\sigma_{TO} = \pm(\gamma_{T} \bullet \tau_{N})$$

where :

σ_{TO}	$[N / mm^2]$	alternating stress in outlet of crankpin oil bore due to torsion
γ_T	$[-]$	stress concentration factor for torsion in outlet of crankpin oil bore (determination- see item M53.3)
τ_N	$[N / mm^2]$	nominal alternating torsional stress related to crankpin diameter

M 53.3 EVALUATION OF STRESS CONCENTRATION FACTORS

3.1. General

The stress concentration factors are evaluated by means of the formulae according to items M53.3.2, M53.3.3 and M53.3.4 applicable to the fillets and crankpin oil bore of solid forged web-type crankshafts and to the crankpin fillets of semi-built crankshafts only. It must be noticed that stress concentration factor formulae concerning the oil bore are only applicable to a radially drilled oil hole. All formulae are based on investigations of FVV (Forschungsvereinigung Verbrennungskraftmaschinen) for fillets and on investigations of ESDU (Engineering Science Data Unit) for oil holes.
All crank dimensions necessary for the calculation of stress concentration factors are shown in figure 7

The stress concentration factor for bending (α_B, β_B) is defined as the ratio of the maximum equivalent stress (VON MISES) – occurring in the fillets under bending load – to the nominal bending stress related to the web cross-section (see Appendix I).

The stress concentration factor for compression (β_Q) in the journal fillet is defined as the ratio of the maximum equivalent stress (VON MISES) – occurring in the fillet due to the radial force – to the nominal compressive stress related to the web cross-section.

The stress concentration factor for torsion (α_T, β_T) is defined as the ratio of the maximum equivalent shear stress – occurring in the fillets under torsional load – to the nominal torsional stress related to the axially bored crankpin or journal cross-section (see Appendix I).

The stress concentration factors for bending (γ_B) and torsion (γ_T) are defined as the ratio of the maximum principal stress – occurring at the outlet of the crankpin oil-hole under bending and torsional loads – to the corresponding nominal stress related to the axially bored crankpin cross section (see Appendix II).

When reliable measurements and/or calculations are available, which can allow direct assessment of stress concentration factors, the relevant documents and their analysis method have to be submitted to Classification Societies in order to demonstrate their equivalence to present rules evaluation.

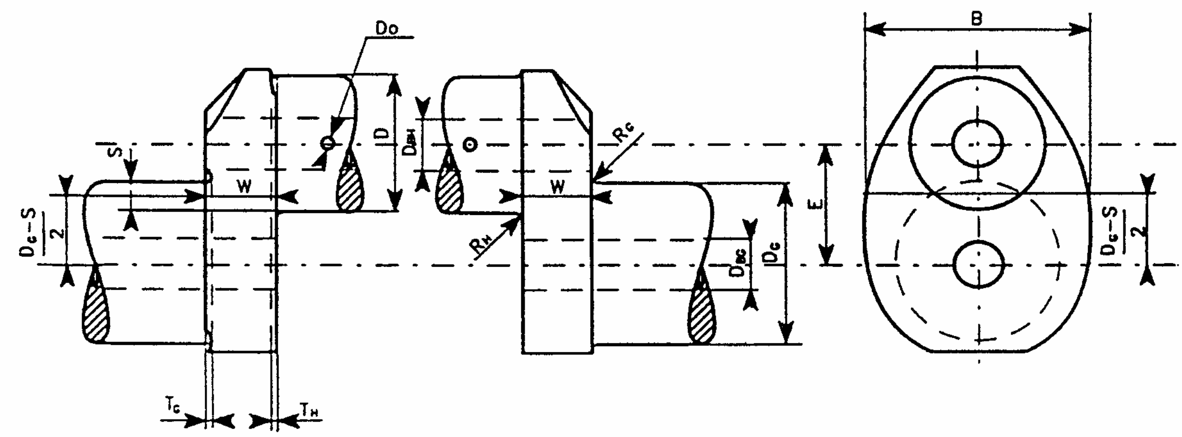


Fig. 7-Crank dimensions

Actual dimensions :

D	[mm]	crankpin diameter
D_{BH}	[mm]	diameter of axial bore in crankpin
D_o	[mm]	diameter of oil bore in crankpin
R_H	[mm]	fillet radius of crankpin
T_H	[mm]	recess of crankpin fillet
D_G	[mm]	journal diameter
D_{BG}	[mm]	diameter of axial bore in journal
R_G	[mm]	fillet radius of journal
T_G	[mm]	recess of journal fillet
E	[mm]	pin eccentricity
S	[mm]	pin overlap $S = \frac{D + D_G}{2} - E$
W (*)	[mm]	web thickness
B (*)	[mm]	web width

(*) In the case of 2 stroke semi-built crankshafts:

- when $T_H > R_H$, the web thickness must be considered as equal to :

  $W_{red} = W - (T_H - R_H)$ [refer to fig. 5]

- web width B must be taken in way of crankpin fillet radius centre according to fig. 5

The following related dimensions will be applied for the calculation of stress concentration factors in :

Crankpin fillet	Journal fillet
$r = R_H / D$	$r = R_G / D$

s	=	S/D
w	=	W/D crankshafts with overlap
		W_{red}/D crankshafts without overlap
b	=	B/D
d_o	=	D_O/D
d_G	=	D_{BG}/D
d_H	=	D_{BH}/D
t_H	=	T_H/D
t_G	=	T_G/D

Stress concentration factors are valid for the ranges of related dimensions for which the investigations have been carried out. Ranges are as follows :

$$
\begin{array}{lllll}
 & & s & \le 0.5 \\
0.2 & \le & w & \le 0.8 \\
1.1 & \le & b & \le 2.2 \\
0.03 & \le & r & \le 0.13 \\
0 & \le & d_G & \le 0.8 \\
0 & \le & d_H & \le 0.8 \\
0 & \le & d_O & \le 0.2
\end{array}
$$

Low range of s can be extended down to large negative values provided that :

- If calculated f (recess) < 1 then the factor f (recess) is not to be considered (f (recess) = 1)
- If s < - 0.5 then f (s,w) and f (r,s) are to be evaluated replacing actual value of s by - 0.5.

3.2. Crankpin fillet

The stress concentration factor for bending (α_B) is :

$$\boldsymbol{\alpha_B = 2.6914 \bullet f(s, w) \bullet f(w) \bullet f(b) \bullet f(r) \bullet f(d_G) \bullet f(d_H) \bullet f(\text{recess})}$$

where :

$$f(s,w) = -4.1883 + 29.2004 \bullet w - 77.5925 \bullet w^2 + 91.9454 \bullet w^3 - 40.0416 \bullet w^4 + (1-s) \bullet (9.5440 - 58.3480 \bullet w + 159.3415 \bullet w^2 - 192.5846 \bullet w^3 + 85.2916 \bullet w^4) + (1-s)^2 \bullet (-3.8399 + 25.0444 \bullet w - 70.5571 \bullet w^2 + 87.0328 \bullet w^3 - 39.1832 \bullet w^4)$$

$$f(w) = 2.1790 \bullet w^{0.7171}$$

$$f(b) = 0.6840 - 0.0077 \bullet b + 0.1473 \bullet b^2$$

$$f(r) = 0.2081 \bullet r^{(-0.5231)}$$

$$f(d_G) = 0.9993 + 0.27 \bullet d_G - 1.0211 \bullet d_G^2 + 0.5306 \bullet d_G^3$$

$$f(d_H) = 0.9978 + 0.3145 \bullet d_H - 1.5241 \bullet d_H^2 + 2.4147 \bullet d_H^3$$

$$f(\text{recess}) = 1 + (t_H + t_G) \bullet (1.8 + 3.2 \bullet s)$$

The stress concentration factor for torsion (α_T) is :

$$\alpha_T = 0.8 \bullet f(r,s) \bullet f(b) \bullet f(w)$$

where :

$f \quad (r,s) \quad = \quad r^{(-0.322 + 0.1015 \bullet (1-s))}$

$f \quad (b) \quad = \quad 7.8955 - 10.654 \bullet b + 5.3482 \bullet b^2 - 0.857 \bullet b^3$

$f \quad (w) \quad = \quad w^{(-0.145)}$

3.3. Journal fillet (not applicable to semi-built crankshaft)

The stress concentration factor for bending (β_B) is :

$$\beta_B = 2.7146 \bullet f_B(s,w) \bullet f_B(w) \bullet f_B(b) \bullet f_B(r) \bullet f_B(d_G) \bullet f_B(d_H) \bullet f(\text{recess})$$

where **:**

$f_B \quad (s,w) \quad = \quad -1.7625 + 2.9821 \bullet w - 1.5276 \bullet w^2 + (1 - s) \bullet (5.1169 - 5.8089 \bullet w + 3.1391 \bullet w^2) + (1 - s)^2 \bullet (-2.1567 + 2.3297 \bullet w - 1.2952 \bullet w^2)$

$f_B \quad (w) \quad = \quad 2.2422 \bullet w^{0.7548}$

$f_B \quad (b) \quad = \quad 0.5616 + 0.1197 \bullet b + 0.1176 \bullet b^2$

$f_B \quad (r) \quad = \quad 0.1908 \bullet r^{(-- 0.5568)}$

$f_B \quad (d_G) \quad = \quad 1.0012 - 0.6441 \bullet d_G + 1.2265 \bullet d_G^2$

$f_B \quad (d_H) \quad = \quad 1.0022 - 0.1903 \bullet d_H + 0.0073 \bullet d_H^2$

$f \quad (\text{recess}) \quad = \quad 1 + (t_H + t_G) \bullet (1.8 + 3.2 \bullet s)$

The stress concentration factor for compression (β_Q) due to the radial force is :

$$\beta_Q = 3.0128 \bullet f_Q(s) \bullet f_Q(w) \bullet f_Q(b) \bullet f_Q(r) \bullet f_Q(d_H) \bullet f(\text{recess})$$

where :

$f_Q \quad (s) \quad = \quad 0.4368 + 2.1630 \bullet (1-s) - 1.5212 \bullet (1-s)^2$

$f_Q \quad (w) \quad = \quad \dfrac{w}{0.0637 + 0.9369 \bullet w}$

$f_Q \quad (b) \quad = \quad -0.5 + b$

$f_Q \quad (r) \quad = \quad 0.5331 \bullet r^{(-0.2038)}$

$f_Q \quad (d_H) \quad = \quad 0.9937 - 1.1949 \bullet d_H + 1.7373 \bullet d_H^2$

$f \quad (\text{recess}) = \quad 1 + (t_H + t_G) \bullet (1.8 + 3.2 \bullet s)$

The stress concentration factor for torsion (β_T) is :

$$\beta_T = \alpha_T$$

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

If crankpin and journal diameters and/or radii are of different sizes

$$\beta_T = 0.8 \bullet f(r,s) \bullet f(b) \bullet f(w)$$

Where :

f (r,s), f (b) and f (w) are to be determined in accordance with item M 53.3.2. (see calculation of α_T), however, the radius of the journal fillet is to be related to the journal diameter :

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

3.4. Outlet of crankpin oil bore

The stress concentration factor for bending (γ_B) is :

$$\gamma_B = 3 - 5.88 \bullet d_O + 34.6 \bullet d_O^2$$

The stress concentration factor for torsion (γ_T) is :

$$\gamma_T = 4 - 6 \bullet d_O + 30 \bullet d_O^2$$

M 53.4 ADDITIONAL BENDING STRESSES

In addition to the alternating bending stresses in fillets (see item M 53.2.1.3) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying σ_{add} as given by table :

Type of engine	σ_{add}[N/mm²]
Crosshead engines	± 30 (*)
Trunk piston engines	± 10

(*) The additional stress of ± 30 N/mm² is composed of two components

1) **an additional stress of ± 20 N/mm² resulting from axial vibration**
2) **an additional stress of ± 10 N/mm² resulting from misalignment / bedplate deformation**

It is recommended that a value of ± 20 N/mm^2 be used for the axial vibration component for assessment purposes 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.

M 53.5 CALCULATION OF EQUIVALENT ALTERNATING STRESS

5.1. General

In the fillets, bending and torsion lead to two different biaxial stress fields which can be represented by a Von Mises equivalent stress with the additional assumptions that bending and torsion stresses are time phased and the corresponding peak values occur at the same location (see Appendix I).
As a result the equivalent alternating stress is to be calculated for the crankpin fillet as well as for the journal fillet by using the Von Mises criterion.

At the oil hole outlet, bending and torsion lead to two different stress fields which can be represented by an equivalent principal stress equal to the maximum of principal stress resulting from combination of these two stress fields with the assumption that bending and torsion are time phased (see Appendix II).

The above two different ways of equivalent stress evaluation both lead to stresses which may be compared to the same fatigue strength value of crankshaft assessed according to Von Mises criterion.

5.2. Equivalent alternating stress

The equivalent alternating stress is calculated in accordance with the formulae given.

For the crankpin fillet :

$$\sigma_V = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3 \bullet \tau_H{}^2}$$

For the journal fillet :

$$\sigma_V = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3 \bullet \tau_G{}^2}$$

For the outlet of crankpin oil bore :

$$\sigma_V = \pm\frac{1}{3}\sigma_{BO} \bullet \left[1 + 2\sqrt{1 + \frac{9}{4}\left(\frac{\sigma_{TO}}{\sigma_{BO}}\right)^2}\right]$$

where :

σ_v $\left[N / mm^2\right]$ equivalent alternating stress

for other parameters see items M53.2.1.3., M53.2.2.3. and M53.4.

M 53.6 CALCULATION OF FATIGUE STRENGTH

The fatigue strength is to be understood as that value of equivalent alternating stress (Von Mises) which a crankshaft can permanently withstand at the most highly stressed points. The fatigue strength may be evaluated by means of the following formulae.

Related to the crankpin diameter :

$$\sigma_{DW} = \pm K \bullet (0.42 \bullet \sigma_B + 39.3\ ) \bullet \left[0.264 + 1.073 \bullet D^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \bullet \sqrt{\frac{1}{R_X}}\ \right]$$

with :

$R_X = R_H$ in the fillet area

$R_X = Do / 2$ in the oil bore area

Related to the journal diameter :

$$\sigma_{DW} = \pm K \bullet (0.42 \bullet \sigma_B + 39.3\ ) \bullet \left[0.264 + 1.073 \bullet D_G^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \bullet \sqrt{\frac{1}{R_G}}\ \right]$$

where :

σ_{DW} $[N / mm^2]$ allowable fatigue strength of crankshaft

K **[-]** **factor for different types of crankshafts without surface treatment. Values greater than 1 are only applicable to fatigue strength in fillet area.**
= 1.05 for continuous grain flow forged or drop-forged crankshafts
= 1.0 for free form forged crankshafts (without continuous grain flow)
factor for cast steel crankshafts with cold rolling treatment in fillet area
= 0.93 for cast steel crankshafts manufactured by companies using a classification society approved cold rolling process

σ_B $[N / mm^2]$ minimum tensile strength of crankshaft material

For other parameters see item M53.3.3

When a surface treatment process is applied, it must be approved by Classification Society.

These formulae are subject to the following conditions :

- surfaces of the fillet, the outlet of the oil bore and inside the oil bore (down to a minimum depth equal to 1.5 times the oil bore diameter) shall be smoothly finished.
- for calculation purposes R_H, R_G or R_X are to be taken as not less than 2 mm.

As an alternative the fatigue strength of the crankshaft can be determined by experiment based either on full size crankthrow (or crankshaft) or on specimens taken from a full size crankthrow.

In any case the experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment have to be submitted for approval to Classification Society (method, type of specimens, number of specimens (or crankthrows), number of tests, survival probability, confidence number,….)

M 53.7 ACCEPTABILITY CRITERIA

The sufficient dimensioning of a crankshaft is confirmed by a comparison of the equivalent alternating stress and the fatigue strength. This comparison has to be carried out for the crankpin fillet, the journal fillet, the outlet of crankpin oil bore and is based on the formula :

$$\mathbf{Q} = \frac{\sigma_{DW}}{\sigma_V}$$

where :

Q [–] acceptability factor

Adequate dimensioning of the crankshaft is ensured if the smallest of all acceptability factors satisfies the criteria :

$$\mathbf{Q \geq 1.15}$$

M 53.8 CALCULATION OF SHRINK-FITS OF SEMI-BUILT CRANKSHAFT

8.1. General

All crank dimensions necessary for the calculation of the shrink-fit are shown in figure 8.

