"Mircea cel Batran" Naval Academy Scientific Bulletin, Volume XVIII – 2015 – Issue 1
Published by "Mircea cel Batran" Naval Academy Press, Constanta, Romania // The journal is indexed in:
PROQUEST SciTech Journals, PROQUEST Engineering Journals, PROQUEST Illustrata: Technology, PROQUEST Technology Journals, PROQUEST Military Collection PROQUEST Advanced Technologies & Aerospace

THE KINEMATIC OF ENGINE DRIVE MECHANISM. CONSIDERATIONS ON BOXER ENGINE CRANK CONFIGURATIONS INFLUENCE ON THE ENGINE EQUILIBRATION

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Abstract: *In the present paper are presented in the first part some theoretical aspects linked by the engine kinematics and defined the dimensional relationships for the kinematic engine drive mechanism, submitted graphically and analytically.*
This are proceeded in the second part with considerations on crank configurations influence on the engine equilibration in general and extensively for BOXER engine configurations. Following these questions, a great importance is attached for the study of the advantages and disadvantages of using the BOXER configurations in naval engines.
Keywords*: kinematics, crank configurations, engine equilibration, BOXER engine.*

Introduction

The work done by the cyclical evolution of the motor fluid in the cylinder of the internal combustion engine is transmitted to the crankshaft via a crank and connecting rod mechanism (motor mechanism). Connecting rod is the liaison between the piston pin (crosshead pin) and time crankpin. The role is fulfilled by the crankshaft elbow.
The motor mechanism is called centered if the engine crankshaft axis intersects the cylinder axis. Motor mechanism is called unbalanced if the engine crankshaft axis does not intersect the crankshaft axis cylinder axis.
For reciprocating internal combustion engines, the transformation of the alternating translation motion of the piston in rotation motion of the crankshaft, is specific. This is achieved through a crank and connecting rod mechanism.

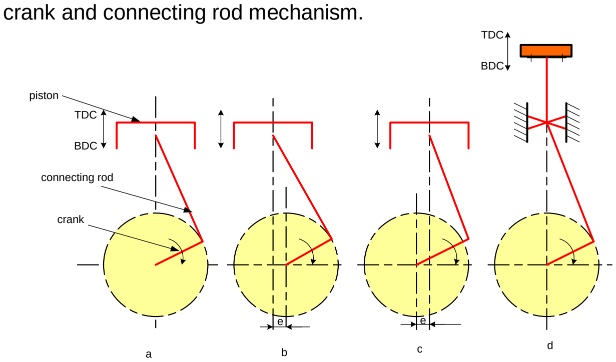

Figure 1.1. Normal type mechanisms.
a. normally centered; b,c. normally with eccentricity; d. cross head mechanism.

Most V engines use articulated mechanisms with adjacent rods (fig.1.2a). Some V engines and engines that each crank is driven by more than two rods (W, X, etc.) use a particular type of mechanism: main and side rods mechanism (fig. 1.2b.). In this case, rod 1, articulated with crankshaft crankpin is called main rod, and the one articulated by the head of the main connecting rod is called secondary rod (auxiliary connecting - rod).

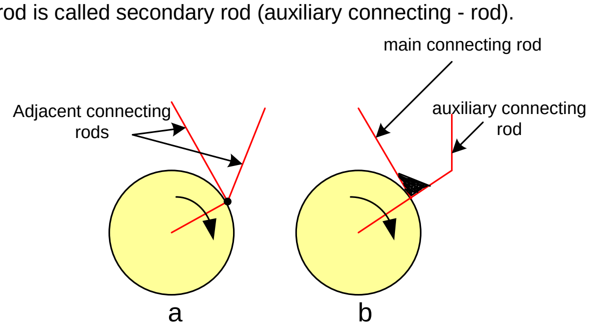

Figure 1.2. Articulated mechanisms
a – articulated mechanism with adjacent connecting rods; b – articulated mechanism with main and auxiliary connecting rods.

