2.5. Transmission of propulsive power

The transmission of power from the prime mover to the propeller is made up of a large set of components, which should deserve proper care and attention from the designer mainly in terms of performance requirements, choice of the most adequate solutions, definition of arrangement and location, and in some cases definition of their dimensions (e.g. shaft's length and diameter).

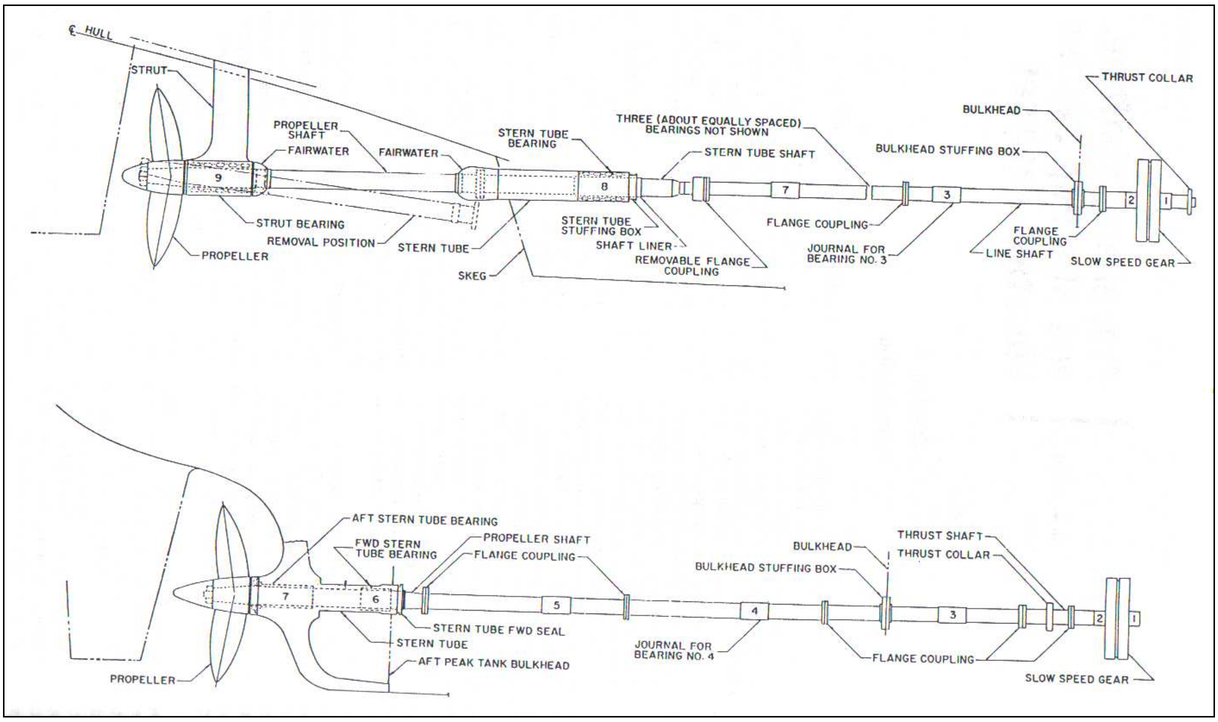


Fig. 2.31 – Shafting arrangement with (above) and without (below) strut bearing.

Figure 2.31 illustrates a typical shafting arrangement with (above) and without (below) strut bearing of a propulsion system.

Major components of power transmission sub-systems are as follows:

a) Gearbox:

a.1) Reduction;

a.2) Reduction inversion (clutching device has to be also provided).

b) Clutch;

c) Couplings:

c.1) Rigid coupling;

c.1.1) Forged bolted flange with holes;

c.1.2) Removable or muff coupling (of the SKF type).

c.2) Elastic coupling.

d) Transmission shafts (thrust or engine shaft, intermediate or line shaft, and tail or propeller shaft);

e) Bearings:

- e.1) Support;
- e.2) Thrust;
- e.3) Combined.

f) Seals:

- f.1) Stern tube seals:
  - f.1.2) Axial sealing:
    - f.1.2.1) Stuffing boxes;
    - f.1.2.2) Lip sealing.
  - f.1.3) Radial face sealing.
- f.2) Bulkhead seal.

For most of the components listed above, the engineer defines their performance requirements during basic design stage, receives proposals from manufacturers, which are then analyzed in order to select the most advantageous technical and economical solution (in terms of acquisition cost and integration cost in the ship system as a whole).

In respect to transmission shafts and rigid couplings (bolted flanges type), in general, the naval engineer is also responsible for the detailed design (location, dimensions and materials selection for shipyard manufacture and installation purposes). There are a few exceptions related to the detail design of components of propulsion systems with very specific propulsors, such as: controllable pitch propellers and waterjets, usually exclusively carried out by their manufacturers.

2.6. Description of the most significant types of propulsion systems and their components:

2.6.1. Prime movers

Introduction:

In general, prime movers can be divided in three distinct groups, as follows:

a) Combustion engines:

> a.1) Internal (e.g. slow and medium speed diesel engines, gasoline engines, and gas turbines);
>
> a.2) External (e.g. steam engine - out of order, nuclear power plants or boilers with steam turbines).

b) Electrical motor.

c) Hydraulic motor.

Since internal combustion engines of the diesel type are the most common in marine applications, a special attention will be given in here to the most relevant technical aspects of this specific prime mover.

Diesel and Gasoline Engine's Working Principle:

The diesel engine is a type of internal combustion engine which ignites the fuel by injecting it into hot, high-pressure air in a combustion chamber. In common with all internal combustion engines the diesel engine operates with a fixed sequence of events, which may be achieved either in four strokes or two, a stroke being the travel of the piston between its extreme points. Each stroke is accomplished in half a revolution of the crankshaft. This is why these engines are also called Compression Ignition (CI) engines.

On the other hand, on a gasoline engine the ignition is initiated by means of an electrical spark. Therefore, these engines are also called Spark Ignition (SI) engines.

Engine Operating Cycles:

In the reciprocating engine, the piston moves back and forth in a cylinder and transmits power through a connecting rod and crank mechanism to the drive shaft as shown in Figure 2.32. The steady rotation of the crank produces a cyclical piston motion. The piston comes to rest at the Top-Center (TC) crank position and Bottom-Center (BT) crank position when the cylinder volume is a minimum or maximum, respectively. The minimum cylinder volume is called clearance volume, V_c. The volume swept out by the piston, the difference between the maximum and or total volume, V_t and the clearance volume, is called the displaced or swept volume, V_d. The ratio of maximum volume to minimum volume is the compression ratio, r_c. Typical values of r_c are 8 to 12 for SI engines and 12 to 24 for CI.

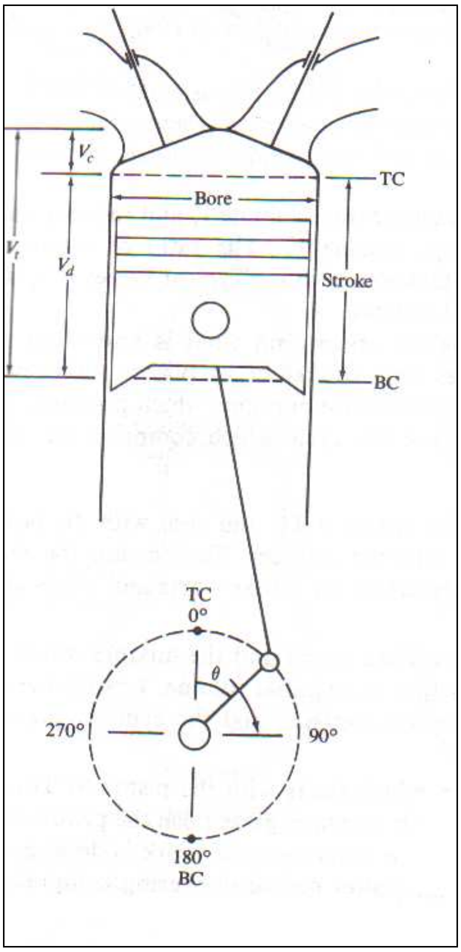


Fig. 2.32 – Basic geometry of the reciprocating internal combustion engine.

a) Four-stroke

The majority of reciprocating engines operate on what is known as the four-stroke cycle. Each cylinder requires four strokes of its piston – two revolutions of the crankshaft – to complete the sequence of events which produces one power stroke. Both SI and CI engines use this cycle which comprises (see Figure 2.33):

a.1) An intake stroke, which starts with piston at TC and ends with the piston at BC;

a.2) A compression stroke, when both valves are closed and the mixture inside the cylinder is compressed to a small fraction of its initial volume;

a.3) A power or expansion stroke, which starts with the piston at TC and ends at BC as the high-temperature, high-pressure gases push the piston down and force the crank to rotate;

a.4) An exhaust stroke, where the remaining burned gases exit the cylinder.

Each engine cycle with one power stroke is completed in two crankshaft revolutions.

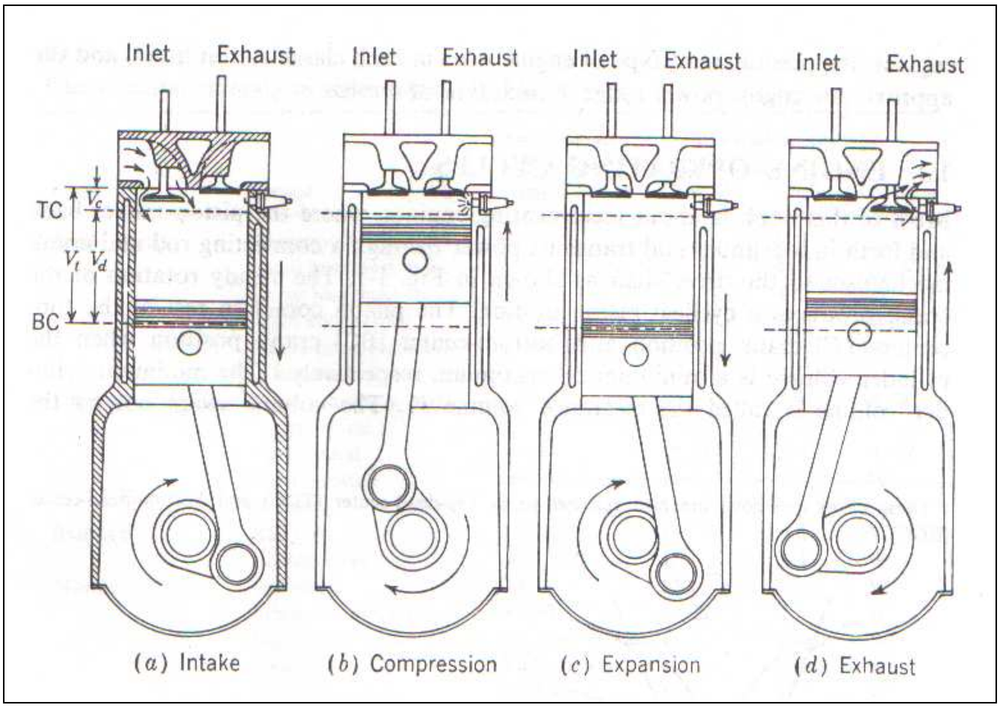


Fig. 2.33 – The four-stroke operating cycle.

b) Two-stroke

Also often called Otto cycle to pay tribute to its inventor Nicolaus Otto, who built the first engine of this type in 1876.

Figure 2.34 shows one of the simplest types of two-strokes engine designs. Ports in cylinder liner, opened and closed by the piston motion, control the exhaust and inlet flows while piston is close to BC. The two strokes are:

b.1) A compression stroke, which starts by closing the inlet and exhaust ports, and then compresses the cylinder contents and draws fresh charge into the crankcase. As the piston approaches TC, combustion is initiated;

b.2) A power or expansion stroke, similar to that in the four-stroke cycle until the piston approaches BC, when first the exhaust ports and then intake ports are uncovered. Most of the burnt gases exit the cylinder in an exhaust blow-down process. When the inlet ports are uncovered, the fresh charge which has been compressed in the crankcase flows into the cylinder. The piston and the ports are generally shaped to deflect the incoming charge from flowing directly into the exhaust ports and to achieve effective scavenging of the residual gases.

Each engine cycle with one power stroke is completed in one crankshaft revolution.

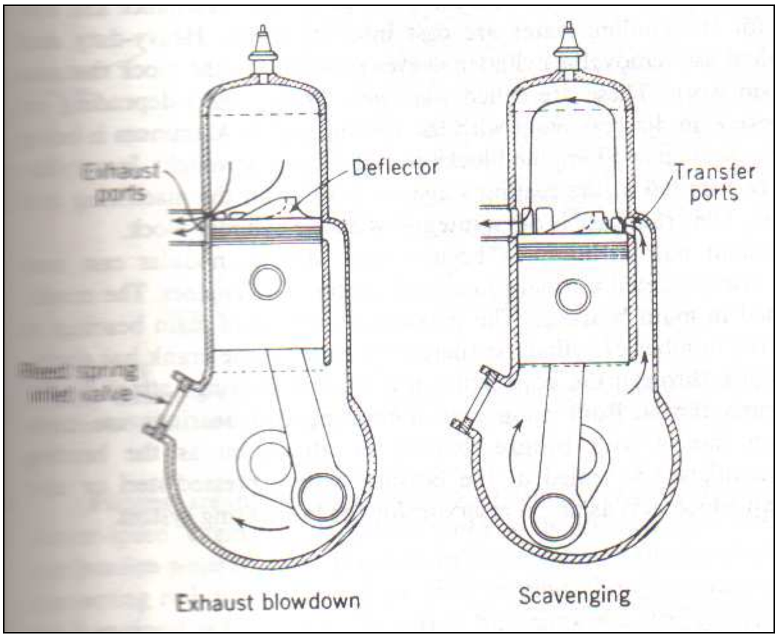


Fig. 2.34 – The two-stroke operating cycle.

Engine Components:

Labeled cutaway drawings of a four-stroke SI engine and a two-stroke CI engine are shown in Figures 2.35.a and 2.35.b, respectively. The spark-ignition engine is four-cylinder in-line automobile engine. The diesel engine is a large V eight-cylinder design with a uniflow scavenging process. The function of the major components of these engines and their construction materials will now be revised.

The engine cylinders are contained in the engine block. The block has traditionally been made of gray cast iron because its good wear resistance and low cost. Passages for the cooling water are cast into the block. Heavy duty marine engines often use removable cylinder sleeves pressed into the block that can be replaced when worn. These are called wet liners or dry liners depending whether the sleeve is in direct contact with cooling water. The crankcase is often integral with the cylinder block.

The crankshaft has traditionally been a steel forging. The crankshaft is supported in main bearings.

Pistons are made of aluminum in small engines or cast iron in large slower-speed engines. The piston both seals the cylinder and transmits the combustion-generated gas pressure to the crank via the connecting rod.

The cylinder head seals off the cylinders and is made of cast iron or aluminum. It must be strong and rigid to distribute the gas forces acting on the head as uniformly as possible through the engine block. The cylinder head contains the spark plug (SI engine) or fuel injector (CI engine), and, in overhead valve engines, parts of the valve mechanism.

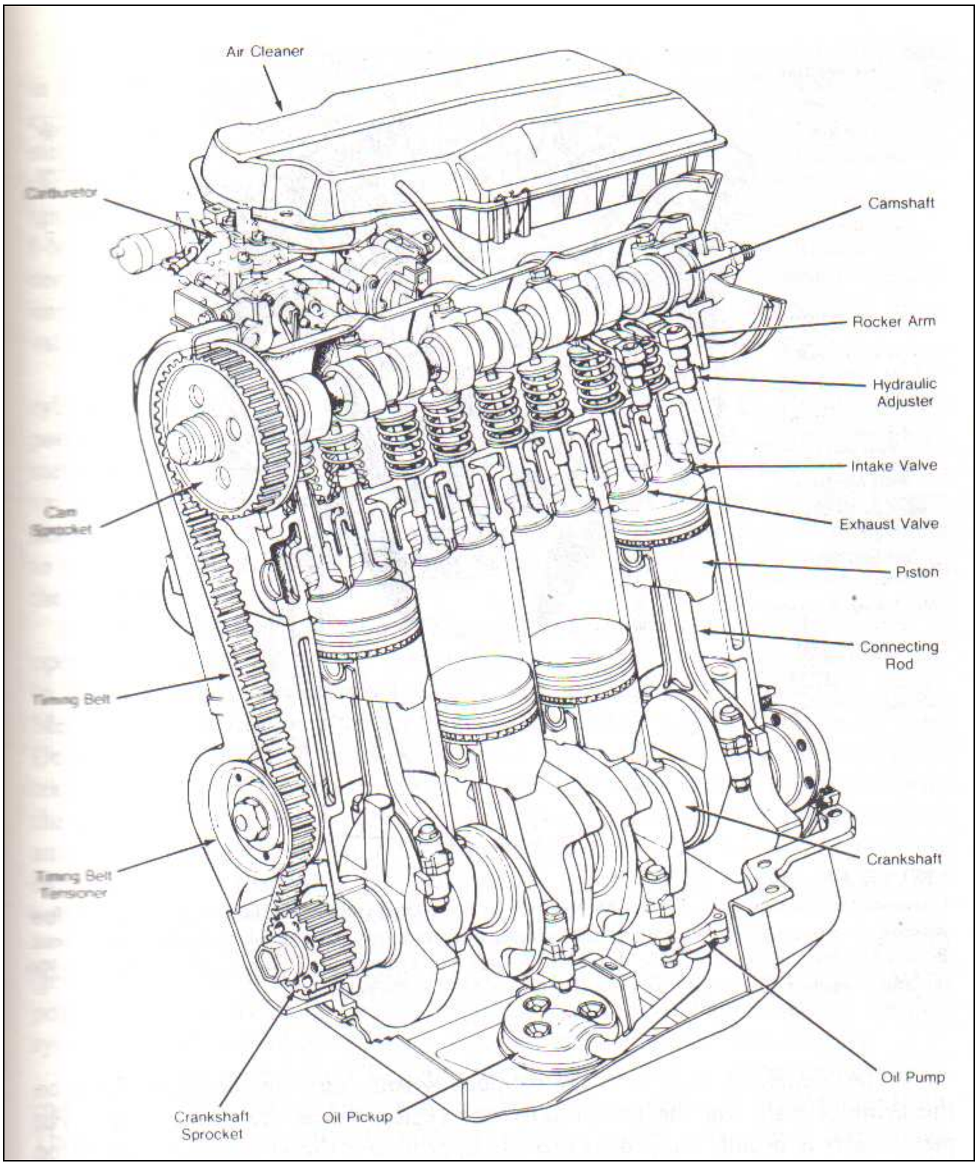


Fig. 2.35.a – Cutaway drawing of a four-stroke cycle spark-ignition engine.

The valve shown in Figure 2.35.a are poppet valves, the valve type normally used in four-stroke engines. Valves are made from forged alloy steel; The cooling of the exhaust valves which operates at about 700º C may be enhanced by using hollow stem partially filled with sodium which by evaporation and condensation dissipates heat more easily. The valve stem moves in a valve guide, which can be integral part of the cylinder head or may be a separate unit.

A camshaft made of cast iron or forged steel with one cam per valve is used to open and close the valves. The cam surfaces are hardened to obtain adequate life.

An intake manifold (aluminum or cast iron) and an exhaust manifold (generally of cast iron) complete the engine assembly. Other engine components specific to SI or CI engines – carburetor, ignition systems, fuel pumps, fuel injectors, etc. – can be found in literature.

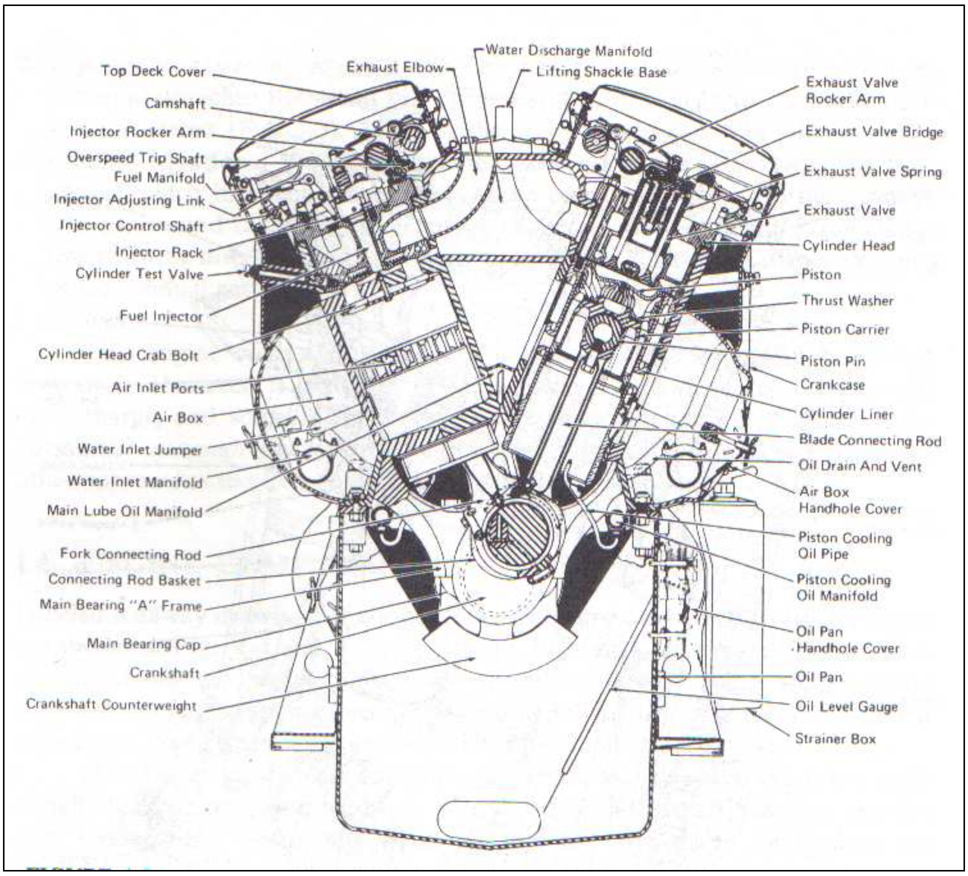


Fig. 2.35.b – Cross-section drawing of a two-stroke cycle diesel engine.

<u>Engine Design and Operating Parameters:</u>

a) Important engine characteristics

Some basic geometrical relationships and the parameters commonly used to characterize engine operation are developed. The factors important to an engine use are:

a.1) The engine's performance over its operating range;

a.2) The engine's fuel consumption within this operating range and the cost of the required fuel;

a.3) The engine's noise and air pollutant emissions within this operating range;

a.4) The initial cost of the engine and its installation;

a.5) The reliability and durability of the engine, its maintenance requirements, and how these affect engine availability and operating costs.

These factors control total engine operating costs – usually the primary consideration of the user – and whether the engine in operation can satisfy environmental regulations. These notes are primarily concerned with performance, efficiency and emission characteristics of diesel engines for marine applications.

Engine performance is more precisely defined by:

- the maximum power (or the maximum torque) available at each speed within the useful engine operating range;

- the range of speed and power over which engine operation is satisfactory.

b) Geometrical properties of engines

The following parameters define the basic geometry of a reciprocating engine (see Figure 2.36):

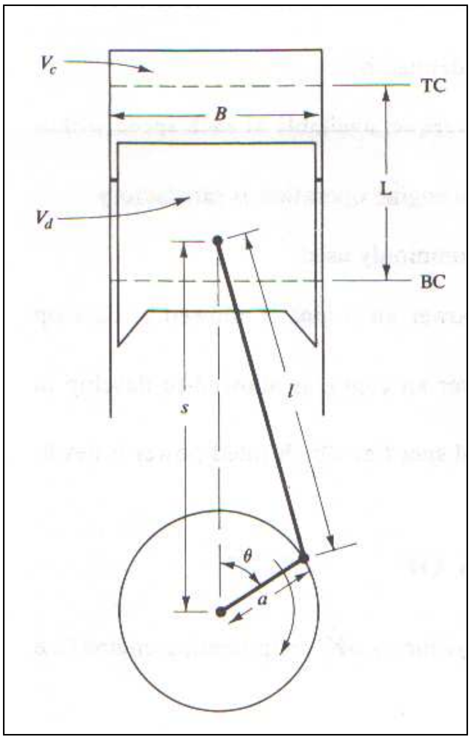


Fig. 2.36 – Geometry of cylinder, piston, connecting rod, and crankshaft.

b.1) Compression ratio, r_c:

$$r_c = \frac{V_d + V_c}{V_c} \tag{2.14}$$

b.2) Ratio of cylinder bore (B) to piston stroke (L), R_{bs}:

$$R_{bs} = \frac{B}{L} \tag{2.15}$$

b.3) Ratio of connecting rod length (l) to crank radius (a), R:

$$R = \frac{l}{a} \tag{2.16}$$

In addition, the stroke and crank radius are related by: $L = 2a$.

b.4) The cylinder volume, V, at any crank position θ is given by:

$$V = V_c + \frac{\pi B^2}{4}\left(l + a - s\right) \tag{2.17}$$

, where s is the distance between the crank axis and the piston pin axis (see Figure 2.36), and is given by:

$$s = a\cos\theta + \sqrt{l^2 - a^2 \sin^2\theta} \tag{2.18}$$

Substituting equation (2.16) and (2.18) into (2.17) we obtain:

$$\frac{V}{V_c} = 1 + \frac{1}{2}(r_c - 1)\left[R + 1 - \cos\theta - \sqrt{R^2 - \sin^2\theta}\right] \tag{2.19}$$

b.5) The combustion chamber surface area, A, at any crank position θ is given by:

$$A = A_{ch} + A_p + \pi B(l + a - s) \tag{2.20}$$

, where A_{ch} is the cylinder head surface area, and A_p is the piston crown surface area. For flat-topped pistons $A_p = \pi B^2/4$, and therefore using equations (2.18) and (2.20) we obtain:

$$A = A_{ch} + A_p + \frac{\pi BL}{2}\left[R + 1 - \cos\theta - \sqrt{R^2 - \sin^2\theta}\right] \tag{2.21}$$

b.6) An important characteristic speed is the mean piston speed, $\overline{S}_p$, given by:

$$\overline{S}_p = 2LN \tag{2.22}$$

, where N is the rotational speed of the crankshaft. Mean piston speed is often a more appropriate parameter than crank rotational speed for correlating engine behavior as a function of speed. For example gas-flow velocities in the intake. The instantaneous piston velocity, S_p, is obtained from:

$$S_p = \frac{ds}{dt} \tag{2.23}$$

The piston velocity is zero at the beginning of the stroke, reaches a maximum near the middle of the stroke, and decreases to zero at the end stroke. Differentiation of equation (2.18) and substitution above gives:

$$S_p = \frac{\pi}{2}\sin\theta\left[1 + \frac{\cos\theta}{\sqrt{R^2 - \sin^2\theta}}\right] \tag{2.24}$$

Resistance to gas flow into the engine or stresses due to inertia of the moving parts limit the maximum mean piston speed to within the range 8 to 15 [m/s], where the lower end is typical of marine diesel engines.

c) Brake torque and power

Engine torque is normally measured with a dynamometer. The engine is clamped on a test bed and the shaft is connected to the dynamometer rotor. Figure 2.37 illustrates the operating principle of a dynamometer. The rotor is coupled electromagnetically, hydraulically, or by mechanical friction to a stator, which is supported in low friction bearings.

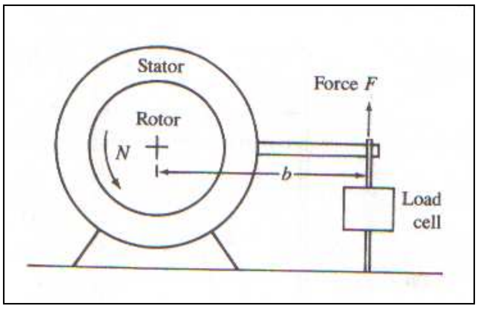


Fig. 2.37 – Schematic of operating principle of a dynamometer.

The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, springs, or pneumatic means.

Using the notation in Figure 2.37, if the torque exerted by the engine is:

$$T = Fb \tag{2.25}$$

, the power delivered P by the engine and absorbed by the dynamometer is the product of the torque by the angular speed:

$$P = 2\pi NT \tag{2.26}$$

The value of engine power measured as described above is called brake power, P_b.

d) Indicated work per cycle

Pressure data for the gas in the cylinder over the operating cycle of the engine can be used to calculate the work transfer from the gas to the piston. The cylinder pressure and corresponding cylinder volume throughout the engine cycle can be plotted on a p-V diagram as shown in Figure 2.38. The indicated work per cycle, $W_{c,i}$ (per cylinder) is obtained by integration around the curve to obtain the area enclosed on the diagram:

$$W_{c,i} = \oint pdV \tag{2.27}$$

With two-strokes cycles (Figure 2.38.a), the application of equation (2.27) is straightforward. With the addition of inlet and exhaust strokes for the four-stroke cycle, some ambiguity is introduced as two definitions of indicated output are in common use. These will be defined as:

d.1) Gross indicated work per cycle – which is the work delivered to the piston over the compression and expansion strokes only (area A + area C);

d.2) Net indicated work per cycle – which is the work delivered to the piston over the entire four-stroke cycle (area A – area B);

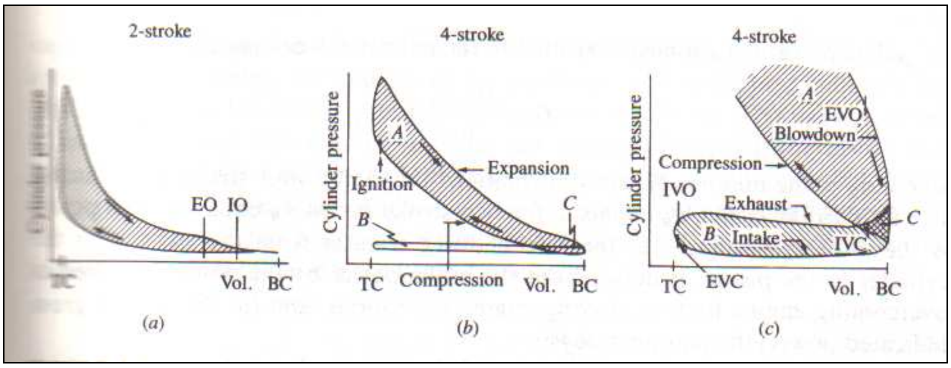


Fig. 2.38 – Examples of p-V diagrams.