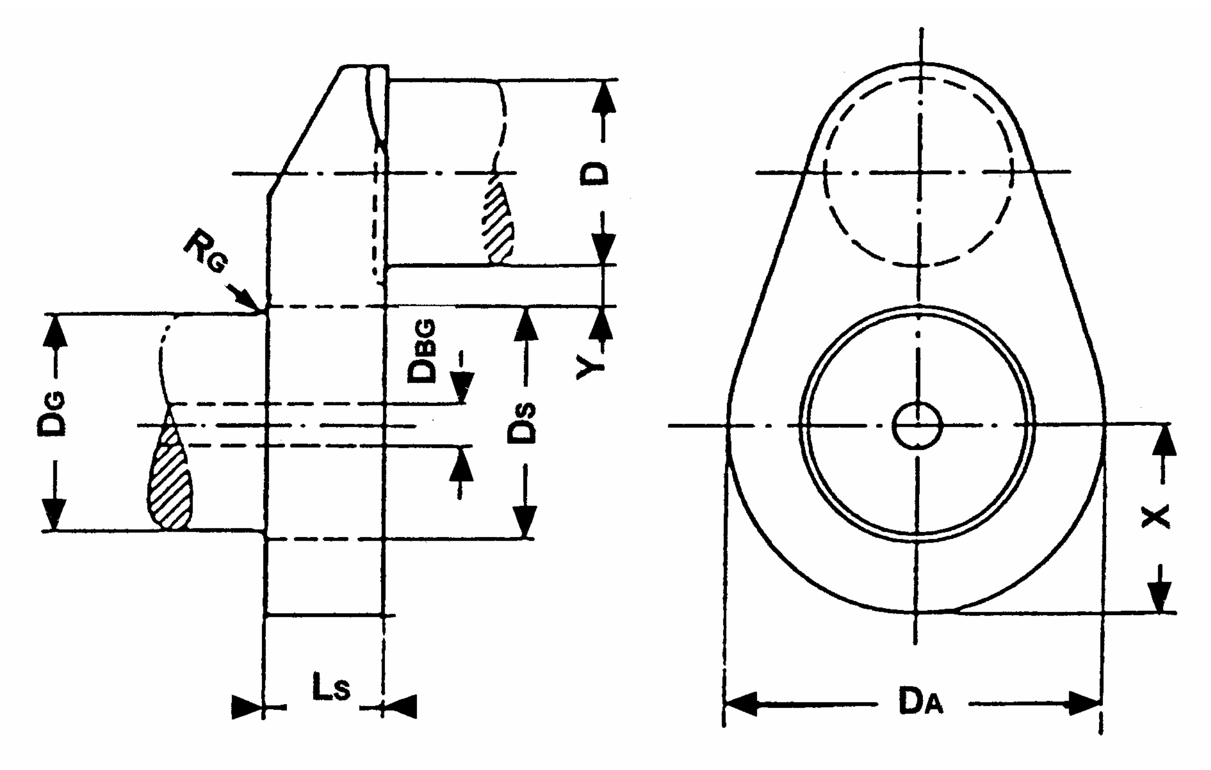


Fig. 8 – Crankthrow of semi-built crankshaft

Where :

D_A [mm] outside diameter of web
or
twice the minimum distance x between centre-line of journals and outer contour of web, whichever is less

D_S [mm] shrink diameter

D_G [mm] journal diameter

D_{BG} [mm] diameter of axial bore in journal

L_S [mm] length of shrink-fit

R_G [mm] fillet radius of journal

y [mm] distance between the adjacent generating lines of journal and pin $y \geq 0.05 \bullet D_S$

Where y is less than $0.1 \bullet D_S$ special consideration is to be given to the effect of the stress due to the shrink-fit on the fatigue strength at the crankpin fillet.

Respecting the radius of the transition from the journal to the shrink diameter, the following should be complied with :

$$R_G \geq 0.015 \bullet D_G$$

and

$$R_G \geq 0.5 \bullet (D_S - D_G)$$

where the greater value is to be considered.

The actual oversize Z of the shrink-fit must be within the limits Z_{min} and Z_{max} calculated in accordance with items M53.8.3 and 8.4.

In the case where 8.2 condition cannot be fulfilled then 8.3 and 8.4 calculation methods of Z_{min} and Z_{max} are not applicable due to multizone-plasticity problems.

In such case Z_{min} and Z_{max} have to be established based on FEM calculations.

8.2. Maximum permissible hole in the journal pin

The maximum permissible hole diameter in the journal pin is calculated in accordance with the following formula :

$$\mathbf{D_{BG}} = \mathbf{D_S} \bullet \sqrt{\mathbf{1} - \frac{\mathbf{4000 \bullet S_R \bullet M_{max}}}{\mu \bullet \pi \bullet \mathbf{D_S^2} \bullet \mathbf{L_S} \bullet \sigma_{\mathbf{SP}}}}$$

where :

S_R [-] safety factor against slipping, however a value not less than 2 is to be taken unless documented by experiments.

M_{max} [Nm] absolute maximum value of the torque M_{Tmax} in accordance with M 53 2.2.2

μ [-] coefficient for static friction, however a value not greater than 0.2 is to be taken unless documented by experiments.

σ_{SP} [N/mm2] minimum yield strength of material for journal pin

This condition serves to avoid plasticity in the hole of the journal pin.

8.3. Necessary minimum oversize of shrink-fit

The necessary minimum oversize is determined by the greater value calculated according to :

$$Z_{min} \geq \frac{\sigma_{SW} \bullet D_S}{E_m}$$

and

$$Z_{min} \geq \frac{4000}{\mu \bullet \pi} \bullet \frac{S_R \bullet M_{max}}{E_m \bullet D_S \bullet L_S} \bullet \frac{1 - Q_A^2 \bullet Q_S^2}{\left(1 - Q_A^2\right) \bullet \left(1 - Q_S^2\right)}$$

where :

Z_{min} [mm] minimum oversize

E_m [N/mm²] Young's modulus

σ_{SW} [N/mm²] minimum yield strength of material for crank web

Q_A [-] web ratio, $Q_A = \frac{D_S}{D_A}$

Q_S [-] shaft ratio, $Q_S = \frac{D_{BG}}{D_S}$

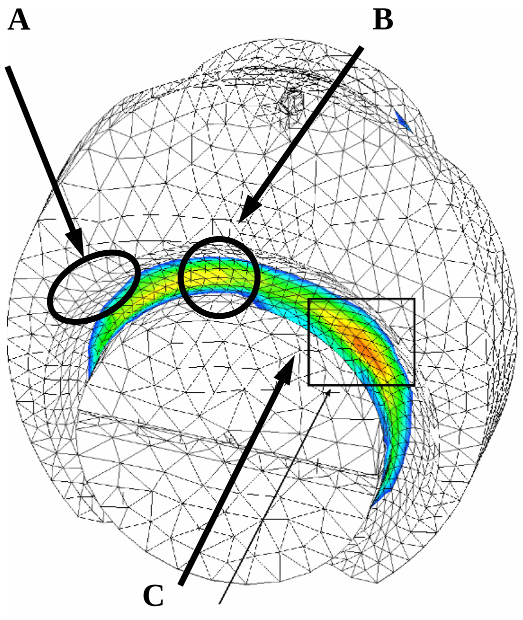
8.4. Maximum permissible oversize of shrink-fit

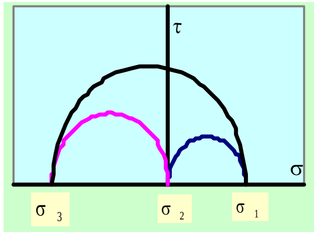
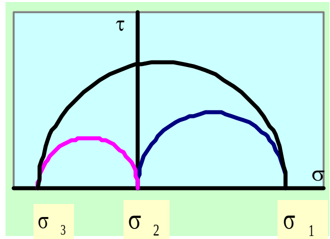
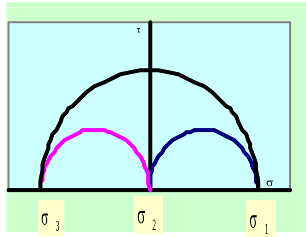
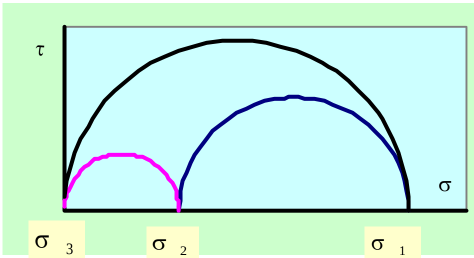
The maximum permissible oversize is calculated according to :

$$Z_{max} \leq D_S \bullet \left(\frac{\sigma_{SW}}{E_m} + \frac{0.8}{1000} \right)$$

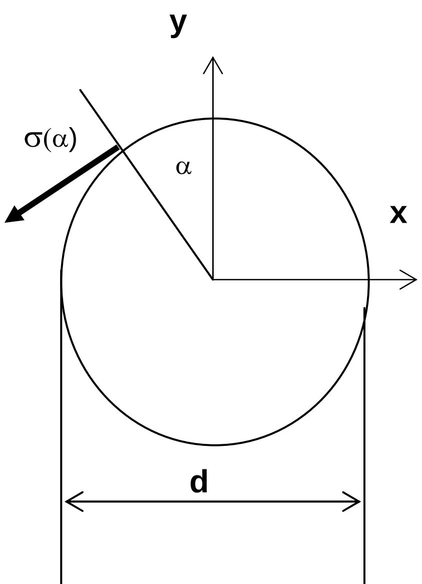
This condition serves to restrict the shrinkage induced mean stress in the fillet.

Definition of Stress Concentration Factors in crankshaft fillets

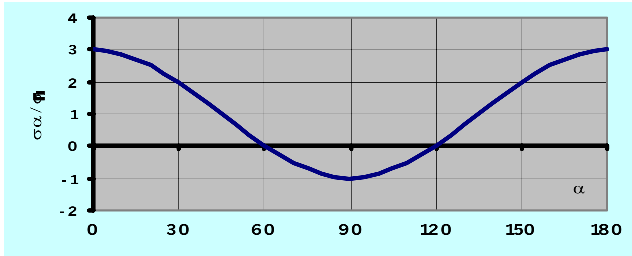
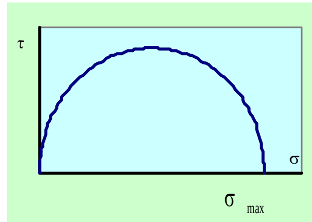
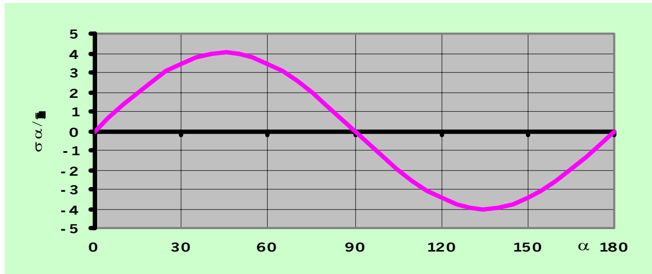
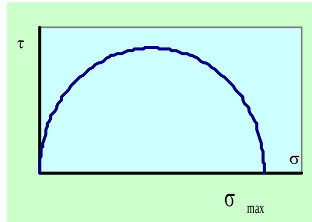
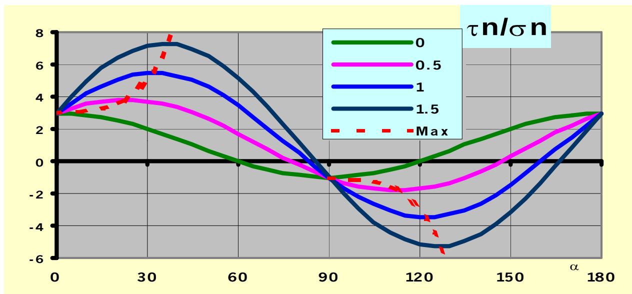
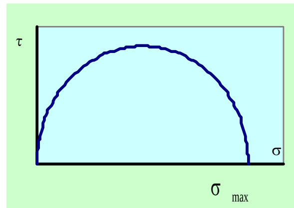


	Stress	Max $\Vert\sigma_3\Vert$	Max σ_1	
Torsional loading	Location of maximal stresses	**A**	**C**	**B**
	Typical principal stress system Mohr's circle diagram with $\sigma_2 = 0$	 $\Vert\sigma_3\Vert > \sigma_1$	 $\sigma_1 > \Vert\sigma_3\Vert$	 $\sigma_1 \approx \Vert\sigma_3\Vert$
	Equivalent stress and **S.C.F.**	$\tau_{equiv} = \frac{\sigma_1 - \sigma_3}{2}$ $\text{S.C.F.} = \frac{\tau_{equiv}}{\tau_n}$ for α_T, β_T		
Bending loading	Location of maximal stresses	**B**	**B**	**B**
	Typical principal stress system Mohr's circle diagram with $\sigma_3 = 0$			$\sigma 2 \neq 0$
	Equivalent stress and **S.C.F.**	$\sigma\,\text{equiv} = \sqrt{\sigma_1{}^2 + \sigma_2{}^2 - \sigma_1 . \sigma_2}$ $\text{S.C.F.} = \frac{\sigma\,\text{equiv}}{\sigma_n}$ for $\alpha_B, \beta_B, \beta_Q$		

Stress Concentration Factors and stress distribution at the edge of oil drillings.



d = hole diameter

Stress type	Nominal stress tensor	Uniaxial stress distribution around the edge	Mohr's circle diagram
Tension	$\begin{bmatrix} \sigma n & 0 \\ 0 & 0 \end{bmatrix}$	$\sigma\alpha = \sigma n\ \gamma_B\ /\ 3\ [1+2\cos(2\ \alpha)]$ 	 $\gamma_B = \sigma_{max}/\sigma_n$ for $\alpha = k\pi$
Shear	$\begin{bmatrix} 0 & \tau n \\ \tau n & 0 \end{bmatrix}$	$\sigma\alpha = \gamma_T\ \tau n \sin(2\alpha)$ 	 $\gamma_T = \sigma_{max}/\tau_n$ for $\alpha = \frac{\pi}{4} + k\frac{\pi}{2}$
Tension + shear	$\begin{bmatrix} \sigma n & \tau n \\ \tau n & 0 \end{bmatrix}$	$\sigma_\alpha = \frac{\gamma_B}{3}\sigma_n\left\{1+2\left[\cos(2\alpha)+\frac{3}{2}\frac{\gamma_T}{\gamma_B}\frac{\tau_n}{\sigma_n}\sin(2\alpha)\right]\right\}$ 	 $\sigma_{max} = \frac{\gamma_B}{3}\sigma_n\left[1+2\sqrt{1+\frac{9}{4}\left(\frac{\gamma_T}{\gamma_B}\frac{\tau_n}{\sigma_n}\right)^2}\right]$ for $\alpha = \frac{1}{2}\mathrm{tg}^{-1}\left(\frac{3\gamma_T\tau_n}{2\gamma_B\sigma_n}\right)$

M54 Deleted

(1986)
(Rev 1
1997)

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M55 ~~Planned maintenance scheme (PMS) for machinery~~

(1988)

Deleted in May 2001

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M56 Marine gears – load capacity of involute parallel axis spur and helical gears

(1990)
(Rev.1
1994)/
Corr.
1996

M56.1 Basic principles – introduction and general influence factors

M56 1.1 Introduction

The following definitions are mainly based on the ISO 6336 standard (hereinafter called "reference standard") for the calculation of load capacity of spur and helical gears.

M56.1.2 Scope and field of application

The following definitions apply to enclosed gears, both intended for main propulsion and for essential auxiliary services, which accumulate a large number of load cycles (several millions), as required by the Rules of the Society.

The following definitions deal with the determination of load capacity of external and internal involute spur and helical gears, having parallel axis, with regard to surface durability (pitting) and tooth root bending strength and to this purpose the relevant basic equations are provided in Parts 2 and 3.

The influence factors common to said equations are described in the present Part 1.

The others, introduced in connection with each basic equation, are described in the following Parts 2 and 3.

All influence factors are defined regarding their physical interpretation. Some of the influence factors are determined by the gear geometry or have been established by conventions. These factors are to be calculated in accordance with the equations provided. Other factors, which are approximations, can be calculated according to methods acceptable to the Society.

M56.1.3 Symbols and units

The main symbols used are listed below.

Other symbols introduced in connection with the definition of influence factors are described in the appropriate sections.

M56
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SI units have been adopted.

a	centre distance	mm
b	common facewidth	mm
bl,2	facewidth of pinion, wheel	mm
d	reference diameter	mm
dl,2	reference diameter of pinion, wheel	mm
dal,2	tip diameter of pinion, wheel	mm
dbl,2	base diameter of pinion, wheel	mm
dfl,2	root diameter of pinion, wheel	mm
dwl	working diameter of pinion, wheel	mm
Ft	nominal tangential load	N
Fbt	nominal tangential load on base cylinder in the transverse section	N
h	tooth depth	mm
mn	normal module	mm
mt	transverse module	mm
nl,2	rotational speed of pinion, wheel	revs/min
P	maximum continuous power transmitted by the gear set	kW
Tl,2	torque in way of pinion, wheel	Nm
u	gear ratio	
v	linear speed at pitch diameter	m/s
xl,2	addendum modification coefficient of pinion, wheel	
z	number of teeth	
zl,2	number of teeth of pinion, wheel	
zn	virtual number of teeth	
αn	normal pressure angle at reference cylinder	o
αt	transverse pressure angle at ref. cylinder	o
αtw	transverse pressure angle at working pitch cylinder	o
β	helix angle at reference	o
βb	helix angle at base cylinder	o
εα	transverse contact ratio	
εβ	overlap ratio	
εγ	total contact ratio	

M56.1.4 Geometrical definitions

For internal gearing z2, a, d2, da2, db2 and dw2 are negative. The pinion is defined as the gear with the smaller number of teeth, therefore the absolute value of the gear ratio, defined as follows, is always greater or equal to the unity:

$u = z2/z1 \quad = dw2/dw1 \quad = d2/d1$

For external gears u is positive, for internal gears u is negative.