Unbalanced type mechanism (figure 1.3) consists of:
- *Crank*, length *R,* performs a rotation motion with constant angular speed, $\omega=\pi n/30$;
- *Connecting rod*, length *L*, performs a plane-parallel motion, being articulated at the ends with crankshaft crankpin and piston;
- *Piston*, which performs an alternating translation movement along the cylinder axis, between TDC and BDC.

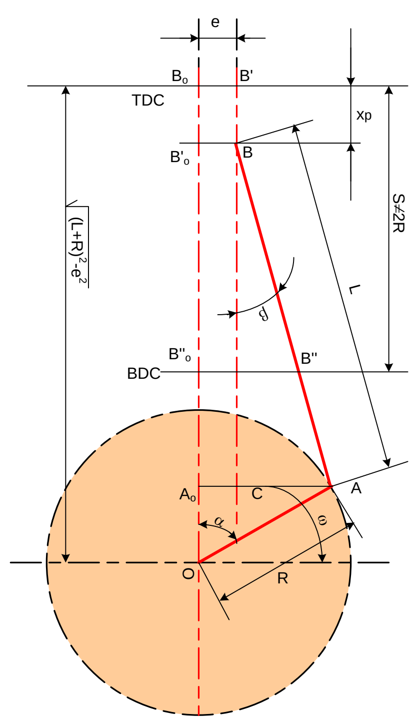

Figure 1.3. Diagram of normal and unbalanced mechanism
R – crank length; ω *- angular velocity; L – connecting rod length;* α *- crankshaft rotation angle;* β *- connecting rod obliquity; S – piston stroke;* x_p *– piston movement, corresponding to rotation angle* α*.; e – eccentricity.*

The ratio between the crank radius *R* and the connecting rod length *L,* is:

$$\lambda_d = R/L \tag{1.1}$$

and represents a mechanism compactness factor; since λ_d is higher, the connecting rod is relatively short compared to the crank and the engine construction becomes more compact

The ratio between the eccentricity and the crank radius is:

$$\delta_d = \frac{|e|}{R} \qquad (1.2)$$

And represents the misalignment of the mechanism;

After analyzing the motor mechanism kinematics, successively results, the movements, velocities and accelerations of the mechanism components. This allows the determination of forces and moments acting on the mentioned components.

Cinematic quantities expressions

Angular velocity

$$\omega = \frac{\pi \cdot n}{30} \qquad \left[\frac{rad}{s}\right] \qquad (1.3)$$

Piston movement

$$x_p = OB_0 - OB_0^{'} \qquad (1.4)$$

$$OB_0^{'} = OA_0 + A_0B_0^{'} \qquad (1.5)$$

$$OB_0 = OB_0^{''} + B_0^{''}B_0 \qquad (1.6)$$

Resulting:

$$x_p = (R + L) - (R\cos\alpha + L\cos\beta) \qquad (1.7)$$

Or:

$$x_p = R\left[(1-\cos\alpha) + \frac{1}{\lambda}(1-\cos\beta)\right] \qquad (1.8)$$

$$\sin\beta = \lambda \sin\alpha\,; \qquad (1.9)$$

$$\cos\beta = \sqrt{1-\lambda^2\sin^2\alpha}\,; \qquad (1.10)$$

It follows the exact phrase of piston displacement:

$$x_p = R\left[(1-\cos\alpha) + \frac{1}{\lambda}\left(1-\sqrt{1-\lambda^2\sin^2\alpha}\right)\right] \qquad (1.11)$$

Exact phrase of piston velocity:

$$w_p = R\varpi\left[\sin\alpha + \frac{\lambda\sin\alpha\cos\alpha}{\sqrt{1-\lambda^2\sin^2\alpha}}\right] \qquad (1.12)$$

Piston acceleration exact phrase:

$$j_p = R\varpi^2\left[\cos\alpha + \lambda\frac{\lambda^2\sin^4\alpha + \left(\cos^2\alpha - \sin^2\alpha\right)}{\left(1-\lambda^2\sin^2\alpha\right)^{\frac{3}{2}}}\right] \qquad (1.13)$$