Area B + area C is the work transfer between the piston and the cylinder gases during the inlet and exhaust strokes and is called the pumping work, W_p. This work will be different for aspirated and turbocharged engines.

The power per cylinder is related to the indicated work per cycle by:

$$P_i = \frac{W_{c,i} N}{n_R} \tag{2.28}$$

, where n_R is the number of crank revolutions for each power stroke per cylinder. For four-strokes cycles, n_R equals 2; For two-strokes cycles, n_R equals 1. This power is the indicated power, i.e., the rate of work transfer from the gas within the cylinder to the piston. It differs from the brake power by the power absorbed in overcoming engine friction, driving engine accessories, and (in the case of gross indicated power) the pumping power.

e) Mechanical efficiency

As discussed above, a part of the gross indicated work per cycle or power is used to expel exhaust gases and induct fresh charge. An additional portion is used to overcome the friction of the bearings, pistons, and other mechanical components of the engine, and to drive the engine accessories. All of these power requirements are grouped together and called friction power, P_f. Thus:

$$P_{ig} = P_b + P_f \tag{2.29}$$

, where P_{ig} is the indicated power.

The ratio of the brake (or useful) power delivered by the engine to the indicated power is called mechanical efficiency, η_m:

$$\eta_m = \frac{P_{ig}}{P_b} \tag{2.30}$$

f) Mean effective pressure

While torque is a valuable measure of a particular engine's ability to do work, it depends on engine size. Hence, a more useful relative engine performance measure is obtained by dividing the work per cycle by the cylinder volume displaced per cycle. The parameter so obtained has units of force per unit area and is called the mean effective pressure (*mep*). From equation (2.28), we obtain $W_{c,i} = \frac{Pn_R}{N}$, and therefore mean effective pressure is given by:

$$mep = \frac{Pn_R}{V_d N} \tag{2.31}$$

Typical values for *bmep* are as follows: for naturally aspirated four-stroke diesels, the maximum *bmep* is in the 700 to 900 [kPa] range. Turbocharged four-stroke diesel maximum *bmep* values are typically in the range 1000 to 1200 [kPa]. Large low-speed two-stroke diesel engines can achieve *bmep* values of about 1600 [kPa]. Brake mean effective pressure or *bmep* is, as usual, calculated from measured dynamometer torque. Indicated mean effective pressure or *imep* is calculated using the indicated power; i.e., the pressure volume integral in the work per cycle equation. Sometimes the term *fmep* (friction mean effective pressure) is used as an indicator of the mean effective pressure lost to friction (or friction torque) and is just the difference between *imep* and *bmep*

g) Specific fuel consumption and efficiency

In engine tests, the fuel consumption is measured as a flow-rate – mass flow per unit time, $\dot{m}_f$. A more useful parameter is the specific fuel consumption (*sfc*) – the fuel flow rate per unit power output. It measures how efficiently an engine is using the fuel supplied to produce work:

$$sfc = \frac{\dot{m}_f}{P} \tag{2.32}$$

Low values of *sfc* are obviously desirable. For CI engines, best values are lower and in large engines can go below 55 [µg/J] = 200 [g/kW.hr].

The fuel energy supplied which can be released by combustion is given by the mass of fuel supplied to the engine per cycle times the heating value of the fuel, Q_{HV}, defines its energy content.

The engine's "efficiency", which will be called the fuel conversion efficiency, η_f, is given by:

$$\eta_f = \frac{W_c}{m_f Q_{HV}} = \frac{\left(Pn_R/N\right)}{\left(\dot{m}_f n_R/N\right)Q_{HV}} = \frac{P}{\dot{m}_f Q_{HV}} \tag{2.33}$$

Substituting equation (2.32) into (2.33) we obtain:

$$\eta_f = \frac{1}{sfc Q_{HV}} \tag{2.34}$$

Typical heating values, Q_{HV}, for the commercial hydrocarbon fuels used in engines are in the range 42 to 44 [MJ/Kg]. Thus specific fuel consumption is inversely proportional to fuel consumption efficiency for normal hydrocarbon fuels. Notice that the fuel energy supplied to the engine per cycle is not fully released as thermal energy in the combustion process because the actual combustion process is incomplete.

h) Air/fuel ratios

In engine testing, both the air mass flow $\dot{m}_a$ and fuel mass flow rate $\dot{m}_f$ are normally measured. The ratio of these flow rates is useful in defining engine operation conditions:

h.1) Air/fuel ratio (A/F):

$$A/F = \frac{\dot{m}_a}{\dot{m}_f} \tag{2.35}$$

h.2) Fuel/air ratio (F/A):

$$F/A = \frac{\dot{m}_f}{\dot{m}_a} \tag{2.36}$$

i) Volumetric efficiency

The intake system – the air filter, carburetor, and throttle plate (SI), intake manifold, intake port valve (CI) – restricts the amount of air which the engine of given displacement can induct. The parameter used to measure the effectiveness of an engine's induction process is the volumetric efficiency, η_v. Volumetric efficiency is only used with four-stroke cycle engines which have a distinct induction process. It is defined as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston:

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} V_d N} \tag{2.37}$$

, where $\rho_{a,i}$ is the inlet air density. An alternative equivalent definition for volumetric efficiency is given by:

$$\eta_v = \frac{m_a}{\rho_{a,i} V_d} \tag{2.38}$$

, where m_a is the mass of air inducted into the cylinder per cycle.

Typical maximum volumetric efficiencies for naturally aspirated engines are in the range 80 to 90%. The volumetric efficiencies for diesel engines is somewhat higher than for SI engines.

j) Engine specific weight and specific volume

Engine weight and bulk volume for a given rated power are important in marine applications. Two parameters useful for comparing these attributes from one engine to another are:

j.1) Specific weight = engine weight/rated power

j.2) Specific volume = engine volume/rated power

For these two parameters to be useful in engine comparisons, a consistent definition of what components and auxiliaries are included in the term "engine" must be adhered to. These parameters indicate the effectiveness with which the engine designer has used the engine materials and packaged the engine components.

k) Specific emissions and emissions index

Levels of emissions oxides of nitrogen (nitric oxide, NO, and nitrogen dioxide, NO_2, usually grouped together as NO_x), carbon monoxide (CO), unburned hydrocarbons (HC), and particles are important engine operating characteristics.

The concentrations of gaseous emissions in the engine exhaust gases are usually measured in parts per million or percent by volume. Normalized indicators of emissions levels are more useful, however, and two of these are in common use. Specific emissions are the mass flow rate of pollutant per unit power output:

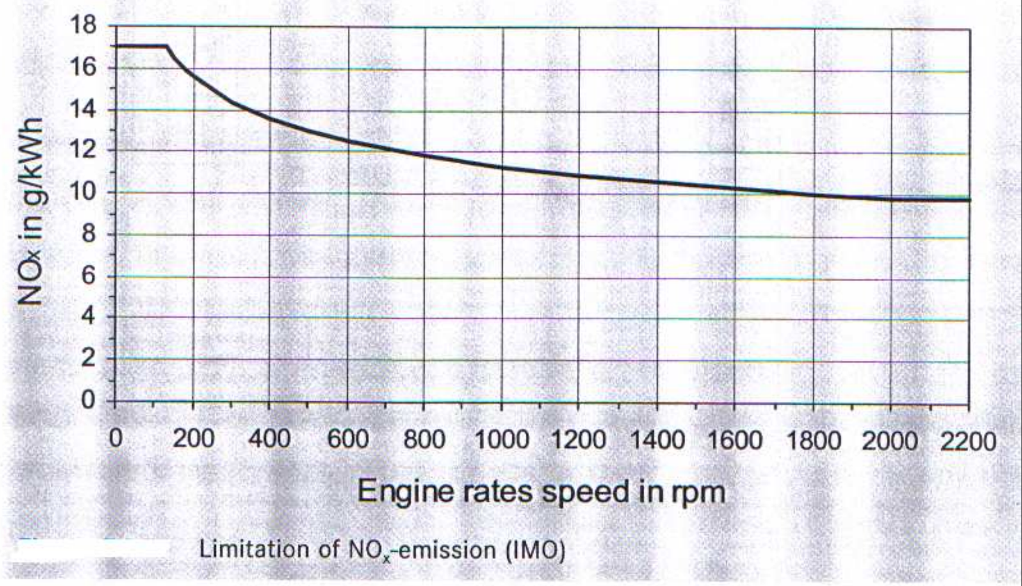
$$sNO_x = \frac{\dot{m}_{NO_x}}{P} \tag{2.39}$$

$$sCO = \frac{\dot{m}_{CO}}{P} \tag{2.40}$$

$$sHC = \frac{\dot{m}_{HC}}{P} \tag{2.41}$$

$$sPart = \frac{\dot{m}_{Part}}{P} \tag{2.42}$$

Indicated and brake specific emissions can be defined. Units in common use are [μg/J] or [g/kW.hr].



The IMO NO_x emission limit depends on the rated diesel engine speed:

$n < 130\ min^{-1}$	$NO_x = 17\ g/kWh$
$n = 130$ to $< 2000\ min^{-1}$	$NO_x = 45\ n^{-0.2}\ g/kWh$
$n \geq 2000\ min^{-1}$	$NO_x = 9.8\ g/kWh$

Fig. 2.39 – Limitation of NOx emissions (IMO).

l) Acoustical emissions

Low noise onboard of yacht, passenger vessels and on naval vessels is an important demand. Noise spectra, i.e. frequency analysis for operating noises, where a clear distinction is made between the different noise sources. As illustrated in Figure 2.40, the noise sources are:

l.1) Air-borne noise, as:

l.1.1) diesel engine free-field noise (see Figure 2.41);

A noise spectrum of the diesel engine operating noise emitted to the environment (free-field) is usually made available by the manufacturer for each diesel engine. These spectra are available on request for project contract-specific purposes. The figures in the noise spectrum are in dB(A) and comply with applicable ISO standards. The datum level is 2.10^{-6} [Pa] and the noise pressures are measured at a distance of 1 meter from the engine block.

l.1.2) undamped exhaust noise (see Figure 2.42);

l.1.3) undamped air intake noise;

To minimize air–borne level the diesel engines can be provided with a noise enclosure, as shown in Figure 2.43.

l.2) Structure-borne noise (see Figure 2.44).

Depending on owner requirements, different “quite systems” will be additionally offered by engine manufacturers. Namely:

- Standard single resilient mounting system;

- Single resilient mounting system with shock;

- Standard double resilient mounting system;

- Double resilient mounting system for low noise;

- Double resilient mounting system for extreme acoustic requirements.

Figure 2.45 shows some examples of different “quiet systems” in respect to a standard, i.e. structure-borne noise levels below the resilient mounting.

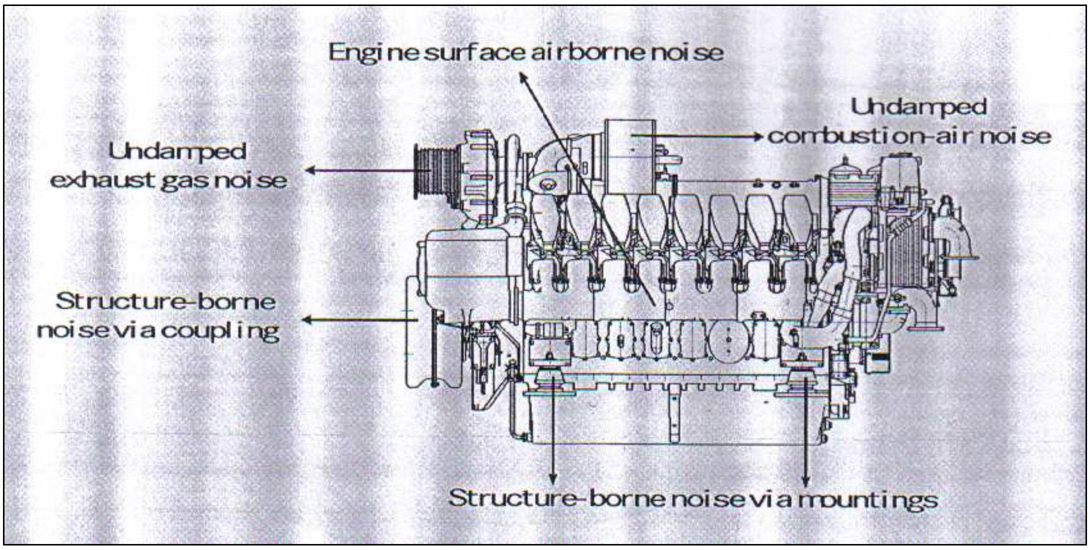
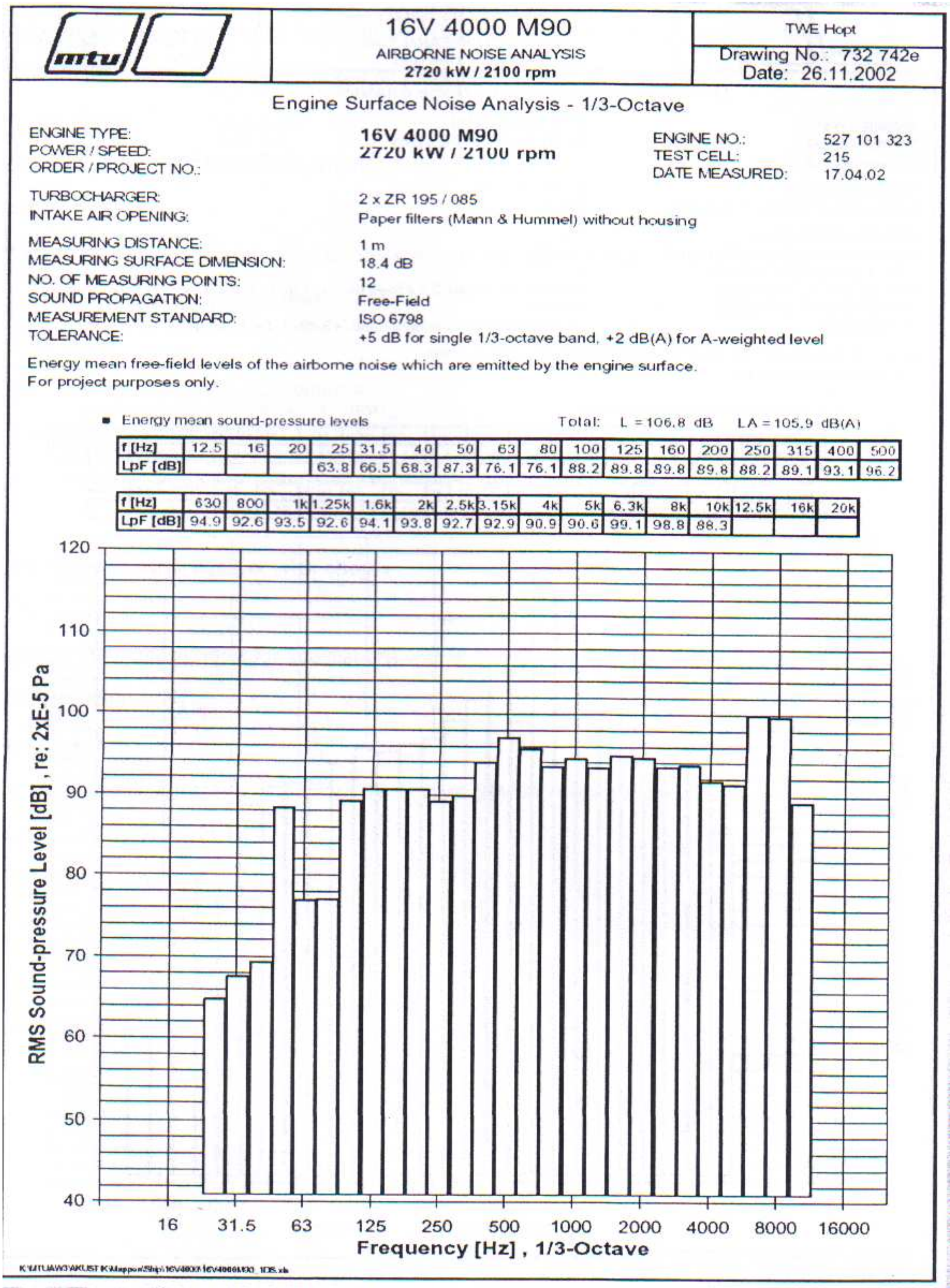


Fig. 2.40 – Diesel engine noise sources.



16V 4000 M90
AIRBORNE NOISE ANALYSIS
2720 kW / 2100 rpm

TWE Hopt
Drawing No.: 732 742e
Date: 26.11.2002

Engine Surface Noise Analysis - 1/3-Octave

ENGINE TYPE: 16V 4000 M90
POWER / SPEED: 2720 kW / 2100 rpm
ORDER / PROJECT NO.:
ENGINE NO.: 527 101 323
TEST CELL: 215
DATE MEASURED: 17.04.02

TURBOCHARGER: 2 x ZR 195 / 085
INTAKE AIR OPENING: Paper filters (Mann & Hummel) without housing

MEASURING DISTANCE: 1 m
MEASURING SURFACE DIMENSION: 18.4 dB
NO. OF MEASURING POINTS: 12
SOUND PROPAGATION: Free-Field
MEASUREMENT STANDARD: ISO 6798
TOLERANCE: +5 dB for single 1/3-octave band, +2 dB(A) for A-weighted level

Energy mean free-field levels of the airborne noise which are emitted by the engine surface.
For project purposes only.

▪ Energy mean sound-pressure levels Total: L = 106.8 dB LA = 105.9 dB(A)

f [Hz]	12.5	16	20	25	31.5	40	50	63	80	100	125	160	200	250	315	400	500
LpF [dB]				63.8	66.5	68.3	87.3	76.1	76.1	88.2	89.8	89.8	89.8	88.2	89.1	93.1	96.2

f [Hz]	630	800	1k	1.25k	1.6k	2k	2.5k	3.15k	4k	5k	6.3k	8k	10k	12.5k	16k	20k
LpF [dB]	94.9	92.6	93.5	92.6	94.1	93.8	92.7	92.9	90.9	90.6	99.1	98.8	88.3			

Fig. 2.41 – Diesel engine surface noise analysis.

mtu	16V 4000 M90 AIRBORNE NOISE ANALYSIS **2720 kW / 2100 rpm**	TWE Hopt Drawing No.: 732 741e Date: 26.11.2002

Undampened Exhaust Gas Noise Analysis - 1/3-Octave

ENGINE TYPE:	**16V 4000 M90**	ENGINE NO.:	527 101 323
POWER / SPEED:	**2720 kW / 2100 rpm**	TEST CELL:	215
ORDER / PROJECT NO.:		DATE MEASURED:	17.04.02

TURBOCHARGER:	2 x ZR 195 / 085
DIAMETER OF EXHAUST GAS PIPE:	400mm
MEASURING DISTANCE:	1 m
MEASURING SURFACE DIMENSION:	12.6 dB
NO. OF MEASURING POINTS:	2
PIPE DESIGN AT TEST CELL:	Manifold, 1 x 90deg, 2 x 45deg bends, 11m straight pipe, horizontal outlet
MEASUREMENT STANDARD:	ISO 6798
TOLERANCE:	+5 dB for single 1/3-octave band, +3 dB(A) for A-weighted level

Energy mean free-field levels of the undampened exhaust gas noise.
For project purposes only.

- Energy mean sound-pressure levels   Total: L = 127.9 dB   LA = 115.6 dB(A)

f [Hz]	12.5	16	20	25	31.5	40	50	63	80	100	125	160	200	250	315	400	500
LpF [dB]				92.5	102.9	102.2	97.7	101.2	111.1	114.1	124.4	124.0	106.2	113.5	109.0	06.3	04.0

f [Hz]	630	800	1k	1.25k	1.6k	2k	2.5k	3.15k	4k	5k	6.3k	8k	10k	12.5k	16k	20k
LpF [dB]	103.4	102.4	102.8	101.0	98.5	98.5	98.0	95.8	91.4	87.4	81.7	74.9	69.8			

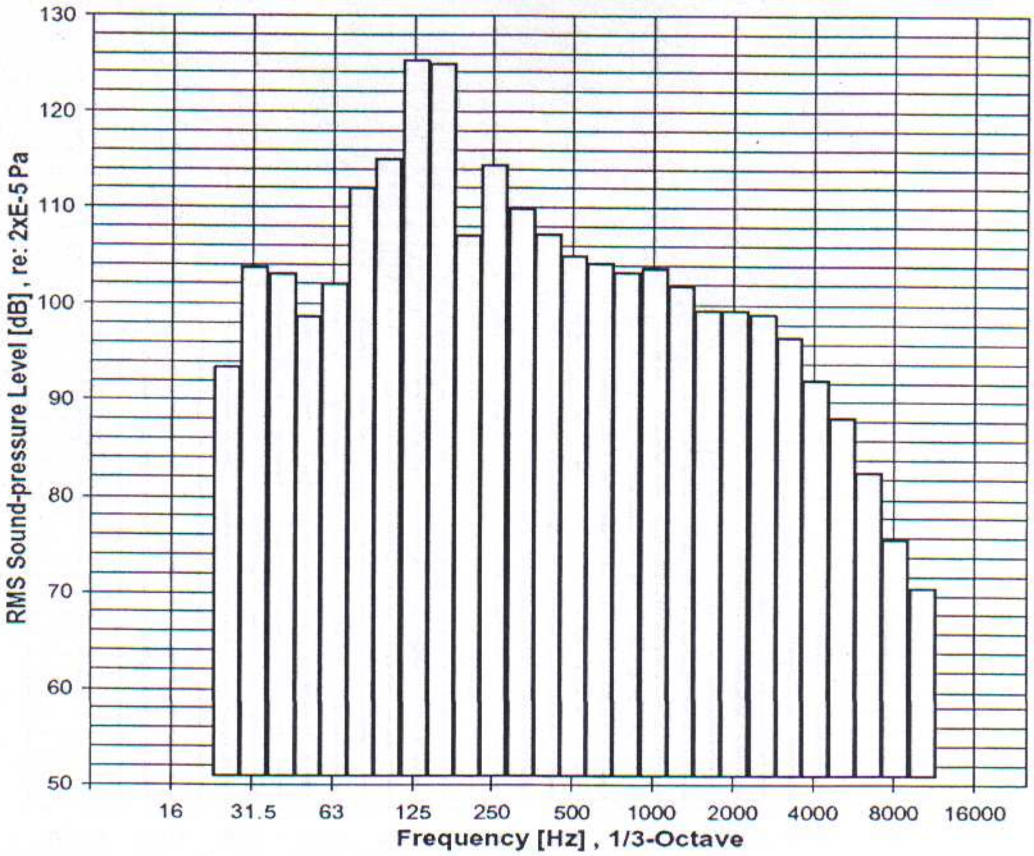


Fig. 2.42 – Diesel engine exhaust gas noise analysis.

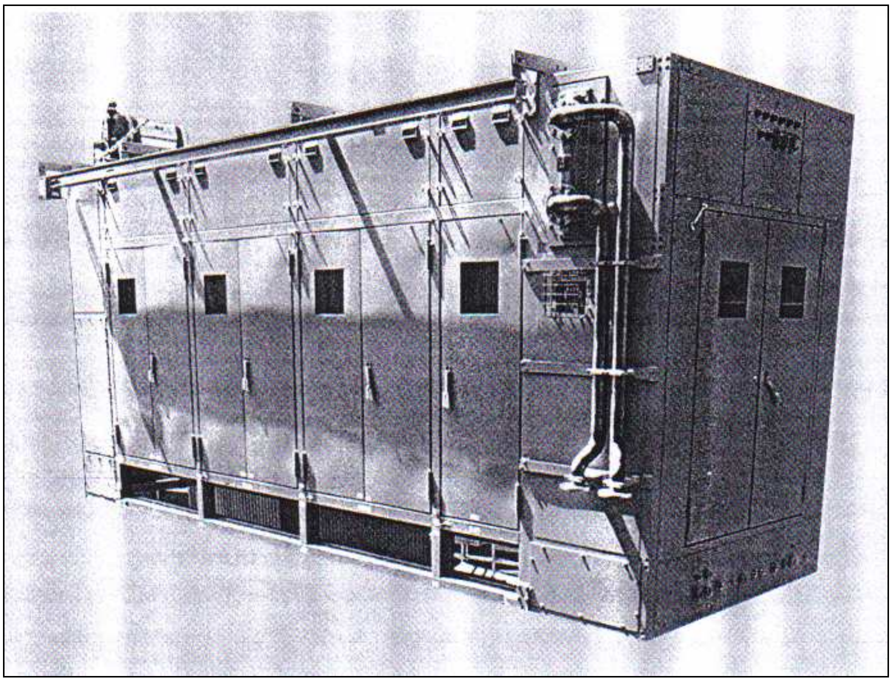


Fig. 2.43 – Diesel engine with noise enclosure.

mtu	16V 4000 M90 STRUCTURE-BORNE NOISE ANALYSIS **2720 kW / 2100 rpm**	TWE Hopt Drawing No.: 732 740e Date: 26.11.2002

Mean Velocity Analysis - 1/3-Octave

ENGINE TYPE:	**16V 4000 M90**	ENGINE NO.:	527 101 323
POWER / SPEED:	**2720 kW / 2100 rpm**	TEST CELL:	215
ORDER / PROJECT NO.:		DATE MEASURED:	17.04.02
MOUNTING:	4 rubber mounts RD 314 (60 shore)		
COUPLING:			
MEASURING DIRECTION:	vertical		
MEASURING STAGE:	At engine feet, above rubber mounts		
NO. OF MEASURING POINTS:	4		
MEASUREMENT STANDARD:	ISO 13 332		
TOLERANCE:	+5 dB for 1/3-octave band, +2 dB for mean excess		

Energy mean velocity analysis of structure-borne noise at engine feet above the rubber mounts.
For project purposes only

- Energy mean velocity levels, vertical Total: L = 110.9 dB

f [Hz]	12.5	16	20	25	31.5	40	50	63	80	100	125	160	200	250	315	400	500
Lv [dB]				75.4	85.1	82.4	94.5	82.7	95.5	99.7	103.5	101.1	95.3	89.7	94.4	103.1	105.0

f [Hz]	630	800	1k	1.25k	1.6k	2k	2.5k	3.15k	4k	5k	6.3k	8k	10k	12.5k	16k	20k
Lv [dB]	98.2	94.3	89.9	89.0	87.3	85.0	84.8	86.7	80.7	81.3	76.3	67.5	64.7			

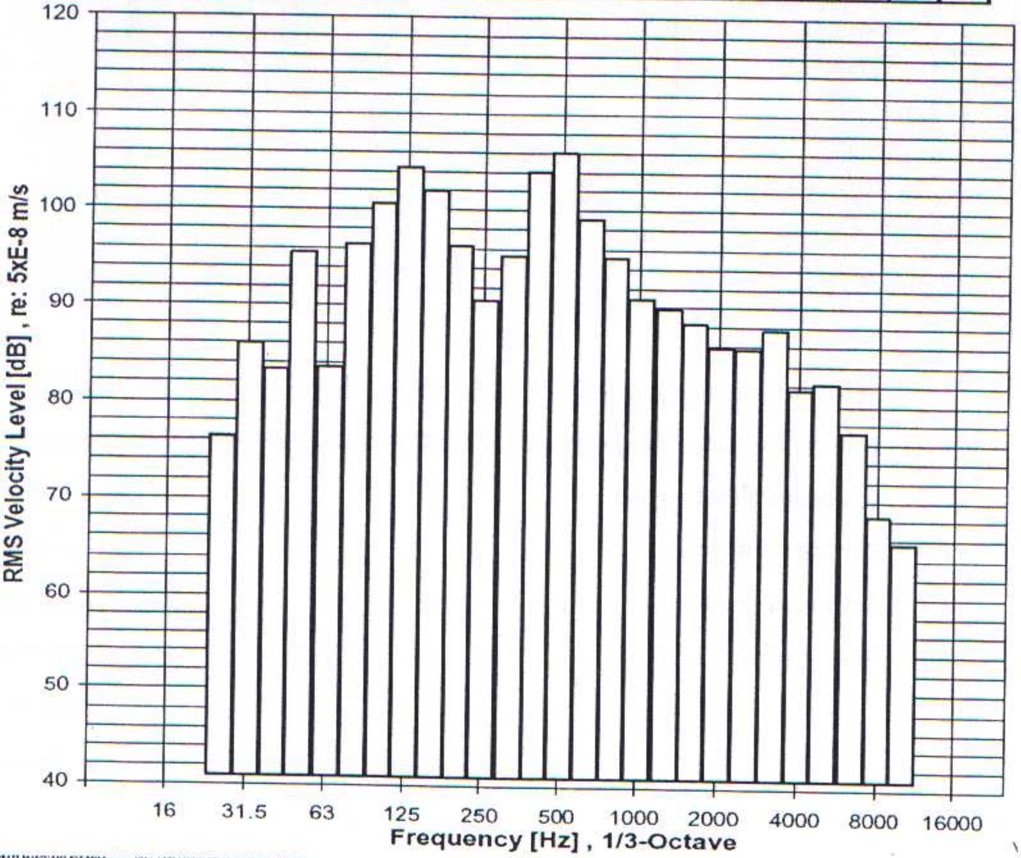


Fig. 2.44 – Structure-borne noise analysis: diesel engine feet above rubber mounts.