In the equation of surface durability b is the common facewidth on the pitch diameter.

In the equation of tooth root bending stress b1 or b2 are the facewidths at the respective tooth roots. In any case, b1 and b2 are not to be taken as greater than b by more than one module (mn) on either side.

The common facewidth b may be used also in the equation of teeth root bending stress if significant crowning or end relief have been adopted.

$$\tan \alpha t = \tan \alpha n / \cos \beta$$
$$\tan \beta b = \tan \beta \cos \alpha t$$
$$d = z\, mn / \cos \beta$$
$$db = d \cos \alpha t = dw \cos \alpha tw$$
$$a = 0.5\,(dwl + dw2)$$
$$zn = z / (\cos^2 \beta b \cos \beta)$$
$$mt = mn / \cos \beta$$
$$\mathrm{inv}\, \alpha = tg\alpha - \pi\alpha / 180;\ \alpha(^\circ)$$

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$$\text{inv}\,\alpha tw = \text{inv}\,\alpha t + 2\,\text{tg}\alpha n\,(x1 + x2)/(zl + z2)$$

$$\varepsilon\alpha = \frac{0{,}5\sqrt{da1^2 - db1^2} \pm 0{,}5\sqrt{da2^2 - db2^2} - a\sin\alpha tw}{\pi mn\cos\alpha t/\cos\beta}$$

the positive sign is used for external gears, the negative sign for internal gears

$$\varepsilon\beta = b\sin\beta/\pi\ (\text{mm})$$

for double helix, b is to be taken as the width of one helix

$$\varepsilon\gamma = \varepsilon\alpha + \varepsilon\beta$$

$$v = d1{,}2\quad nl,2/19099$$

M56.1.5 Nominal tangential load, Ft

The nominal tangential load, Ft, tangential to the reference cylinder and perpendicular to the relevant axial plane, is calculated directly from the maximum continuous power transmitted by the gear set by means of the following equations:

$$T1{,}2 = 9549\,P/nl{,}2$$

$$Ft = 2000\,T1{,}2/d1{,}2$$

M56.1.6 General influence factors

M56.1.6.1 Application factor, KA[1]

The application factor, KA, accounts for dynamic overloads from sources external to the gearing.

KA, for gears designed for infinite life is defined as the ratio between the maximum repetitive cyclic torque applied to the gear set and the nominal rated torque.

The nominal rated torque is defined by the rated power and speed and is the torque used in the rating calculations.

The factor mainly depends on:

– characteristics of driving and driven machines;
– ratio of masses;
– type of couplings;
– operating conditions (overspeeds, changes in propeller load conditions, ...).

When operating near a critical speed of the drive system, a careful analysis of conditions must be made.

The application factor, KA, should be determined by measurements or by system analysis acceptable to the Society. Where a value determined in such a way cannot be supplied, the following values can be considered :

a) Main propulsion

- diesel engine with hydraulic or electromagnetic slip coupling : 1.00

- diesel engine with high elasticity coupling : 1.30

1 Where the vessel, on which the reduction gear is being used, is receiving an Ice Class notation, the Application Factor or the Nominal Tangential Force should be adjusted to reflect the ice load associated with the requested Ice Class.

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- diesel engine with other couplings : 1.50

b) Auxiliary gears

- electric motor, diesel engine with hydraulic or electromagnetic slip coupling : 1.00
- diesel engine with high elasticity coupling : 1.20
- diesel engine with other couplings : 1.40

M56.1.6.2 Load sharing factor, $K\gamma$

The load sharing factor, $K\gamma$ accounts for the maldistribution of load in multiple path transmissions (dual tandem, epicyclic, double helix, etc.)

$K\gamma$ is defined as the ratio between the maximum load through an actual path and the evenly shared load. The factor mainly depends on accuracy and flexibility of the branches.

The load sharing factor, $K\gamma$, should be determined by measurements or by system analysis. Where a value determined in such a way cannot be supplied, the following values can be considered for epicyclic gears :

- up to 3 planetary gears : 1.00
- 4 planetary gears : 1.20
- 5 planetary gears : 1.30
- 6 planetary gears and over : 1.40

M56.1.6.3 Dynamic factor, Kv

The dynamic factor, Kv, accounts for internally generated dynamic loads due to vibrations of pinion and wheel against each other.

Kv is defined as the ratio between the maximum load which dynamically acts on the tooth flanks and the maximum externally applied load (Ft KA $K\gamma$).
The factor mainly depends on:
– transmission errors (depending on pitch and profile errors);
– masses of pinion and wheel;
– gear mesh stiffness variation as the gear teeth pass through the meshing cycle;
– transmitted load including application factor;
– pitch line velocity;
– dynamic unbalance of gears and shaft;
– shaft and bearing stiffnesses;
– damping characteristics of the gear system.

The dynamic factor, Kv, can be calculated as follows:

The method may be applied only to cases where all the following conditions are satisfied:

a) steel gears of heavy rims sections
b) $Ft/b > 150$ N/mm
c) $z1 < 50$

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d) running speed in the subcritical range:

- for helical gears $(vz1)/100<14$
- for spur gears $(vz1)/100<10$

This method may be applied to all types of gears if

$$(vz1)/100<3$$

For gears other than the above, reference can be made to method B outlined in the reference standard.

For helical gears of overlap ratio > unity Kv is obtained from F ig. 1.1.

For spur gears Kv is obtained from Fig. 1.2.

For helical gears of overlap ratio < unity Kv is obtained by means of linear interpolation between the values obtained from Fig. 1.1 and 1.2 :

$$Kv = Kv2 - \varepsilon\beta\,(Kv2 - Kv1)$$

Where :

Kv1 is the Kv value for helical gears, given by Fig. 1.1

Kv2 is the Kv value for spur gears, given by Fig. 1.2

Kv can also be determined as follows :

$$Kv = 1 + K1\,(vz1)/100$$

K1 values are specified in the following Table 1.1

	K1 ISO GRADES OF ACCURACY 2					
	3	**4**	**5**	**6**	**7**	**8**
Spur gears	0.022	0.030	0.043	0.062	0.092	0.125
Helical gears	0.0125	0.0165	0.0230	0.0330	0.0480	0.0700

Table 1.1 Values of the factor K1 for the calculation of Kv

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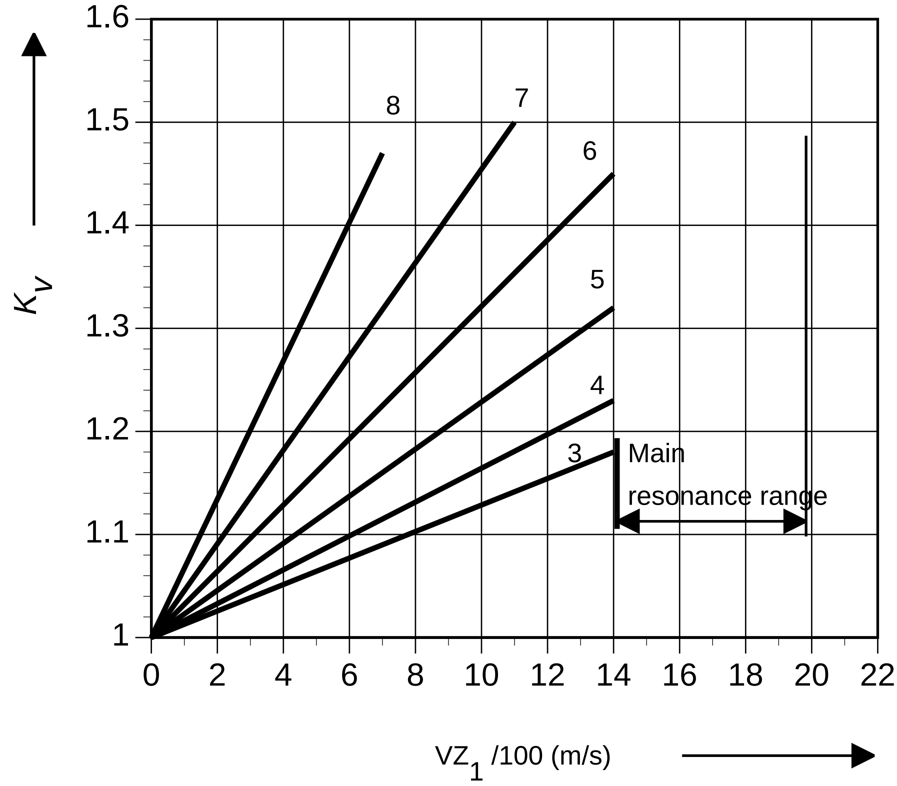

Fig. 1.1

Dynamic factor for helical gear. ISO grades of accuracy 3 - 8 [2]

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[2] ISO grades of accuracy according to ISO 1328. In case of mating gears with different grades of accuracy the grade corresponding to the lower accuracy should be used.

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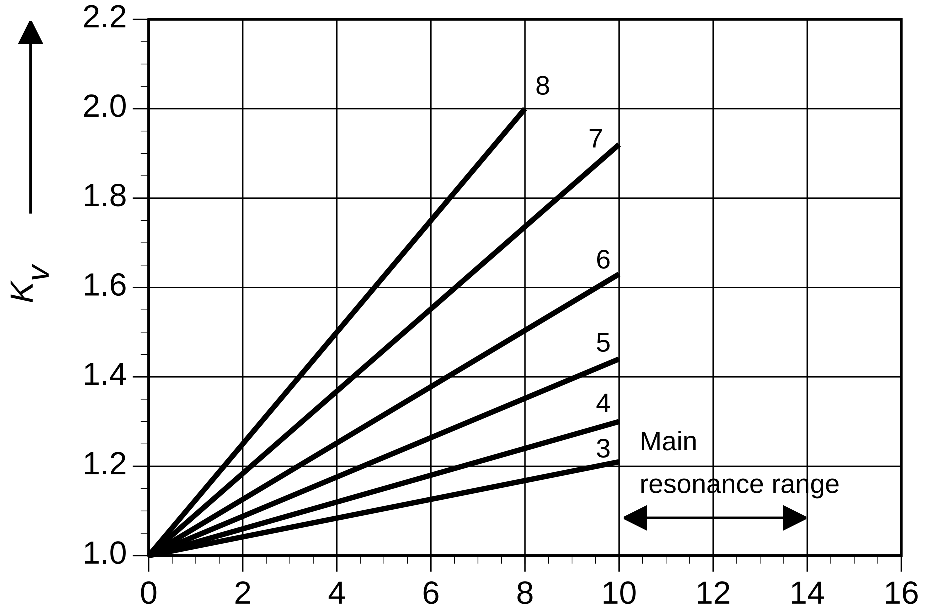

Y- Z_1 /100 (m/s)

Fig. 1.2 dynamic factor for spur gear. ISO grades of accuracy 3 - 8 [2]

M56.1.6.4 Face load distribution factors, KHβ and KFβ

The face load distribution factors, KHβ for contact stress, KFβ for tooth root bending stress, account for the effects of non-uniform distribution of load across the facewidth.

KHβ is defined as follows:

$$KH\beta = \frac{\text{maximum load per unit facewidth}}{\text{mean load per unit facewidth}}$$

KFβ is defined as follows:

$$KF\beta = \frac{\text{maximum bending stress at tooth root per unit facewidth}}{\text{mean bending stress at tooth root per unit facewidth}}$$

The mean bending stress at tooth root relates to the considered facewidth bl resp. b2.
KFβ can be expressed as a function of the factor KHβ.
The factors KHβ and KFβ mainly depend on:

- gear tooth manufacturing accuracy;
- errors in mounting due to bore errors;
- bearing clearances;
- wheel and pinion shaft alignment errors;
- elastic deflections of gear elements, shafts, bearings, housing and foundations which support the gear elements;
- thermal expansion and distortion due to operating temperature;
- compensating design elements (tooth crowning, end relief, etc.).

The face load distribution factors, KHβ for contact stress, and KFβ for tooth root bending stress, can be determined according to the method C2 outlined in the ISO 6336/1 standard.

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Alternative methods acceptable to the Society may be applied.

a) In case the hardest contact is at the end of the facewidth $K_{F\beta}$ is given by the following equations:

$$K_{F\beta} = K_{H\beta}^{N}$$

$$N = \frac{(b/h)^2}{1 + (b/h) + (b/h)^2}$$

(b/h) = facewidth/tooth height ratio, the minimum of b1/h1 or b2/h2. For double helical gears, the facewidth of only one helix is to be used.

b) In case of gears where the ends of the facewidth are lightly loaded or unloaded (end relief or crowning):

$$K_{F\beta} = K_{H\beta}$$

M56.1.6.5 Transverse load distribution factors, $K_{H\alpha}$ and $K_{F\alpha}$

The transverse load distribution factors, $K_{H\alpha}$ for contact stress and $K_{F\alpha}$ for tooth root bending stress, account for the effects of pitch and profile errors on the transversal load distribution between two or more pairs of teeth in mesh.
The factors $K_{H\alpha}$ and $K_{F\alpha}$ mainly depend on:
– total mesh stiffness;
– total tangential load F_t, K_A, K_γ, K_v, $K_{H\beta}$;
– base pitch error;
– tip relief;
– running-in allowances.

The transverse load distribution factors, $K_{H\alpha}$ for contact stress and $K_{F\alpha}$ for tooth root bending stress, can be determined according to method B outlined in the reference standard.

M56.2 Surface durability (pitting)

M56.2.1 Scope and general remarks
The criterion for surface durability is based on the Hertz pressure on the operating pitch point or at the inner point of single pair contact. The contact stress σ_H must be equal to or less than the permissible contact stress σ_{HP}.

M56.2.2 Basic equations

M56.2.2.1 Contact stress

$$\sigma_H = \sigma_{HO}\sqrt{K_A\, K_\gamma K_v\, K_{H\alpha}\, K_{H\beta}} \;\leq\; \sigma_{HP}$$

where:

σ_{HO} = basic value of contact stress for pinion and wheel

$$\sigma_{HO} = Z_B Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}} \qquad \text{for pinion}$$

$$\sigma_{HO} = Z_D Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}} \qquad \text{for wheel}$$

where:
Z_B = single pair mesh factor for pinion (*see* clause 2.3)
Z_D = single pair mesh factor for wheel (*see* clause 2.3)
Z_H = zone factor (*see* clause 2.4)

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Z_E	=	elasticity factor	(*see* clause 2.5)
Z_ε	=	contact ratio factor	(*see* clause 2.6)
Z_β	=	helix angle factor	(*see* clause 2.7)
F_t	=	nominal tangential load at reference cylinder in the transverse section	(*see* Part 1).
b	=	common facewidth	
d_1	=	reference diameter of pinion	
u	=	gear ratio	

(for external gears u is positive, for internal gears u is negative)

Regarding factors K_A, K_γ, K_v, $K_{H\alpha}$ and $K_{H\beta}$, *see* Part 1.

M56.2.2.2 Permissible contact stress

The permissible contact stress σ_{HP} is to be evaluated separately for pinion and wheel:

$$\sigma_{HP} = (\sigma_{H1im}\, Z_N / S_H) \times Z_L\, Z_v\, Z_R\, Z_W\, Z_X$$

where:

σ_{Hlim}	=	endurance limit for contact stress	(*see* clause 2.8)
Z_N	=	life factor for contact stress	(*see* clause 2.9)
Z_L	=	lubrication factor	(*see* clause 2.10)
Z_v	=	speed factor	(*see* clause 2.10)
Z_R	=	roughness factor	(*see* clause 2.10)
Z_W	=	hardness ratio factor	(*see* clause 2.11)
Z_X	=	size factor for contact stress	(*see* clause 2.12)
S_H	=	safety factor for contact stress	(see clause 2.13)

M56.2.3 Single pair mesh factors, Z_B and Z_D

The single pair mesh factors, Z_B for pinion and Z_D for wheel, account for the influence on contact stresses of the tooth flank curvature at the inner point of single pair contact in relation to Z_H.

The factors transform the contact stresses determined at the pitch point to contact stresses considering the flank curvature at the inner point of single pair contact.

The single pair mesh factors, Z_B for pinions and Z_D for wheels, can be determined as follows:

For spur gears, $\varepsilon_\beta = 0$

$Z_B = M_1$ or 1 whichever is the larger value

$Z_D = M_2$ or 1 whichever is the larger value

$$M_1 = \frac{\tan\alpha_{tw}}{\sqrt{\left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - \left(\frac{2\pi}{z_1}\right)\right]\left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\left(\frac{2\pi}{z_2}\right)\right]}}$$

$$M_2 = \frac{\tan\alpha_{tw}}{\sqrt{\left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - \left(\frac{2\pi}{z_2}\right)\right]\left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\left(\frac{2\pi}{z_1}\right)\right]}}$$

For helical gears when $\varepsilon_\beta \geq 1$

$$Z_B = Z_D = 1$$

For helical gears when $\varepsilon_\beta < 1$ the values of Z_B, Z_D are determined by linear interpolation between Z_B, Z_D for spur gears and Z_B, Z_D for helical gears having $\varepsilon_\beta \geq 1$.