From equation (1.11) can be obtained approximate relations for piston displacement by developing the following expression:

$$\sqrt{1-\lambda^2\sin^2\alpha}\,;$$

In power series:

$$\sqrt{1-\lambda^2\sin^2\alpha} = 1 - \frac{1}{2}\lambda^2\sin^2\alpha - \frac{1}{8}\lambda^4\sin^4\alpha - \frac{1}{16}\lambda^6\sin^6\alpha - \frac{5}{128}\lambda^8\sin^8\alpha - \frac{7}{256}\lambda^{10}\sin^{10}\alpha - \frac{21}{1024}\lambda^{12}\sin^{12}\alpha - \frac{33}{2048}\lambda^{14}\sin^{14}\alpha - \frac{1043}{32768}\lambda^{16}\sin^{16}\alpha - \frac{715}{65536}\lambda^{18}\sin^{18}\alpha - \frac{2431}{262144}\lambda^{20}\sin^{20}\alpha - \frac{46189}{524288}\lambda^{22}\sin^{22}\alpha - \frac{323323}{41943040}\lambda^{24}\sin^{24}\alpha ... \qquad (1.14)$$

Using expression:

$$\sin^{2k}\alpha = \frac{1}{2^{2k-1}}\sum_{j=1}^{k}(-1)^j C_{2k}^{k+j}\cos 2j\alpha + \frac{1}{2^{2k}}C_{2k}^{k} \qquad (1.15)$$

Is obtained:

$$\sin^2\alpha = \frac{1-\cos 2\alpha}{2} \qquad (1.16)$$

$$\sin^4\alpha = \frac{\cos 4\alpha - 4\cos 2\alpha + 3}{8} \qquad (1.17)$$

$$\sin^6\alpha = \frac{-\cos 6\alpha + 6\cos 4\alpha - 15\cos 2\alpha + 10}{32} \qquad (1.18)$$

$$\sin^8\alpha = \frac{\cos 8\alpha - 8\cos 6\alpha + 28\cos 4\alpha - 56\cos 2\alpha + 35}{128} \qquad (1.19)$$

$$\sin^{10}\alpha = \frac{-10\cos\alpha + 10\cos 8\alpha - 45\cos 6\alpha + 120\cos 4\alpha - 210\cos 2\alpha + 126}{512} \qquad (1.20)$$

Expression of piston displacement can be written as:

$$x_p = R\begin{pmatrix} A_0 + A_1\cos\alpha + \\ A_2\cos 2\alpha + A_4\cos 4\alpha + A_6\cos 6\alpha + A_8\cos 8\alpha + ... \end{pmatrix} \qquad (1.21)$$

Where

$$A_0 = 1 + \frac{1}{4}\lambda + \frac{3}{64}\lambda^3 + \frac{5}{256}\lambda^5 + \frac{175}{16384}\lambda^7 + ...$$

$$A_1 = -1$$

$$A_2 = -\frac{1}{4}\lambda - \frac{1}{16}\lambda^3 - \frac{15}{512}\lambda^5 - \frac{280}{16384}\lambda^7 \;.......$$

$$A_4 = \frac{1}{64}\lambda^3 + \frac{6}{512}\lambda^5 + \frac{140}{16384}\lambda^7 \;......$$

$$A_6 = -\frac{1}{512}\lambda^5 - \frac{40}{16384}\lambda^7 - ....$$

If in the (1.21) expression, first terms are retained, results:

$$x_p = R(A_0 + A_1\cos\alpha + A_2\cos 2\alpha)$$

Or

$$x_p = R\left[(1-\cos\alpha) + \frac{\lambda}{4}(1-\cos 2\alpha)\right] \qquad (1.22)$$

Using the (1.22) expression, results:

$$x_p = R(1 - \cos\alpha + 0{,}5\cdot\lambda_d\cdot\sin^2\alpha) \qquad [m]$$

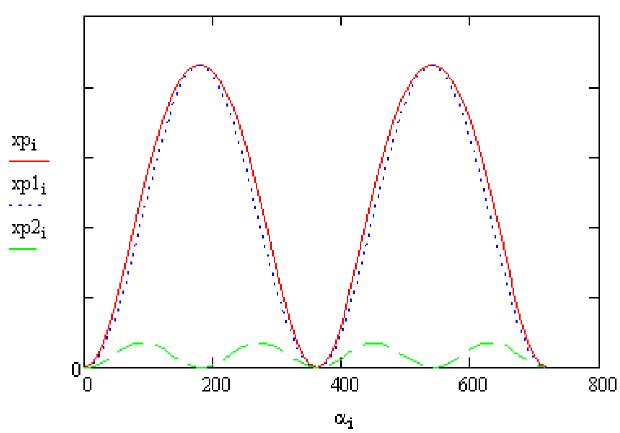

Figure 1.4. Piston movement diagram and his harmonics.