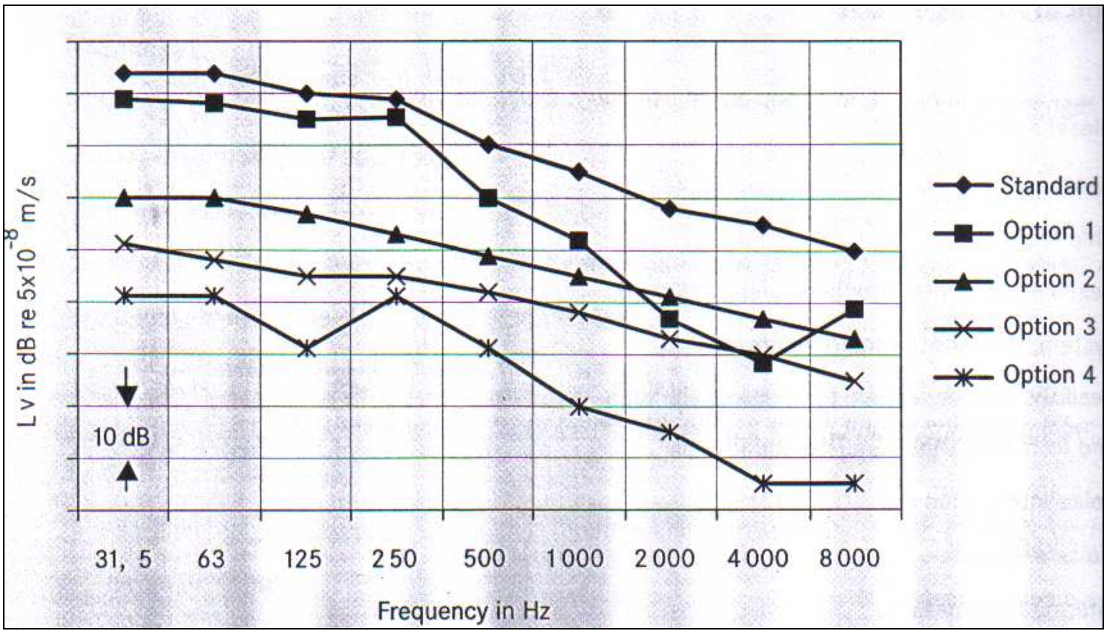


Fig. 2.45 – Four different "quiet systems" above standard.

m) Relations between performance parameters

The importance of the parameters above defined to engine performance becomes evident when power, torque, and mean effective pressure are expressed in terms of these parameters.

For power P:

$$P = \frac{\eta_f m_a N Q_{HV} (F/A)}{n_R} \tag{2.43}$$

For four-stroke diesel engines, volumetric efficiency can be introduced:

$$P = \frac{\eta_f \eta_v N V_d Q_{HV} \rho_{a,i} (F/A)}{2} \tag{2.44}$$

For torque T:

$$T = \frac{\eta_f \eta_v V_d Q_{HV} \rho_{a,i} (F/A)}{4\pi} \tag{2.45}$$

For mean effective pressure *mep*:

$$mep = \eta_f \eta_v Q_{HV} \rho_{a,i} (F/A) \tag{2.46}$$

The power per unit piston area, often called the specific power, is a measure of the engine designer's success in using the available piston area regardless of cylinder size. From equation (2.44), the specific power is given by:

$$\frac{P}{A_P} = \frac{\eta_f \eta_v N L Q_{HV} \rho_{a,i} (F/A)}{2} \tag{2.47}$$

Mean piston speed can be introduced with equation (2.24) to obtain:

$$\frac{P}{A_P} = \frac{\eta_f \eta_v \overline{S}_p Q_{HV} \rho_{a,i} (F/A)}{4} \tag{2.48}$$

Specific power is thus proportional to the product of mean effective pressure and mean piston speed.

These relationships illustrate the direct importance to engine performance of:

m.1) High fuel conversion efficiency;

m.2) High volumetric efficiency;

m.3) Increasing the output of a given displacement engine by increasing the inlet air density;

m.4) Maximum fuel/air ratio that can be usefully burned in the engine;

m.5) High mean piston speed.

n) Engine design and performance data

Engine ratings usually indicate the highest power at which manufacturers expect their products to give satisfactory economy, reliability, and durability under service conditions. Maximum torque, and the speed at which it is achieved, is usually given also. Since both of these quantities depend on displacement volume, for comparative analyses between engines of different displacements in a given category normalized performance parameters are more useful. The following measures, at the operating points indicated, have most significance:

n.1) at maximum or normal rated point:

Mean piston speed – measures comparative success in handling loads due to inertia of the parts and/or friction;

Brake mean effective pressure – in naturally aspirated engines bmep is not stress limited. It then reflects the product of volumetric efficiency, fuel/air ratio and fuel conversion efficiency. In supercharged engines bmep reflects the degree of success in handling higher gas pressures and thermal loading;

Power per unit piston area – measures the effectiveness with which the piston area is used, regardless the cylinder size;

Specific weight – indicates relative economy with which materials are used;

Specific volume – indicates relative economy with which engine space has been used.

n.2) At all speeds at which the engine will be used with full throttle or with maximum fuel-pump setting:

Brake mean effective pressure – measures ability to obtain/provide high air flow and use it effectively over the full range.

n.3) At all useful regimes of operation and particularly in those regimes where the engine is run for long periods of time:

Brake specific fuel consumption;

Brake specific gas and acoustic emissions.

2.6.2. Gearboxes

Introduction:

Steam turbines operate at speeds up to 6000 [rpm]. Medium-speed diesel engines operate up to about 750 [rpm]. The best propeller speed for efficient operation is in the region of 80 to 100 [rpm]. The turbine or engine shaft speed is reduced to that of the propeller by the use of a system of gearing. Their use also permits more than one prime mover to be coupled to the same propeller.

Gearboxes are basically constituted by pinions and gear wheels connected by transmission shafts, which are supported by bearings fixed to the gear case. The bearing may be of the type journal or roller bearing. Roller bearings are usually a self contained piece and therefore their replacement involves dismounting the gear case to remove gear wheels and transmission shafts outside, which represents a major problem for repair of gearboxes with large dimensions. Journal plain bearings are more easily repaired since the anti-friction bearing shells can be rapidly replaced.

In order to reduce friction onto the moving parts of a gearbox there is a clear need for lubrication. The lubricating oil usually is pressurized by a pump and circulates along the bearings, is sprayed over the pinion and gear teeth and then is drained to the oil sump on a closed circuit fitted with a cleaning filter. Hence, the lubricating oil must be contained within the gear case, which should be fitted with seals or glands on the bearings where input and output shafts are installed. For naval applications, it is standard practice to drive the main lubricating oil pump by a train of gearing taking its power from one of the intermediate shafts. This so called “attached pump” furnishes oil for all purposes during normal operations; however, separately electric driven pumps are required for low-speed, standby, and astern operation because the attached pump cannot supply an adequate oil flow during these conditions. An attached pump has the advantage of protecting against the loss of oil supply due to an interruption of electric power or the inadvertent securing of a motor-driven pump. This feature, however, is seldom applied in merchant service. A secondary function of the circulating lubricant is to carry away the heat losses of the gearing and its bearings and also prevent rusting of the interior surfaces of the gear. Therefore, a cooling circuit provided with a heat exchanger (cooler) is installed onboard, where hot lubricating oil is cooled by sea water.

The reverse mode on reverse/reduction gearbox can be attained by various arrangements depending on the required location of the engine input or drive shaft and the driven or output shaft. Figure 2.46 shows a simplified, flat arrangement for ease explanation, and Figures 2.46a, b and c show different sectional views of a typical reverse/reduction gearbox.

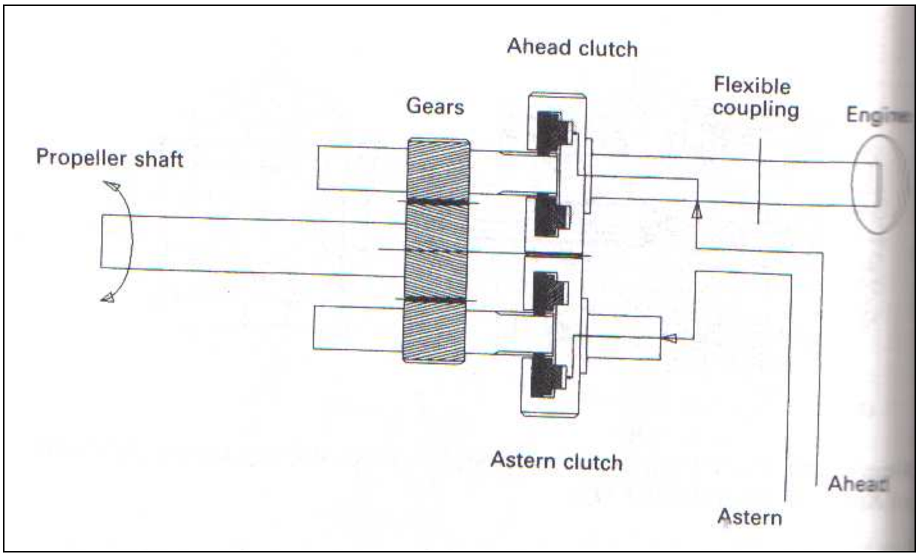


Fig. 2.46 – Typical reverse/reduction gearbox arrangement.

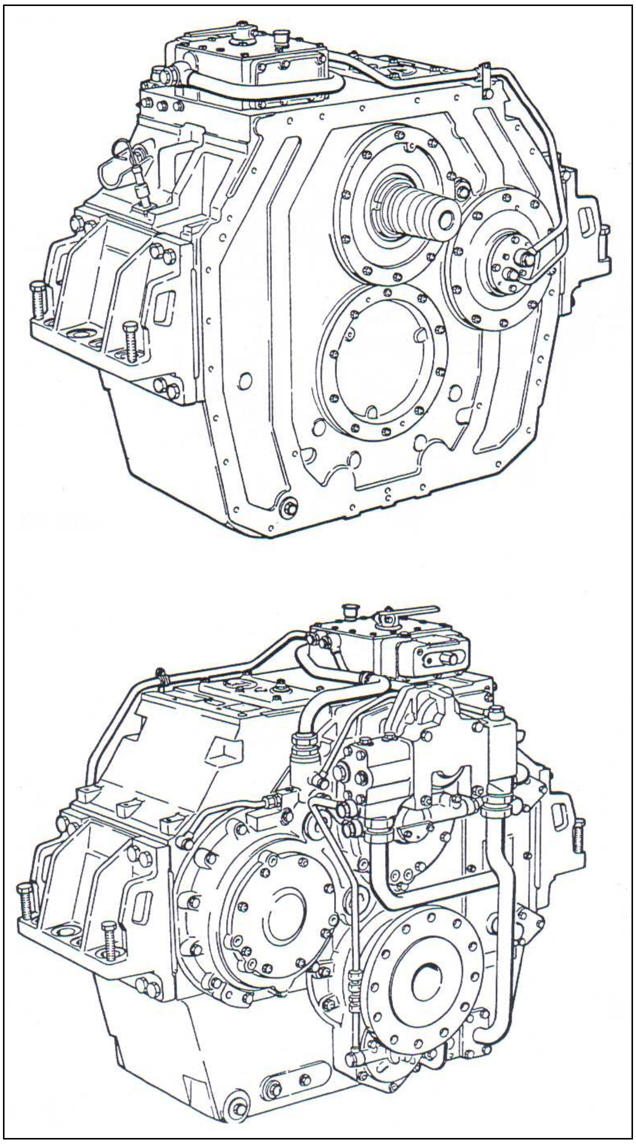


Fig. 2.46.a – Typical reverse/reduction gearbox arrangement.

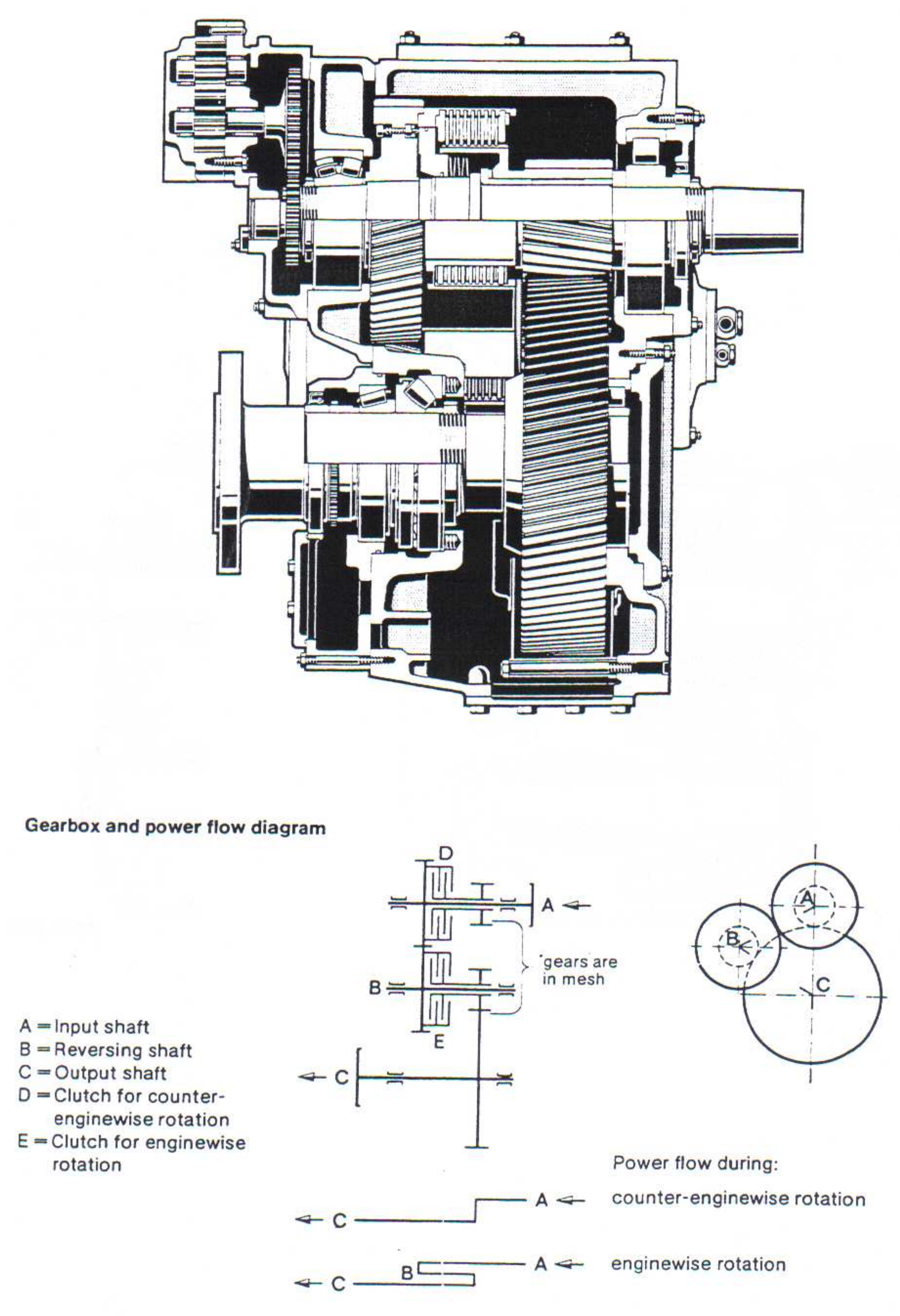


Fig. 2.46.b – Typical reverse/reduction gearbox arrangement.

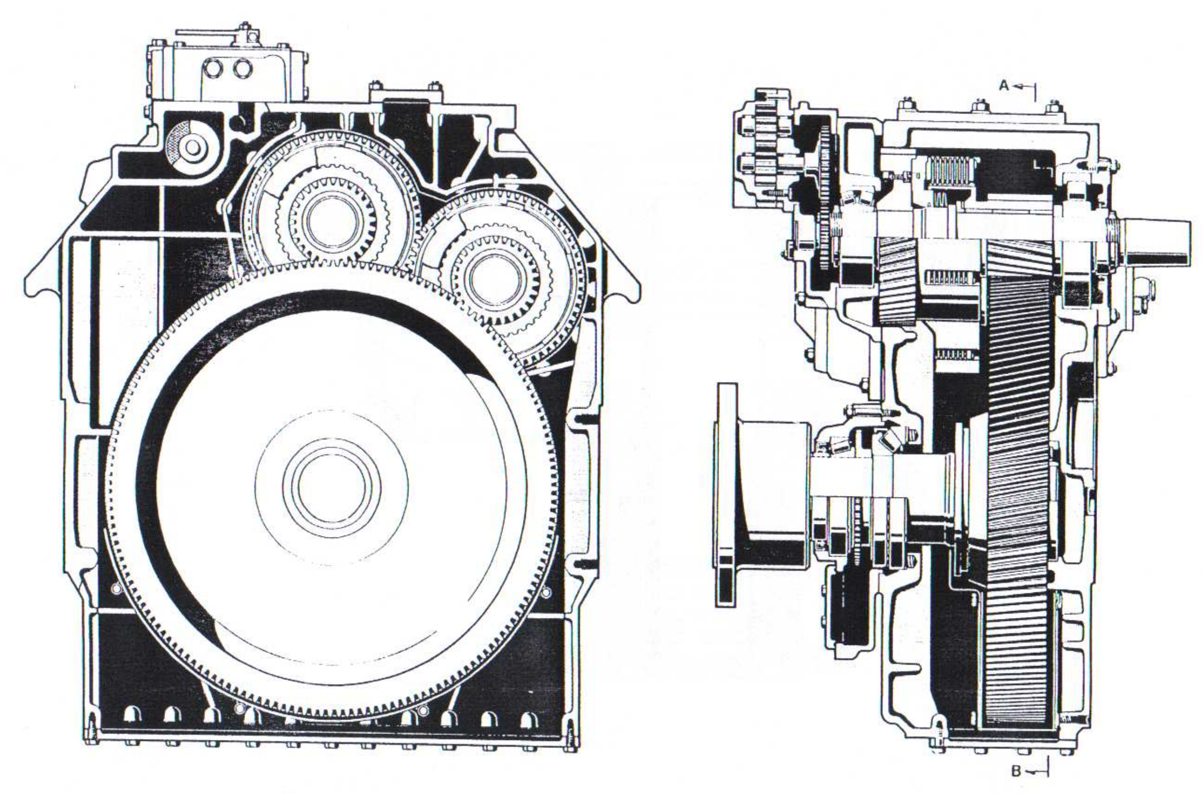


Fig. 2.46.c – Typical reverse/reduction gearbox arrangement.

The drive from the engine input shaft to the counter shaft is through teeth on the outsides of both clutch housings, which are in continuous mesh. When control lever is set for ahead running, the control valve supplies oil pressure to the ring piston of the ahead clutch. When control lever is set for astern running, the control valve supplies oil pressure to the ring piston of the astern clutch. When either clutch is engaged, its pinion provides a drive to the large gear wheel of the driven shaft and the other pinion rotates freely. Oil pressure required for the clutch operation is built up by a gear pump driven from the input shaft.

The propeller thrust on the driven shaft is taken up by the thrust bearing, which might be of the built-in type or placed outside the gearbox. The driven (propeller) shaft, for ahead running, rotates in the opposite direction to the drive shaft or input shaft. For astern running, the driven (propeller) shaft rotates in the same direction as the drive shaft. To stop the propeller shaft, the control is moved to the neutral position and both clutches are disengaged.

There are different types of gearboxes with more than one input shaft connected to prime movers whose output shaft is to be coupled to the same propeller. These reduction gearboxes may also have more than one output shaft, which can be coupled to electric generators or hydraulic pumps. These are usually designated as power take-offs.

Gearboxes manufacturers, based on pre-defined user requirements and joint selection process usually supply the gearbox and all the accessories as a compact unit in standard sizes.

Selection of Gearbox:

Exception for particular applications, gearboxes are available from manufacturers in standard sizes. Main selection parameters to be considered are:

- power to be transmitted;

- maximum rotational speed of the input shaft;
- reduction ratio;
- number of input shafts;
- number of power-take-offs and required power and speed;
- orientation and sense of rotation of input and output shafts;
- reverse rotation requirements;
- built-in thrust bearing requirements.

Other selection elements to be considered during ship design are:

- weight and dimensions;
- position of input and output flanges relatively to gearbox seats;
- location of oil lubricating drain and level indicator;
- depth of the bottom casing relatively to gearbox seats level;
- cooling requirements;
- supply of external electrical power requirements.

Nomenclature of Gear Teeth:

The terminology of gear teeth is illustrated in Figure 2.47. The following basic definitions and equations should be considered:

- The *pitch circle* is a theoretical circle upon which all the calculations are usually based. The pitch circles of a pair of mating gears are tangent to each other;
- A *pinion* is the smaller of two mating gears. The larger is often called the *gear*;
- The *circular pitch* (p) is the distance, measured on the pitch circle, from a point on one tooth to a corresponding point on an adjacent tooth. Thus the circular pitch is equal to the sum of the *tooth thickness* and the *width of space*;
- The *module* (m) is the ratio of the pitch diameter to the number of teeth. The customary unit of length used is the millimeter. The module is the index of tooth size in SI. A pair of meshing gears, consisting of a pinion and a gear, must have exactly the same module, of course, to mesh properly;
- The *addendum* (a) is the radial distance between the *top land* and the pitch circle;
- The *dedendum* (b) is the radial distance between the *bottom land* and the pitch circle;
- The *whole depth* (h_t) is the sum of the addendum and dedendum.

- The *clearance circle* is a circle that is tangent to the addendum circle of the matching gear. The *clearance* (c) is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear;

- The *backlash* is the amount by which the width of a tooth space exceeds the thickness of the engaging tooth measured on the pitch circle.

$$m = \frac{d}{N} \tag{2.49}$$

, where:

m = module, [mm];

d = pitch diameter, [mm];

N = number of teeth.

$$p = \frac{\pi d}{N} = \pi m \tag{2.50}$$

, where:

p = circular pitch, [mm];

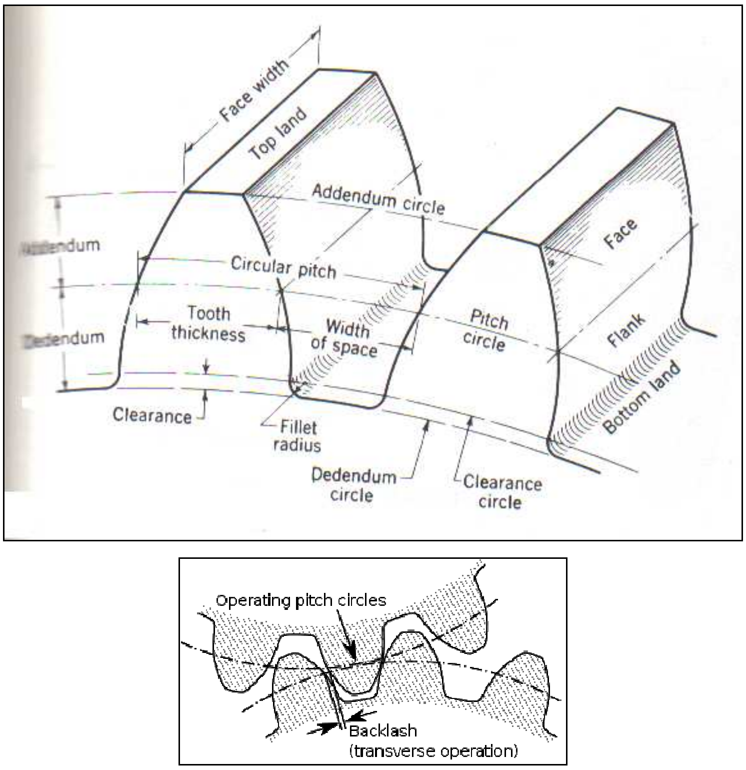


Fig. 2.47 – Nomenclature of gear teeth.

<u>Force Analysis on Gear Teeth:</u>

The notation to be used for force analysis on gear teeth is as follows: beginning with numeral 1 for the frame of the machine, we shall designate the input gear as gear 2, and then number of gears successively 3, 4, etc., until we arrive at the last gear in the train. Next, there may be several shafts involved, and usually one or two gears are mounted on each shaft as well as other elements. We shall designate the shafts, using lowercase letters of the alphabet, a, b, c, etc.

With this notation we can now speak of force exerted by gear 2 against gear 3 as F_{23}. The force of gear 2 against shaft *a* is F_{2a}. We can also write F_{a2} to mean the force of the shaft against gear 2. Unfortunately, it is also necessary to use subscripts to indicate directions. The coordinate directions will usually be indicated by x, y, z coordinates, and the radial and tangential directions by subscripts r and t. With this notation F_{43}^{t} is the tangential component of the force of gear 4 acting against gear 3.

Figure 2.48.a shows a pinion mounted on a shaft a rotating clockwise at n_2 speed and driving a gear on shaft *b* at n_3. The reactions between the mating teeth occur along the pressure line. In Figure 2.48.b the pinion has been separated from the gear and from the shaft, and their effects replaced by forces. F_{a2} and T_{a2} are the force and torque, respectively, exerted by gear 3 against the shaft *a* pinion 2. F_{32} is the force exerted by gear 3 against the pinion. Using a similar approach, we obtain the free-body diagram of the gear shown in Figure 2.48.c.

In Figure 2.49 the free-body diagram of the pinion has been redrawn and the forces resolved into tangential and radial components. We now define the transmitted load (W_t) as:

$$W_t = F_{32}^{t} \tag{2.51}$$

The transmitted load is really the useful component because the radial component serves no useful purpose. It does not transmit power. The applied torque (*T)* and the transmitted load are seen to be related by the following equation:

$$T = \frac{d}{2} W_t \tag{2.52}$$

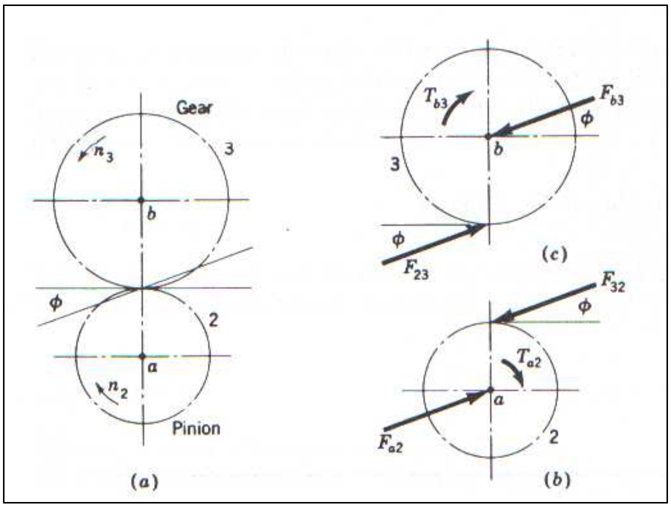


Fig. 2.48 – Free-body diagram of the forces acting upon two gears in a simple gear train.

The pitch line velocity (*V*) is given by:

$$V = \frac{\pi d n}{60} \tag{2.53}$$

, where:

V = pitch-line velocity, [m/s];

d = pitch diameter, [m];

n = speed, [rpm].

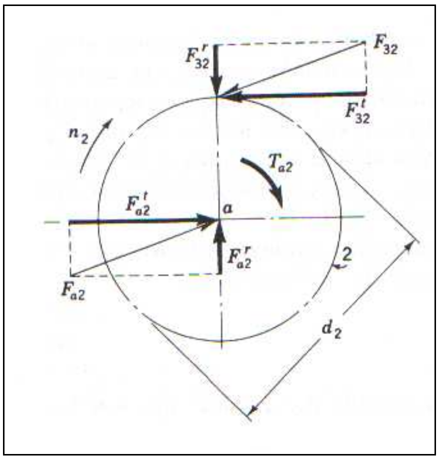


Fig. 2.49 – Resolution of gear forces.

With these units, the tangential load (W_t) can be also obtained from the equation below:

$$H = W_t V \tag{2.54}$$

, where H is the power transmitted in watts.

<u>Tooth Stresses:</u>

The following considerations must be treated as important limiting design factors in specifying the capacity of any gear drive:

- the heat generated during operation;
- failure of the teeth by breakage;
- fatigue failure of the tooth surfaces;
- abrasive wear of the tooth surfaces;
- noise as a result of high speeds, heavy loads, or mounting inaccuracies.

In these notes only static failures due to bending stress, dynamic effects on bending stress, and surface failure due to contact or Hertzian stress will be considered. Therefore, cumulative fatigue effects on bending and surface loads will not be discussed.

a) Bending stress

The particular purpose of these notes is to obtain relations for the bending stresses in a loaded tooth. Wilfred Lewis presented a formula for computing the bending stress in gear teeth in which the tooth form entered into equation. The formula was announced in 1892 and it still remains the basis for most design today.

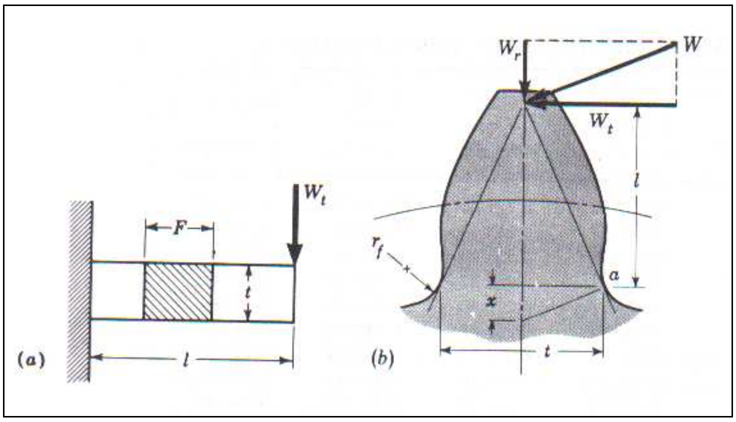


Fig. 2.50 – Cantilever beam analogy for tooth bending stress.