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Thus: $ZB = M1 - \varepsilon\beta\ (M1 - 1)$ and $ZB \geq 1$

$ZD = M2 - \varepsilon\beta\ (M2 - 1)$ and $ZD \geq 1$

M56.2.4 Zone factor, ZH

The zone factor, ZH, accounts for the influence on the Hertzian pressure of tooth flank curvature at pitch point and relates the tangential force at the reference cylinder to the normal force at the pitch cylinder.

The zone factor, ZH, can be calculated as follows :

$$ZH = \sqrt{\frac{2\cos\beta b\ \cos\alpha tw}{\cos^2\alpha t\ \sin\alpha tw}}$$

M56.2.5 Elasticity factor, ZE

The elasticity factor, ZE, accounts for the influence of the material properties E (modulus of elasticity) and ν (Poisson's ratio) on the Hertz pressure.

The elasticity factor, ZE, for steel gears (E = 206000 N/mm^2, ν = 0.3) is equal to:

$$ZE = 189.8\ (N^{1/2}/mm)$$

In other cases, reference can be made to the reference standard.

M56.2.6 Contact ratio factor, Zε

The contact ratio factor, Zε, accounts for the influence of the transverse contact ratio and the overlap ratio on the specific surface load of gears.

The contact ratio factor, Zε, can be calculated as follows:

Spur gears :

$$Z\varepsilon = \sqrt{\frac{4 - \varepsilon\alpha}{3}}$$

Helical gears :

- for $\varepsilon\beta < 1$

$$Z\varepsilon = \sqrt{\frac{4 - \varepsilon\alpha}{3}(1 - \varepsilon\beta) + \frac{\varepsilon\beta}{\varepsilon\alpha}}$$

- for $\varepsilon\beta \geq 1$

$$Z\varepsilon = \sqrt{\frac{1}{\varepsilon\alpha}}$$

M56.2.7 Helix angle factor, Zβ

The helix angle factor, Zβ, accounts for the influence of helix angle on surface durability, allowing for such variables as the distribution of load along the lines of contact. Zβ is dependent only on the helix angle.

The helix angle factor, Zβ, can be calculated as follows :

$$Z\beta = \sqrt{\cos\beta}$$

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Where β is the reference helix angle.

M56.2.8 Endurance limit for contact stress, σHlim

For a given material, σHlim is the limit of repeated contact stress which can be permanently endured. The value of σHlim can be regarded as the level of contact stress which the material will endure without pitting for at least 50 x 10^6 load cycles.

For this purpose, pitting is defined by:
– for not surface hardened gears:
  pitted area > 2% of total active flank area
  for surface hardened gears:
  pitted area > 0,5% of total active flank area, or
  > 4% of one particular tooth flank area.

The σHlim values are to correspond to a failure probability of 1% or less.

The endurance limit mainly depends on:

– material composition, cleanliness and defects;
– mechanical properties;
– residual stresses;
– hardening process, depth of hardened zone, hardness gradient;
– material structure (forged, rolled bar, cast).

The endurance limit for contact stress σHlim, can be determined, in general, making reference to values indicated in ISO 6336/5, quality MQ.

M56.2.9 Life factor, ZN

The life factor, ZN, accounts for the higher permissible contact stress in case a limited life (number of cycles) is required.

The factor mainly depends on:

– material and hardening;
– number of cycles;
– influence factors (ZR, Zv, ZL, ZW, ZX).

The life factor, ZN, can be determined according to method B outlined in the ISO 6336/2 standard.

M56.2.10 Influence factors on lubrication film, ZL, Zv and ZR

The lubricant factor, ZL, accounts for the influence of the type of lubricant and its viscosity, the speed factor, Zv, accounts for the influence of the pitch line velocity and the roughness factor, ZR, accounts for the influence of the surface roughness on the surface endurance capacity.

The factors may be determined for the softer material where gear pairs are of different hardness.
The factors mainly depend on:

– viscosity of lubricant in the contact zone;
_ the sum of the instantaneous velocities of the tooth surfaces;
_ load;
_ relative radius of curvature at the pitch point;
_ surface roughnesses of teeth flanks;
_ hardness of pinion and gear.

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The lubricant factor, ZL, the speed factor, ZV, and the roughness factor ZR can be calculated as follows :

a) Lubricant factor, ZL

The factor, ZL, can be calculated from the following equation:

$$ZL = CZL + \frac{4(1.0 - CZL)}{(1.2 + 134 / v40)^2}$$

In the range 850 N/mm²

$$\text{CZL} = \left(\frac{\sigma \text{H lim} - 850}{350} 0.08 \right) + 0.83$$

If σHlim < 850 N/mm², take CZL = 0.83
If σHlim > 1200 N/mm², take CZL = 0.91

Where :

υ40 = nominal kinematic viscosity of the oil at 40°C, mm²/s

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b) Speed factor, Zv

The speed factor, Zv, can be calculated from the following equations:

$$Zv = CZv + \frac{2(1.0 - CZv)}{\sqrt{0.8 + 32/v}}$$

In the range 850 N/mm² ≤ σHlim ≤ 1200 N/mm², CZv can be calculated as follows :

$$CZv = \left(\frac{\sigma H\lim - 850}{350} 0.08\right) + 0.85$$

c) Roughness factor, ZR

The roughness factor, ZR, can be calculated from the following equations:

$$ZR = \left(\frac{3}{R_{Z10}}\right)^{CZR}$$

Where :

$$R_Z = \frac{R_{Z1} + R_{Z2}}{2}$$

The peak-to-valley roughness determined for the pinion R_{Z1} and for the wheel R_{Z2} are mean values for the peak-to-valley roughness R_Z measured on several tooth flanks (R_Z as defined in the reference standard).

$$R_{Z10} = R_Z \sqrt[3]{\frac{10}{\rho_{red}}}$$

relative radius of curvature:

$$\rho_{red} = \frac{\rho_1 \cdot \rho_2}{\rho_1 + \rho_2}$$

Wherein:

$$\rho_{1,2} = 0.5 \cdot d_{b1,2} \cdot \tan\alpha tw$$

(also for internal gears, d_b negative sign)

If the roughness stated is an Ra value (= CLA value) (= AA value) the following approximate relationship can be applied :

$$Ra = CLA = AA = R_Z/6$$

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In the range 850 N/mm² ≤ σHlim ≤ 1200 N/mm², CZR can be calculated as follows :

$$CZR = 0.32 - 0.0002\ \sigma Hlim$$

If σHlim < 850 N/mm², take CZR = 0.150
If σHlim > 1200 N/mm², take CZR = 0.080

M56.2.11 Hardness ratio factor, ZW

The hardness ratio factor, ZW, accounts for the increase of surface durability of a soft steel gear meshing with a significantly harder gear with a smooth surface.

ZW apply to the soft gear only.

The factor mainly depends on:
– hardness of the soft gear;
– alloying elements of the soft gear;
– tooth flank roughness of the harder gear.

The hardness ratio factor, ZW, can be calculated as follows :

$$ZW = 1.2 - \frac{HB - 130}{1700}$$

Where:

HB = Brinell hardness of the softer material

For HB < 130, ZW = 1.2 will be used.

For HB > 470, ZW = 1.0 will be used.

M56.2.12 Size factor, ZX

The size factor, ZX, accounts for the influence of tooth dimensions on permissible contact stress and reflects the non-uniformity of material properties.

The factor mainly depends on:

– material and heat treatment;
– tooth and gear dimensions;
– ratio of case depth to tooth size.
– ratio of case depth to equivalent radius of curvature.

For through-hardened gears and for surface-hardened gears with adequate casedepth relative to tooth size and radius of relative curvature ZX = 1. When the casedepth is relatively shallow then a smaller value of ZX should be chosen.

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M56.2.13 Safety factor for contact stress, SH

The safety factor for contact stress, SH, can be assumed by the Society taking into account the type of application.

The following guidance values can be adopted :

- Main propulsion gears : 1.20 ÷ 1.40
- Auxiliary gears : 1.15 ÷ 1.20

For gearing of duplicated independent propulsion or auxiliary machinery, duplicated beyond that required for class, a reduced value can be assumed at the discretion of the Society.

M56.3 Tooth root bending strength

M56.3.1 Scope and general remarks

The criterion for tooth root bending strength is the permissible limit of local tensile strength in the root fillet. The root stress σF and the permissible root stress σFP shall be calculated separately for the pinion and the wheel.
σF must not exceed σFP.

The following formulae and definitions apply to gears having rim thickness greater than 3.5 mn.

The result of rating calculations made by following this method are acceptable for normal pressure angles up to 25° and reference helix angles up to 30°.

For larger pressure angles and large helix angles, the calculated results should be confirmed by experience as by method A of the reference standard.

M56.3.2 Basic equations

M56.3.2.1 Tooth root bending stress for pinion and wheel

$$\sigma F = (Ft / bmn) \quad YF\ YS\ Y\beta\ KA\ K\gamma\ Kv\ KF\alpha\ KF\beta \leq \sigma FP$$

where:

YF =	tooth form factor	(*see* clause 3.3)
YS =	stress correction factor	(*see* clause 3.4)
Yβ =	helix angle factor	(*see* clause 3.5)
Ft, KA, Kγ, Kv, KFα KFβ		(*see* Part 1)
b		(*see* Part 1, clause 1.4)
mn		(*see* Part 1, clause 1.3)

M56.3.2.2 Permissible tooth root bending stress for pinion and wheel

$$\sigma FP = (\sigma FE\ Yd\ YN/SF)\ Y\ \sigma\ reITYRreITYX$$

where:

σFE	=	bending endurance limit
Yd	=	design factor
YN	=	life factor
YσreIT	=	relative notch sensitive factor
YRreIT	=	relative surface factor
YX	=	size factor
SF	=	safety factor for tooth root bending stress

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M56.3.3 Tooth form factor, YF

The tooth form factor, YF, represents the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact. YF shall be determined separately for the pinion and the wheel. In the case of helical gears, the form factors for gearing shall be determined in the normal section, i.e. for the virtual spur gear with virtual number of teeth zn.

The tooth form factor, YF, can be calculated as follows:

$$\mathrm{YF} = \frac{6\dfrac{\mathrm{hF}}{\mathrm{mn}}\cos\alpha\mathrm{Fen}}{\left(\dfrac{\mathrm{sFn}}{\mathrm{mn}}\right)^2\cos\alpha\mathrm{n}}$$

Where :

hF	=	bending moment arm for tooth root bending stress for application of load at the outer point of single tooth pair contact	mm
sFn	=	tooth root chord in the critical section	mm
αFen	=	pressure angle at the outer point of single tooth pair contact in the normal section	°

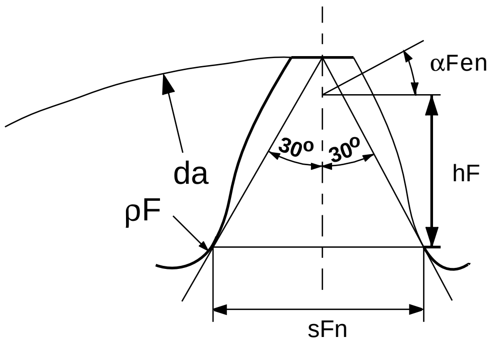

Fig. 3.1 For the calculation of hF, sFn and αFen,
the procedure outlined in the referencestandard can be used.

M56.3.4 Stress correction factor, YS

The stress correction factor, YS, is used to convert the nominal bending stress to the local tooth root stress, taking into account that not only bending stresses arise at the root.

YS applies to the load application at the outer point of single tooth pair contact.

YS shall be determined separately for the pinion and for the wheel.
The stress correction factor, YS, can be determined with the following equation (having range of validity: 1 ≤ qs < 8):

$$YS = (1.2 + 0.13L)qs^{\left(\frac{1}{1.21+2.3/L}\right)}$$

Where :

$$\mathrm{qs} = \frac{\mathrm{sFn}}{2\,\rho\mathrm{F}}$$

qs = notch parameter,
pF = root fillet radius in the critical section mm
L = sFn/hf
For hF and SFn see clause 3.1
For the calculation of pF the procedure outlined in the reference standard can be used. ▶

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M56.3.5 Helix angle factor, $Y\beta$

The helix angle factor, $Y\beta$, converts the stress calculated for a point loaded cantilever beam representing the substitute gear tooth to the stress induced by a load along an oblique load line into a cantilever plate which represents a helical gear tooth.

The helix angle factor, $Y\beta$ can be calculated as follows :

$$Y\beta = 1 - \varepsilon\beta \frac{\beta}{120}$$

where : β = reference helix angle in degrees.

One (1.0) is substituted for $\varepsilon\beta$ when $\varepsilon\beta > 1.0$, and 30° is substituted for $\beta > 30°$.

M56.3.6 Bending endurance limit, σFE

For a given material, σFE is the local tooth root stress which can be permanently endured.
According to the reference standard the number of 3×10^6 cycles is regarded as the beginning of the endurance limit.

σFE is defined as the unidirectional pulsating stress with a minimum stress of zero (disregarding residual stresses due to heat treatment). Other conditions such as alternating stress or prestressing etc. are covered by the design factor Yd.

The σFE values are to correspond to a failure probability 1% or less.

The endurance limit mainly depends on:

- material composition, cleanliness and defects;
- mechanical properties;
- residual stresses;
- hardening process, depth of hardened zone, hardness gradient;
- material structure (forged, rolled bar, cast).

The bending endurance limit, σFE can be determined, in general, making reference to values indicated in ISO 6336/5, quality MQ.

M56.3.7 Design factor, Yd

The design factor, Yd, takes into account the influence of load reversing and shrinkfit prestressing on the tooth root strength, relative to the tooth root strength with unidirectional load as defined for σFE.

The design factor, Yd, for load reversing, can be determined as follows :

Yd = 1,00 in general;

Yd = 0.9 for gears with occasional part load in reversed direction, such as main wheel in reversing gearboxes;

Yd = 0.7 for idler gears

M56.3.8 Life factor, YN

The life factor, YN, accounts for the higher tooth root bending stress permissible in case a limited life (number of cycles) is required.

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The factor mainly depends on:
– material and hardening;
– number of cycles;
– influence factors ($Y\delta e1T$, YRre1T, YX).

The life factor, YN, can be determined according to method B outlined in ISO 6336/3 standard.

M56.3.9 Relative notch sensitivity factor, $Y\delta re1T$

The relative notch sensitivity factor, $Y\delta re1T$, indicates the extent to which the theoretically concentrated stress lies above the fatigue endurance limit.
The factor mainly depends on material and relative stress gradient.

The relative notch sensitivity factor, $Y\delta re1T$, can be determined as follows :

- for notch parameter values (see clause 3.2) included in the range 1.5 < qs < 4, it can be assumed :

$$Y\delta re1T = 1.0$$

- for notch parameter outside said rangeYRre1T can be calculated as outlined in the reference standard.

M56.3.10 Relative surface factor, YRrelT

The relative surface factor, YRre1T, takes into account the dependence of the root strength on the surface condition in the tooth root fillet, mainly the dependence on the peak to valley surface roughness.

The relative surface factor, YRre1T can be determined as follows :

$R_z < 1$	$1 \leq R_z \leq 40$	
1.120	$1.675 - 0.53\ (R_z + 1)^{0.1}$	case hardened steels through - hardened steels ($\sigma B \geq 800$ N/mm^2)
1.070	$5.3 - 4.2\ (R_z + 1)^{0.01}$	normalised steels ($\sigma B < 800$ N/mm^2)
1.025	$4.3 - 3.26\ (R_z + 1)^{0.005}$	nitrided steels

Where :

R_z = mean peak to-valley roughness of tooth root fillets μm

σB = tensile strength in N/mm^2

The method applied here is only valid when scratches or similar defects deeper than 2 R_z are not present.

If the roughness stated is an Ra value (= CLA value) (= AA value) the following approximate relationship can be applied :

$$Ra = CLA = AA = R_z/6$$

M56.3.11 Size factor, YX

The size factor, YX, takes into account the decrease of the strength with increasing size.

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The factor mainly depends on:
– material and heat treatment;
– tooth and gear dimensions;
– ratio of case depth to tooth size.

The size factor, YX, can be determined as follows :

YX = 1.00	for mn ≤ 5	generally
YX = 1.03 - 0.06 mn	for 5 < mn < 30	normalised and through -
YX - 0.85	for mn ≥ 30	hardened steels
YX = 1.05 - 0.010 mn	for 5 < mn < 25	surface
YX = 0.80	for mn ≥ 25	hardened steels

M56.3.12 Safety factor for tooth root bending stress, SF

The safety factor for tooth root bending stress, SF, can be assumed by the Society taking into account the type of application.

The following guidance values can be adopted :

- Main propulsion gears: 1.55 ÷ 2.00
- Auxiliary gears: 1.40 ÷1.45

For gearing of duplicated independent propulsion or auxiliary machinery, duplicated beyond that required for class, a reduced value can be assumed at the discretion of the Society.

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M57 Use of Ammonia as a refrigerant
(1993)

1. Ammonia refrigerating machinery shall be installed in dedicated gastight compartments. Except for small compartments, at least two access doors are to be provided.