Piston velocity

$$w_p = R \cdot \omega (\sin \alpha + \frac{\lambda_d}{2} \cdot \sin 2 \cdot \alpha) \qquad \left[\frac{m}{s}\right] \qquad (1.23)$$

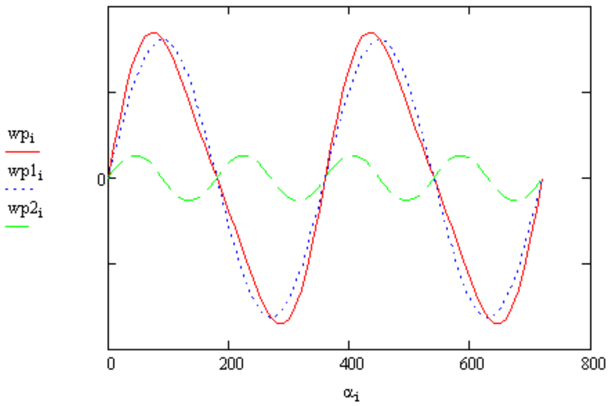

Figure 1.5. Piston velocity diagram and his harmonics.

Piston acceleration

$$j_p = R \cdot \omega^2 (\cos \alpha + \lambda_d \cdot \cos 2 \cdot \alpha) \qquad \left[\frac{m}{s^2}\right] \qquad (1.24)$$

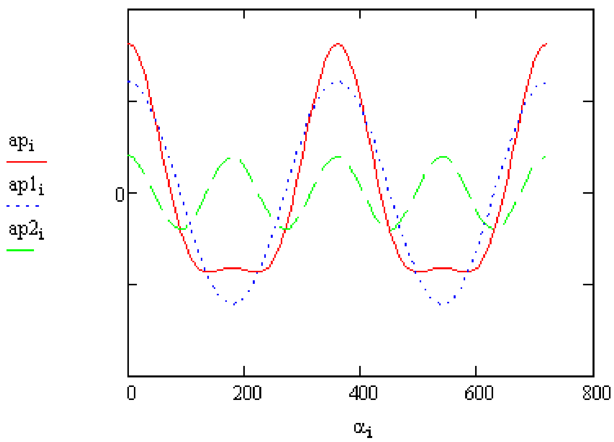

Figure 1.6. Piston acceleration diagram and his harmonics.

Connecting rod obliquity

$$\beta = \arcsin(\lambda_d \cdot \sin \alpha) \qquad [grd] \qquad (1.25)$$

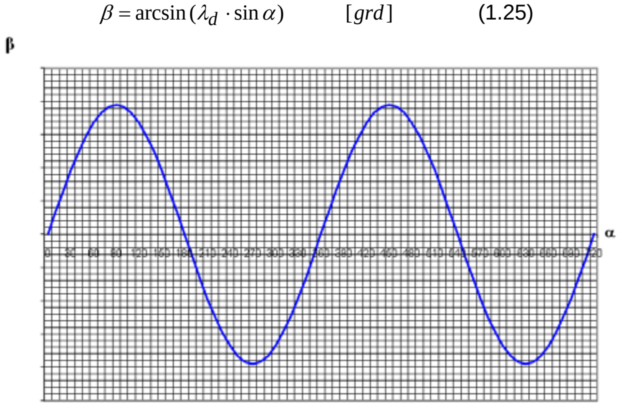

Figure 1.7. Connecting rod obliquity diagram.