To derive the basic Lewis equation, refer to Figure 2.50 which shows a cantilever of cross-sectional dimensions *F* and *t*, having a length *l* and a load W_t uniformly distributed across the distance *F*. The section modulus is $I/c = Ft^2/6$ and therefore the bending stress is given by:

$$\sigma = \frac{M}{I/c} = \frac{6W_t l}{Ft^2} \tag{2.55}$$

Referring to Figure 2.50.b, we assume that the maximum stress in a gear tooth occurs at a point a. By similar triangles, we can write:

$$\frac{t/2}{x} = \frac{l}{t/2} \text{ or } x = \frac{t^2}{4l} \tag{2.56}$$

If we now substitute the value of *x* from equation (2.56) into equation (2.55) and multiply the numerator and denominator by the circular pitch *p*, we find:

$$\sigma = \frac{W_t}{F\left(\frac{2}{3}\right)xp} \tag{2.57}$$

Letting $y = 2x/3p$ and substituting $m = p/\pi$ and $Y = \pi y$, we obtain:

$$\sigma = \frac{W_t}{FmY} \tag{2.58}$$

Table 2.3 below shows the values of Y.

Tab. 2.3 – Lewis form factor for tooth design.

Number of teeth	$\phi = 20°$ $a = 0.8m$* $b = m$	$\phi = 20°$ $a = m$ $b = 1.25m$	$\phi = 25°$ $a = m$ $b = 1.25m$	$\phi = 25°$ $a = m$ $b = 1.35m$†
12	0.335 12	0.229 60	0.276 77	0.254 73
13	0.348 27	0.243 17	0.292 81	0.271 77
14	0.359 85	0.255 30	0.307 17	0.287 11
15	0.370 13	0.266 22	0.320 09	0.301 00
16	0.379 31	0.276 10	0.331 78	0.313 63
17	0.387 57	0.285 08	0.342 40	0.325 17
18	0.395 02	0.293 27	0.352 10	0.335 74
19	0.401 79	0.300 78	0.360 99	0.345 46
20	0.407 97	0.307 69	0.369 16	0.354 44
21	0.413 63	0.314 06	0.376 71	0.362 76
22	0.418 83	0.319 97	0.383 70	0.370 48
24	0.428 06	0.330 56	0.396 24	0.384 39
26	0.436 01	0.339 79	0.407 17	0.396 57
28	0.442 94	0.347 90	0.416 78	0.407 33
30	0.449 02	0.355 10	0.425 30	0.416 91
34	0.459 20	0.367 31	0.439 76	0.433 23
38	0.467 40	0.377 27	0.451 56	0.446 63
45	0.478 46	0.390 93	0.467 74	0.465 11
50	0.484 58	0.398 60	0.476 81	0.475 55
60	0.493 91	0.410 47	0.490 86	0.491 77
75	0.503 45	0.422 83	0.505 46	0.508 77
100	0.513 21	0.435 74	0.520 71	0.526 65
150	0.523 21	0.449 30	0.536 68	0.545 56
300	0.533 48	0.463 64	0.553 51	0.565 70
Rack	0.544 06	0.478 97	0.571 39	0.587 39

* Stub teeth.
† Large fillet.
Source: R. G. Mitchiner and H. H. Mabie, "The Determination of the Lewis Form Factor and the AGMA Geometry Factor *J* for External Spur Gear Teeth," *ASME Journal of Mechanical Design*, vol. 104, no. 1, Jan. 1982, pp. 148–158.

The use of this original Lewis form factor implies that the teeth do not share the load and the greatest force is exerted at the tip of the tooth. However, in practice the contact ratio is larger than unity to achieve a quality gearset (i.e. another pair of teeth will be in contact simultaneously to the tip of the tooth contact). Therefore, more recently AGMA developed a new Y form factor with corrections for the above mentioned limitation on the original Lewis form factor.

b) Stress concentration

When Wilfred Lewis first proposed the formula for bending stress, stress concentration factors were not in use. But it is known now that there are a great many situations in which they must be used. Photoelastic investigations constitute the primary source of information on stress concentration in gear teeth. These results were interpreted in terms of the fatigue stress concentration factor K_f as:

$$K_f = H + \left(\frac{t}{r}\right)^L + \left(\frac{t}{l}\right)^M \tag{2.59}$$

, where:

$H = 0.34 - 0.4583662\phi$;

$L = 0.316 - 0.4583662\phi$;

$M = 0.290 + 0.4583662\phi$;

$$r = \frac{r_f + (b - r_f)^2}{(d/2) + b - r_f}.$$

In these equations l and t are found from the layout of Figure 2.51, ϕ is the pressure angle, r_f is the fillet radius, b is the addendum, and d is the pitch diameter.

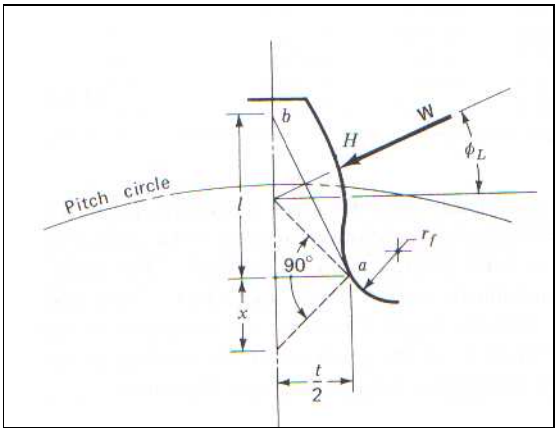


Fig. 2.51 – Graphical layout to obtain x and t.

c) Geometry factor

The AGMA established a factor J, called the *geometry factor*, which uses the modified form factor Y of equation (2.58), the fatigue stress-concentration factor K_f of equation (2.59), and a *load-sharing ratio* based on the proportion of the total load carried by the most heavily loaded tooth. The AGMA equation of factor J is as follows:

$$J = \frac{Y}{K_f m_N} \tag{2.60}$$

With this definition of geometry factor, we can now write equation (2.58) in the form:

$$\sigma = \frac{W_t}{FmJ} \tag{2.61}$$

, which gives the normal stress corresponding to the total load W acting at the highest point of single-tooth contact including the effects of stress concentration. Values of geometry factor J are given in Tables 2.4.a (for $\phi = 20^{\circ}$) and 2.4.b (for $\phi = 25^{\circ}$).

Tab. 2.4 – AGMA geometry form factor for tooth design.

AGMA GEOMETRY FACTOR J FOR TEETH HAVING $\phi = 20°$, $a = 1m$, $b = 1.25m$, AND $r_f = 0.300m$

Number of teeth	Number of teeth in mating gear: 1	17	25	35	50	85	300	1000
18	0.244 86	0.324 04	0.332 14	0.338 40	0.344 04	0.350 50	0.355 94	0.361 12
19	0.247 94	0.330 29	0.338 78	0.345 37	0.351 34	0.358 22	0.364 05	0.369 63
20	0.250 72	0.336 00	0.344 85	0.351 76	0.358 04	0.365 32	0.371 51	0.377 49
21	0.253 23	0.341 24	0.350 44	0.357 64	0.364 22	0.371 86	0.378 41	0.384 75
22	0.255 52	0.346 07	0.355 59	0.363 06	0.369 92	0.377 92	0.384 79	0.391 48
24	0.259 51	0.354 68	0.364 77	0.372 75	0.380 12	0.388 77	0.396 26	0.403 60
26	0.262 89	0.362 11	0.372 72	0.381 15	0.388 97	0.398 21	0.406 25	0.414 18
28	0.265 80	0.368 60	0.379 67	0.388 51	0.396 73	0.406 50	0.415 04	0.423 51
30	0.268 31	0.374 62	0.385 80	0.395 00	0.403 59	0.413 83	0.422 83	0.431 79
34	0.272 47	0.383 94	0.396 71	0.405 94	0.415 17	0.426 24	0.436 04	0.445 86
38	0.275 75	0.391 70	0.404 46	0.414 80	0.424 56	0.436 33	0.446 80	0.457 35
45	0.280 13	0.402 23	0.415 79	0.426 85	0.437 35	0.450 10	0.461 52	0.473 10
50	0.282 52	0.408 08	0.422 08	0.435 55	0.444 48	0.457 78	0.469 75	0.481 93
60	0.286 13	0.417 02	0.431 73	0.443 83	0.455 42	0.469 60	0.482 43	0.495 57
75	0.289 79	0.426 20	0.441 63	0.454 40	0.466 68	0.481 79	0.495 54	0.509 70
100	0.293 53	0.435 61	0.451 80	0.465 27	0.478 27	0.494 37	0.509 09	0.524 35
150	0.297 38	0.445 30	0.462 26	0.476 45	0.490 23	0.507 36	0.523 12	0.539 54
300	0.301 41	0.455 26	0.473 04	0.487 98	0.502 56	0.520 78	0.537 65	0.555 33
Rack	0.305 71	0.465 54	0.484 15	0.499 88	0.515 29	0.534 67	0.552 72	0.571 73

AGMA GEOMETRY FACTOR J FOR TEETH HAVING $\phi =$ [illegible]
$a = 1m$, $b = 1.25m$, AND $r_f = 0.300m$

Number of teeth	Number of teeth in mating gear: 1	17	25	35	50	85	300	[illegible]
13	0.286 65	0.346 84	0.352 92	0.357 44	0.361 38	0.365 72	0.369 25	[illegible]
14	0.293 64	0.359 24	0.365 87	0.370 81	0.375 14	0.379 94	0.383 86	[illegible]
15	0.300 09	0.370 27	0.377 40	0.382 75	0.387 44	0.392 67	0.396 94	[illegible]
16	0.305 58	0.380 16	0.387 75	0.393 46	0.398 49	0.404 11	0.408 73	[illegible]
17	0.310 43	0.389 07	0.397 09	0.403 14	0.408 49	0.414 48	0.419 41	[illegible]
18	0.314 75	0.397 14	0.405 56	0.411 93	0.417 56	0.423 90	0.429 13	[illegible]
19	0.318 62	0.404 49	0.413 28	0.419 94	0.425 85	0.432 50	0.438 01	[illegible]
20	0.322 11	0.411 21	0.420 34	0.427 27	0.433 44	0.440 39	0.446 16	[illegible]
21	0.325 28	0.417 38	0.426 82	0.434 01	0.440 42	0.447 65	0.453 67	[illegible]
22	0.328 16	0.423 06	0.432 80	0.440 23	0.446 86	0.454 36	0.460 60	[illegible]
24	0.333 22	0.433 18	0.443 46	0.451 32	0.458 36	0.466 35	0.473 01	[illegible]
26	0.337 52	0.441 93	0.452 68	0.460 93	0.468 33	0.476 74	0.483 78	[illegible]
28	0.341 22	0.449 57	0.460 75	0.469 33	0.477 05	0.485 85	0.493 23	[illegible]
30	0.344 43	0.456 31	0.467 85	0.476 75	0.484 75	0.493 89	0.501 57	[illegible]
34	0.349 76	0.467 63	0.479 81	0.489 23	0.497 72	0.507 46	0.515 66	[illegible]
38	0.354 00	0.476 78	0.489 48	0.499 33	0.508 24	0.518 47	0.527 10	[illegible]
45	0.359 67	0.489 19	0.502 61	0.513 05	0.522 52	0.533 44	0.542 68	[illegible]
50	0.362 78	0.496 08	0.509 91	0.520 68	0.530 47	0.541 77	0.551 36	[illegible]
60	0.367 50	0.506 83	0.521 09	0.532 38	0.542 67	0.554 57	0.564 69	[illegible]
75	0.372 32	0.517 47	0.532 57	0.544 40	0.555 20	0.567 73	0.578 42	[illegible]
100	0.377 26	0.528 60	0.544 36	0.556 76	0.568 10	0.581 29	0.592 57	[illegible]
150	0.382 37	0.540 05	0.556 51	0.569 51	0.581 38	0.595 26	0.607 16	[illegible]
300	0.387 72	0.551 85	0.569 02	0.582 59	0.595 07	0.609 67	0.622 22	[illegible]
Rack	0.393 42	0.564 05	0.581 94	0.596 13	0.609 21	0.624 56	0.637 78	[illegible]

d) Dynamic effects

When a pair of gears is driven at moderate or high speeds and noise is generated, it is certain that dynamic effects are present. One of the earliest efforts to account for an increase in dynamic load due to velocity employed a number of gears of the same size, materials, and strength which were subjected to destructive testing. For example, if a pair of gears failed at 3

[kN] at zero velocity, and at 1.5 [kN] at velocity V_1, then a velocity factor, designated as K_v, of 0.5 was specified for the gears at velocity V_1. Thus another identical pair of gears running at pitch-velocity V_1 could be assumed to have a dynamic load equal to twice the transmitted load.

Carl Barth in the XIX century first expressed velocity factor, also called *dynamic factor*, by the equation:

$$K_v = \frac{3}{3+V} \tag{2.62}$$

These tests were performed on teeth having a cycloidal profile, instead of an involute; Cycloidal teeth were in quite general use in those days because they were easier to cast than involute teeth.

For cut or milled teeth or for gears not carefully generated:

$$K_v = \frac{6}{6+V} \tag{2.63}$$

For spur gears whose teeth are finished by hobbing or shaping, AGMA recommends the formula:

$$K_v = \frac{50}{30+\sqrt{200V}} \tag{2.64}$$

If the gears have high-precision shaved or ground teeth and if an appreciable dynamic load is developed, then the AGMA dynamic factor is:

$$K_v = \sqrt{\frac{78}{78+\sqrt{200V}}} \tag{2.65}$$

If the gears have high-precision shaved or ground teeth and if there is no appreciable dynamic load is developed, then the AGMA recommends the dynamic factor $K_v=1$.

Introducing the dynamic factor into equation (2.61) gives:

$$\sigma = \frac{W_t}{K_v F m J} \tag{2.66}$$

e) Surface durability

To assure a satisfactory life, the gears must be designed so that the dynamic surface stresses are within the surface endurance-limit of the material. In many cases the first visible evidence of wear is seen near the pitch line; this seems reasonable because the maximum dynamic load occurs in this area.

To obtain an expression for the surface-contact stress, we shall employ the Hertz theory. The contact stress between two cylinders may be computed by the equation:

$$p_{\max} = \frac{2F}{\pi b l} \tag{2.67}$$

, where:

$p_{\max}$ = surface pressure, [Pa];

F = force pressing the two cylinders together, [N];

l = length of cylinders, [m];

$$b = \sqrt{\frac{2F}{\pi l}\frac{\left[(1-\nu_1)^2/E_1\right]+\left[(1-\nu_2)^2/E_2\right]}{(1/d_1)+(1/d_2)}}\text{, [m].}$$

, where: ν_1 , ν_2 and E_1, E_2 are the elastic constants and d_1 and d_2 are the diameters, respectively, of the two cylinders.

To adapt these relations to the notation used in gearing, we replace F by $W_t/\cos\phi$, d by $2r$, and l by the face width F. With these changes we can substitute the expression of b into equation (2.67). Replacing $p_{\max}$ by σ_H, the surface compressive stress (Hertzian stress) is found to be:

$$\sigma_H{}^2 = \frac{W_t}{F\cos\phi}\frac{(1/r_1)+(1/r_2)}{\left[(1-\nu_1)^2/E_1\right]+\left[(1-\nu_2)^2/E_2\right]} \tag{2.68}$$

, where r_1 and r_2 are the instantaneous values of the radii of curvature on the pinion-gear tooth profiles, respectively, at the point of contact. By accounting for loading sharing in the value of W_t, equation (2.68) can be solved for Hertzian stress for any point from the beginning to the end of tooth contact (where sliding and rolling are neglected).

As an example of the use of this equation, let us find the contact stress when a pair of teeth is in contact at the pitch point. The radii of curvature r_1 and r_2 of the tooth profiles, when they are in contact at the pitch point, are:

$$r_1 = \frac{d_P\sin\phi}{2} \qquad r_2 = \frac{d_G\sin\phi}{2} \tag{2.69}$$

, where ϕ is the pressure angle. Then:

$$\frac{1}{r_1}+\frac{1}{r_2} = \frac{2}{\sin\phi}\left(\frac{1}{d_P}+\frac{1}{d_G}\right) \tag{2.70}$$

Defining the *speed ratio* m_G as:

$$m_G = \frac{N_G}{N_P} = \frac{d_G}{d_P} \tag{2.71}$$

, equation (2.69) can be re-written as:

$$\frac{1}{r_1}+\frac{1}{r_2} = \frac{2}{\sin\phi}\frac{m_G+1}{m_G d_P} \tag{2.72}$$

Combining equations (2.68) and (2.72) we obtain the compressive stress (-):

$$\sigma_H = -\sqrt{\frac{W_t}{Fd_P}\frac{1}{\pi\left(\frac{1-\nu_P{}^2}{E_P}+\frac{1-\nu_G{}^2}{E_G}\right)}\frac{1}{\frac{\cos\phi\sin\phi}{2}\frac{m_G}{m_G+1}}} \tag{2.73}$$

The subscripts *P* and *G* in equation (2.73) designate pinion and gear, respectively.

The second term in radical of equation (2.73) is called the elastic coefficient C_P. Thus C_P is given by:

$$C_P = \sqrt{\frac{1}{\pi\left(\frac{1-\nu_P{}^2}{E_P}+\frac{1-\nu_G{}^2}{E_G}\right)}} \tag{2.74}$$

Values of C_P have been worked out for various combinations of material, and these are listed in Table 2.5.

Tab. 2.5 – Values of the elastic coefficient C_P for spur and helical gears with non-localized contact and for $\nu = 0.3$.

Pinion	Modulus of elasticity *E*, GPa	Gear: Steel	Malleable iron	Nodular iron	Cast iron	Aluminum bronze	Tin bronze
Steel	200	191	181	179	174	162	158
Mall. iron	170	181	174	172	168	158	154
Nod. iron	170	179	172	170	166	156	152
Cast iron	150	174	168	166	163	154	149
Al. bronze	120	162	158	156	154	145	[illegible]
Tin bronze	110	158	154	152	149	141	[illegible]

Source: AGMA 218.91.

The geometry factor *I* for spur gears is the denominator of the third term under the radical of equation (2.73). Thus:

$$I = \frac{\cos\phi\sin\phi}{2}\frac{m_G}{m_G+1} \tag{2.75}$$

Now recall that a velocity factor K_v was used in the bending-stress equation to account for the fact that the force between the teeth is actually more than the transmitted load because of the dynamic effect. This factor must also be used in the equation for surface-compressive stress for exactly the same reasons, i.e. $C_v = K_v$, given by equation (2.61).

Finally, combining equations (2.73), (2.74) and (2.75) we obtain:

$$\sigma_H = -C_p\sqrt{\frac{W_t}{C_v F d_p I}} \tag{2.76}$$

To assure stresses are within the endurance-limit of the material, the gears must be designed according to the face width parametric variation method for estimating gear size shown on Annex A.

Helical Gears – Kynematics and Force Analysis:

The helical gears are used to transmit motion between parallel shafts. However, in the case of helical gears the teeth are not parallel to the axis of rotation like in spur gears.

The helix angle is the same on each gear, but one gear must have right-hand and the other a left hand helix. The shape of the tooth is an involute helicoid and is illustrated in Figure 2.52.

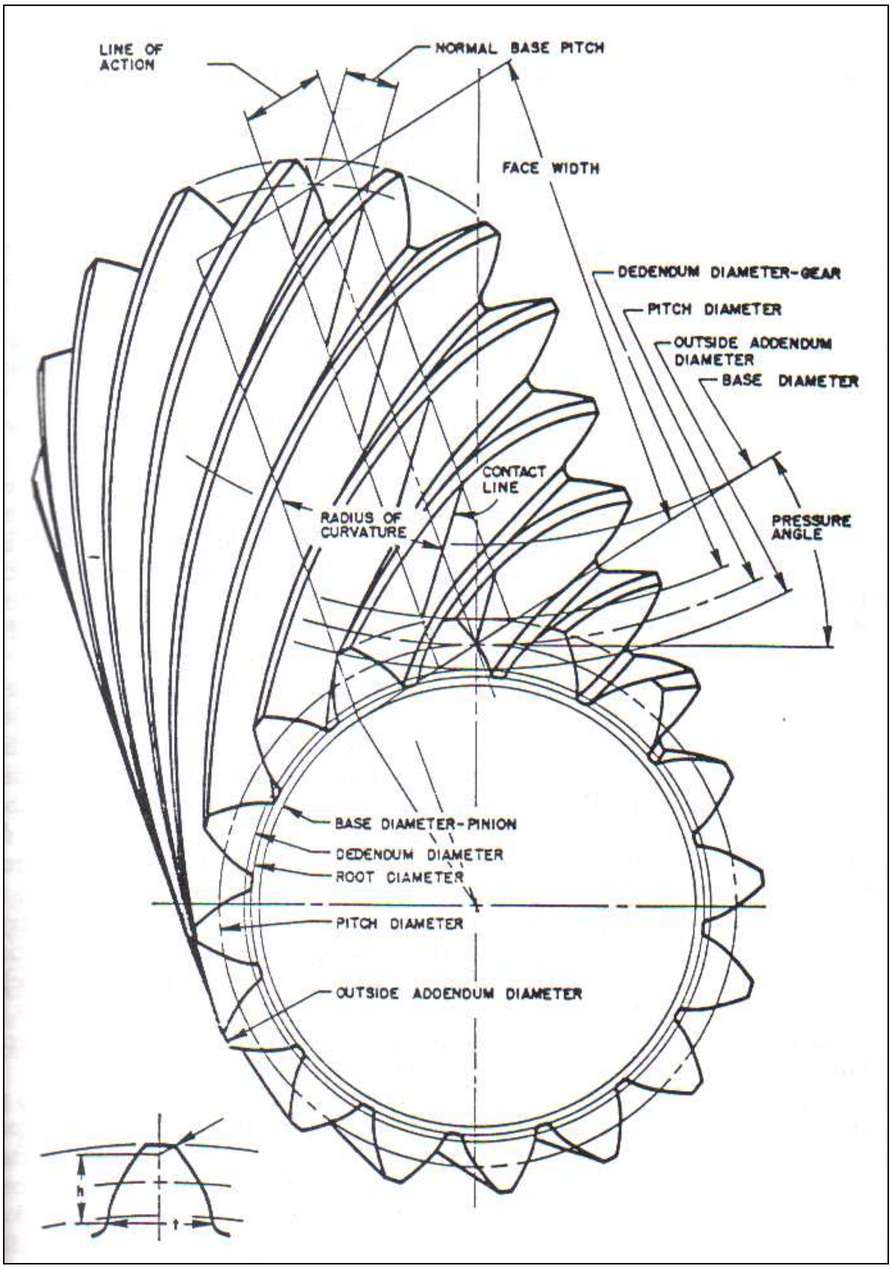


Fig. 2.52 – Involute helicoid geometry.

The initial contact of spur-gear is a line extending all the way across the face tooth. The initial contact of a helical-gear teeth is a point which changes into a line as the teeth come into more engagement. In spur gears the line of contact is parallel to the axis of rotation; in helical gears the line is diagonal across the face of the tooth. It is thus gradual engagement of the teeth and the smooth transfer of load from one tooth to another which give helical gears the ability to transmit heavy loads at high speeds. Because of the nature of contact between helical gears, the contact ratio is only of minor importance, and it is the contact area, which is proportional to the face width of the gear, that becomes significant.

Helical gears subject the shaft bearings to both radial and thrust loads. When the thrust load become high or are objectionable for other reasons, it may be desirable to use double helical gears. A double helical gear (herringbone) is equivalent to two helical gears of opposite hand, mounted side by side on the same shaft. They develop opposite thrust reactions and thus cancel out the thrust load.

Figure 2.53 represents a portion of the top view of a helical-rack. Lines *ab* and *cd* are the centerlines of two adjacent helical teeth taken on the pitch plane. The angle ψ is the helix angle. The distance *ac* is the *transverse circular pitch* p_t in the plane of rotation (usually called the circular pitch). The distance *ae* is the *normal circular pitch* p_n in the plane of rotation, and is related to the transverse circular pitch as follows:

$$p_n = p_t \cos\psi \tag{2.77}$$

The distance *ad* is called the *axial pitch* p_x, and is related to the transverse circular pitch as follows:

$$p_x = \frac{p_t}{\tan\psi} \tag{2.78}$$

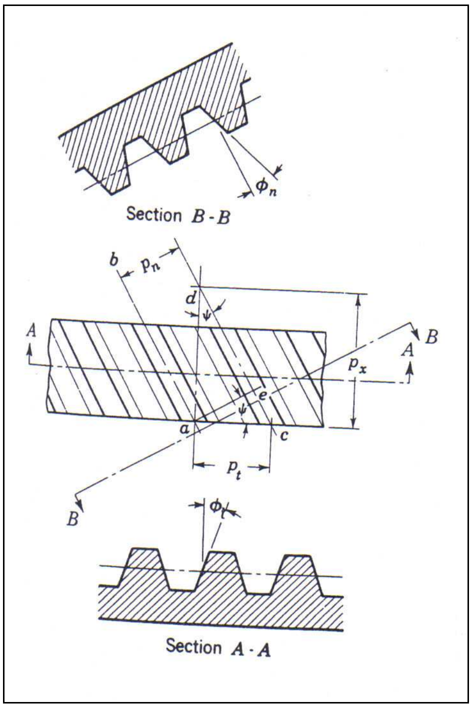


Fig. 2.53 – Nomenclature of helical gears.

The *normal module* m_n, is given by:

$$m_n = m\cos\psi \tag{2.79}$$

The pressure angle ϕ_n in the normal direction is different from the pressure angle ϕ_t in the direction of rotation, because of the angularity of the teeth. These angles are related by the equation:

$$\cos\psi = \frac{\tan\phi_n}{\tan\phi_t} \tag{2.80}$$

Figure 2.54 is a three-dimensional view of the forces acting against a helical-gear. The point of application of the forces is in the pitch plane and in the center of the gear face.

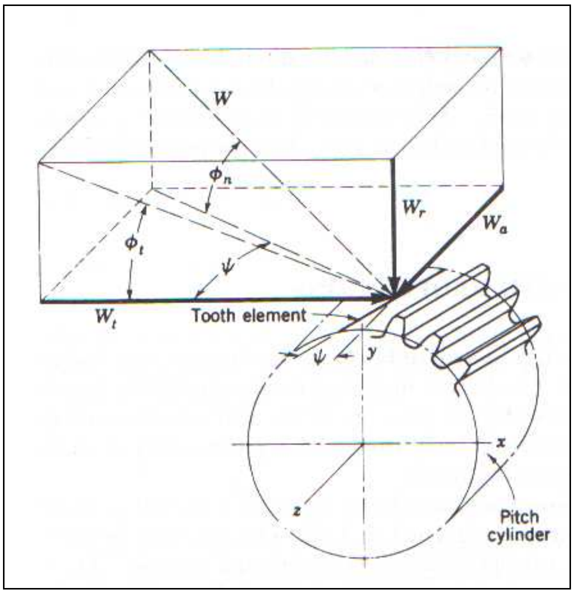


Fig. 2.54 – Tooth forces acting on right-hand helical gear.

From the geometry of the figure, the three components of the total (normal) tooth force *W* are:

$$
\begin{aligned}
W_r &= W \sin\phi_n \\
W_t &= W \cos\phi_n \cos\psi \\
W_a &= W \cos\phi_n \sin\psi
\end{aligned}
\tag{2.81}
$$

, where:

W = total force;

W_r = radial component;

W_t = tangential component, also called transmitted load;

W_a = axial component, also called thrust load.

Usually W_t is given and the other forces are desired. In this case it is not difficult to discover that:

$$
\begin{aligned}
W_r &= W_t \tan\phi_t \\
W_a &= W_t \tan\psi \\
W &= \frac{W_t}{\cos\phi_n \cos\psi}
\end{aligned}
\tag{2.82}
$$

<u>Helical Gears – Strength Analysis:</u>

The same equations for bending stress (2.66) and surface stress (2.76) in spur gears could be repeated in here because they also apply to helical gears. For helical gears the velocity factor is usually given by equation (2.65).

Geometry factors for helical gears must account for the fact that contact takes place along a diagonal line across the tooth face and that we are usually dealing with the transverse pitch instead of normal pitch. The worst loading occurs when the line of contact intersects the tip of the tooth, but the unloaded end strengthens the tooth. The *J* factor for $\phi_n = 20^\circ$ can be found in Table 2.6. The AGMA also published *J* factors for $\phi_n = 15^\circ$ and $\phi_n = 22^\circ$.

Geometry factors *I* for helical and herringbone gears are calculated from the equation:

$$I = \frac{\cos\phi_t \sin\phi_t}{2m_N} \frac{m_G}{m_G + 1} \tag{2.83}$$

In this equation ϕ_t is the transverse pressure angle and m_N is the load-sharing ratio and is found from equation:

$$m_n = \frac{p_N}{0.95Z} \tag{2.84}$$

Here p_N is the normal base pitch; it is related to the normal circular pitch p_N by the relation:

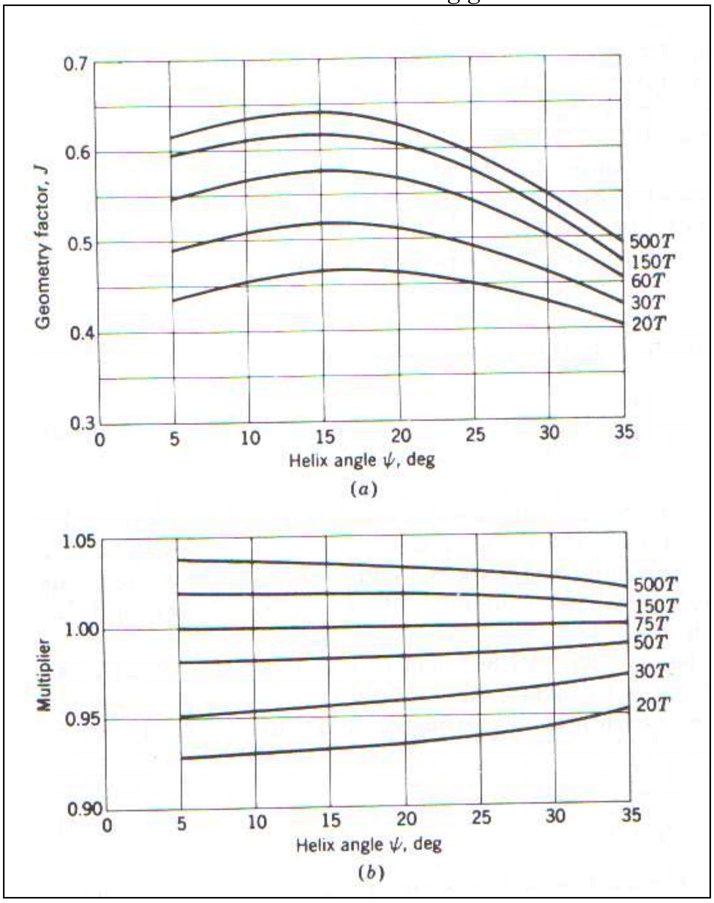
$$p_N = p_n \cos\phi_n \tag{2.85}$$

The quantity Z is the length of the line of action in the transverse plane. It is best obtained from layout of the two gears, but may also be found from the equation:

$$Z = \sqrt{(r_P + a)^2 - r_{bP}^2} - \sqrt{(r_G + a)^2 - r_{bG}^2} - (r_P + r_G)\sin\phi_t \tag{2.86}$$

, where r_P and r_G are the pitch radii and r_{bP} and r_{bG} are the base-circle radii, respectively, of the pinion and gear.