2. Compartments containing ammonia machinery (including process vessels) are to be fitted with :

   a) a negative ventilation system independent of ventilation systems serving other ship spaces and having a capacity not less than 30 changes per hour based upon the total volume of the space; other suitable arrangements which ensure an equivalent effectiveness may be considered;
   b) a fixed ammonia detector system with alarms inside and outside the compartment;
   c) water screens above all access doors, operable manually from outside the compartment;
   d) an independent bilge system.

3. At least two sets of breathing apparatus and protective clothings are to be available.

4. Ammonia piping is not to pass through accommodation spaces.

5. In case of ammonia plants of fishing vessels under 55 m in length or other ammonia plants with a quantity of ammonia not greater than 25 kg said plants are allowed to be located in the machinery space.
   The area where the ammonia machinery is installed is to be served by a hood with a negative ventilation system, so as not to permit any leakage of ammonia from dissipating into other areas in the space.
   A water spray system is to be provided for the said area.
   In addition previous items 2 b), 3 and 4 apply.

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M58 Charge Air Coolers
(1994)

1. **Plan approval**

   For charge air coolers, plans are not required for approval.

2. **Welding and materials**

   Materials are to be supplied with work certificates.
   Welding procedures and welders qualified by a recognised body are to be employed.

3. **Testing**

   Hydrostatic test on charge air cooler water side at 0.4 Nmm^2 (but not less than 1.5 times the maximum working pressure) is required.

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M59 (1996) Control and Safety Systems for Dual Fuel Diesel Engines

M59.1 Application

In addition to the requirements for oil firing diesel engines by the Classification Societies, and the requirements contained in chapter 5 and 16 of the IGC Code*, as far as found applicable, the following requirements are to be applied to dual-fuel diesel engines utilising high pressure Methane gas (NG: Natural Gas) fuel injection (hereinafter referred to as DFD engines).

M59.2 Operation mode

2.1 DFD engines are to be of the dual-fuel type employing pilot fuel ignition and to be capable of immediate change-over to oil fuel only.

2.2 Only oil fuel is to be used when starting the engine.

2.3 Only oil fuel is, in principle, to be used when the operation of an engine is unstable, and/or during manoeuvring and port operations.

2.4 In case of shut-off of the gas fuel supply, the engines are to be capable of continuous operation by oil fuel only.

M59.3 Protection of crankcase

3.1 Crankcase relief valves are to be fitted in way of each crankthrow. The construction and operating pressure of the relief valves are to be determined considering explosions due to gas leaks.

3.2 If a trunk piston type engine is used as DFD engine, the crankcase is to be protected by the following measures.

(1) Ventilation is to be provided to prevent the accumulation of leaked gas, the outlet for which is to be led to a safe location in the open through flame arrester.

(2) Gas detecting or equivalent equipment. (It is recommended that means for automatic injection of inert gas are to be provided).

(3) Oil mist detector.

3.3 If a cross-head type engine is used as DFD, the crankcase is to be protected by oil mist detector or bearing temperature detector.

M59.4 Protection for piston underside space of cross-head type engine

4.1 Gas detecting or equivalent equipment is to be provided for piston underside space of cross-head type engine.

M59.5 Engine Exhaust System

5.1 Explosion relief valves or other appropriate protection system against explosion are to be provided in the exhaust, scavenge and air inlet manifolds.

* International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk, mandatory under the 1983 amendments to 1974 SOLAS Convention.

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M59
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5.2 The exhaust gas pipes from DFD engines are not to be connected to the exhaust pipes of other engines or systems.

M59.6 Starting air line

6.1 Starting air branch pipes to each cylinder are to be provided with effective flame arresters.

M59.7 Combustion Monitoring

7.1 A failure mode and effect analysis (FMEA) examining all possible faults affecting the combustion process is to be submitted.

Details of required monitoring will be determined based on the outcome of the analysis. However, the following table may serve as guidance:

Faulty condition	Alarm	Aut. shut-off of the interlocked valves*
Function of gas fuel injection valves and pilot oil fuel injection valves	X	X
Exhaust gas temperature at each cylinder outlet and deviation from average	X	X
Cylinder pressure or ignition failure of each cylinder	X	X

* It is recommended that the gas master valve is also closed.

M59.8 Gas fuel supply to engine

8.1 Flame arresters are to be provided at the inlet to the gas supply manifold for the engine.

8.2 Arrangements are to be made so that the gas supply to the engine can be shut-off manually from starting platform or any other control position.

8.3 The arrangement and installation of the gas piping are to provide the necessary flexibility for the gas supply piping to accommodate the oscillating movements of DFD engine, without risk of fatigue failure.

8.4 The connecting of gas line and protection pipes or ducts regulated in 9.1 to the gas fuel injection valves are to provide complete coverage by the protection pipe or ducts.

M59.9 Gas fuel supply piping systems

9.1 Gas fuel piping may pass through or extend into machinery spaces or gas-safe spaces other than accommodation spaces, service spaces and control stations provided that they fulfil one of the following :

(1) The system complying with 16.3.1.1 of the IGC Code, and in addition, with (a), (b) and (c) given below.

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M59
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(a) The pressure in the space between concentric pipes is monitored continously. Alarm is to be issued and automatic valves specified in 16.3.6 of the IGC Code (hereinafter referred to as "interlocked gas valves") and the master gas fuel valves specified in 16.3.7 of the IGC Code (hereinafter referred to as "master gas valve") are to be closed before the pressure drops to below the inner pipe pressure (however, an interlocked gas valve connected to vent outlet is to be opened).

(b) Construction and strength of the outer pipes are to comply with the requirements of 5.2 of the IGC Code.

(c) It is to be so arranged that the inside of the gas fuel supply piping system between the master gas valve and the DFD engine is to be automatically purged with inert gas, when the master gas valve is closed; or

(2) The system complying with 16.3.1.2 of the IGC Code, and in addition, with (a) through (d) given below.

(a) Materials, construction and strength of protection pipes or ducts and mechanical ventilation systems are to be sufficiently durable against bursting and rapid expansion of high pressure gas in the event of gas pipe burst.

(b) The capacity of mechanical ventilating system is to be determined considering the flow rate of gas fuel and construction and arrangement of protective pipes or ducts, as deemed appropriate by the Classification Society.

(c) The air intakes of mechanical ventilating systems are to be provided with non-return devices effective for gas fuel leaks. However, if a gas detector is fitted at the air intakes, these requirements may be dispensed with.

(d) The number of flange joints of protective pipes or ducts is to be minimised; or

(3) Alternative arrangements to those given in paragraph 9.1(1) and (2) will be specially considered based upon an equivalent level of safety.

9.2 High pressure gas piping system are to be ensured to have sufficient constructive strength by carrying out stress analysis taking into account the stresses due to the weight of the piping system including acceleration load when significant, internal pressure and loads induced by hog and sag of the ships.

9.3 All valves and expansion joints used in high pressure gas fuel supply lines are to be of an approved type.

9.4 Joints on entire length of the gas fuel supply lines are to be butt-welded joints with full penetration and to be fully radiographed, except where specially approved by the Classification Society.

9.5 Pipe joints other than welded joints at the locations specially approved by the society are to comply with the appropriate standards recognised by the society, or those whose structural strength has been verified through tests and analysis as deemed appropriate by the Classification Society.

9.6 For all butt-welded joints of high pressure gas fuel supply lines, post-weld heat treatment are to be performed depending on the kind of material.

M59.10 Shut-off of gas fuel supply

10.1 In addition to the causes specified in 16.3.6 of the IGC Code, supply of gas fuel to DFD engines is to be shut off by the interlocked gas valves in case following abnormality occurs;

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M59
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(1) Abnormality specified in 7.1
(2) DFD engine stops from any cause
(3) Abnormality specified in 9.1 (1)(a)

10.2 In addition to the causes specified in 16.3.7 of IGC Code, the master gas valve is to be closed in case of any of the following:

(1) Oil mist detector or bearing temperature detector specified in 3.2(3) and 3.3 detects abnormality.
(2) Any kind of gas fuel leakage is detected.
(3) Abnormality specified in 9.1(1)(a)
(4) Abnormality specified in 11.1

10.3 The master gas valve is recommended to close automatically upon activation of the interlocked gas valves.

M59.11 Emergency stop of the DFD engines

11.1 DFD engine is to stopped before the gas concentration detected by the gas detectors specified in 16.2.2 of the IGC Code reached 60% of lower flammable limit.

M59.12 Gas fuel make-up plant and related storage tanks

12.1 Construction, control and safety system of high pressure gas compressors, pressure vessels and heat exchangers constituting a gas fuel make-up plant are so arranged as to the satisfaction of the Classification Society.

12.2 The possibility for fatigue failure of the high pressure gas piping due to vibration is to be considered.

12.3 The possibility for pulsation of gas fuel supply pressure caused by the high pressure gas compressor is to be considered.

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M60 (1997) Control and Safety of Gas Turbines for Marine Propulsion Use

M60.1 Governor and Over speed protective devices

M60.1.1

Main gas turbines are to be provided with over speed protective devices to prevent the turbine speed from exceeding more than 15% of the maximum continuous speed.

M60.1.2

Where a main gas turbine incorporates a reverse gear, electric transmission, controllable pitch propeller or other free-coupling arrangement, a speed governor independent of the over speed protective device is to be fitted and is to be capable of controlling the speed of the unloaded gas turbine without bringing the over speed protective device into action.

M60.2 Miscellaneous automatic safety devices

M60.2.1

Details of the manufacturer's proposed automatic safety devices to safeguard against hazardous conditions arising in the event of malfunctions in the gas turbine installation are to be submitted to the Classification Society together with the failure mode and effect analysis.

M60.2.2

Main gas turbines are to be equipped with a quick closing device (shut-down device) which automatically shuts off the fuel supply to the turbines at least in case of:

a) Over speed
b) Unacceptable lubricating oil pressure drop
c) Loss of flame during operation
d) Excessive vibration
e) Excessive axial displacement of each rotor (Except for gas turbines with rolling bearings)
f) Excessive high temperature of exhaust gas
g) Unacceptable lubricating oil pressure drop of reduction gear
h) Excessive high vacuum pressure at the compressor inlet

M60.2.3

The following turbine services are to be fitted with automatic temperature controls so as to maintain steady state conditions throughout the normal operating range of the main gas turbine:

a) Lubricating oil supply
b) Oil fuel supply (or automatic control of oil fuel viscosity as alternative)
c) Exhaust gas

M60.2.4

Automatic or interlocked means are to be provided for clearing all parts of the main gas turbine of the accumulation of liquid fuel or for purging gaseous fuel, before ignition commences on starting or recommences after failure to start.

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M60.2.5

Hand trip gear for shutting off the fuel in an emergency is to be provided at the manoeuvring station.

M60.2.6

Starting devices are to be so arranged that firing operation is discontinued and main fuel valve is closed within pre-determined time, when ignition is failed.

M60.3 Alarming devices

M60.3.1

Alarming devices listed in table 1 are to be provided.

M60.3.2

Alarms marked with "*" in Table 1 are to be activated at the suitable setting points prior to arriving the critical condition for the activation of shutdown devices.

M60.3.3

Suitable alarms are to be operated by the activation of shutdown devices.

M60
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Table 1 List of alarm and shutdown

Monitoring parameter	Alarm	Shutdown
Turbine speed	◓	❐
Lubricating oil pressure	◒ *	❐
Lubricating oil pressure of reduction gear	◒ *	❐
Differential pressure across lubricating oil filter	◓	
Lubricating oil temperature	◓	
Oil fuel supply pressure	◒	
Oil fuel temperature	◓	
Cooling medium temperature	◓	
Bearing temperature	◓	
Flame and ignition Failure	○	❐
Automatic starting Failure	○	
Vibration	◓ *	❐
Axial displacement of rotor	◓	❐
Exhaust gas temperature	◓ *	❐
Vacuum pressure at the compressor inlet	◓ *	❐
Loss of control system	○	

◓ Alarm for high value
◒ Alarm for low value
○ Alarm activated
❐ Shut down

◀◀

M61 (Dec 2003) Starting Arrangements of Internal Combustion Engines

M61.1 Mechanical starting arrangements

M61.1.1 The arrangement for air starting is to be such that the necessary air for the first charge can be produced on board without external aid.

M61.1.2 Where the main engine is arranged for starting by compressed air, two or more air compressors are to be fitted. At least one of the compressors is to be driven independent of the main propulsion unit and is to have the capacity not less than 50 % of the total required.

M61.1.3 The total capacity of air compressors is to be sufficient to supply within one hour the quantity of air needed to satisfy M61.1.5 by charging the receivers from atmospheric pressure. The capacity is to be approximately equally divided between the number of compressors fitted, excluding an emergency compressor which may be installed to satisfy M61.1.1.

M61.1.4 Where the main engine is arranged for starting by compressed air, at least two starting air receivers of about equal capacity are to be fitted which may be used independently.

M61.1.5 The total capacity of air receivers is to be sufficient to provide, without their being replenished, not less than 12 consecutive starts alternating between Ahead and Astern of each main engine of the reversible type, and not less than six starts of each main non-reversible type engine connected to a controllable pitch propeller or other device enabling the start without opposite torque. The number of starts refers to engine in cold and ready to start conditions. Additional number of starts may be required when the engine is in the warm running condition. When other consumers such as auxiliary engines starting systems, control systems, whistle, etc., are to be connected to starting air receivers, their air consumption is also to be taken into account.

Regardless of the above, for multi-engine installations the number of starts required for each engine may be reduced upon the agreement with the Classification Society depending upon the arrangement of the engines and the transmission of their output to the propellers.

M61.2 Electrical starting

M61.2.1 Where the main engine is arranged for electric starting, two separate batteries are to be fitted. The arrangement is to be such that the batteries cannot be connected in parallel. Each battery is to be capable of starting the main engine when in cold and ready to start conditions.The combined capacity of the batteries is to be sufficient without recharging to provide within 30 minutes the number of starts of main engines are required above in case of air starting.

M61.2.2 Electric starting arrangements for auxiliary engines are to have two separate batteries or may be supplied by separate circuits from the main engine batteries when such are provided. In the case of a single auxiliary engine only one battery may be required. The capacity of the batteries for starting the auxiliary engines is to be sufficient for at least three starts for each engine.

M61.2.3 The starting batteries are to be used for starting and the engine's own monitoring purposes only. Provisions are to be made to maintain continuously the stored energy at all times.

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M62 (Feb. 2002) Rooms for emergency fire pumps in cargo ships

The room(s) where the pump and prime mover are installed is/are to have adequate space for maintenance work and inspections.

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M63 Alarms and safeguards for emergency diesel engines

(Jan 2005)

1. Field of application

These requirements apply to diesel engines required to be immediately available in an emergency and capable of being controlled remotely or automatically operated.

2. Information to be submitted

Information demonstrating compliance with these requirements is to be submitted to the relevant Classification Society. The information is to include instructions to test the alarm and safety systems.

3. Alarms and safeguards

.1 Alarms and safeguards are to be fitted in accordance with Table 1.

.2 The safety and alarm systems are to be designed to ‘fail safe’. The charateristics of the ‘fail safe’ operation are to be evaluated on the basis not only of the system and its associated machinery, but also the complete installation, as well as the ship.

.3 Regardless of the engine output, if shutdowns additional to those specified in Table 1 are provided except for the overspeed shutdown, they are to be automatically overridden when the engine is in automatic or remote control mode during navigation.

.4 The alarm system is to function in accordance with M29, with additional requirements that grouped alarms are to be arranged on the bridge.

.5 In addition to the fuel oil control from outside the space, a local means of engine shutdown is to be provided.

.6 Local indications of at least those parameters listed in Table 1 are to be provided within the same space as the diesel engines and are to remain operational in the event of failure of the alarm and safety systems.

Table 1

Parameter	≥ 220kW	<220kW
Fuel oil leakage from pressure pipes	❍	❍
Lubricating oil temperature	◓	
Lubricating oil pressure	◒	◒
Oil mist concentration in crankcase[1]	◓	
Pressure or flow of cooling water	◒	
Temperature of cooling water (or cooling air)	◓	◓
Overspeed activated	❍ + ❒	

Note:

1 for engines having a power of more than 2250 kW or a cylinder bore of more than 300mm.

◒ **Alarm for low value**
◓ **Alarm for high value**
❍ **Alarm activated**
❒ **Shut down**

END

M64 Design of integrated cargo and ballast systems on tankers

(April 2003) (Rev.1 July 2004)

1. Application

These requirements are applicable to integrated cargo and ballast systems installed on tankers (i.e. cargo ships constructed or adapted for the carriage of liquid cargoes in bulk) contracted for construction on or after 1 January 2004, irrespective of the size or type of the tanker.
Within the scope of these requirements, integrated cargo and ballast system means any integrated hydraulic and/or electric system used to drive both cargo and ballast pumps (including active control and safety systems and excluding passive components, e.g. piping).

2. Functional Requirements

The operation of cargo and/or ballast systems may be necessary, under certain emergency circumstances or during the course of navigation, to enhance the safety of tankers.
As such, measures are to be taken to prevent cargo and ballast pumps becoming inoperative simultaneously due to a single failure in the integrated cargo and ballast system, including its control and safety systems.