Tab. 2.6 – Geometry factors for helical and herringbone gears having a normal pressure angle of 20º. (a) Geometry factors for gears mating with a 75-tooth gear. (b) J-factor multipliers when tooth numbers other than 75 are used in the mating gear.



Gear Arrangements:

The early reduction gear designs incorporated many devices to minimize the effects of bending and torsion of the pinion and of inaccuracies in machining and alignment. However, experience has demonstrated that such devices are unnecessary, and gear elements can be so proportioned and machined that uniform tooth pressures can be obtained without use of mechanical devices to compensate for pinion deflections. Hence, helical gears have been used for many years and remain a part of most systems of gearing.

Figure 2.55a represents the simplest arrangement of a marine reduction gear, i.e., the pinion meshing with gear, as used, for instance, for connecting a propeller to a diesel engine or to an electric motor.

Figure 2.55b is a drive with two pinions as used frequently with diesel engines of comparatively large power.

Figure 2.55c represents the early-type of single-reduction gear for a turbine drive, the principal difference between this reduction gear and the one shown in Figure 2.55b being in the number of pinion bearings. It was used for speed ratios up to 20:1.

Figure 2.55d is the usual arrangement of a double-reduction gear for a turbine driven ship. The two input pinions are driven by the two elements (high-pressure and low-pressure turbines) of a cross-compound turbine. Power is divided between the two input pinions. The terms "tandem" and "articulated" are also applied to this arrangement because of the disposition of the first and second reductions, and a flexible coupling is generally provided between the first and second reductions.

Figure 2.55e represents the nested-type double-reduction gear, which has also been used with cross-compound turbines.

Figure 2.55f illustrates the type of gear referred as a locked-train double-reduction gear.

Figure 2.55g is a locked-train type double-reduction gear for a cross-compound turbine or for two gas turbines prime movers. This arrangement has become standard for high-powered naval vessels and high-powered merchant vessels because it minimizes the weight and size of the assembly.

Figure 2.55h is a planetary gear. It has a single input “sun pinion” which drives three or more “planet gears”. These planet gears are mounted on a planet carrier which is solidly connected to the output coupling. The outer “ring gear” is held stationary in the gear housing. This type of type of gear has been applied to turbine-generator drive gears.

Many other reduction gear arrangements are possible and have been used. For example, an epicyclic gear arrangement is shown in Figure 2.56. Epicyclic gears with their compact, lightweight, construction are being increasingly used in marine transmissions.

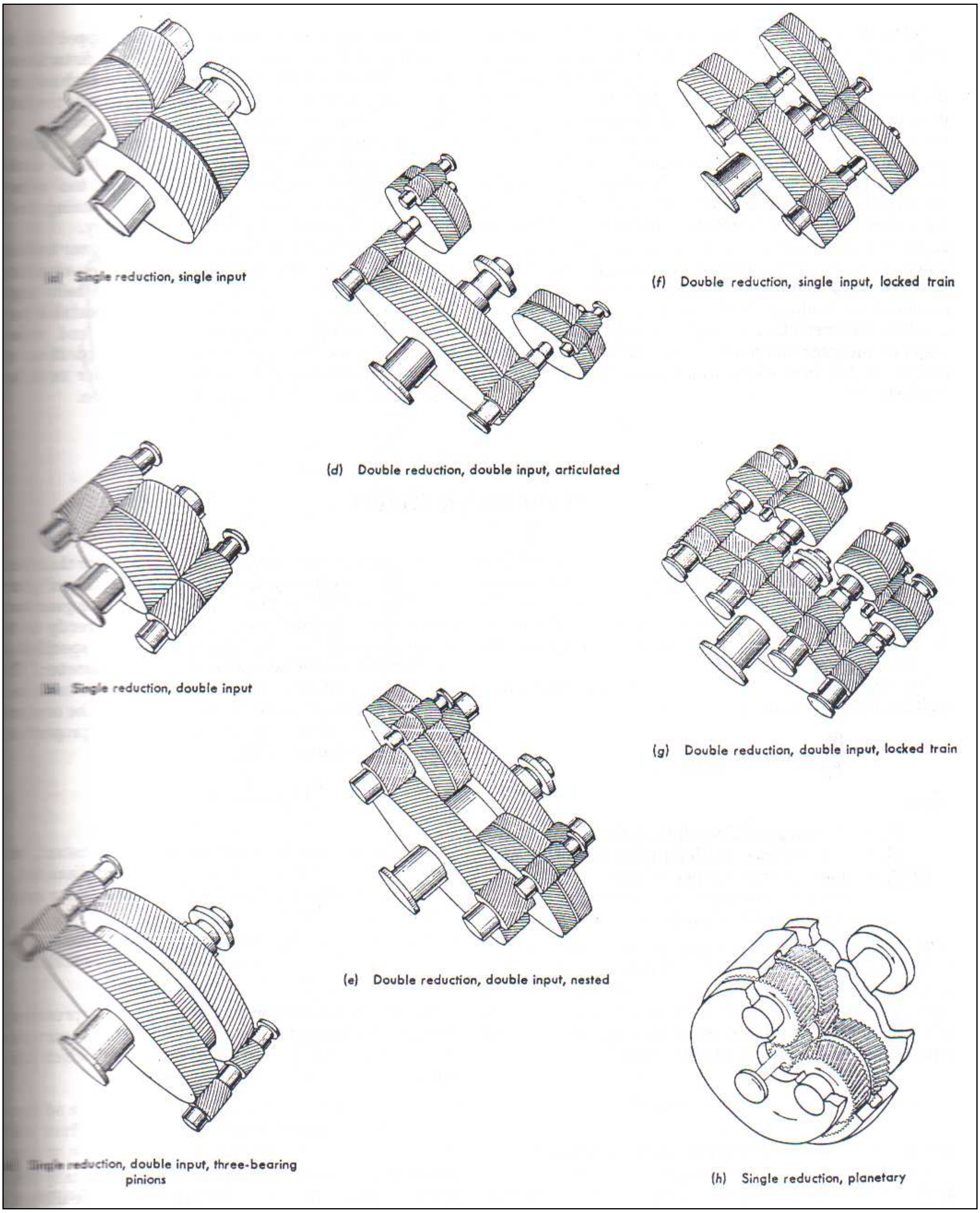


Fig. 2.55 – Gear arrangements.

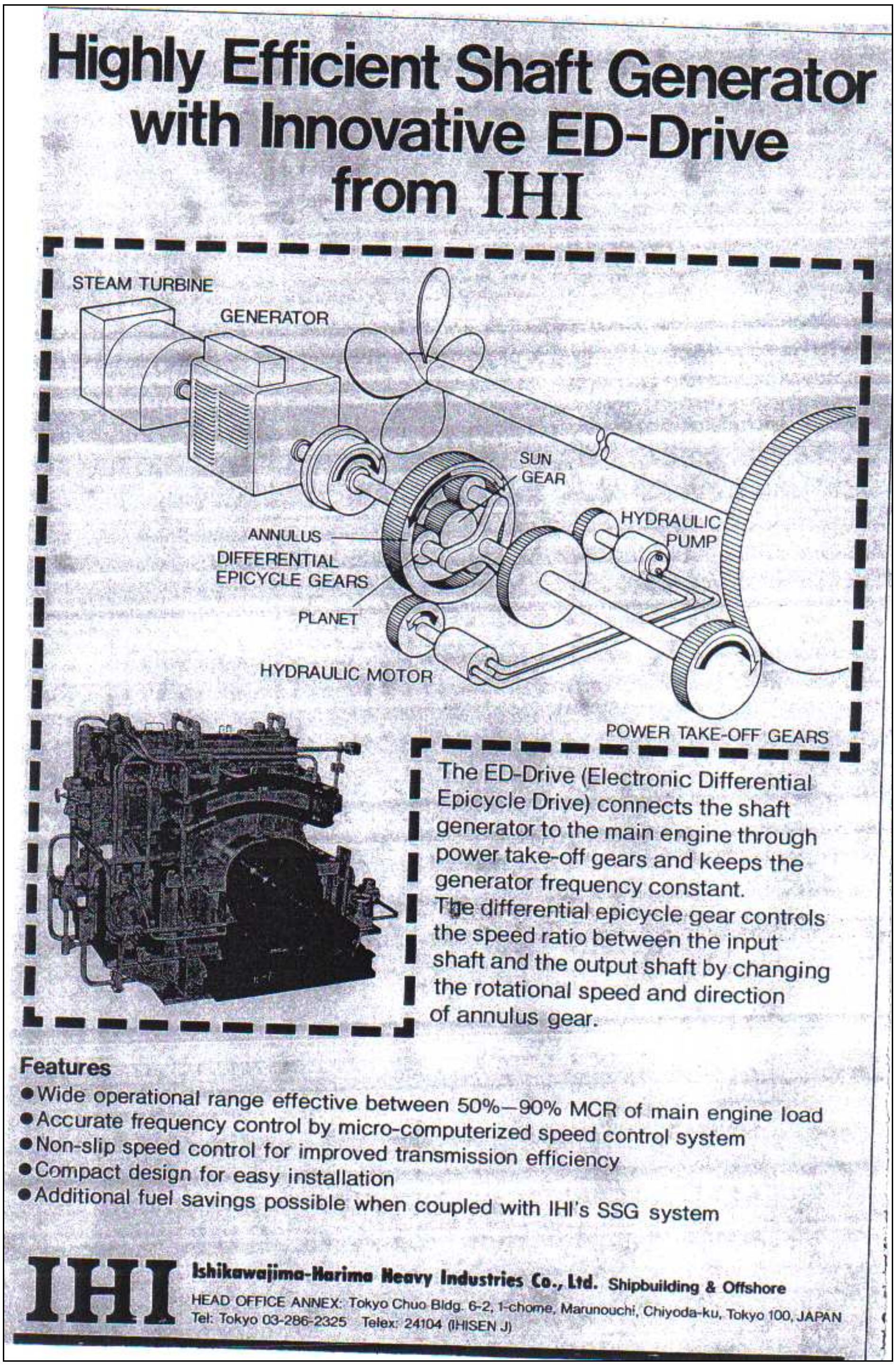


Fig. 2.56 – Epicyclic gear arrangement arrangements.

2.6.3. Propulsion shafts

Introduction:

A propulsion shaft is a rotating member, usually of circular cross section, having mounted upon it several power-transmission elements (such as gears, flywheels and propellers). Shafts may be subjected to bending, tension, compression, or torsional loads, acting singly or in combination with one another.

Design Considerations:

In general, the dimensions of shafting are predicted on the basis of strength requirements so that deflection is not too large; however, it is occasionally to modify an otherwise satisfactory shafting system design due to vibration considerations. Shafting diameters usually have only a minor impact on the longitudinal vibration characteristics, due to the fact that both the stiffness and weight of the shafting change proportionally; but the whirling and torsional modes of vibration are sensitive to shaft diameters. Shafting vibration, as such, is discussed in great detail on the Ship Vibration course.

Propulsion shafting is subjected to a variety of steady and alternating loads, which induce torsional shear, axial thrust, and bending stresses between the shafting and the mating elements (such as between propeller and shaft) which, when coupled with axial strains from bending stress, are very important from a fatigue standpoint.

The steady loads represent average conditions and can be estimated with a degree of certainty as they are directly derived from the main engine torque and the propeller thrust. On the other hand, vibratory loads do not lend themselves to a precise evaluation and are difficult to treat in an absolute sense.

Whenever possible, the power-transmission elements, such as clutches, gears or propellers should be located close to the supporting bears. This reduces the bending moment, and hence the deflection and bending stress.

Design for Static Load:

The stresses at the surface of a solid round shaft subjected to combined loading of bending and torsion are:

$$\sigma_x = \frac{32M}{\pi d^3} \quad \tau_{xy} = \frac{16T}{\pi d^3} \tag{2.87}$$

, where:

σ_x = bending stress;

τ_{xy} = torsional stress;

d = shaft diameter;

M = bending moment at critical section;

T = torsional moment at critical section.

By the use of a Mohr's circle it is found that the maximum shear stress is given by:

$$\tau_{max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^{\;2}} \tag{2.88}$$

Substituting equation (2.87) into (2.88) we obtain:

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{M^2 + T^2} \tag{2.89}$$

The maximum shear stress theory of static failure states that $S_{xy} = S_y/2$. By employing a factor of safety *n*, we can now write equation (2.89) as:

$$\frac{S_y}{2n} = \frac{16}{\pi d^3}\sqrt{M^2 + T^2} \tag{2.90}$$

, or:

$$d = \left[\frac{32n}{\pi S_y}\sqrt{M^2 + T^2}\right]^{1/3} \tag{2.91}$$

It is important to note that these relations are only valid when the stresses are truly non-variable. Classification Societies established codes for the design of transmission shafting where combined stress concentration and load factors, depending on the conditions of particular application, are introduced into the basic equation above.

<u>Shaft Design (according to Rules and Regulations issued by LRS, 1987, Part 5, Chap. 6):</u>

Shafting for merchant vessels is required to meet the minimum standards set by the Classification Society which classes the vessel. Classification Societies use rather simple formulas to compute the minimum shaft diameter. These formulas contain coefficients which are changed from time to time in recognition of experience or advancements in technology.

According to Lloyd's Register of Ships, the diameter of the shaft is to be not less than given by the following formula:

$$d = FK\sqrt[3]{\frac{P}{R}\left(\frac{560}{\sigma_u + 160}\right)} \tag{2.92}$$

, where:

$400 < \sigma_u < 600$ [N/mm^2] = ultimate stress of a carbon or carbon manganese steel;

$\sigma_u < 800$ [N/mm^2] = ultimate stress of other alloys (should be submitted for consideration);

d = shaft diameter, [mm];

R = shaft speed, [rpm];

P = power delivered, [kW];

K = stress concentration factor depending on the type of shaft (see Table 2.7);

F = load factor depending on the type of application (see Table 2.7).

Hollow shafts (with central holes), the outside diameters of the shafts are to be not less than given by the following formula:

$$d_o = d \sqrt[3]{\frac{1}{\left[1 - \left(\frac{d_i}{d_o}\right)^4\right]}} \tag{2.93}$$

, where:

d_o = outer shaft diameter, [mm];

d_i = inner shaft diameter, [mm], if $d_i <= 0.4d_o$ then $d_o = d$

Tab. 2.7 – Load factor (F) and stress concentration factor (K), according to Rules and Regulations issued by LRS, 1987, Part 5, Chap. 6.

<table>
<tr><th></th><th></th><th>K</th><th>F</th></tr>
<tr><td rowspan="2">Thrust Shaft (Propeller Shafts and Stern Shafts)</td><td>With key</td><td>1.22</td><td>100</td></tr>
<tr><td>Without key</td><td>1.26</td><td rowspan="6">100
(engines)
95
(turbines, electric motors, and engines with a friction clutch allowing for slippage)</td></tr>
<tr><td rowspan="4">Intermediate Shafts</td><td>With integral forged coupling flanges or with shrink-fitted keyless coupling flanges</td><td>1.00</td></tr>
<tr><td>With key</td><td>1.10</td></tr>
<tr><td>Flange, where diameter of the radial coupling bolts are less than 0.3d</td><td>1.10</td></tr>
<tr><td>Fitted with longitudinal slots where width c<1.4d and length l<0.2d, (where d is determined for K=1)</td><td>1.20</td></tr>
</table>

<u>Shaft Materials:</u>

With the exception of vessels of very high power, mild steel is used for both inboard and outboard shafting. In the case of high-powered ships, the inboard shafting may be made of high-strength steel; however, high-strength steel is not recommended for outboard applications. Because of the seawater environment, as well as the fretting corrosion conditions that exist at shaft sleeves and the propeller interface, the fatigue limit of high-strength steel is not really greater than that of mild steel, nor is the endurance limit in a fretting condition better than that of mild steel. Hence, except in the case of designs in which all bearings are of the oil-lubricated type, outboard shafting is provided with sleeves that shrunk on the shafting in way of bearings, stuffing boxes, and fairings. Shaft sleeves are made of bronze or other materials, such as rubber, FGRP or epoxy resins, which are resistant to attack by seawater. Moreover, before sleeves have been applied the propeller shafts are commonly cold rolled for a distance forward and aft of the shaft taper, where the propellers hub mates, and in way of the ends of the shaft liners so that compressive stresses will retard formation of fatigue cracks at the surfaces where bending stresses are higher.

Shaft Balance:

Solid shafting is inherently balanced, but hollow shafting requires attention in this regard. The balance of hollow shafting is accomplished during machining operation by shifting lathe centers prior to the final machining cuts.

Shaft Alignment:

The simplistic view of the main propulsion shaft installation is that the system is set up with initial straight alignment and remains in that state during the lifetime of the ship, unless affected by accident or wear. The reality is that there are many factors which can affect and alter alignment during building and throughout the working lifetime of the vessel.

Shaft alignment should be performed relatively to both vertical and horizontal planes. Horizontal alignment (in respect to vertical plane) is usually conducted by align all the bearing concentrically and relatively to the center of the gearbox flange.

Vertical alignment is much more complex, and requires a particular methodology. The following steps should be taken on implementing the method of fair curve alignment:

a) The initial calculation is to determine the vertical “cold” load on each bearing (total weight of the shaft divided by the number of bearings), assuming all the bearings to be in straight line;

b) Calculation or estimation of differential thermal expansion at the reduction gearbox bearings and, eventually, at other line shaft bearings when the installation goes from cold to the operating condition;

c) Estimation of the stern tube bearing wear down at the aft edge (closer to propeller) for designs in which the bearing is non-metallic of the anti-friction type. In the case of designs in which the bearing is metallic of the oil-lubricated type, this procedure is unnecessary. Propeller weight and buoyancy should be taken into account in this estimation;

d) In case where the hull is distorted by hog or sag due to different conditions of loading, the corresponding hull shapes should be calculated and the associated external height variations on each bearing should be also verified in respect to the initial assumed loading condition;

e) Influence numbers, in terms of load change for each height variation (e.g. 0.1 [mm] up or down), should be used to go from “cold” to the operating condition;

f) Considering the above mentioned loading conditions in “hot” operation, the corresponding vertical positions of all the bearings and consequent distorted shaft line can then be determined.

It should be noticed, that this procedure of calculating the vertical loads on each bearing is approximate not only for reasons associated with difficulties in estimating the hull shapes for different conditions of loading and differential thermal expansion for “hot” and “cold” operating conditions, but also because high deck temperatures and low sea temperatures or rough weather conditions (large head waves) can cause differential expansion and alternate hogging sagging of the hull, respectively.

The intention of this fair curve (rational) alignment is to ensure that bearings are correctly loaded and that the shaft is maintained in a straight line so that is not severely stressed under operating conditions.

Fitting of the stern tube and strut bearing should be carried out at a later stage of construction of the heavily framed stern, on a controlled environment to prevent thermal expansions. It is normal practice to install the shafting from the tailshaft to the engine, using the axis of the stern tube or the strut bearing as a reference point for all the alignment checks.

After the ship has been launched an alignment check should be performed to confirm all the assumptions made during the rational alignment and the shaft installation is satisfactory. The procedure may involve the use of hydraulic jacks placed on each side of the bearing, to lift the shaft just clear.

There are basically three ways that the alignment of a completely installed shafting system can be checked:

- Hydraulic jack method to measure the actual bearing load;

- Strain gauges method to monitor in service shaft stresses;

- Drop-and-gap method, which involves to remove the bolts from selected couplings and compare the relative positions of the mating flanges with calculated values;

- Measurement of bearings weardown method, this is a practical method of measuring alignment since weardown of anti-friction material at the bearings is close related with changes of shaft alignment. In this case the gap between the bearing and the shaft is measured by a poker gauge or by inserting a wedge between the shaft and bearing from the outside. Then wear measurements are compared with a data bank of the maximum permissible wear.

<u>Hull Flexibility, Shaft Length and the Engine Position:</u>

The conventional midships position for the engines of older vessels, with the exception of the tankers, was based on low engine power and strong hull construction. Shafts were long, but being of moderate diameter, were able to flex with the hull as loading or other conditions changed (and in heavy weather). A loading or ballast condition which changed hull shape and shaft alignment to an unusual degree, sometimes caused higher temperature in some bearings due to uneven load distribution. Shaft stress was the hidden factor.

The trend towards higher engine powers and the positioning of engines aft, gave rise to large diameter, short length shafts of increased stiffness. Excessive vibration and resulting damage in many spaces is a common feature as result. Hull detuners intended to reduce vibration have been fitted in steering gear compartments but the improvement to many ships seems to be marginal.

Therefore, it is recalled that hull flexural vibration seems to be less of a problem in ships where the propulsion engine is placed close to a node of the hull's first mode shape, and avoid to select propeller frequencies (shaft revolutions per unit of time times the number of blades) resonant to the hull's natural frequencies of vibration.

2.6.4. Shaft bearings

Main propulsion shafting is supported by bearings that maintain the shafting in proper alignment. As shown in Figure 2.31, these propulsion shaft bearings are naturally divided into two groups: bearings inside the watertight boundary of the hull and bearings outside the hull watertight boundary. Bearings located inside the watertight boundary are called line shaft bearings.

Due to the nature of loadings, adverse working conditions (e.g. low rotational speed) and a direct impact on ship navigation safety aspects, the requirements imposed upon the design of propulsion shaft bearings are extremely severe. Reliability is heavily emphasized in the design of bearings because there is no redundancy for bearings, and a single failure can incapacitate the entire propulsion system.

In addition to the radial bearings that support the shafting (whose surfaces of contact are placed on a radial line parallel to the shaft axis), a main thrust bearing is located inside the ship and transmits the propeller thrust from the shafting to the hull structure; in this case, the surfaces of contact are placed perpendicular to shaft axis.

The intermediate shafting between tailshaft and main engine, gearbox or thrust block may be supported in plain, tilting pad or roller bearings. The two former types usually have individual oil sumps, the oil being circulated by a collar and scraper device; roller bearings are grease lubricated. The individual oil sumps usually have cooling water coils or a simple water chamber fitted. Cooling water is provided from a service main connected to seawater circulating system. The cooling water passes directly overboard. Roller bearings have been used in the smaller shaft sizes, but the advantage of lighter weight and lower friction have in general not been sufficient to offset the higher reliability and lower maintenance requirements of the plain type (note that roller bearings are a self contained unit which must be tightly mounted or dismounted by the end of the shaft).

Plain Bearings:

Any oil between a static shaft and a plain journal bearing in which rests, tends to be squeezed out so that there is metal to metal contact. At the start rotation the journal is inclined to roll up the bearing surface against the direction of rotation until friction slip occurs. Then, provided there is oil in the clearance space, this will cling to the moving surface and be dragged between the shaft and the bearing. Shaft rotation, as it speeds up, continues to carry oil to the shaft underside so developing a film with sufficient pressure to hold the shaft clear of the bearing. The pressure build-up is related to the speed of rotation. Thus oil delivered as the shaft turns at normal speed, will form a layer or film, separating shaft and bearing, and so prevent direct wear of metal to metal. Pressure generated in the oil film, is most effective over about one third of the bearing area (see Figure 2.57) because of oil loss at the bearing ends and peripherally. Load is supported and transmitted to the journal, mainly by the area where the film is generated.

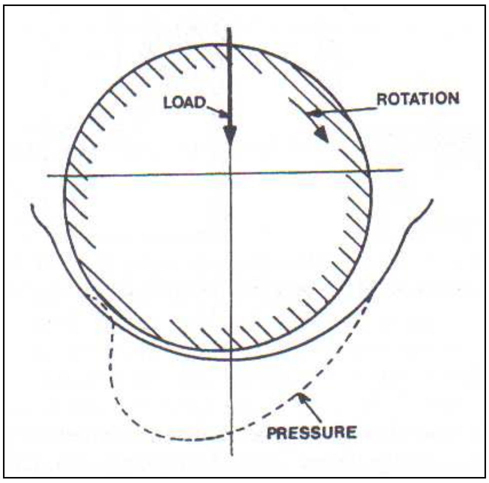


Fig. 2.57 – Fluid film pressure in plain bearing.

As shown in Figure 2.58, alignment is also critical for babbitt surface lined to an heavy steel removable shell which is then supported by bearing housing. Therefore, the bearing shell can be made with self-aligning feature by providing a spherical or crowned seat at the interface between the shell and housing. This allows the axis of the bearing shell to align exactly with that of the shaft.

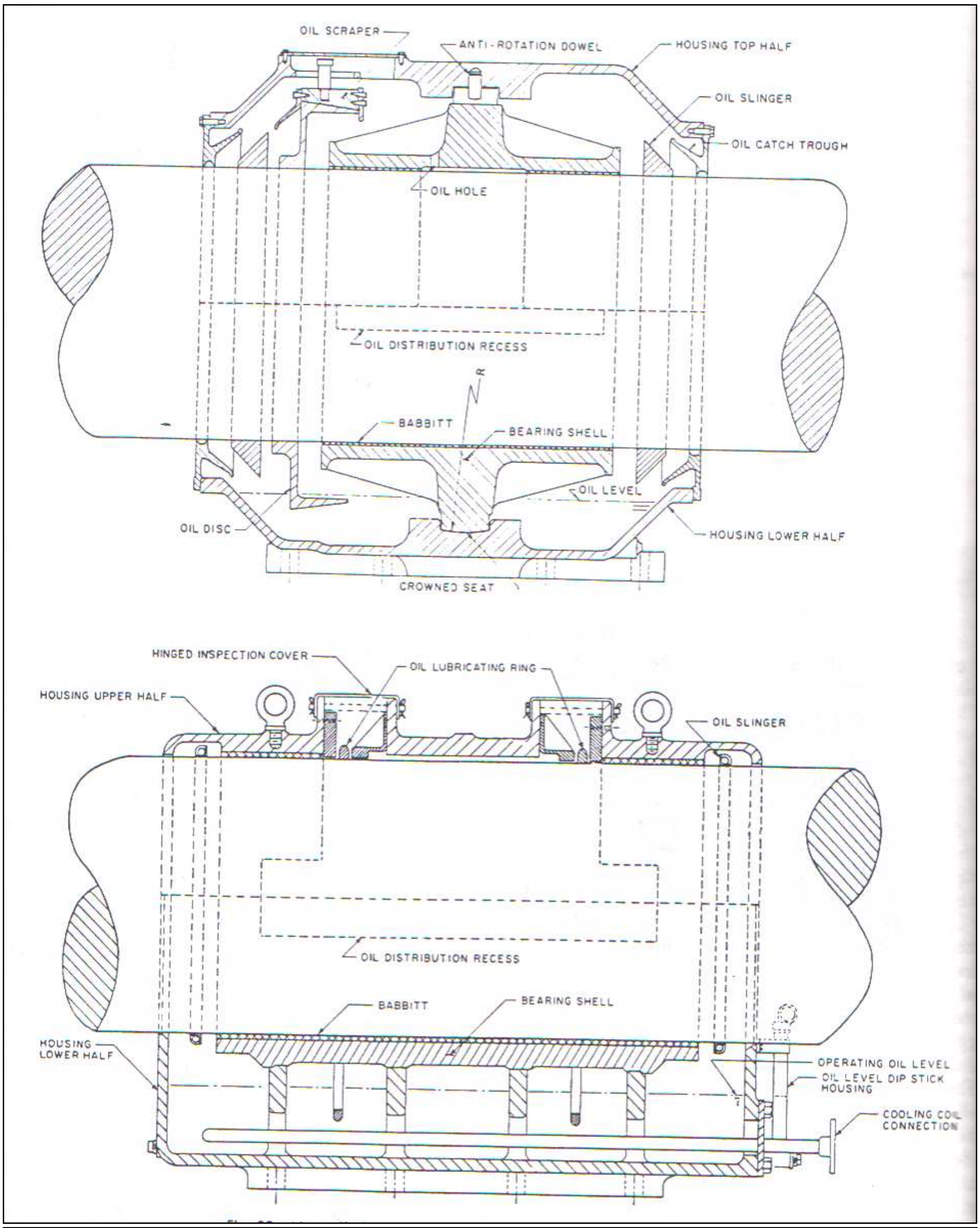


Fig. 2.58 – Self-alignment line shaft bearing with oil-disk lubrication (above) and non-self aligning line shaft bearing with oil-ring lubrication (below).

Usually for plain and tilting pad bearings, only a bottom bearing half is provided, the top acting purely as a cover. The aftermost plumber block however, always has a full bearing. This bearing and any bearing in the forward end of the stern tube, may be subjected to negative vertical loading.

Tilting Pad Bearings:

Replacement of the ineffective side portions of the journal by pads capable of carrying load will considerably increase its capacity. Tilting pads based on those developed by Michell for thrust blocks (see Figure 2.59) are used for the purpose. Each pad tilts as oil is delivered to it, so that a wedge of oil is formed. The three pressure wedges give a larger total support area than that obtained with a plain bearing. The arrangement of pads in a bearing is shown in Figure 2.60.

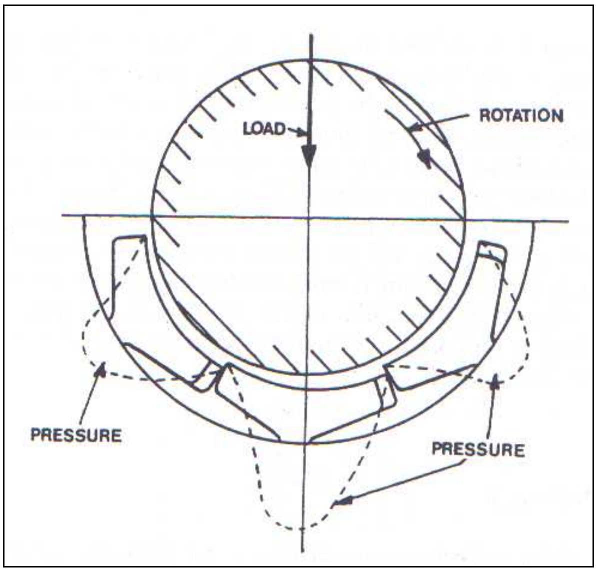


Fig. 2.59 – Fluid film pressure in tilting pad bearing.

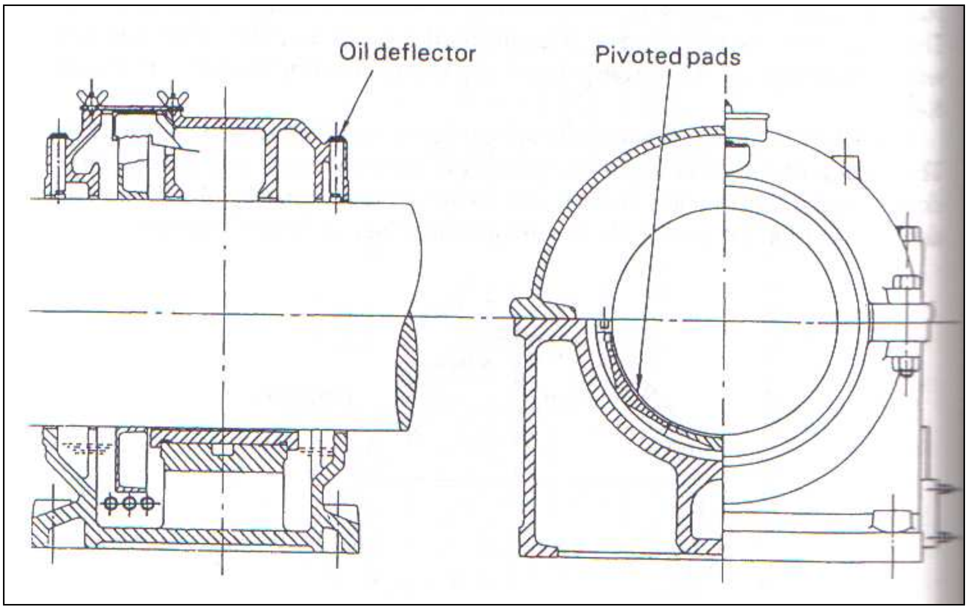


Fig. 2.60 – Tilting pad bearing.

Roller Bearings:

Roller bearings (see Figure 2.61) are supplied in sizes to suit shafts to the largest diameter. Flange couplings dictate that roller bearing races must be in two parts for fitting.

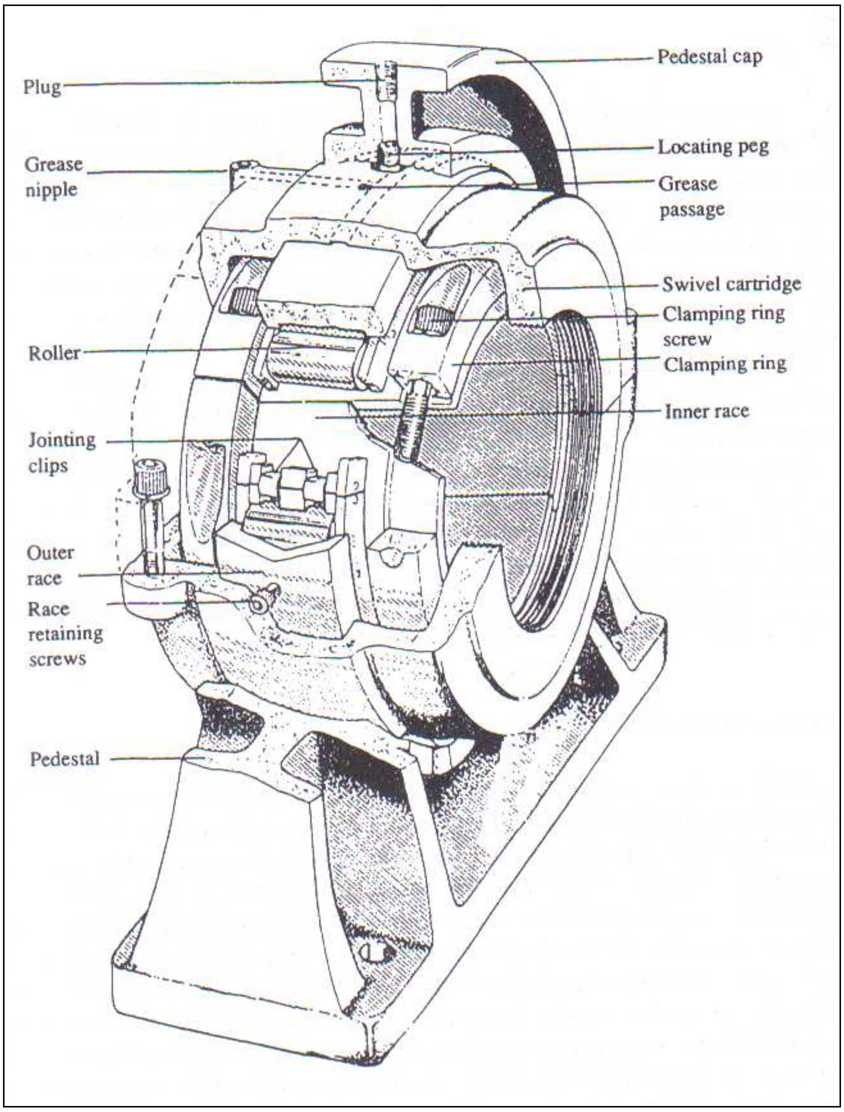


Fig. 2.61 – Roller bearing.

The section of the shaft where the split roller bearing is to be fitted must be machined very accurately and with good finishing. The two halves of the inner and outer races are fitted and held with clamping rings.

Adequate speed for build-up of fluid film pressure is vital for journal bearings. At low speed there may be metal to metal contact with wear and damage. Friction at low rotational speeds is high. Roller bearings are not dependent on speed for effective lubrication. Friction is low at all speeds. This makes them suitable for steam turbine installations and in ships where slow steaming may be necessary. Roller bearings, where fitted, are grease lubricated.

Thrust Blocks:

The main thrust block transfers forward or astern propeller thrust to the hull and limits axial movement of the shaft. Some axial clearance is essential to allow formation of an oil film in the wedge shape between the collar and the thrust pads (see Figure 2.62a). This clearance is also needed to allow for expansion as parts warm up to operating temperature. The actual clearance required, depends on dimensions of pads, speed, thrust load and the type of oil employed. High temperature, power loss and failure can result if axial clearance is too small.

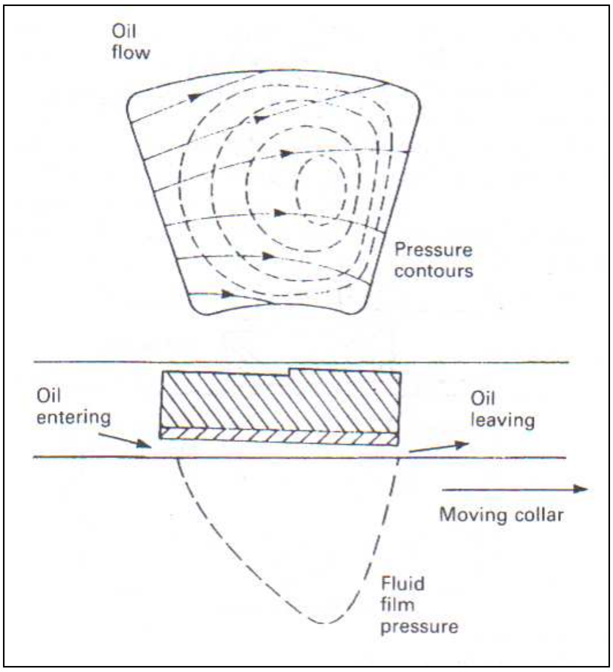


Fig. 2.62a – Michell thrust pad.

A larger than necessary clearance will not cause harm to the thrust bearing pads, but axial movement of the shaft must be limited for protection of main machinery.

The accepted method of checking thrust clearance, involves jacking the shaft axially to the end of its travel in one direction and then back to the limit of travel in the other. Total movement of the thrust shaft (about 1 [mm] being typical) is registered on a dial gauge. Feelers can be used as an alternative, between thrust ring and casing. Use of feelers in the thrust pad/collar gap is likely to cause damage and may give false readings.

The sitting of the main block close to the propulsion machinery reduces any problems due to differential expansion of the shaft and the hull. The low hull temperature of midships engine refrigerated cargo ships, caused a contraction relative to the shaft of perhaps 20 [mm]. Other problems associated with the stern tube end of the shafting system include whirl of the tailshaft, relative movement of the hull and misalignment due to droop from the propeller weight. Some thrusts are housed in the after end of large slow speed diesels or against gearboxes. Deformation produced by the thrust load, can cause misalignment problems, unless suitable stiffening is employed (particularly with an end of gearbox installation).

The substantial double bottom structure under the main propulsion machinery provides an ideal foundation for the thrust block and a further reason for sitting it close to the engine. The upright thrust block and any supporting stool, must have adequate strength to withstand the effect of loading which tends to cause a forward tilt. This results in lift of the aft journal of the block (unless not fitted) and misalignment of the shaft.

Axial vibration of the shaft system, caused by slackening of the propeller blade load as it turns in the stern frame or by the splay of diesel engine crankwebs, is normally damped by the thrust block. Serious vibration problems have sometimes caused thrust block rock, panting of the tank top and structural damage.

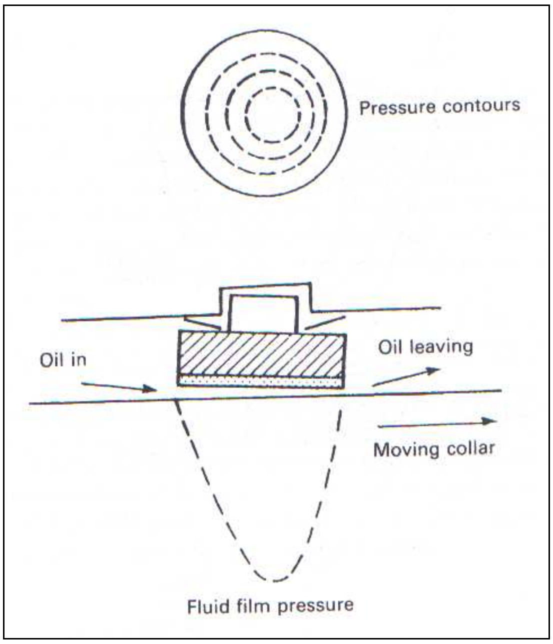


Fig. 2.62b – Circular thrust pad.

The pivot position of thrust pads may be central or offset. As illustrated in Figure 2.63, offset pads are interchangeable in thrust blocks for direct reversing engines, where the direction of load and rotation changes. Offset pads for non-reversing engine and controllable pitch propeller installations are not interchangeable. Two sets are required. Pads with a central pivot position are interchangeable. Some modern thrust blocks are fitted with circular pads (see Figure 2.62b) instead of those with familiar kidney shape. A comparison of the pressure contours on the conventional kidney shaped pads and the circular type shows why the latter are effective.

Fig. 2.63 – Self-equalizing main thrust bearing, fitted with kidney shaped interchangeable thrust pads.

Outboard Bearings:

Outboard bearings can be further classified as stern tube or strut bearings. Figure 2.31 shows the locations of these bearings relative to the ship arrangement.

Outboard bearings can be either water or oil lubricated. Nearly all outboard bearings (specially the stern tube bearings) were water lubricated until about 1960, when a transition to oil-lubricating bearings began.

The traditional stern bearing (see Figure 2.64) is water-lubricated and consists of a number of lignum vitae staves held by bronze retaining strips, in a gunmetal bush. The staves are shaped with "V" or "U" grooves between them at the surface, to allow access for cooling warter. More recently, as an alternative to wood, reinforced rubber or Tufnol has been applied to staves. As a general rule of thumb, bearing length is equal to four times shaft diameter.

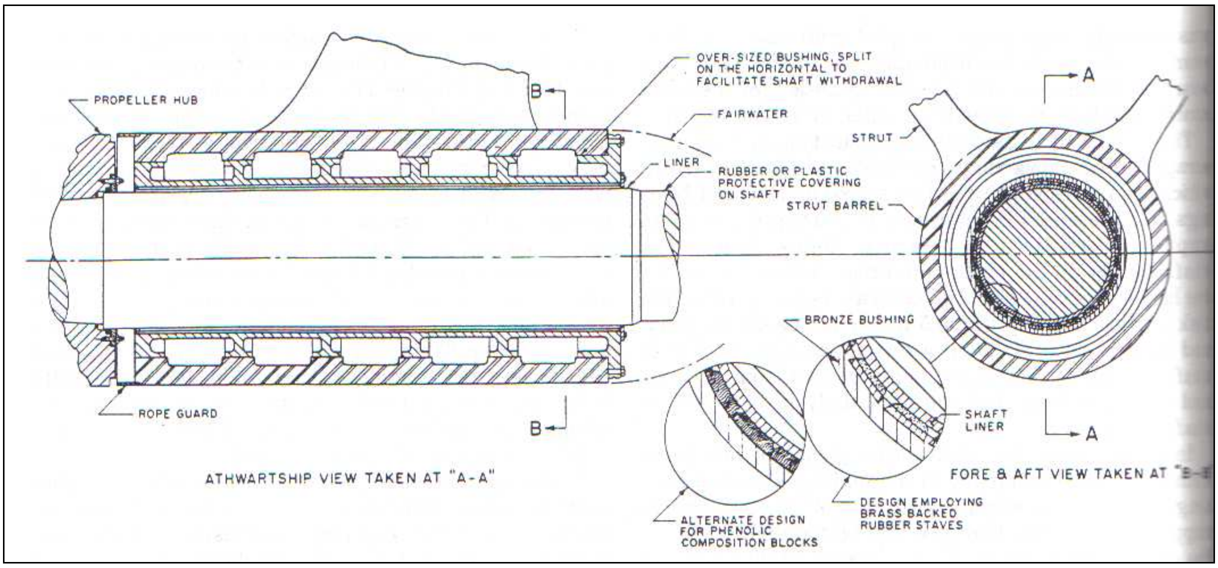


Fig. 2.64 – Water lubricated strut bearing.

Stern tubes (see Figure 2.65) are supported at the after end by the stern frame boss and at the forward end in the aft peak bulkhead. Their cast iron construction requires strong support in way of the bearing, from the stern frame boss. On a seawater lubricated stern tube, a steel nut at the outboard retains the tube in position, with its collar hard against the sternframe and the bearing section firm within the stern frame boss. Seawater enters the aft end or from the circulation system to cool and lubricate, which is an electrolyte and will support galvanic corrosion. On an oil-lubricated stern tube such as the one shown in Figure 2.65, two seals: one mounted after (outboard) and another forward of the tube prevent the ingress of seawater and leakage of oil, respectively. Oil enters the oil channels from a circulating system to cool and lubricate all the inner parts of the bearing tube and then leaves by the oil drain fitted at the bottom. The pressure differential between the oil in the stern tube and the ambient seawater can be accomplished by means of a head tank that is located about 3 meters above the full-load waterline.

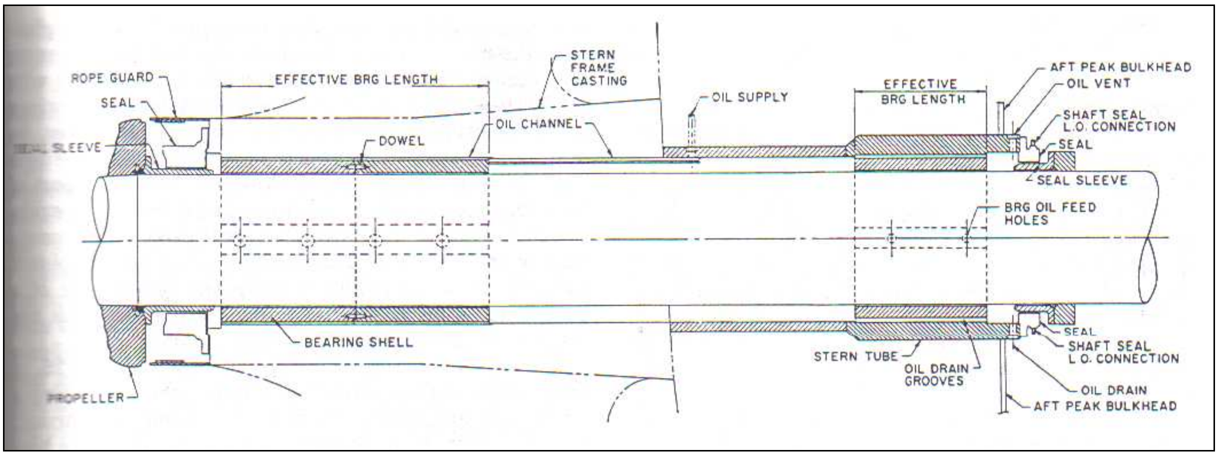


Fig. 2.65 – Typical oil-lubricated stern tube bearing.

<u>Stern Tube Sealing Arrangements</u>

There are basically three sealing arrangements used for stern bearings. These are:

a) Simple stuffing boxes filled with proprietary packing material;

b) Lip seals, in which a number of flexible membranes in contact with the shaft, prevent the passage of fluid along the shaft (see Figure 2.66 and Figure 2.66a);

c) Radial face seals, in which a wear-resistant face fitted radially around the shaft, is in contact with similar faces fitted to the after bulkhead and to the after end of the stern tube. A spring system is necessary to keep the two faces in contact.

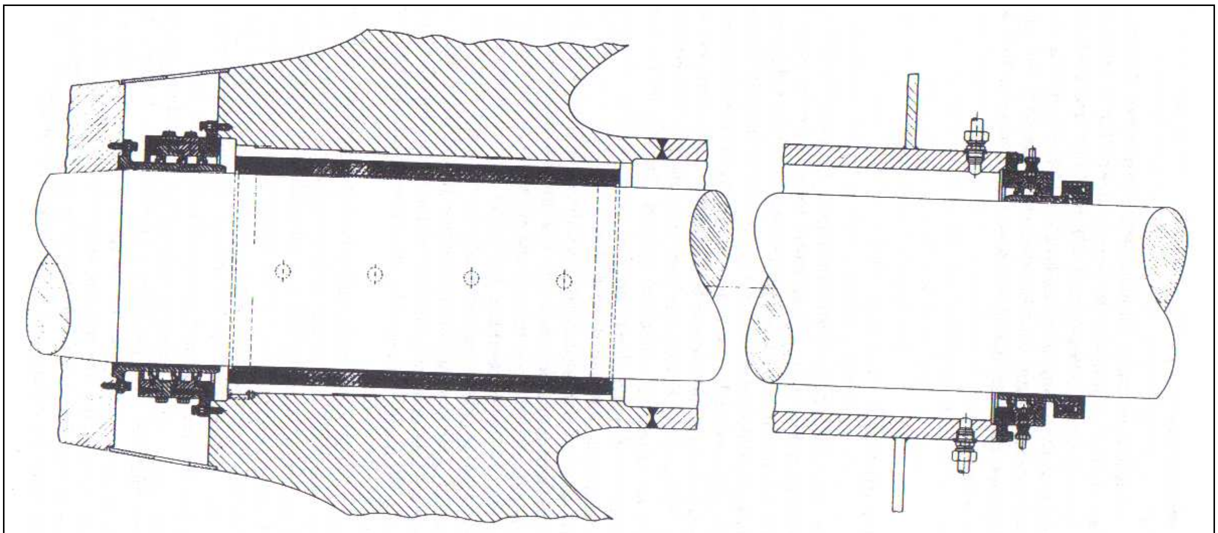


Fig. 2.66 –Oil-lubricated stern tube bearing (Simplex type).

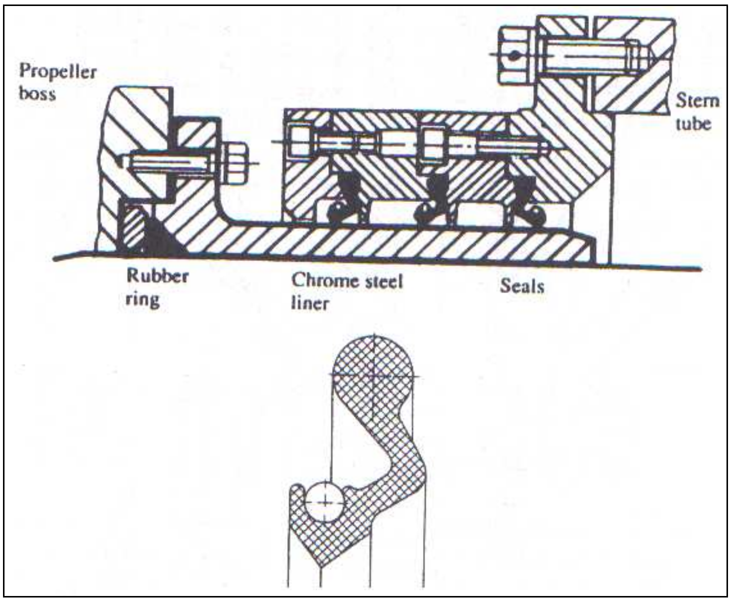


Fig. 2.66a –Lip seal assembly (nitrile rubber ring cross-section).

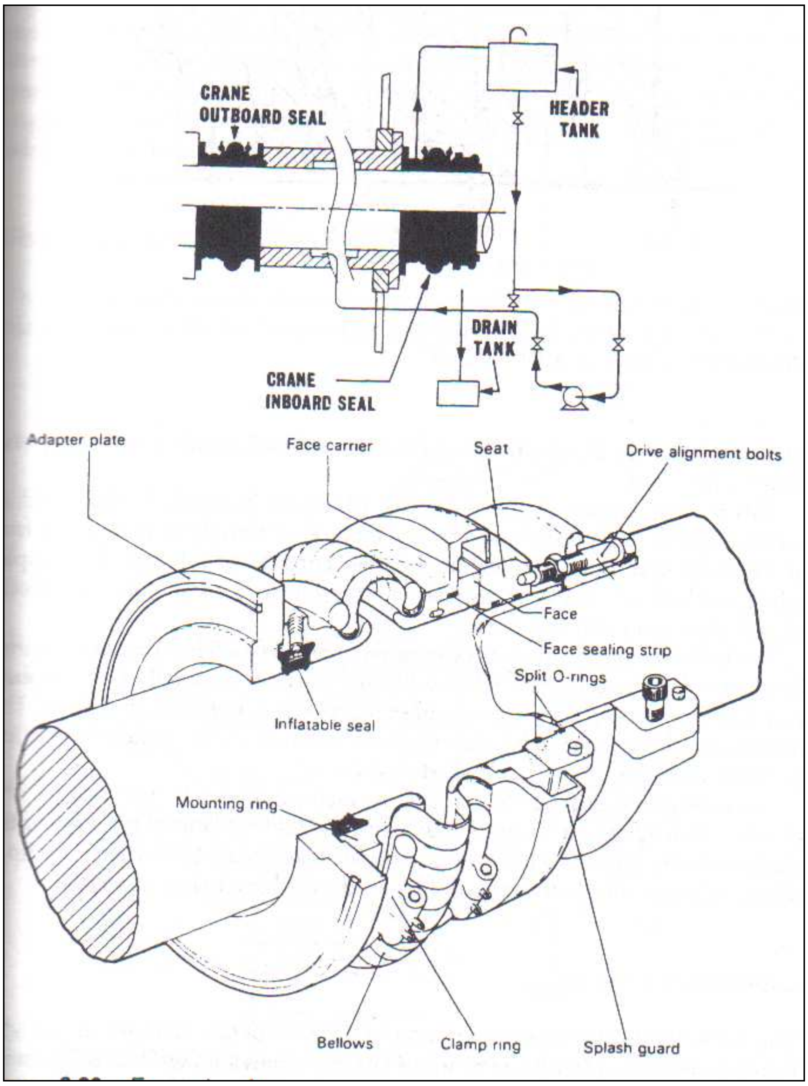


Fig. 2.67 – Radial face seal assembly.

<u>Oil-Lubricated vs Water-Lubricated Stern Tube Bearings:</u>

Therefore, oil-lubricated stern tube bearings present the following advantages in comparison with water-lubricated stern tube bearings:

- Lower levels of vibrations associated with smaller clearances and a more efficient cooling system;

- Enhanced stern tube bearing lubrication conditions and therefore less weardown;

- Lower friction losses and therefore an increased mechanical propulsive efficiency;

In terms of disadvantages of oil-lubricated stern tube bearings relatively to water-lubricated, these are associated with higher acquisition costs.

<u>Shaft Bearings Design Considerations:</u>

Shaft bearings both inside and outside the stern tube are to be so disposed that, when the plant is hot and irrespective of the condition of loading of the ship, each bearing is subjected to approximately equal positive reaction forces. Guiding values of maximum pressure at the bearing, based on projected area (shaft diameter times bearing length), is normally under 10 [Kg/cm^2] and number of bearings (R) can be determined on a first approach as follows:

$$R = \frac{W}{pD^2 \frac{L}{D}} \tag{2.94}$$

, where:

W = total weight to be supported (other weight such as gearing and stern tube must be subtracted), [kg];

p = maximum pressure at the bearing, [kg/cm^2];

D = shaft diameter (usually increased by 3 to 6 [mm] at the bearing zone, [cm];

$\frac{L}{D}$ = length-diameter ratio of the bearing.

By appropriate spacing of the bearing and by the alignment of the shafting in relation to the coupling flange at the engine or gearing, care is to be taken to ensure that no undue transverse forces or bending moments are exerted on the crankshaft or gear shafts when plant is hot. By spacing the bearings sufficiently apart, steps are also be taken to ensure that the reaction forces of the line or gear shaft bearings are not appreciably affected should the alignment of one or more bearings be altered by hull deflections or by displacement or wear of the bearings themselves.

Guide values for the maximum permissible distance (d) between bearings are given by:

$$\begin{aligned} \frac{d}{D} &= 14 \quad (250\text{ mm} < \text{D} < 405\text{ mm}) \\ \frac{d}{D} &= 12 \quad (405\text{ mm} < \text{D} < 760\text{ mm}) \end{aligned} \tag{2.95}$$

2.6.5. Couplings

Couplings are semi-permanent connections between two shafts or shafting sections; there are three different groups of types of couplings:

a) Rigid couplings;

b) Flexible couplings;

c) Hydraulic couplings.

a) <u>Rigid Couplings:</u>

Except in instances where special considerations preclude their use, shafting sections are connected by means of rigid couplings; these can be divided into four different types:

a.1) Flange with bolts coupling (see Figure 2.69):

a.2) Flanged compression coupling (see Figure 2.68):

a.3) Ribbed coupling (see Figure 2.70):

a.4) Seller conevise coupling (see Figure 2.68):

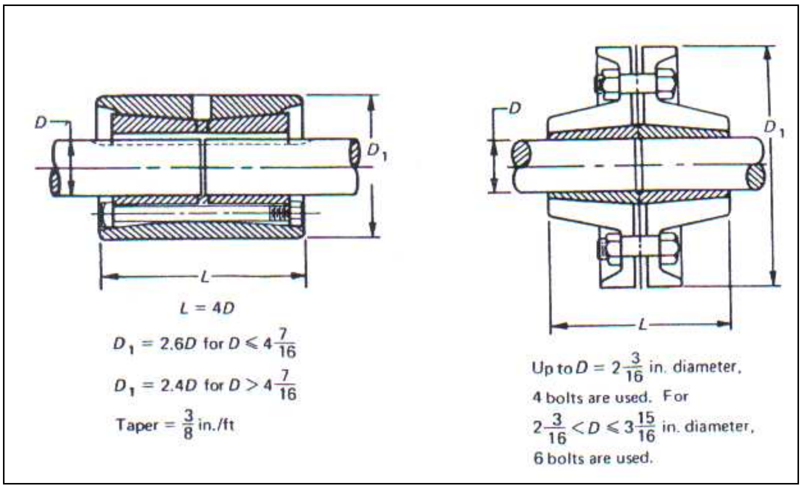


Fig. 2.68 – Seller conevise (left) and flanged compression (right) couplings.

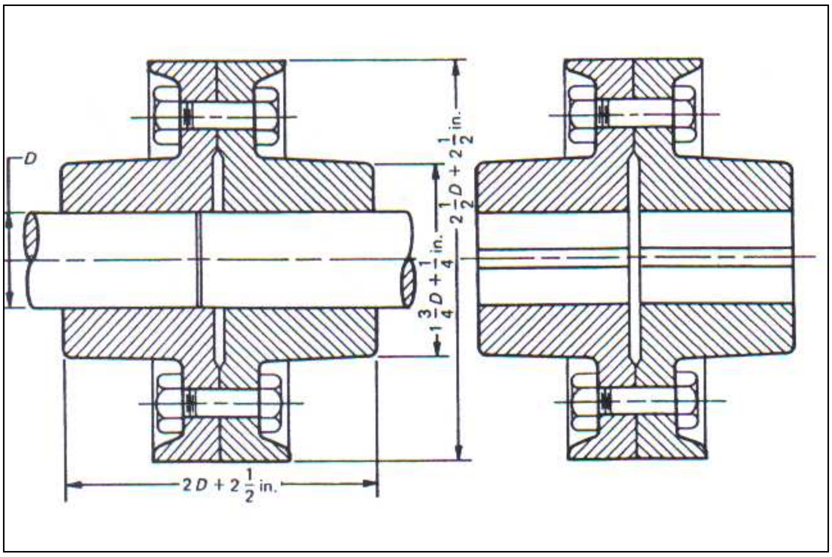


Fig. 2.69 – Flange with bolts coupling.