3. Design features

The following design features are, inter alia, to be fitted:

.1 the emergency stop circuits of the cargo and ballast systems are to be independent from the circuits for the control systems. A single failure in the control system circuits or the emergency stop circuits are not to render the integrated cargo and ballast system inoperative;

.2 manual emergency stops of the cargo pumps are to be arranged in a way that they are not to cause the stop of the power pack making ballast pumps inoperable;

.3 the control systems are to be provided with backup power supply, which may be satisfied by a duplicate power supply from the main switch board. The failure of any power supply is to provide audible and visible alarm activation at each location where the control panel is fitted.

.4 in the event of failure of the automatic or remote control systems, a secondary means of control is to be made available for the operation of the integrated cargo and ballast system. This is to be achieved by manual overriding and/or redundant arrangements within the control systems.

Note:

1. This UR is to be uniformly implemented by all IACS Societies on tankers (as defined in M64.1) contracted for construction on or after 1 January 2004

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

END

M66

(Jan 2005) (Corr.1 Nov 2005) (Rev.1 Oct 2006) (Corr.1 Mar 2007) (Rev.2 Sept 2007) (Corr.1 Oct 2007)

Type Testing Procedure for Crankcase Explosion Relief Valves

1. Scope

1.1 To specify type tests and identify standard test conditions using methane gas and air mixture to demonstrate that classification society requirements are satisfied for crankcase explosion relief valves intended to be fitted to engines and gear cases.

1.2 This test procedure is only applicable to explosion relief valves fitted with flame arresters.

Note:

Where internal oil wetting of a flame arrester is a design feature of an explosion relief valve, alternative testing arrangements that demonstrate compliance with this UR may be proposed by the manufacturer. The alternative testing arrangements are to be agreed by the classification society.

2. Recognised Standards

2.1 EN 12874:2001: Flame arresters – Performance requirements, test methods and limits for use.

2.2 ISO/IEC EN 17025:2005: General requirements for the competence of testing and calibration laboratories.

2.3 EN 1070:1998: Safety of Machinery – Terminology.

2.4 VDI 3673: Part 1: Pressure Venting of Dust Explosions.

2.5 IMO MSC/Circular 677 – Revised Standards for the Design, Testing and Locating of Devices to Prevent the Passage of Flame into Cargo Tanks in Tankers

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Note:

1) Engines are to be fitted with components and arrangements complying with this UR when:

i) the engine is installed on existing ships (i.e. ships for which the date of contract for construction is before 1 January 2008) and the date of application for certification of the engine is on or after 1 January 2008; or

ii) the engine is installed on new ships (i.e. ships for which the date of contract for construction is on or after 1 January 2008).

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

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3. Purpose

3.1 The purpose of type testing crankcase explosion relief valves is fourfold:

3.1.1 To verify the effectiveness of the flame arrester.

3.1.2 To verify that the valve closes after an explosion.

3.1.3 To verify that the valve is gas/air tight after an explosion.

3.1.4 To establish the level of over pressure protection provided by the valve.

4. Test facilities

4.1 Test houses carrying out type testing of crankcase explosion relief valves are to meet the following requirements:

4.1.1 The test houses where testing is carried out are to be accredited to a National or International Standard, e.g. ISO/IEC 17025, and are to be acceptable to the classification societies.

4.1.2 The test facilities are to be equipped so that they can perform and record explosion testing in accordance with this procedure.

4.1.3 The test facilities are to have equipment for controlling and measuring a methane gas in air concentration within a test vessel to an accuracy of ± 0.1%.

4.1.4 The test facilities are to be capable of effective point-located ignition of a methane gas in air mixture.

4.1.5 The pressure measuring equipment is to be capable of measuring the pressure in the test vessel in at least two positions, one at the valve and the other at the test vessel centre. The measuring arrangements are to be capable of measuring and recording the pressure changes throughout an explosion test at a frequency recognising the speed of events during an explosion. The result of each test is to be documented by video recording and by recording with a heat sensitive camera.

4.1.6 The test vessel for explosion testing is to have documented dimensions. The dimensions are to be such that the vessel is not "pipe like" with the distance between dished ends being not more than 2.5 times its diameter. The internal volume of the test vessel is to include any standpipe arrangements.

4.1.7 The test vessel is to be provided with a flange, located centrally at one end perpendicular to the vessel longitudinal axis, for mounting the explosion relief valve. The test vessel is to be arranged in an orientation consistent with how the valve will be installed in service, i.e., in the vertical plane or the horizontal plane.

4.1.8 A circular plate is to be provided for fitting between the pressure vessel flange and valve to be tested with the following dimensions:

a) Outside diameter of 2 times the outer diameter of the valve top cover.

b) Internal bore having the same internal diameter as the valve to be tested.

M66
(cont'd)

4.1.9 The test vessel is to have connections for measuring the methane in air mixture at the top and bottom.

4.1.10 The test vessel is to be provided with a means of fitting an ignition source at a position specified in item 5.3.

4.1.11 The test vessel volume is to be as far as practicable, related to the size and capability of the relief valve to be tested. In general, the volume is to correspond to the requirement in UR M9.3 for the free area of explosion relief valve to be not less than $115cm^2/m^3$ of crankcase gross volume.

Notes:

1. This means that the testing of a valve having $1150cm^2$ of free area, would require a test vessel with a volume of $10m^3$.

2. Where the free area of relief valves is greater than 115 cm^2/m^3 of the crankcase gross volume, the volume of the test vessel is to be consistent with the design ratio.

3. In no case is the volume of the test vessel to vary by more than +15% to -15% from the design cm^2/m^3 volume ratio.

5. Explosion test process

5.1 All explosion tests to verify the functionality of crankcase explosion relief valves are to be carried out using an air and methane mixture with a volumetric methane concentration of 9.5% ±0.5%. The pressure in the test vessel is to be not less than atmospheric and is not to exceed the opening pressure of the relief valve.

5.2 The concentration of methane in the test vessel is to be measured at the top and bottom of the vessel and these concentrations are not to differ by more than 0.5%.

5.3 The ignition of the methane and air mixture is to be made at the centreline of the test vessel at a position approximately one third of the height or length of the test vessel opposite to where the valve is mounted.

5.4 The ignition is to be made using a maximum 100 joule explosive charge.

6. Valves to be tested

6.1 The valves used for type testing (including testing specified in item 6.3) are to be selected from the manufacturer's normal production line for such valves by the classification society witnessing the tests.

6.2 For approval of a specific valve size, three valves are to be tested in accordance with 6.3 and 7. For a series of valves item 9 refers.

6.3 The valves selected for type testing are to have been previously tested at the manufacturer's works to demonstrate that the opening pressure is in accordance with the specification within a tolerance of ± 20% and that the valve is air tight at a pressure below the opening pressure for at least 30 seconds.

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Note:

This test is to verify that the valve is air tight following assembly at the manufacturer's works and that the valve begins to open at the required pressure demonstrating that the correct spring has been fitted.

6.4 The type testing of valves is to recognise the orientation in which they are intended to be installed on the engine or gear case. Three valves of each size are to be tested for each intended installation orientation, i.e. in the vertical and/or horizontal positions.

7. Method

7.1 The following requirements are to be satisfied at explosion testing:

7.1.1 The explosion testing is to be witnessed by a classification society surveyor.

7.1.2 Where valves are to be installed on an engine or gear case with shielding arrangements to deflect the emission of explosion combustion products, the valves are to be tested with the shielding arrangements fitted.

7.1.3 Successive explosion testing to establish a valve's functionality is to be carried out as quickly as possible during stable weather conditions.

7.1.4 The pressure rise and decay during all explosion testing is to be recorded.

7.1.5 The external condition of the valves is to be monitored during each test for indication of any flame release by video and heat sensitive camera.

7.2 The explosion testing is to be in three stages for each valve that is required to be approved as being type tested.

7.2.1 Stage 1:

7.2.1.1 Two explosion tests are to be carried out in the test vessel with the circular plate described in 4.1.8 fitted and the opening in the plate covered by a 0.05mm thick polythene film.

Note:

These tests establish a reference pressure level for determination of the capability of a relief valve in terms of pressure rise in the test vessel, see 8.1.6.

7.2.2 Stage 2:

7.2.2.1 Two explosion tests are to be carried out on three different valves of the same size. Each valve is to be mounted in the orientation for which approval is sought i.e., in the vertical or horizontal position with the circular plate described in 4.1.8 located between the valve and pressure vessel mounting flange.

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7.2.2.2 The first of the two tests on each valve is to be carried out with a 0.05mm thick polythene bag, having a minimum diameter of three times the diameter of the circular plate and volume not less than 30% of the test vessel, enclosing the valve and circular plate. Before carrying out the explosion test the polythene bag is to be empty of air. The polythene bag is required to provide a readily visible means of assessing whether there is flame transmission through the relief valve following an explosion consistent with the requirements of the standards identified in Section 2.

Note:

During the test, the explosion pressure will open the valve and some unburned methane/air mixture will be collected in the polythene bag. When the flame reaches the flame arrester and if there is flame transmission through the flame arrester, the methane/air mixture in the bag will be ignited and this will be visible.

7.2.2.3 Provided that the first explosion test successfully demonstrated that there was no indication of combustion outside the flame arrester and there are no visible signs of damage to the flame arrester or valve, a second explosion test without the polythene bag arrangement is to be carried out as quickly as possible after the first test. During the second explosion test, the valve is to be visually monitored for any indication of combustion outside the flame arrester and video records are to be kept for subsequent analysis. The second test is required to demonstrate that the valve can still function in the event of a secondary crankcase explosion.

7.2.2.4 After each explosion, the test vessel is to be maintained in the closed condition for at least 10 seconds to enable the tightness of the valve to be ascertained. The tightness of the valve can be verified during the test from the pressure/time records or by a separate test after completing the second explosion test.

7.2.3 Stage 3:

7.2.3.1 Carry out two further explosion tests as described in Stage 1. These further tests are required to provide an average baseline value for assessment of pressure rise, recognising that the test vessel ambient conditions may have changed during the testing of the explosion relief valves in Stage 2.

8. Assessment and records

8.1 For the purposes of verifying compliance with the requirements of this UR, the assessment and records of the valves used for explosion testing is to address the following:

8.1.1 The valves to be tested are to have evidence of design appraisal/approval by the classification society witnessing tests.

8.1.2 The designation, dimensions and characteristics of the valves to be tested are to be recorded. This is to include the free area of the valve and of the flame arrester and the amount of valve lift at 0.2bar.

8.1.3 The test vessel volume is to be determined and recorded.

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8.1.4 For acceptance of the functioning of the flame arrester there is not to be any indication of flame or combustion outside the valve during an explosion test. This should be confirmed by the test laboratory taking into account measurements from the heat sensitive camera.

8.1.5 The pressure rise and decay during an explosion is to be recorded, with indication of the pressure variation showing the maximum overpressure and steady under-pressure in the test vessel during testing. The pressure variation is to be recorded at two points in the pressure vessel.

8.1.6 The effect of an explosion relief valve in terms of pressure rise following an explosion is ascertained from maximum pressures recorded at the centre of the test vessel during the three stages. The pressure rise within the test vessel due to the installation of a relief valve is the difference between average pressure of the four explosions from Stages 1 and 3 and the average of the first tests on the three valves in Stage 2. The pressure rise is not to exceed the limit specified by the manufacturer.

8.1.7 The valve tightness is to be ascertained by verifying from the records at the time of testing that an underpressure of at least 0.3bar is held by the test vessel for at least 10 seconds following an explosion. This test is to verify that the valve has effectively closed and is reasonably gas-tight following dynamic operation during an explosion.

8.1.8 After each explosion test in Stage 2, the external condition of the flame arrester is to be examined for signs of serious damage and/or deformation that may affect the operation of the valve.

8.1.9 After completing the explosion tests, the valves are to be dismantled and the condition of all components ascertained and documented. In particular, any indication of valve sticking or uneven opening that may affect operation of the valve is to be noted. Photographic records of the valve condition are to be taken and included in the report.

9. Design series qualification

9.1 The qualification of quenching devices to prevent the passage of flame can be evaluated for other similar devices of identical type where one device has been tested and found satisfactory.

9.2 The quenching ability of a flame arrester depends on the total mass of quenching lamellas/mesh. Provided the materials, thickness of materials, depth of lamellas/thickness of mesh layer and the quenching gaps are the same, then the same quenching ability can be qualified for different sizes of flame arresters subject to (a) and (b) being satisfied.

(a) $$\frac{n_1}{n_2} = \sqrt{\frac{S_1}{S_2}}$$

(b) $$\frac{A_1}{A_2} = \frac{S_1}{S_2}$$

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(cont'd)

Where:

n_1 = total depth of flame arrester corresponding to the number of lamellas of size 1 quenching device for a valve with a relief area equal to S_1

n_2 = total depth of flame arrester corresponding to the number of lamellas of size 2 quenching device for a valve with a relief area equal to S_2

A_1 = free area of quenching device for a valve with a relief area equal to S_1

A_2 = free area of quenching device for a valve with a relief area equal to S_2

9.3 The qualification of explosion relief valves of larger sizes than that which has been previously satisfactorily tested in accordance with Sections 7 and 8 can be evaluated where valves are of identical type and have identical features of construction subject to the following:

9.3.1 The free area of a larger valve does not exceed three times + 5% that of the valve that has been satisfactorily tested.

9.3.2 One valve of the largest size, subject to 9.3.1, requiring qualification is subject to satisfactory testing required by 6.3 and 7.2.2 except that a single valve will be accepted in 7.2.2.1 and the volume of the test vessel is not to be less than one third of the volume required by 4.1.11.

9.3.3 The assessment and records are to be in accordance with Section 8 noting that 8.1.6 will only be applicable to Stage 2 for a single valve.

9.4 The qualification of explosion relief valves of smaller sizes than that which has been previously satisfactorily tested in accordance with Sections 7 and 8 can be evaluated where valves are of identical type and have identical features of construction subject to the following:

9.4.1 The free area of a smaller valve is not less than one third of the valve that has been satisfactorily tested.

9.4.2 One valve of the smallest size, subject to 9.4.1, requiring qualification is subject to satisfactory testing required by 6.3 and 7.2.2 except that a single valve will be accepted in 7.2.2.1 and the volume of the test vessel is not to be more than the volume required by 4.1.11.

9.4.3 The assessment and records are to be in accordance with Section 8 noting that 8.1.6 will only be applicable to Stage 2 for a single valve.

10. The report

10.1 The test facility is to deliver a full report that includes the following information and documents:

10.1.1 Test specification.

10.1.2 Details of test pressure vessel and valves tested.

10.1.3 The orientation in which the valve was tested, (vertical or horizontal position).

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10.1.4 Methane in air concentration for each test.

10.1.5 Ignition source.

10.1.6 Pressure curves for each test.

10.1.7 Video recordings of each valve test.

10.1.8 The assessment and records stated in 8.

11. Approval

11.1 The approval of an explosion relief valve is at the discretion of individual classification societies based on the appraisal plans and particulars and the test facility's report of the results of type testing.

End Of Document

M65 Draining and Pumping Forward Spaces in Bulk Carriers

(Feb 2004)
(Rev.1 July 2004)

Application

1. This requirement applies to bulk carriers constructed generally with single deck, top-side tanks and hopper side tanks in cargo spaces intended primarily to carry dry cargo in bulk, and includes such types as ore carriers and combination carriers, which are contracted for construction on or after 1 January 2005.

Dewatering capacity

2. The dewatering system for ballast tanks located forward of the collision bulkhead and for bilges of dry spaces any part of which extends forward of the foremost cargo hold[1] is to be designed to remove water from the forward spaces at a rate of not less than 320Am^3/h, where A is the cross-sectional area in m^2 of the largest air pipe or ventilator pipe connected from the exposed deck to a closed forward space that is required to be dewatered by these arrangements

[1]. Reference is made to SOLAS regulation XII/13 and Unified Interpretation SC 179 "Dewatering of forward spaces of bulk carriers".

Note:

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

END

M66 Type Testing Procedure for Crankcase Explosion Relief Valves

(Jan 2005)
(Corr.1 Nov 2005)
(Rev.1 Oct 2006)
(Corr.1 Mar 2007)
(Rev.2 Sept 2007)
(Corr.1 Oct 2007)
(Rev.3 Jan 2008)

1. Scope

1.1 To specify type tests and identify standard test conditions using methane gas and air mixture to demonstrate that classification society requirements are satisfied for crankcase explosion relief valves intended to be fitted to engines and gear cases.

1.2 This test procedure is only applicable to explosion relief valves fitted with flame arresters.

Note:

Where internal oil wetting of a flame arrester is a design feature of an explosion relief valve, alternative testing arrangements that demonstrate compliance with this UR may be proposed by the manufacturer. The alternative testing arrangements are to be agreed by the classification society.

2. Recognised Standards

2.1 EN 12874:2001: Flame arresters – Performance requirements, test methods and limits for use.

2.2 ISO/IEC EN 17025:2005: General requirements for the competence of testing and calibration laboratories.

2.3 EN 1070:1998: Safety of Machinery – Terminology.

2.4 VDI 3673: Part 1: Pressure Venting of Dust Explosions.

2.5 IMO MSC/Circular 677 – Revised Standards for the Design, Testing and Locating of Devices to Prevent the Passage of Flame into Cargo Tanks in Tankers.