Fig. 2.70 – Ribbed coupling.

b) Flexible Couplings:

Flexible couplings are able to withstand slight lateral or angular shaft misalignments. Apart from withstanding slight shaft misalignments, flexible coupling are able to protect the gearboxes. For example, where a gearbox is fitted, a torsionally flexible coupling is necessary between the medium-speed diesel and the reduction gear. The coupling is necessary because the periodic application and reduction of torque as engine cylinders fire in turn, tends to result in alternate loading and unloading of the gear teeth. The torsional vibration effect is sufficient to cause serious tooth damage. Flexible couplings may be installed as separate entities or in conjunction with air or oil operated clutches. Flexible couplings may be built in common casing with the clutch.

Several types of flexible coupling are shown in Figures 2.71 to 2.86.

The Geislinger coupling shown in Figure 2.77 has a housing and a hub connected by leaf springs, which flex in service to absorb torsional effects from the engine.

For larger angular shaft misalignments, Hooke's universal joints or also designated cardan joints are utilized to connect two shafts (see Figure 2.86). These type of joints can withstand angular shaft misalignments up to 30º. Moreover, using intermediate shafts it is possible to transmit power without changing the angular speed between to parallel shafts.

c) <u>Hydraulic Couplings:</u>

On hydraulic couplings there is no mechanical connection between two shafts or shafting sections. Power is transmitted by means of kinetic energy of the fluid actuating between the impeller and the drum.

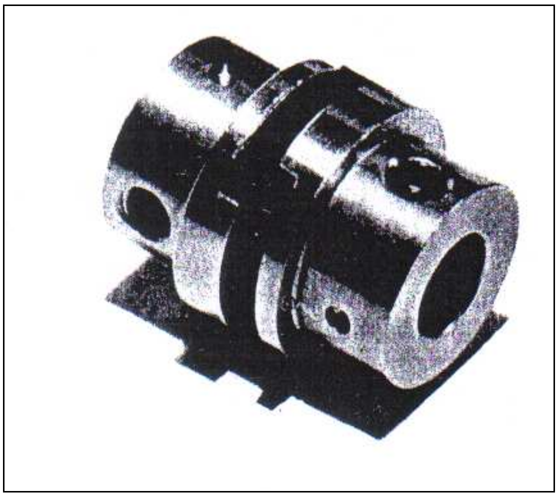


Fig. 2.71 Double slider coupling.

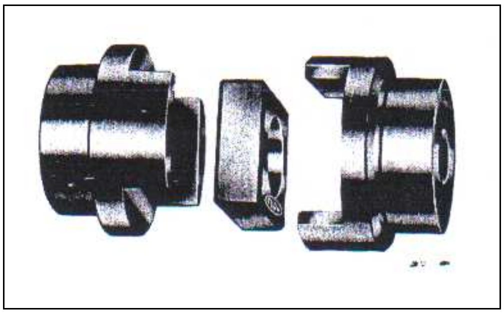


Fig. 2.72 Slider block coupling.

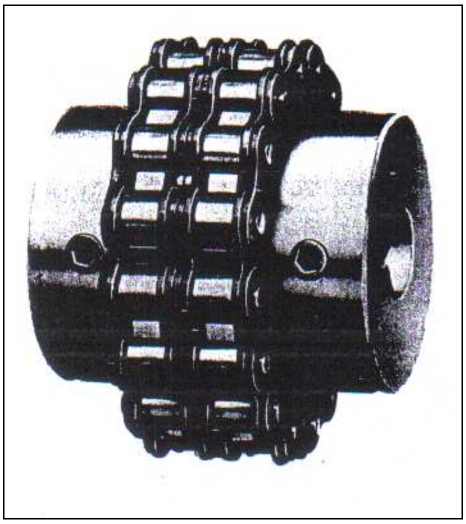


Fig. 2.73 Double roller chain coupling.

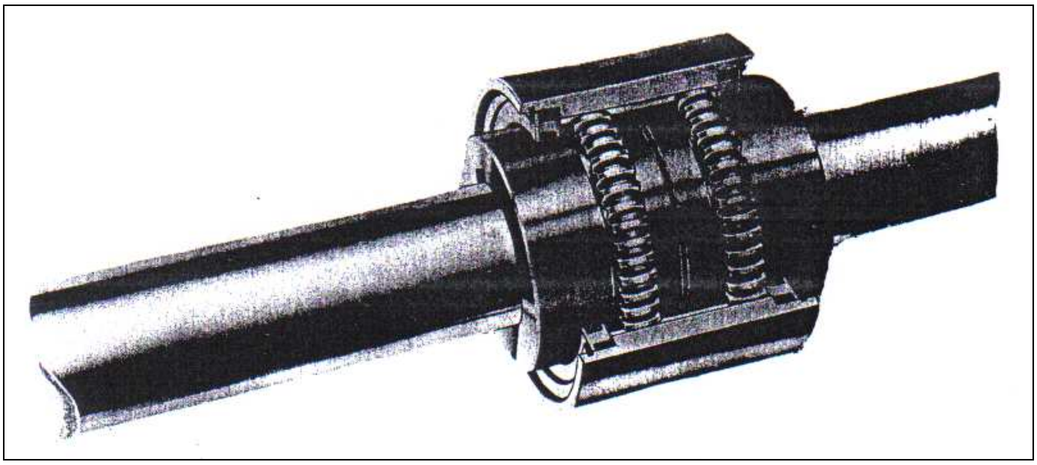


Fig. 2.74 Gear coupling.

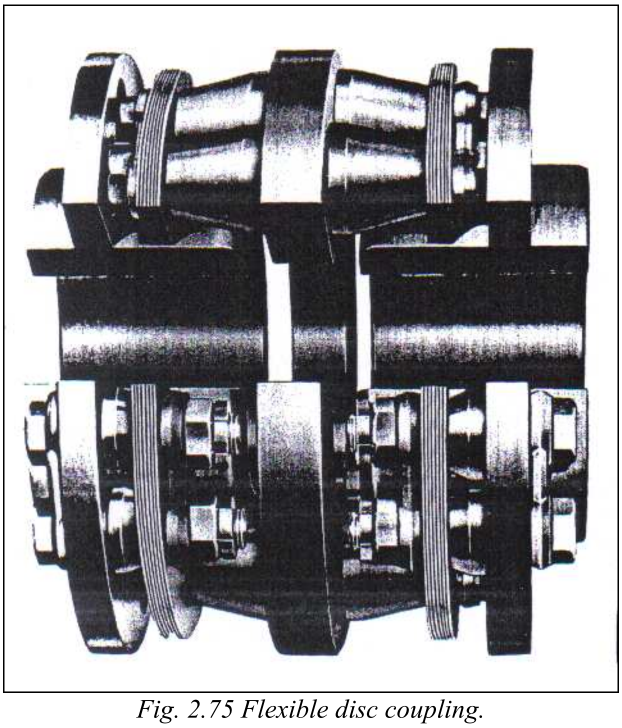


Fig. 2.75 Flexible disc coupling.

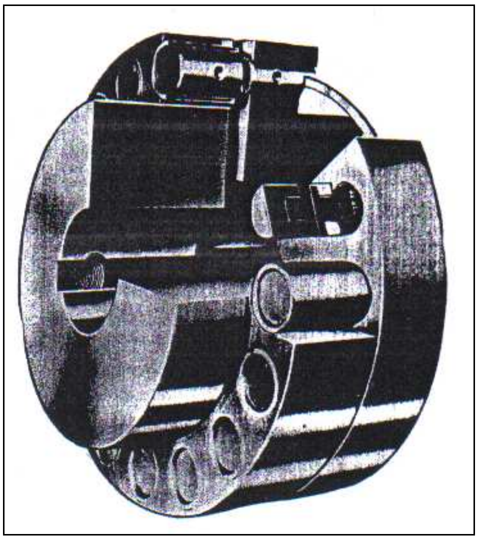


Fig. 2.76 Franke pin coupling.

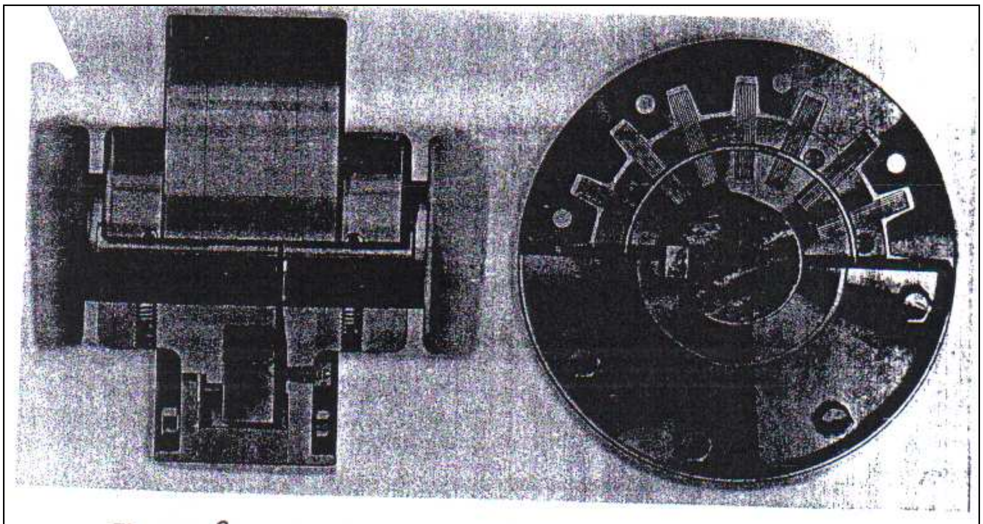


Fig. 2.77 Laminated metal radial spoke coupling.

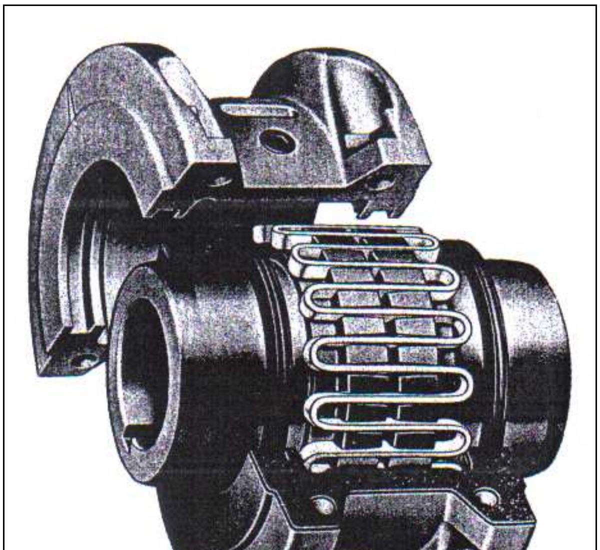


Fig. 2.78 Steelflex coupling.

Fig. 2.78 Sure-flex coupling.

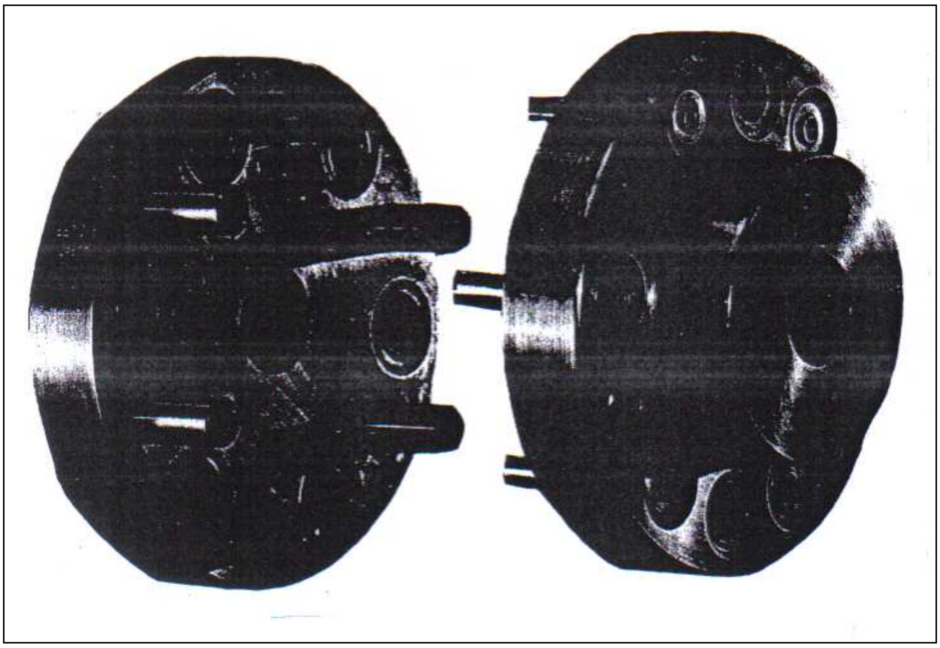


Fig. 2.80 Ajax rubber-cushioned sleeve bearing coupling.

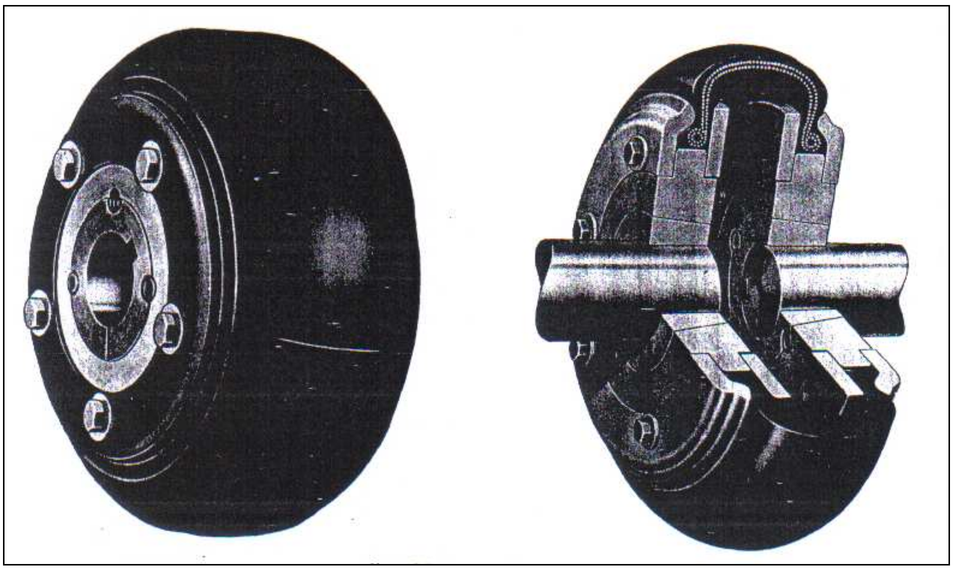


Fig. 2.81 Para-flex coupling.

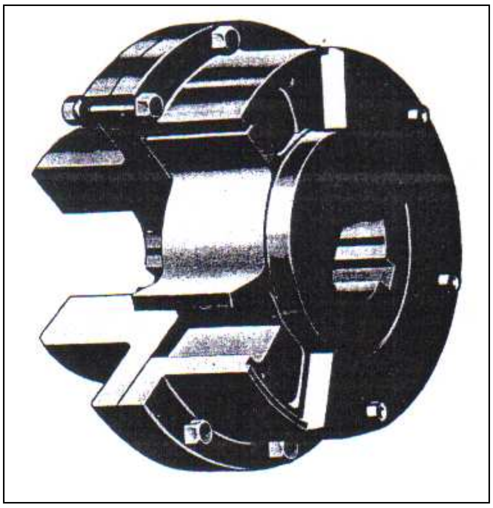


Fig. 2.82 Rubber insert coupling.

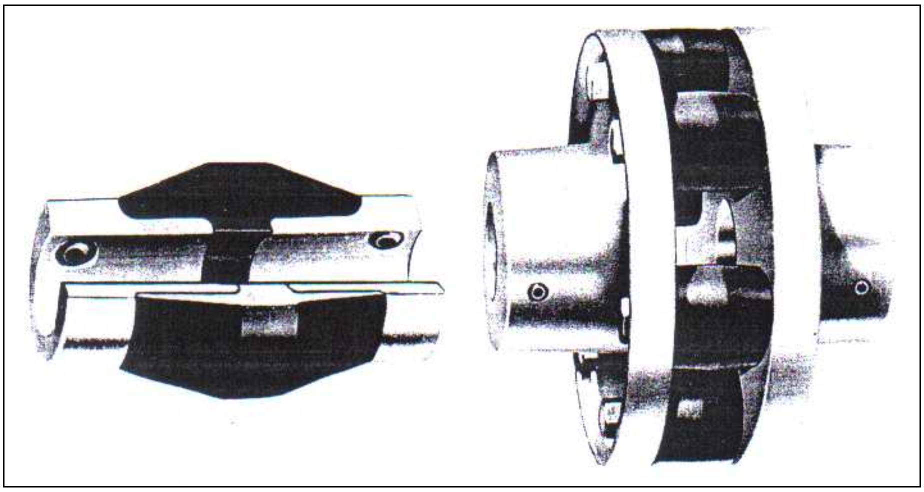


Fig. 2.83 Bonded rubber disc coupling.

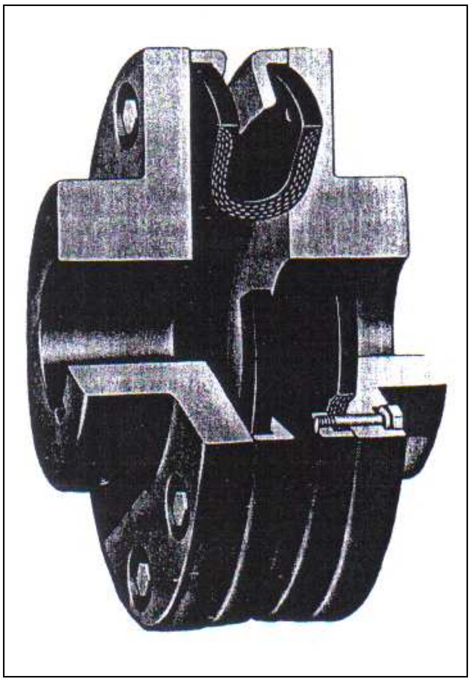


Fig. 2.84 Airflex coupling.

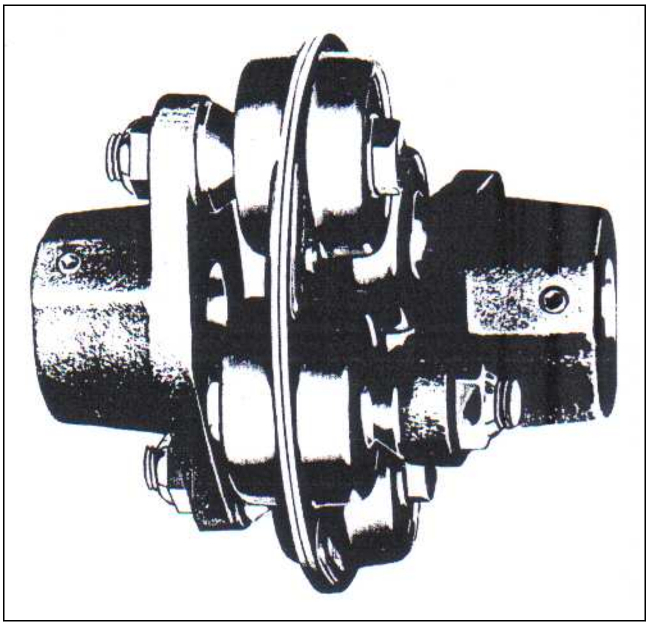


Fig. 2.85 Morflex coupling.

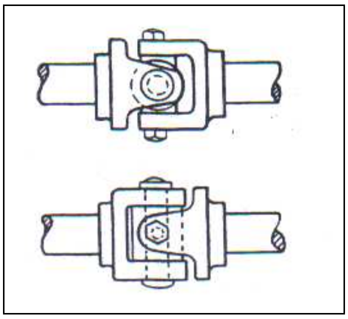


Fig. 2.86 Hooke's univerasal coupling.

2.6.6. Clutches

Clutches are movable connections between two shafts which allow them to be disconnected when rotating, and therefore can be used to control power transmission; there are four different groups of types of clutches:

a) Positive contact clutches;

b) Friction clutches;

c) Electromagnetic clutches;

d) Other types of clutches.

a) <u>Positive Contact Clutches:</u>

Positive contact clutches allow torque transmission without slippage; the most common are the teeth clutches, which can be sub-divided into square-jaw clutches (see Figures 2.87 and 2.89), these can be activated on both senses of rotation, and spiral-jaw clutches (see Figures 2.88), these can be only activated on a single sense of rotation; the engagement speed is limited to 10 [rpm] for the first and to 150 [rpm] for the latter type; for disengagement of the clutch under loading conditions, teeth should have an improved finishing and must be lubricated.

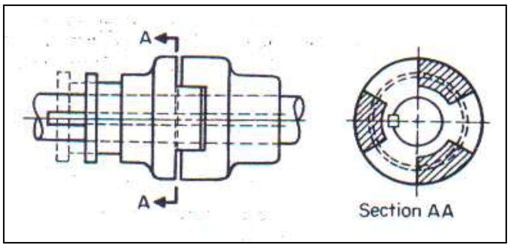


Fig. 2.87 Square-jaw clutch.

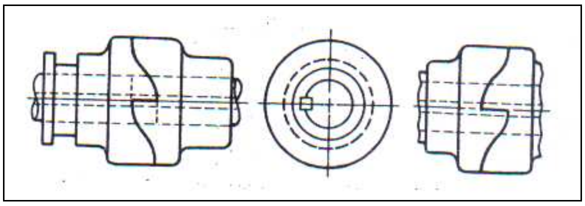


Fig. 2.88 Spiral-jaw clutch.

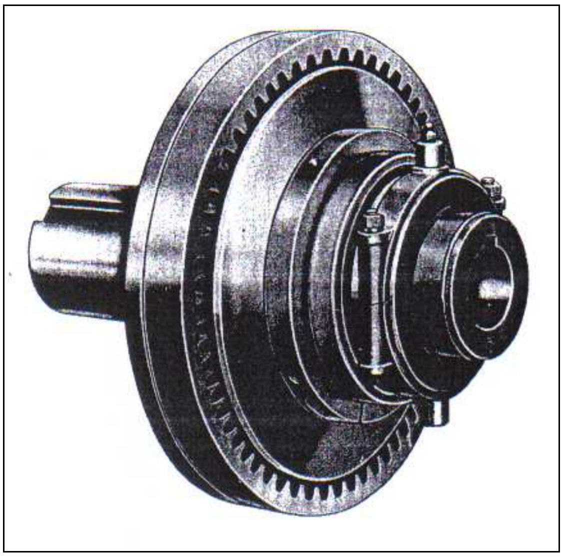


Fig. 2.89 Serrated tooth positive contact clutch.

Apart from having a low engagement speed, other disadvantages of the teeth clutches are the possibility of occurrence of shock on gears and the need to have a certain relative motion between the two rotating elements in order to allow for their engagement while they are stopped. The advantages of these types of clutches are the simplicity of construction, the absence of slippage when engaged, and the reduced heat dissipation under operation.

b) Friction Clutches:

Friction clutches are specially designed to reduce the engagement shock by means of slippage during this engagement stage; these can be also used as a safety device to prevent damage from a large torsional vibration load exceeding the maximum allowable value also by means of slippage.

Friction clutches, which are the most common, can be sub-divided into radial and axial contact clutches, depending on the direction of the exerted contact pressure.

Cone clutches (see Figures 2.91 and 2.92) and disc type clutches (see Figure 2.90) are examples of axial contact clutches; the disc type clutches can be either single disc or multiple discs type.

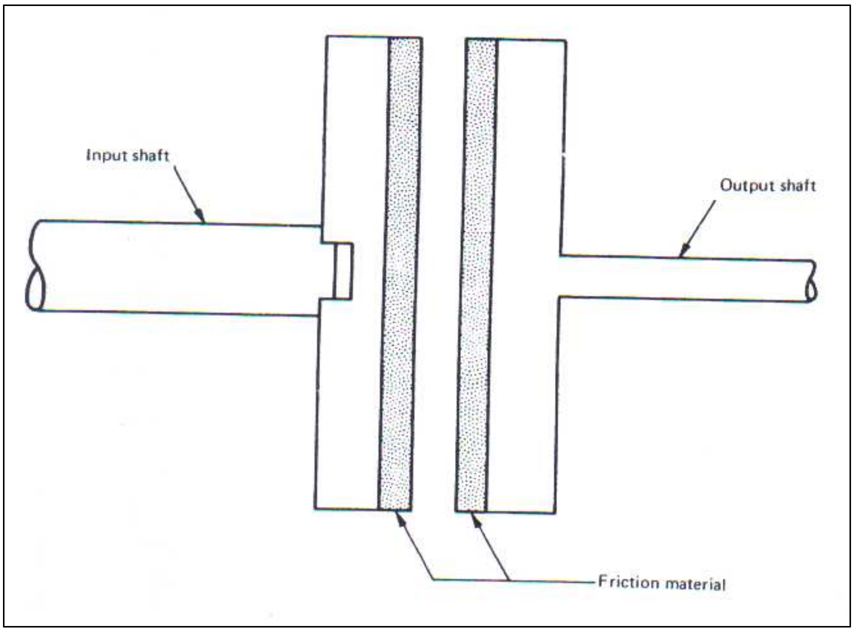


Fig. 2.90 Disc type clutch.

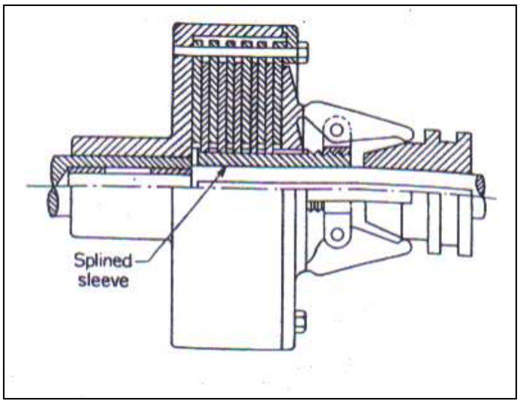


Fig. 2.91 Multidisc clutch.

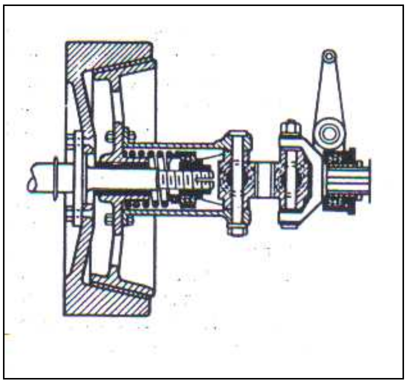


Fig. 2.92 Cone clutch.

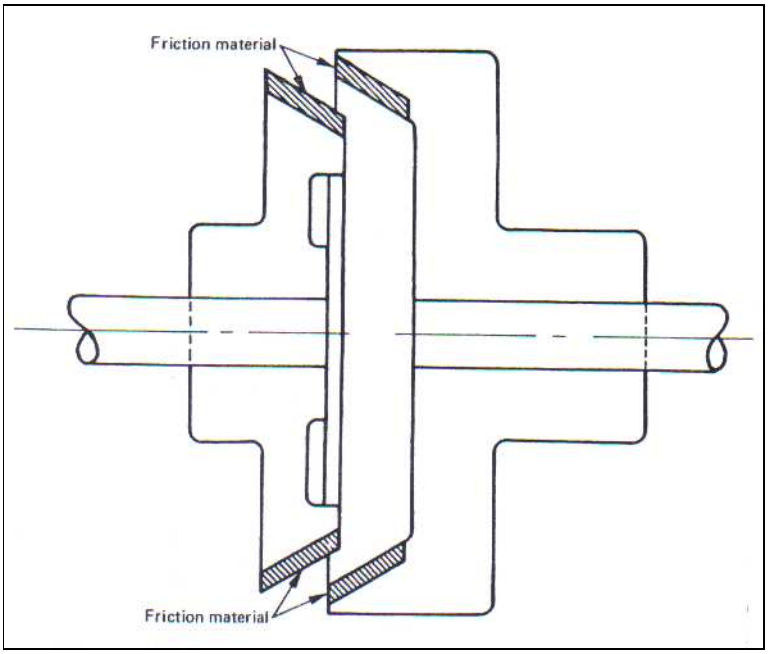


Fig. 2.93 Cone clutch (schematic).

Radial contact clutches can be further sub-divided into band clutch (see Figure 2.94), rim or drum type (see Figures 2.96 and 2.99) and overrunning clutches (see Figures 2.95 and 2.98), that use centrifugal compressive forces applied to the spheres or rollers; the latter automatically transmit the torque on a single sense of rotation and allow free disengagement on the opposite sense.

On friction clutches the maximum torque that can be transmitted depends on friction forces applied to contact surfaces (pads) which is transmitted by different possible mechanical devices.

Major advantages of friction clutches are the absence of shocks when engaged and the possibility of using these clutches for large rotational speeds without limitations; the major disadvantages of the friction clutches are the occurrence of slippage, the occurrence of wear on contact surfaces, which also implies a significant heat dissipation and thus requires the installation of an external cooling circuit for adequate operation.