Note:

1) Engines are to be fitted with components and arrangements complying with this UR when:

i) the engine is installed on existing ships (i.e. ships for which the date of contract for construction is before 1 July 2008) and the date of application for certification of the engine (i.e. the date of whatever document the Classification Society requires/accepts as an application or request for certification of an individual engine) is on or after 1 July 2008; or

ii) the engine is installed on new ships (i.e. ships for which the date of contract for construction is on or after 1 July 2008).

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

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3. Purpose

3.1 The purpose of type testing crankcase explosion relief valves is fourfold:

3.1.1 To verify the effectiveness of the flame arrester.

3.1.2 To verify that the valve closes after an explosion.

3.1.3 To verify that the valve is gas/air tight after an explosion.

3.1.4 To establish the level of over pressure protection provided by the valve.

4. Test facilities

4.1 Test houses carrying out type testing of crankcase explosion relief valves are to meet the following requirements:

4.1.1 The test houses where testing is carried out are to be accredited to a National or International Standard, e.g. ISO/IEC 17025, and are to be acceptable to the classification societies.

4.1.2 The test facilities are to be equipped so that they can perform and record explosion testing in accordance with this procedure.

4.1.3 The test facilities are to have equipment for controlling and measuring a methane gas in air concentration within a test vessel to an accuracy of ± 0.1%.

4.1.4 The test facilities are to be capable of effective point-located ignition of a methane gas in air mixture.

4.1.5 The pressure measuring equipment is to be capable of measuring the pressure in the test vessel in at least two positions, one at the valve and the other at the test vessel centre. The measuring arrangements are to be capable of measuring and recording the pressure changes throughout an explosion test at a frequency recognising the speed of events during an explosion. The result of each test is to be documented by video recording and by recording with a heat sensitive camera.

4.1.6 The test vessel for explosion testing is to have documented dimensions. The dimensions are to be such that the vessel is not "pipe like" with the distance between dished ends being not more than 2.5 times its diameter. The internal volume of the test vessel is to include any standpipe arrangements.

4.1.7 The test vessel is to be provided with a flange, located centrally at one end perpendicular to the vessel longitudinal axis, for mounting the explosion relief valve. The test vessel is to be arranged in an orientation consistent with how the valve will be installed in service, i.e., in the vertical plane or the horizontal plane.

4.1.8 A circular plate is to be provided for fitting between the pressure vessel flange and valve to be tested with the following dimensions:

a) Outside diameter of 2 times the outer diameter of the valve top cover.

b) Internal bore having the same internal diameter as the valve to be tested.

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4.1.9 The test vessel is to have connections for measuring the methane in air mixture at the top and bottom.

4.1.10 The test vessel is to be provided with a means of fitting an ignition source at a position specified in item 5.3.

4.1.11 The test vessel volume is to be as far as practicable, related to the size and capability of the relief valve to be tested. In general, the volume is to correspond to the requirement in UR M9.3 for the free area of explosion relief valve to be not less than $115cm^2/m^3$ of crankcase gross volume.

Notes:

1. This means that the testing of a valve having $1150cm^2$ of free area, would require a test vessel with a volume of $10m^3$.

2. Where the free area of relief valves is greater than 115 cm^2/m^3 of the crankcase gross volume, the volume of the test vessel is to be consistent with the design ratio.

3. In no case is the volume of the test vessel to vary by more than +15% to -15% from the design cm^2/m^3 volume ratio.

5. Explosion test process

5.1 All explosion tests to verify the functionality of crankcase explosion relief valves are to be carried out using an air and methane mixture with a volumetric methane concentration of 9.5% ±0.5%. The pressure in the test vessel is to be not less than atmospheric and is not to exceed the opening pressure of the relief valve.

5.2 The concentration of methane in the test vessel is to be measured at the top and bottom of the vessel and these concentrations are not to differ by more than 0.5%.

5.3 The ignition of the methane and air mixture is to be made at the centreline of the test vessel at a position approximately one third of the height or length of the test vessel opposite to where the valve is mounted.

5.4 The ignition is to be made using a maximum 100 joule explosive charge.

6. Valves to be tested

6.1 The valves used for type testing (including testing specified in item 6.3) are to be selected from the manufacturer's normal production line for such valves by the classification society witnessing the tests.

6.2 For approval of a specific valve size, three valves are to be tested in accordance with 6.3 and 7. For a series of valves item 9 refers.

6.3 The valves selected for type testing are to have been previously tested at the manufacturer's works to demonstrate that the opening pressure is in accordance with the specification within a tolerance of ± 20% and that the valve is air tight at a pressure below the opening pressure for at least 30 seconds.

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Note:

This test is to verify that the valve is air tight following assembly at the manufacturer's works and that the valve begins to open at the required pressure demonstrating that the correct spring has been fitted.

6.4 The type testing of valves is to recognise the orientation in which they are intended to be installed on the engine or gear case. Three valves of each size are to be tested for each intended installation orientation, i.e. in the vertical and/or horizontal positions.

7. Method

7.1 The following requirements are to be satisfied at explosion testing:

7.1.1 The explosion testing is to be witnessed by a classification society surveyor.

7.1.2 Where valves are to be installed on an engine or gear case with shielding arrangements to deflect the emission of explosion combustion products, the valves are to be tested with the shielding arrangements fitted.

7.1.3 Successive explosion testing to establish a valve's functionality is to be carried out as quickly as possible during stable weather conditions.

7.1.4 The pressure rise and decay during all explosion testing is to be recorded.

7.1.5 The external condition of the valves is to be monitored during each test for indication of any flame release by video and heat sensitive camera.

7.2 The explosion testing is to be in three stages for each valve that is required to be approved as being type tested.

7.2.1 Stage 1:

7.2.1.1 Two explosion tests are to be carried out in the test vessel with the circular plate described in 4.1.8 fitted and the opening in the plate covered by a 0.05mm thick polythene film.

Note:

These tests establish a reference pressure level for determination of the capability of a relief valve in terms of pressure rise in the test vessel, see 8.1.6.

7.2.2 Stage 2:

7.2.2.1 Two explosion tests are to be carried out on three different valves of the same size. Each valve is to be mounted in the orientation for which approval is sought i.e., in the vertical or horizontal position with the circular plate described in 4.1.8 located between the valve and pressure vessel mounting flange.

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7.2.2.2 The first of the two tests on each valve is to be carried out with a 0.05mm thick polythene bag, having a minimum diameter of three times the diameter of the circular plate and volume not less than 30% of the test vessel, enclosing the valve and circular plate. Before carrying out the explosion test the polythene bag is to be empty of air. The polythene bag is required to provide a readily visible means of assessing whether there is flame transmission through the relief valve following an explosion consistent with the requirements of the standards identified in Section 2.

Note:

During the test, the explosion pressure will open the valve and some unburned methane/air mixture will be collected in the polythene bag. When the flame reaches the flame arrester and if there is flame transmission through the flame arrester, the methane/air mixture in the bag will be ignited and this will be visible.

7.2.2.3 Provided that the first explosion test successfully demonstrated that there was no indication of combustion outside the flame arrester and there are no visible signs of damage to the flame arrester or valve, a second explosion test without the polythene bag arrangement is to be carried out as quickly as possible after the first test. During the second explosion test, the valve is to be visually monitored for any indication of combustion outside the flame arrester and video records are to be kept for subsequent analysis. The second test is required to demonstrate that the valve can still function in the event of a secondary crankcase explosion.

7.2.2.4 After each explosion, the test vessel is to be maintained in the closed condition for at least 10 seconds to enable the tightness of the valve to be ascertained. The tightness of the valve can be verified during the test from the pressure/time records or by a separate test after completing the second explosion test.

7.2.3 Stage 3:

7.2.3.1 Carry out two further explosion tests as described in Stage 1. These further tests are required to provide an average baseline value for assessment of pressure rise, recognising that the test vessel ambient conditions may have changed during the testing of the explosion relief valves in Stage 2.

8. Assessment and records

8.1 For the purposes of verifying compliance with the requirements of this UR, the assessment and records of the valves used for explosion testing is to address the following:

8.1.1 The valves to be tested are to have evidence of design appraisal/approval by the classification society witnessing tests.

8.1.2 The designation, dimensions and characteristics of the valves to be tested are to be recorded. This is to include the free area of the valve and of the flame arrester and the amount of valve lift at 0.2bar.

8.1.3 The test vessel volume is to be determined and recorded.

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8.1.4 For acceptance of the functioning of the flame arrester there is not to be any indication of flame or combustion outside the valve during an explosion test. This should be confirmed by the test laboratory taking into account measurements from the heat sensitive camera.

8.1.5 The pressure rise and decay during an explosion is to be recorded, with indication of the pressure variation showing the maximum overpressure and steady under-pressure in the test vessel during testing. The pressure variation is to be recorded at two points in the pressure vessel.

8.1.6 The effect of an explosion relief valve in terms of pressure rise following an explosion is ascertained from maximum pressures recorded at the centre of the test vessel during the three stages. The pressure rise within the test vessel due to the installation of a relief valve is the difference between average pressure of the four explosions from Stages 1 and 3 and the average of the first tests on the three valves in Stage 2. The pressure rise is not to exceed the limit specified by the manufacturer.

8.1.7 The valve tightness is to be ascertained by verifying from the records at the time of testing that an underpressure of at least 0.3bar is held by the test vessel for at least 10 seconds following an explosion. This test is to verify that the valve has effectively closed and is reasonably gas-tight following dynamic operation during an explosion.

8.1.8 After each explosion test in Stage 2, the external condition of the flame arrester is to be examined for signs of serious damage and/or deformation that may affect the operation of the valve.

8.1.9 After completing the explosion tests, the valves are to be dismantled and the condition of all components ascertained and documented. In particular, any indication of valve sticking or uneven opening that may affect operation of the valve is to be noted. Photographic records of the valve condition are to be taken and included in the report.

9. Design series qualification

9.1 The qualification of quenching devices to prevent the passage of flame can be evaluated for other similar devices of identical type where one device has been tested and found satisfactory.

9.2 The quenching ability of a flame arrester depends on the total mass of quenching lamellas/mesh. Provided the materials, thickness of materials, depth of lamellas/thickness of mesh layer and the quenching gaps are the same, then the same quenching ability can be qualified for different sizes of flame arresters subject to (a) and (b) being satisfied.

(a) $$\frac{n_1}{n_2} = \sqrt{\frac{S_1}{S_2}}$$

(b) $$\frac{A_1}{A_2} = \frac{S_1}{S_2}$$

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Where:

n_1 = total depth of flame arrester corresponding to the number of lamellas of size 1 quenching device for a valve with a relief area equal to S_1

n_2 = total depth of flame arrester corresponding to the number of lamellas of size 2 quenching device for a valve with a relief area equal to S_2

A_1 = free area of quenching device for a valve with a relief area equal to S_1

A_2 = free area of quenching device for a valve with a relief area equal to S_2

9.3 The qualification of explosion relief valves of larger sizes than that which has been previously satisfactorily tested in accordance with Sections 7 and 8 can be evaluated where valves are of identical type and have identical features of construction subject to the following:

9.3.1 The free area of a larger valve does not exceed three times + 5% that of the valve that has been satisfactorily tested.

9.3.2 One valve of the largest size, subject to 9.3.1, requiring qualification is subject to satisfactory testing required by 6.3 and 7.2.2 except that a single valve will be accepted in 7.2.2.1 and the volume of the test vessel is not to be less than one third of the volume required by 4.1.11.

9.3.3 The assessment and records are to be in accordance with Section 8 noting that 8.1.6 will only be applicable to Stage 2 for a single valve.

9.4 The qualification of explosion relief valves of smaller sizes than that which has been previously satisfactorily tested in accordance with Sections 7 and 8 can be evaluated where valves are of identical type and have identical features of construction subject to the following:

9.4.1 The free area of a smaller valve is not less than one third of the valve that has been satisfactorily tested.

9.4.2 One valve of the smallest size, subject to 9.4.1, requiring qualification is subject to satisfactory testing required by 6.3 and 7.2.2 except that a single valve will be accepted in 7.2.2.1 and the volume of the test vessel is not to be more than the volume required by 4.1.11.

9.4.3 The assessment and records are to be in accordance with Section 8 noting that 8.1.6 will only be applicable to Stage 2 for a single valve.

10. The report

10.1 The test facility is to deliver a full report that includes the following information and documents:

10.1.1 Test specification.

10.1.2 Details of test pressure vessel and valves tested.

10.1.3 The orientation in which the valve was tested, (vertical or horizontal position).

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10.1.4 Methane in air concentration for each test.

10.1.5 Ignition source.

10.1.6 Pressure curves for each test.

10.1.7 Video recordings of each valve test.

10.1.8 The assessment and records stated in 8.

11. Approval

11.1 The approval of an explosion relief valve is at the discretion of individual classification societies based on the appraisal of plans and particulars and the test facility's report of the results of type testing.

End Of Document

M67
(Jan 2005) (Corr.1 Nov 2005) (Rev.1 Oct 2006) (Corr.1 Oct 2007)

Type Testing Procedure for Crankcase Oil Mist Detection and Alarm Equipment

1. Scope

1.1 To specify the tests required to demonstrate that crankcase oil mist detection and alarm equipment intended to be fitted to diesel engines satisfy classification society requirements.

Note:
This test procedure is also applicable to oil mist detection and alarm equipment intended for gear cases.

2. Recognised Standards

2.1 IACS Unified Requirement E10 Type Test Specification.

3. Purpose

3.1 The purpose of type testing crankcase oil mist detection and alarm equipment is seven fold:

3.1.1 To verify the functionality of the system.

3.1.2 To verify the effectiveness of the oil mist detectors.

3.1.3 To verify the accuracy of oil mist detectors.

3.1.4 To verify the alarm set points.

3.1.5 To verify time delays between oil mist leaving the source and alarm activation.

3.1.6 To verify functional failure detection.

3.1.7 To verify the influence of optical obscuration on detection.

Note:

1) Engines are to be fitted with crankcase oil mist detection and alarm equipment complying with this UR when:

   i) an application for certification of an engine is dated on/after 1 January 2007; or

   ii) installed in new ships for which the date of contract for construction is on or after 1 January 2007.

   The requirements of 6.7 and 6.8 are to be uniformly implemented by IACS Societies from 1 January 2008.

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

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4. Test facilities

4.1 Test houses carrying out type testing of crankcase oil mist detection and alarm equipment are to satisfy the following criteria:

4.1.1 A full range of facilities for carrying out the environmental and functionality tests required by this procedure shall be available and be acceptable to the classification societies.

4.1.2 The test house that verifies the functionality of the equipment is to be equipped so that it can control, measure and record oil mist concentration levels in terms of mg/l to an accuracy of ± 10% in accordance with this procedure.

5. Equipment testing

5.1 The range of tests is to include the following:

5.1.1 For the alarm/monitoring panel:

(a) Functional tests described in Section 6.

(b) Electrical power supply failure test.

(c) Power supply variation test.

(d) Dry heat test.

(e) Damp heat test.

(f) Vibration test.

(g) EMC test.

(h) Insulation resistance test.

(i) High voltage test.

(j) Static and dynamic inclinations, if moving parts are contained.

5.1.2 For the detectors:

(a) Functional tests described in Section 6.

(b) Electrical power supply failure test.

(c) Power supply variation test.

(d) Dry heat test.

(e) Damp heat test.

(f) Vibration test.

(g) EMC test where susceptible

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(cont'd)

(h) Insulation resistance test.

(i) High voltage test.

(j) Static and dynamic inclinations.

6. Functional tests

6.1 All tests to verify the functionality of crankcase oil mist detection and alarm equipment are to be carried out in accordance with 6.2 to 6.6 with an oil mist concentration in air, known in terms of mg/l to an accuracy of ±10%.

6.2 The concentration of oil mist in the test chamber is to be measured in the top and bottom of the chamber and these concentrations are not to differ by more than 10%. See also 8.1.1.1.

6.3 The oil mist monitoring arrangements are to be capable of detecting oil mist in air concentrations of between 0 and 10% of the lower explosive limit (LEL) or between 0 and a percentage corresponding to a level not less than twice the maximum oil mist concentration alarm set point.

Note: The LEL corresponds to an oil mist concentration of approximately 50mg/l (~4.1% weight of oil in air mixture).

6.4 The alarm set point for oil mist concentration in air is to provide an alarm at a maximum level corresponding to not more than 5% of the LEL or approximately 2.5mg/l.

6.5 Where alarm set points can be altered, the means of adjustment and indication of set points are to be verified against the equipment manufacturer's instructions.

6.6 Where oil mist is drawn into a detector via piping arrangements, the time delay between the sample leaving the crankcase and operation of the alarm is to be determined for the longest and shortest lengths of pipes recommended by the manufacturer. The pipe arrangements are to be in accordance with the manufacturer's instructions/recommendations.

6.7 Detector equipment that is in contact with the crankcase atmosphere and may be exposed to oil splash and spray from engine lubricating oil is to be demonstrated as being such, that openings do not occlude or become blocked under continuous oil splash and spray conditions. Testing is to be in accordance with arrangements proposed by the manufacturer and agreed by the classification society.

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6.8 Detector equipment may be exposed to water vapour from the crankcase atmosphere which may affect the sensitivity of the equipment and it is to be demonstrated that exposure to such conditions will not affect the functional operation of the detector equipment. Where exposure to water vapour and/or water condensation has been identified as a possible source of equipment malfunctioning, testing is to demonstrate that any mitigating arrangements such as heating are effective. Testing is to be in accordance with arrangements proposed by the manufacturer and agreed by the classification society.

Note:
This testing is in addition to that required by 5.1.2(e) and is concerned with the effects of condensation caused by the detection equipment being at a lower temperature than the crankcase atmosphere.

7. Detectors and alarm equipment to be tested

7.1 The detectors and alarm equipment selected for the type testing are to be selected from the manufacturer's normal production line by the classification society witnessing the tests.

7.2 Two detectors are to be tested. One is to be tested in clean condition and the other in a condition representing the maximum level of lens obscuration specified by the manufacturer.

8. Method

8.1 The following requirements are to be satisfied at type testing:

8.1.1 Oil mist generation is to satisfy 8.1.1.1 to 8.1.1.5.

8.1.1.1 Oil mist is to be generated with suitable equipment using an SAE 80 monograde mineral oil or equivalent and supplied to a test chamber having a volume of not less than $1m^3$. The oil mist produced is to have a maximum droplet size of 5 μm.

Note:
The oil droplet size is to be checked using the sedimentation method.

8.1.1.2 The oil mist concentrations used are to be ascertained by the gravimetric deterministic method or equivalent.

Note:
For this test, the gravimetric deterministic method is a process where the difference in weight of a 0.8 μm pore size membrane filter is ascertained from weighing the filter before and after drawing 1 litre of oil mist through the filter from the oil mist test chamber. The oil mist chamber is to be fitted with a recirculating fan.

8.1.1.3 Samples of oil mist are to be taken at regular intervals and the results plotted against the oil mist detector output. The oil mist detector is to be located adjacent to where the oil mist samples are drawn off.

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8.1.1.4 The results of a gravimetric analysis are considered invalid and are to be rejected if the resultant calibration curve has an increasing gradient with respect to the oil mist detection reading. This situation occurs when insufficient time has been allowed for the oil mist to become homogeneous. Single results that are more than 10% below the calibration curve are to be rejected. This situation occurs when the integrity of the filter unit has been compromised and not all of the oil is collected on the filter paper.

8.1.1.5 The filters require to be weighed to a precision of 0.1mg and the volume of air/oil mist sampled to 10ml.

8.1.2 The testing is to be witnessed by authorised personnel from classification societies where type testing approval is required by a classification society.

8.1.3 Oil mist detection equipment is to be tested in the orientation (vertical, horizontal or inclined) in which it is intended to be installed on an engine or gear case as specified by the equipment manufacturer.

8.1.4 Type testing is to be carried out for each type of oil mist detection and alarm equipment for which a manufacturer seeks classification approval. Where sensitivity levels can be adjusted, testing is to be carried out at the extreme and mid-point level settings.

9. Assessment

9.1 Assessment of oil mist detection equipment after testing is to address the following:

9.1.1 The equipment to be tested is to have evidence of design appraisal/approval by the classification society witnessing tests.

9.1.2 Details of the detection equipment to be tested are to be recorded such as name of manufacturer, type designation, oil mist concentration assessment capability and alarm settings.

9.1.3 After completing the tests, the detection equipment is to be examined and the condition of all components ascertained and documented. Photographic records of the monitoring equipment condition are to be taken and included in the report.

10. Design series qualification

10.1 The approval of one type of detection equipment may be used to qualify other devices having identical construction details. Proposals are to be submitted for consideration.

11. The report

11.1 The test house is to provide a full report which includes the following information and documents:

11.1.1 Test specification.

11.1.2 Details of equipment tested.

11.1.3 Results of tests.

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12. Acceptance

12.1 Acceptance of crankcase oil mist detection equipment is at the discretion of individual classification societies based on the appraisal plans and particulars and the test house report of the results of type testing.

12.2 The following information is to be submitted to classification societies for acceptance of oil mist detection equipment and alarm arrangements:

12.2.1 Description of oil mist detection equipment and system including alarms.

12.2.2 Copy of the test house report identified in 11.

12.2.3 Schematic layout of engine oil mist detection arrangements showing location of detectors/sensors and piping arrangements and dimensions.

12.2.4 Maintenance and test manual which is to include the following information:

(a) Intended use of equipment and its operation.

(b) Functionality tests to demonstrate that the equipment is operational and that any faults can be identified and corrective actions notified.

(c) Maintenance routines and spare parts recommendations.

(d) Limit setting and instructions for safe limit levels.

(e) Where necessary, details of configurations in which the equipment is and is not to be used.

End of Document

M68 (Feb. 2005) Dimensions of propulsion shafts and their permissible torsional vibration stresses

M68.1 Scope

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

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

Explicitly the following applications are not covered by this UR:

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

M68.2 Alternative calculation methods

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

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

Notes:

1. This UR M 68 replaces URs M33, M37, M38, M39 and M48.
2. This UR M 68 applies to ships contracted for construction on or after 1 July 2006.
3. The “contracted for construction” date means the date on which the contract to build the vessel is signed between the prostective owner and the shipbuilder. For further details regarding the date of “contracted for construction”, refer to IACS Procedural Requirement (PR) No.29.

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M68.3 Material limitations

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

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

- For carbon and carbon manganese steels, a minimum specified tensile strength not exceeding 600 N/mm^2 for use in M68.5 and not exceeding 760 N/mm^2 in M68.4.
- For alloy steels, a minimum specified tensile strength not exceeding 800 N/mm^2.
- For propeller shafts in general a minimum specified tensile strength not exceeding 600 N/mm^2 (for carbon, carbon manganese and alloy steels).

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

M68.4 Shaft diameters

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

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

where:
d = minimum required diameter in mm
di = actual diameter in mm of shaft bore
d_o = outside diameter in mm of shaft. If the bore of the shaft is ≤0.40d_o, the expression

$1 - d_i^4 / d_o^4$ may be taken as 1.0

F = factor for type of propulsion installation
= 95 for intermediate shafts in turbine installation, diesel installations with hydraulic (slip type) couplings, electric propulsion installations
= 100 for all other diesel installations and all propeller shafts
k = factor for the particular shaft design features, see M68.6
n_0 = speed in revolutions per minute of shaft at rated power
p = rated power in kW transmitted through the shaft (losses in gearboxes and bearings are to be disregarded)
σ_B = specified minimum tensile strength in N/mm^2 of the shaft material, see M68.3

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

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M68.5 Permissible torsional vibration stresses

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

Torsional vibration calculations are to include normal operation and operation with any one cylinder misfiring (i.e. no injection but with compression) giving rise to the highest torsional vibration stresses in the shafting.

For continuous operation the permissible stresses due to alternating torsional vibration are not to exceed the values given by the following formulae:

$$\pm\tau_C = \frac{\sigma_B + 160}{18} \bullet c_K \bullet c_D \bullet \left(3 - 2 \bullet \lambda^2\right) \qquad \text{for} \quad \lambda < 0.9$$

$$\pm\tau_C = \frac{\sigma_B + 160}{18} \bullet c_K \bullet c_D \bullet 1.38 \qquad \text{for} \quad 0.9 \le \lambda < 1.05$$

where:

τ_C = permissible stress amplitude in N/mm² due to torsional vibration for continuous operation
σ_B = specified minimum ultimate tensile strength in N/mm² of the shaft material, see also M68.3
c_K = factor for the particular shaft design features, see M68.6

c_D = size factor

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

d_o = shaft outside diameter in mm

λ = speed ratio = n/n_0
n = speed in revolutions per minute under consideration
n_0 = speed in revolutions per minute of shaft at rated power

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

Restricted speed ranges in normal operating conditions are not acceptable above λ = 0.8.

Restricted speed ranges in one-cylinder misfiring conditions of single propulsion engine ships are to enable safe navigation.

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

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

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

$$\frac{16 \bullet n_c}{18 - \lambda_c} \le n \le \frac{(18 - \lambda_c) \bullet n_c}{16}$$

where:
n_C = critical speed in revolutions per minute (resonance speed)
λ_C = speed ratio = n_c / n_o

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

$$\pm\tau_T = 1.7 \bullet \tau_c / \sqrt{c_K}$$

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

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

intermediate shafts with						thrust shafts external to engines		propeller shafts		
integral coupling flange [1] and straight sections	shrink fit coupling [2]	Keyway, tapered connection [3][4]	Keyway, cylindrical connection [3][4]	radial hole [5]	longitudinal slot [6]	on both sides of thrust collar [1]	in way of bearing when a roller bearing is used	Flange mounted or keyless taper fitted propellers [8]	Key fitted propellers [8]	Between forward end of aft most bearing and forward stern tube seal
k= 1.0	1.0	1.10	1.10	1.10	1.20	1.10	1.10	1.22	1.26	1.15
c_K=1.0	1.0	0.60	0.45	0.50	0.30 [7]	0.85	0.85	0.55	0.55	0.80

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

Footnotes

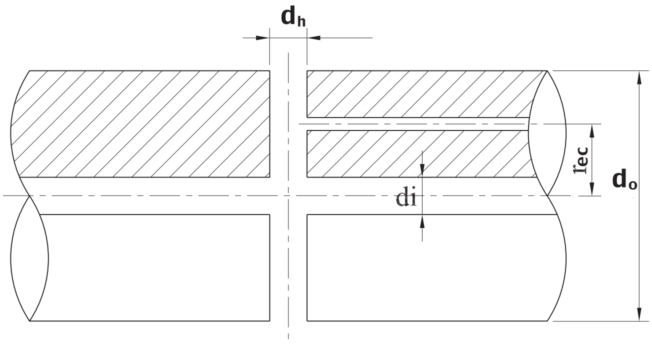
1) Fillet radius is not to be less than 0.08d.

2) k and c_k refer to the plain shaft section only. Where shafts may experience vibratory stresses close to the permissible stresses for continuous operation, an increase in diameter to the shrink fit diameter is to be provided, e.g. a diameter increase of 1 to 2 % and a blending radius as described in the table note.

3) At a distance of not less than $0.2d_o$ from the end of the keyway the shaft diameter may be reduced to the diameter calculated with k=1.0.

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

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



6) Subject to limitations as slot length (l)/outside diameter < 0.8 and inner diameter (d_i)/outside diameter < 0.8 and slot width (e)/outside diameter > 0.10. The end rounding of the slot is not to be less than e/2. An edge rounding should preferably be avoided as this increases the stress concentration slightly.
The k and c_K values are valid for 1, 2 and 3 slots, i.e. with slots at 360 respectively 180 and respectively 120 degrees apart.

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

$$c_K = 1.45/scf$$

Note that the scf is defined as the ratio between the maximum local principal stress and $\sqrt{3}$ times the nominal torsional stress (determined for the bored shaft without slots).

8) Applicable to the portion of the propeller shaft between the forward edge of the aftermost shaft bearing and the forward face of the propeller hub (or shaft flange), but not less than 2.5 times the required diameter.

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M68.7 Notes

1. Shafts complying with this UR satisfy the following:

1. Low cycle fatigue criterion (typically < 10^4), i.e. the primary cycles represented by zero to full load and back to zero, including reversing torque if applicable.
   This is addressed by the formula in M68.4.

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

3. The accumulated fatigue due to torsional vibration when passing through a barred speed range or any other transient condition with associated stresses beyond those permitted for continuous operation is addressed by the criterion for transient stresses in M68.5.

2. Explanation of k and c_K.

The factors k (for low cycle fatigue) and c_K (for high cycle fatigue) take into account the influence of:

- The stress concentration factors (scf) relative to the stress concentration for a flange with fillet radius of $0.08d_o$ (geometric stress concentration of approximately 1.45).

$$c_K \approx \frac{1.45}{scf} \qquad \text{and} \qquad k \approx \left[\frac{scf}{1.45}\right]^x$$

  where the exponent *x* considers low cycle notch sensitivity.

- The notch sensitivity. The chosen values are mainly representative for soft steels (σ_B < 600), while the influence of steep stress gradients in combination with high strength steels may be underestimated.

- The size factor c_D being a function of diameter only does not purely represent a statistical size influence, but rather a combination of this statistical influence and the notch sensitivity.

The actual values for k and c_K are rounded off.

3. Stress concentration factor of slots

The stress concentration factor (scf) at the end of slots can be determined by means of the following empirical formulae using the symbols in footnote 6):

$$scf = \alpha_{t(hole)} + 0.57 \bullet \frac{(l-e)/d}{\sqrt{\left(1-\frac{d_i}{d}\right) \bullet \frac{e}{d}}}$$

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This formula applies to:

- slots at 120 or 180 or 360 degrees apart.
- slots with semicircular ends. A multi-radii slot end can reduce the local stresses, but this is not included in this empirical formula.
- slots with no edge rounding (except chamfering), as any edge rounding increases the scf slightly.

$\alpha_{t(hole)}$ represents the stress concentration of radial holes (in this context e = hole diameter) and can be determined as :

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

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

END

M69 (June 2008) Qualitative Failure Analysis for Propulsion and Steering on Passenger Ships

1. Scope

Detailing a qualitative failure analysis for propulsion and steering for new passenger ships including those having a length of 120 m or more or having three or more main vertical zones.

2. Note

This may be considered as the first step for demonstrating compliance with the revised SOLAS Chapter II-2, Regulation 21 – SOLAS 2006 Amendments, Resolution MSC.216(82), annex 3.

3. Objectives

3.1 For ships having at least two independent means of propulsion and steering to comply with SOLAS requirements for a safe return to port, items (a) and (b) below are applicable:

(a) Provide knowledge of the effects of failure in all the equipment and systems due to fire in any space, or flooding of any watertight compartment that could affect the availability of the propulsion and steering.

(b) Provide solutions to ensure the availability of propulsion and steering upon such failures in item (a).

3.2 Ships not required to satisfy the safe return to port concept will require the analysis of failure in single equipment and fire in any space to provide knowledge and possible solutions for enhancing availability of propulsion and steering.

4. Systems to be considered

4.1 The qualitative failure analysis is to consider the propulsion and steering equipment and all its associated systems which might impair the availability of propulsion and steering.

4.2 The qualitative failure analysis should include:

(a) Propulsion and electrical power prime movers, e.g.,
- Diesel engines
- Electric motors

Note:

1. This UR is to be uniformly implemented by IACS Societies for Passenger Ships contracted for construction on or after 1 January 2010.

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

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(b) Power transmission systems, e.g.,
- Shafting
- Bearings
- Power converters
- Transformers
- Slip ring systems

(c) Steering gear
- Rudder actuator or equivalent for azimuthing propulsor
- Rudder stock with bearings and seals
- Rudder
- Power unit and control gear
- Local control systems and indicators
- Remote control systems and indicators
- Communication equipment

(d) Propulsors, e.g.,
- Propeller
- Azimuthing thruster
- Water jet

(e) Main power supply systems, e.g.,
- Electrical generators and distribution systems
- Cable runs
- Hydraulic
- Pneumatic

(f) Essential auxiliary systems, e.g.,
- Compressed air
- Oil fuel
- Lubricating oil
- Cooling water
- Ventilation
- Fuel storage and supply systems

(g) Control and monitoring systems, e.g.,
- Electrical auxiliary circuits
- Power supplies
- Protective safety systems
- Power management systems
- Automation and control systems

(h) Support systems, e.g.,
- Lighting
- Ventilation

To consider the effects of fire or flooding in a single compartment, the analysis is to address the location and layout of equipment and systems.

5. Failure Criteria

5.1 Failures are deviations from normal operating conditions such as loss or malfunction of a component or system such that it cannot perform an intended or required function.

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5.2 The qualitative failure analysis should be based on single failure criteria, (not two independent failures occurring simultaneously).

5.3 Where a single failure cause results in failure of more than one component in a system (common cause failure), all the resulting failures are to be considered together.

5.4 Where the occurrence of a failure leads directly to further failures, all those failures are to be considered together.

6. Verification of Solutions

6.1 The shipyard is to submit a report to class societies that identifies how the objectives have been addressed. The report is to include the following information:

(a) Identify the standards used for analysis of the design.

(b) Identify the objectives of the analysis.

(c) Identify any assumptions made in the analysis.

(d) Identify the equipment, system or sub-system, mode of operation of the equipment.

(e) Identify probable failure modes and acceptable deviations from the intended or required function.

(f) Evaluate the local effects (e.g. fuel injection failure) and the effects on the system as a whole (e.g. loss of propulsion power) of each failure mode as applicable.

(g) Identify trials and testing necessary to prove conclusions.

Note: All stakeholders (e.g., class, owners, shipyard and manufacturers) should as far as possible be involved in the development of the report.

6.2 The report is to be submitted prior to approval of detail design plans. The report may be submitted in two parts:

(a) A preliminary analysis as soon as the initial arrangements of different compartments and propulsion plant are known which can form the basis of discussion. This is to include a structured assessment of all essential systems supporting the propulsion plant after a failure in equipment, fire or flooding in any compartment casualty.

(b) A final report detailing the final design with a detailed assessment of any critical system identified in the preliminary report.

6.3 Verification of the report findings are to be agreed between the class society and the shipyard.

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