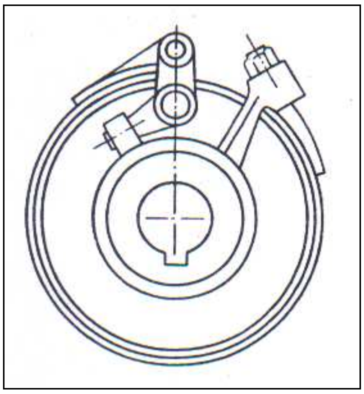


Fig. 2.94 Band clutch.

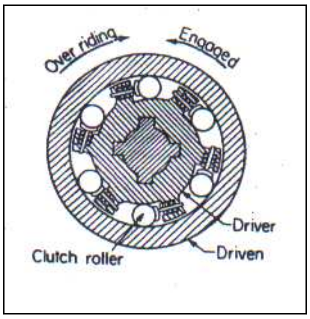


Fig. 2.95 Overruning clutch.

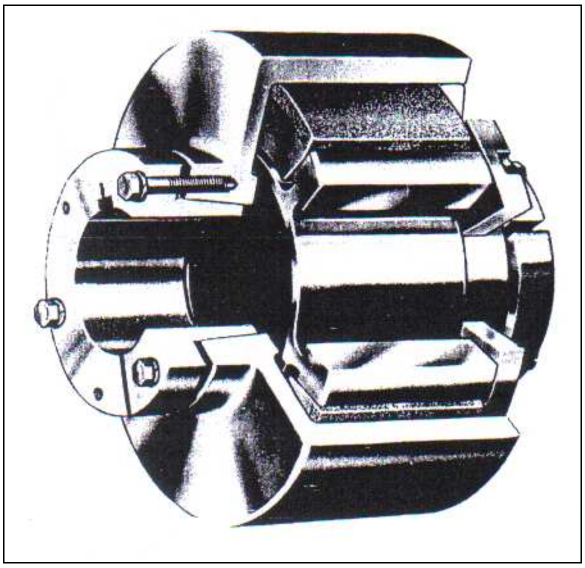


Fig. 2.96 Friction clutch of the rim or drum type.

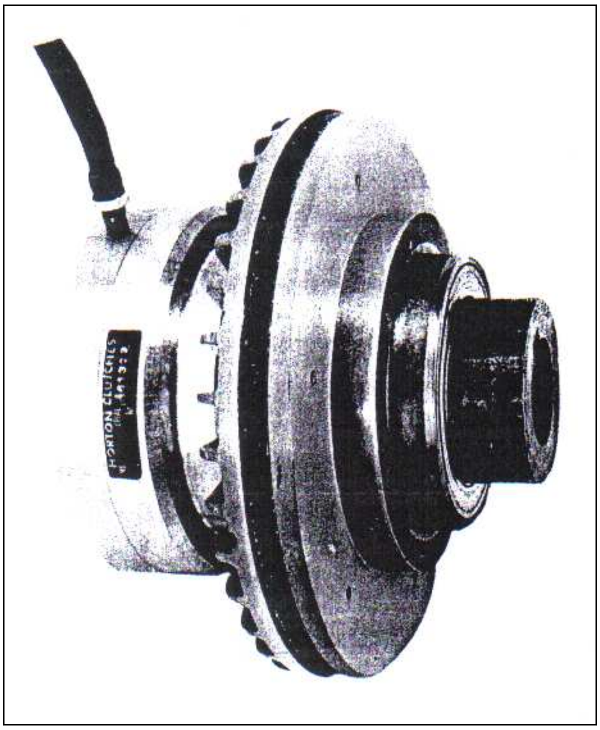


Fig. 2.97 Pneumatic air type clutch.

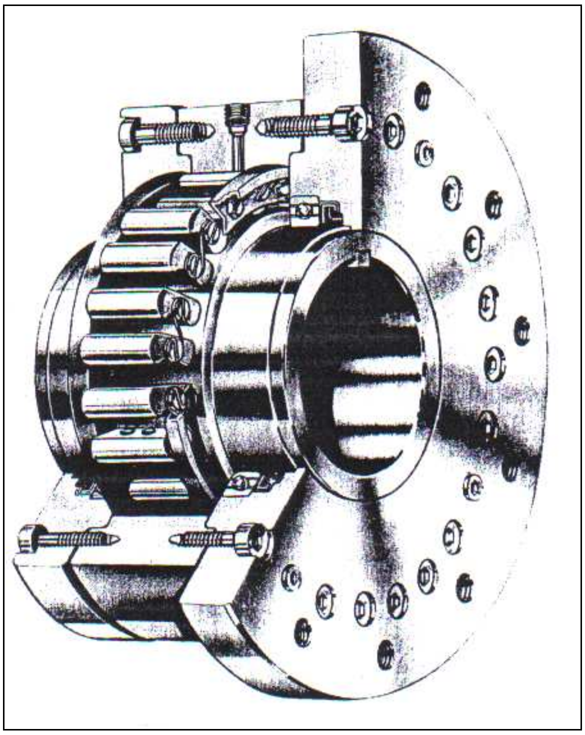


Fig. 2.98 Radial overruning clutch.

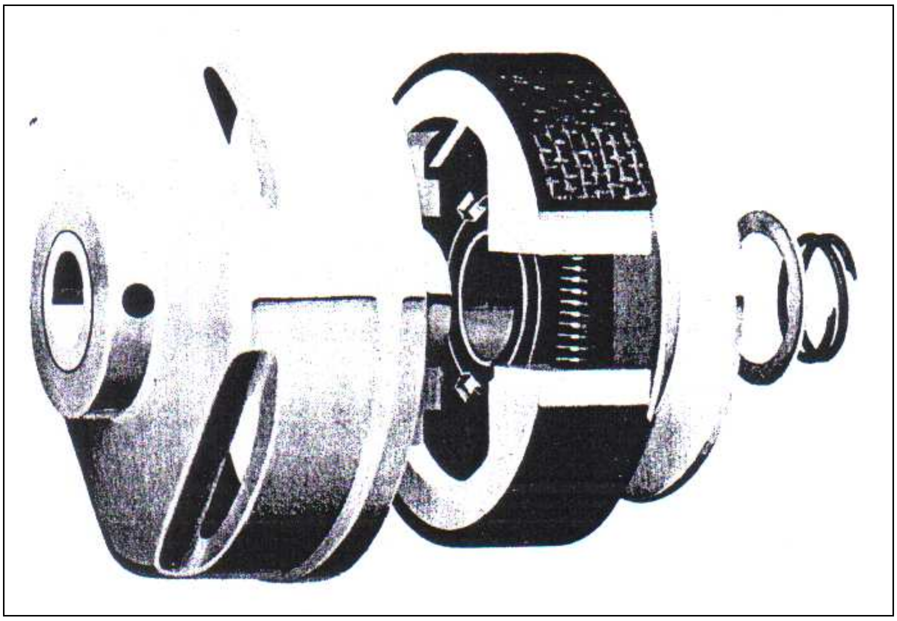


Fig. 2.99 Centrifugal type clutch.

c) Electromagnetic Clutches:

Electromagnetic clutches use a magnetic field to transmit torque between a stator and a rotor (see Figures 2.100 and 2.101).

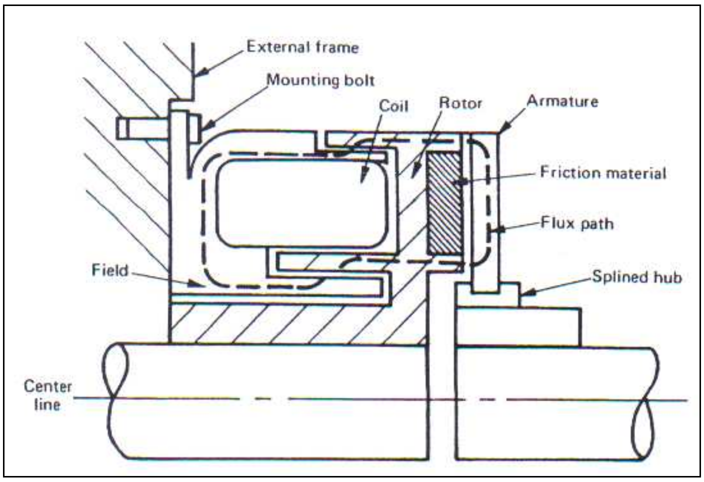


Fig. 2.100 Electromagnetic clutch (schematic).

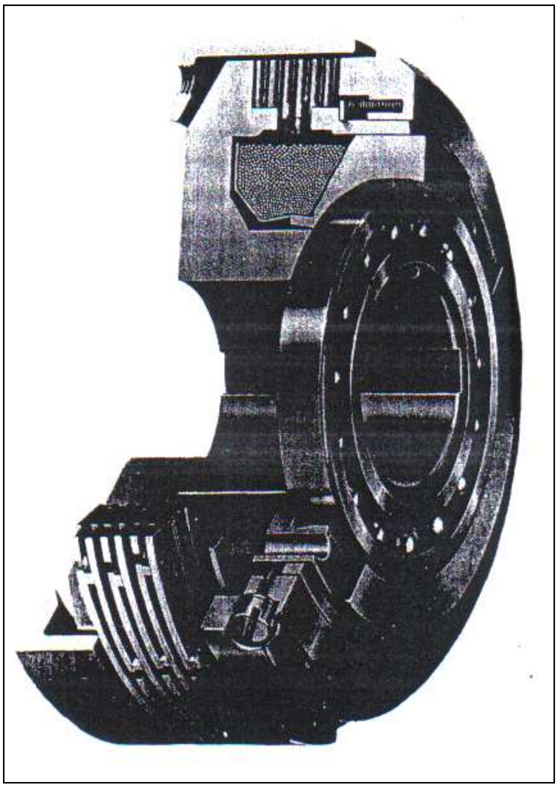


Fig. 2.101 Electromagnetic stationary coil tooth type clutch.

c) <u>Other Types of Clutches:</u>

There are also hydraulic and pneumatic clutches, which have the advantage of transmitting torque with reduced shock and vibration, relatively to mechanical clutches, and in addition allow simple actuation and remote control.

2.6.7. Propellers

a) <u>Fixed-Pitch Propeller:</u>

The propeller consists of a boss with several blades of helicoidal form attached to it. When rotating it “screws” or thrusts its way through the water by giving momentum to the column of water passing through it. The thrust is transmitted along the shafting to the thrust block and finally to the ship’s structure.

On a solid fixed-pitch propeller although usually described as fixed, the pitch does vary with increasing radius from the boss.

b) Propeller Mounting:

The propeller is fitted onto a taper on the tailshaft and a key may be inserted between the two (see Figure 2.102.); alternatively a keyless arrangement may be used. A large nut is fastened and locked in place on the end of the tailshaft; a cone is then bolted over the end of the tailshaft to provide a smooth flow of water from the propeller.

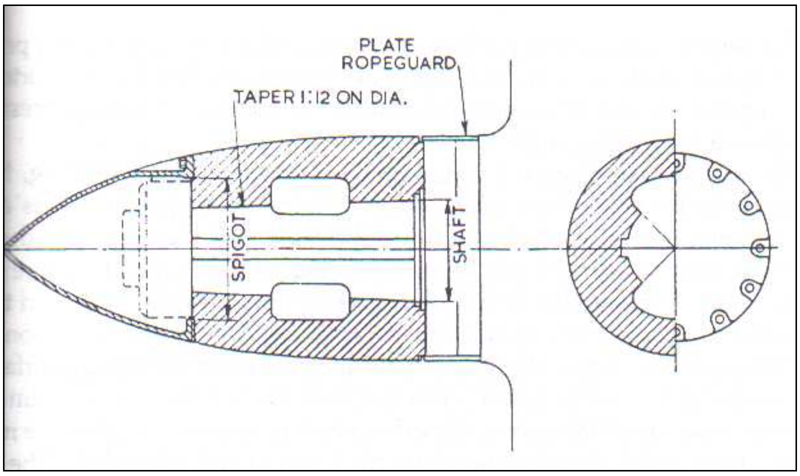


Fig. 2.102 Typical arrangement of solid propeller boss.

One method of keyless propeller fitting is the oil injection system. The propeller bore has a series of axial and circumferential grooves machined into it. High-pressure oil is injected between the tapered section of the tailshaft and the propeller. This reduces the friction between the two parts and the propeller is pushed up the shaft taper by a hydraulic jacking ring. Once the propeller is positioned the oil pressure is released and the oil runs back, leaving the shaft and propeller securely fastened together.

The Pilgrim Nut is a patented device which provides a pre-determined frictional grip between the propeller and its shaft. With this arrangement the engine torque may be transmitted without loading the key, where it is fitted. The Pilgrim Nut is, in effect, a threaded hydraulic jack which receives thrust form a hydraulic pressurized nitrile rubber tyre. This thrust is applied to the propeller to force it onto the tapered tailshaft. Propeller removal is achieved by reversing the Pilgrim Nut and using a withdrawal plate which is fastened to the propeller boss by studs. When the tyre is pressurized the propeller is drawn off the taper. Assembly and withdrawal are shown in Figure 2.103.

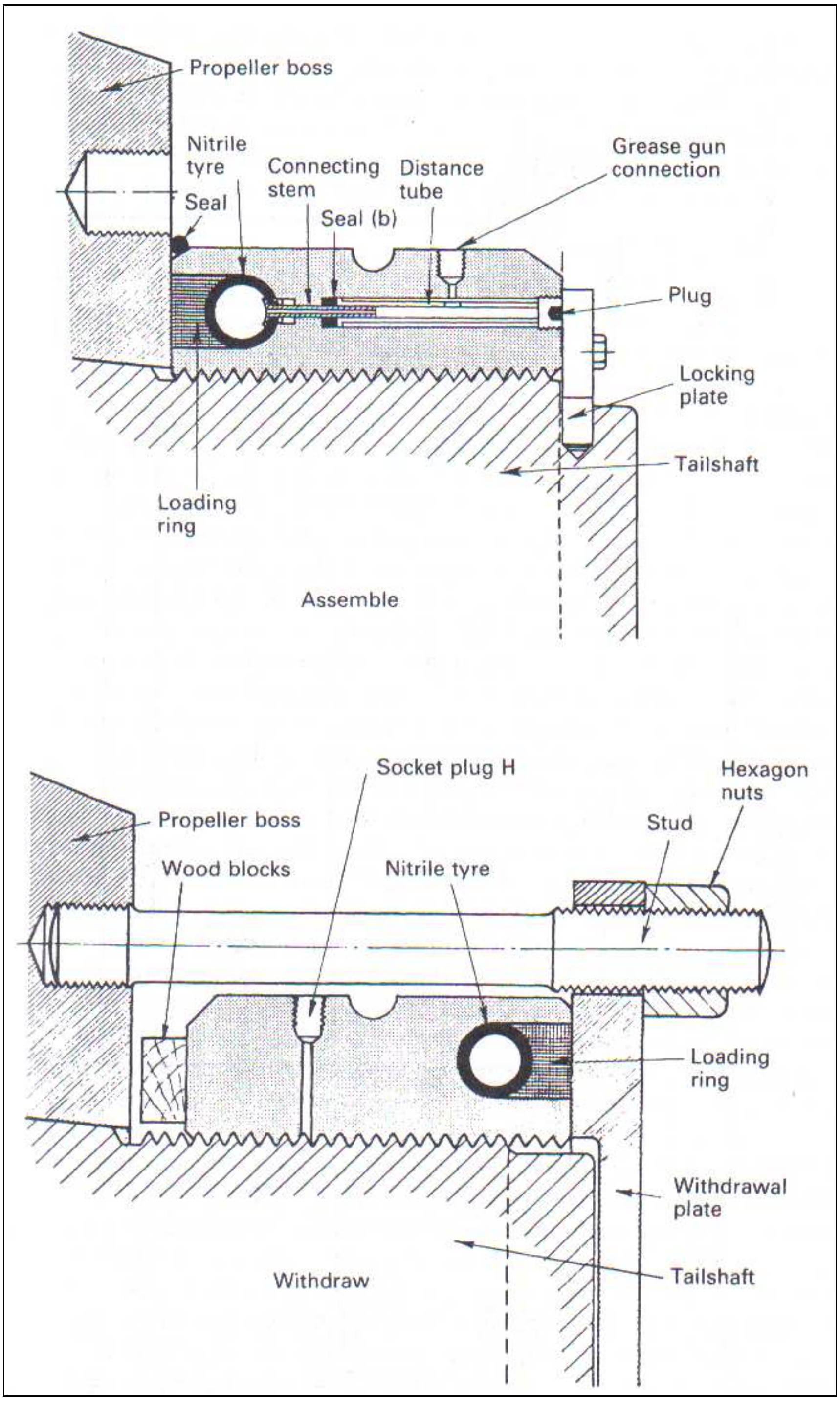


Fig. 2.103 The Pilgrim Nut. Mounting (top) and dismounting (bottom) arrangements.

c) <u>Controllable-Pitch Propeller:</u>

A controllable-pitch propeller is made up of a boss with separate blades mounted to it. An internal mechanism enables the blade to be moved simultaneously through an arc to change pitch angle and therefore the pitch. A typical arrangement is shown in Figure 2.104.

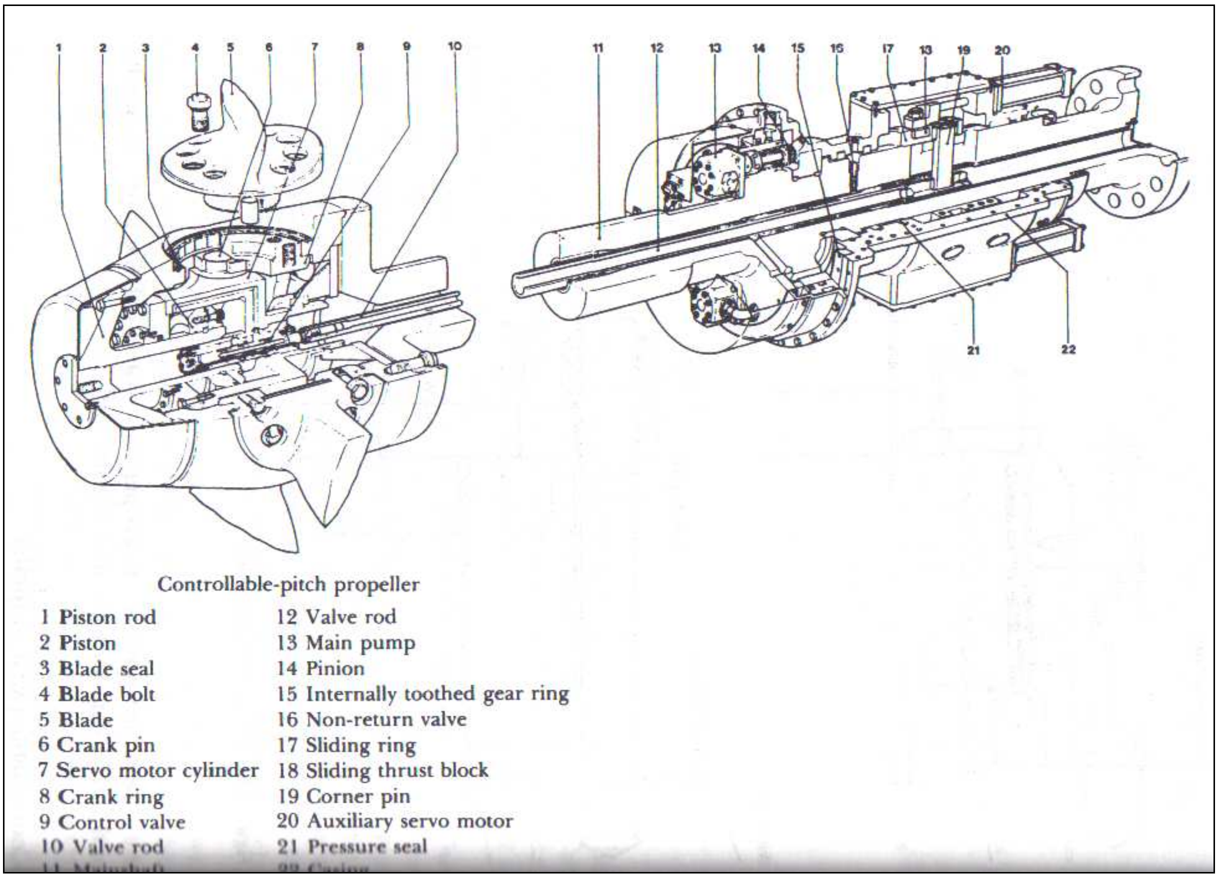


Fig. 2.104 Controllable-pitch propeller.

When a pitch demand signal is received a spool valve is operated which controls the supply of low-pressure oil to the auxiliary servo motor. The servo motor moves the sliding thrust block assembly to position the valve rod into the propeller hub where the cylinder is actuated. The cylinder movement is transferred by a crank pin and ring to the propeller blades.

The control mechanism, which is usually hydraulic, passes through the tailshaft and operation is usually from the bridge. Varying the pitch will vary the thrusts provided, and since a zero pitch position exists the engine shaft may turn continuously. The blades may rotate to provide astern thrust and therefore the engine does not require to be reversed.

d) Propeller Maintenance:

When a ship is in dry dock the opportunity should be taken to thoroughly examine the propeller, and any repair necessary should be carried out by skilled dockyard staff.

A careful examination should be made around the blade edges for signs of cracks. Edge cracks should be welded up with suitable electrodes.

Bent blades, particularly at the tips, should receive attention as soon as possible. Application of heat will be required followed by stress relieve around the repair.

Surface roughness caused by slight pitting can be lightly ground out and the area polished. More serious damage should be repaired by welding and subsequent heat treatment. A temporary repair for deep pits or holes could be done with a suitable resin filler.

2.7. Electrical propulsion

Alternating Current (AC) has now all but replaced Direct Current (DC) as the standard supply for all marine installations. The use of AC has a number of major advantages: reduced first cost, less weight, less space requirement and a reduction of maintenance requirements. Temperature affects performance of all electrical equipment and also the usefull life of the insulation and thus the equipment itself. Adequate ventilation of electric equipment is therefore essential.

As illustrated in Figure 2.105, this electric propulsion system uses AC pure sinusoidal electric power for the main switchboard, the electric motors and other electric consumers. It is available in models from 500 [kW] to 50 [MW], with low and medium voltages up to 15 [kV]. The system will reduce fuel consumption to a minimum, and leads to a major reduction of pollution from NO_x, SO_x and CO_2. It also performs silently.

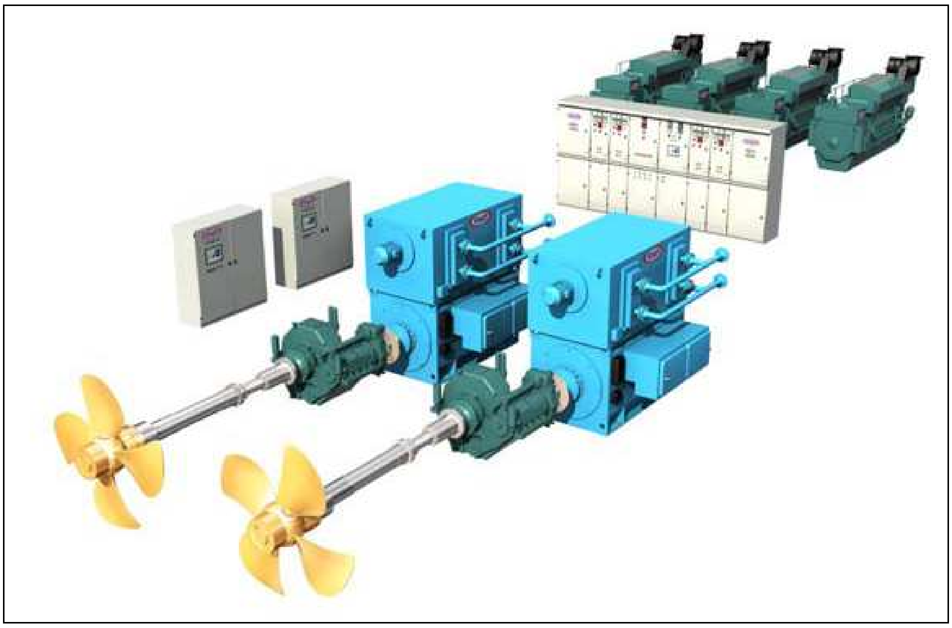


Fig. 2.105 – Fully electrical propulsion plant.

2.8. Configuration of the propulsion installation. Combined plants.

The basic choices when selecting a ship's prime mover are a diesel engine, gas turbine, or steam turbine; however, for some service requirements it may be a practical impossibility for any single prime mover to acceptably satisfy all of the operating requirements. As examples, Offshore Patrol Vessels (OPV) and naval combatants may be required to operate for extended periods of time at low, economical cruising speeds, but upon command may be required to reach maximum power, which can be several hundred percent higher, in a matter of minutes to respond to an emergency. A single prime mover may not be suitable for such extreme service requirements, and a combined plant may be a preferred choice. In combined plants, two different prime movers (although they may be different sizes of the same type) are usually connected to the propeller shaft through a common transmission system, as illustrated by Figure 2.106, to take advantage of the desirable features of each prime mover. A combined diesel or gas turbine (CODOG) plant may be a preferred choice for diverse service requirements such as those associated with OPV and small combatant vessels. During low-speed, high-endurance cruising mode, a relatively small diesel engine (the "cruise" engine) would be used; or, for maximum-power requirements, the higher-power gas turbine (the “boost” engine) would be brought on line. A mechanism such as an overriding clutch would be used to ensure that either the diesel or the gas turbine, but not both, drive the propeller. While the fuel economy and endurance of the gas turbine would be less than those of the diesel, these would be secondary considerations because of the gas turbine infrequent use. CODOG plants make it possible to minimize the operating hours on the large gas turbine, which is required only for high power requirements, and the presence of two prime movers provides a degree of redundancy. These features are advantageous in many practical situations, and CODOG plants are commonly used.

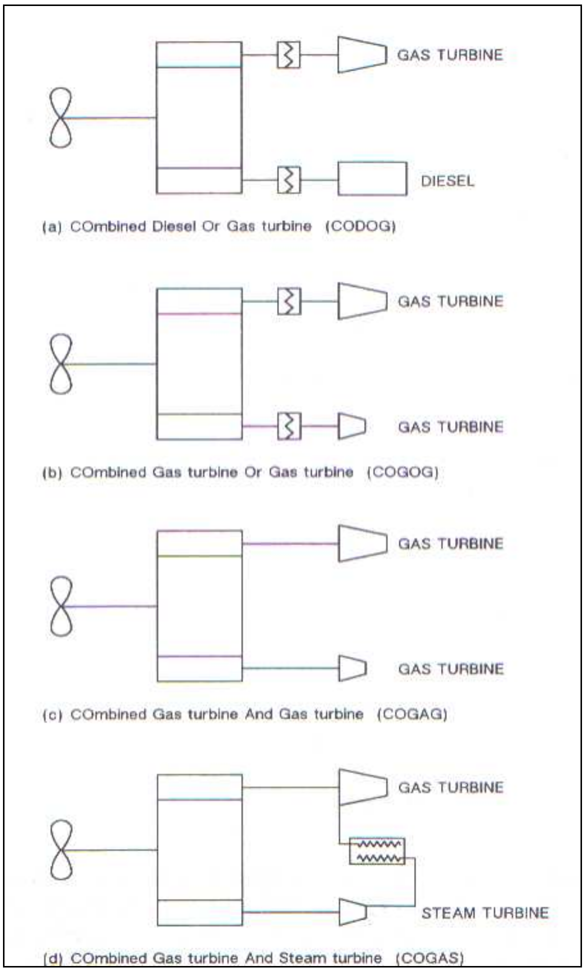


Fig. 2.106 – Typical combined propulsion plant alternatives.

In addition to CODOG plants, other typical combined plants are as follows:

- Combined gas turbine or gas turbine (COGOG);

- Combined gas turbine and gas turbine (COGAG);

- Combined gas turbine and steam turbine (COGAS);

Noteworthy features of three other combined plants are as follows:

- Combined diesel or diesel (CODOD), usually a “father” and “son” plant permits generation of propulsion power required in two distinct operating modes at engine loadings close to the optimum fuel consumption loadings;

- Combined diesel and diesel (CODAD), the flexibility to run at any speed close to the optimum fuel consumption loadings and the redundancy of this plant are highly appreciated. Four engines, twin screw, and twin gearboxes installed in two separated machinery spaces also presents a very high survivability arrangement for vessel damaged condition;

- Combined diesel and gas (CODAG), quite common in naval vessels. Potentially provides the economy of a diesel engine and the almost instant boost power of a gas turbine on demand, but coordination of the two dissimilar types of power plants may present a complex mechanical problem to solve.

Another way to achieve flexibility is to install diesel-electric and gas turbine solution (CODLAG) or a diesel-electric and gas turbine-electric solution (CODLAGL), where the propulsion power comes from a common “power station”.

2.9. Analysis of different types of configurations, identification of main advantages and disadvantages

The means selected to stop and reverse a ship are closely related to the choice of prime mover. Most direct-drive diesel engines are readily reversible. Steam turbines and gas turbines on the other hand, cannot be directly reversed and require a reversing reduction gearbox.

Electric drivers are reversed by dynamically breaking the propulsion motor and energizing the electric motor in the reverse direction.

Controllable-pitch propellers provide a more responsive reversing capability than do the alternatives mentioned above.

Another way to achieve economy savings is to install an integrated plant. The arrangement normally consists of two or more internal combustion engines coupled to shaft alternators. This integrated internal combustion-electric plant feeds both the electrical users and main propulsion system.

<u>Advantages of an Integrated Plant:</u>

- Simplified machinery operations onboard;

- Reduction of the amount of operating equipments;

- Maintenance cost reduction;

- Fuel economy;

- Noise reduction;

- Recovery of heat losses from exhaust gases of turbocharged engines.

<u>Disadvantages of an Integrated Plant:</u>

- Larger initial cost;

- Reduction of operational flexibility;

- In case of propulsion system failure, this can lead to a possible vessel black-out if no autonomous redundancy is provided;

- Reduced shaft reversing capabilities;

- Reduction of the power available for propulsion.