1.2.6. Connecting rod angular velocity

$$\omega_b = \frac{\lambda_d \cdot \omega \cdot \cos \alpha}{\cos \beta} \qquad [s^{-1}] \qquad (1.26)$$

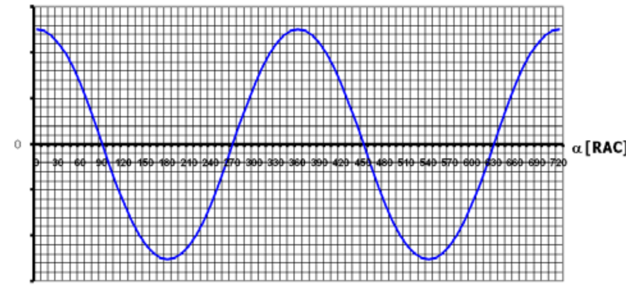

Figure 1.8. Connecting rod angular velocity diagram.

Connecting rod angular acceleration

$$\theta = \frac{\lambda_d (\lambda_d^2 - 1) \cdot \omega \cdot \cos \alpha}{\cos^3 \beta} \qquad [s^{-2}] \qquad (1.27)$$

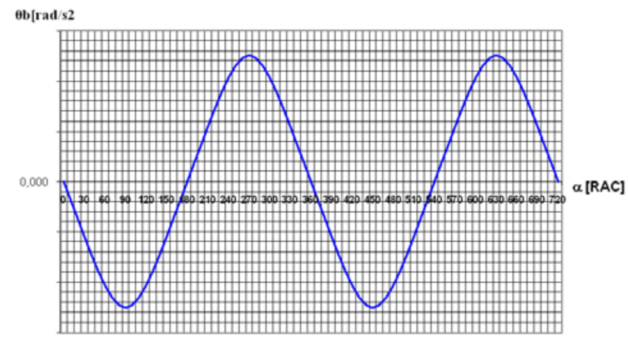

Figure 1.9. Connecting rod angular acceleration diagram.

Crank configurations influence on the engine balance

For a smoother running of the engine, the crankshaft elbows angular differences must be equal.

These engines, who meet the above condition, are called uniformly distributed ignition engines.

Crank configuration represents the geometrical configuration obtained by projecting the crank plans on a normal plan at the crankshaft axis.

For the single-acting engines, angular difference between two successive ignitions results from dividing the motor cycle:

$$\Theta_{ciclu} = \tau \cdot \pi \qquad [^0RAC] \qquad (1.28)$$

for a *i* numbers of engine cylinders:

$$\Delta_\alpha = \frac{\Theta_{ciclu}}{i} = \frac{\tau \cdot \pi}{i} \qquad [^0RAC] \qquad (1.29)$$

Ignitions order and crankshaft configuration are chosen sometimes considering other criteria than usual, such as: a better admission of fresh load, reduced stress caused by transverse vibrations, torsional vibrations etc., according to the specific requirements of the engine use.

For cranks star configuration shown in the figure below, both ignition orders, 1-2-4-3 and 1-3-4-2 are equivalent, from the point of view of minimum bearings load and torsional vibration of the crankshaft.

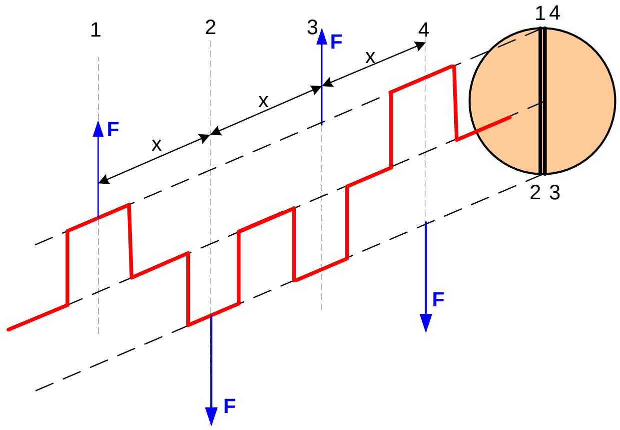

Figure 1.10. Crank plans projections on a crankshaft axis normal plan, resultant force that acting on the crankpin and the distance between crankpins.

For cranks configuration:

$$\Delta\alpha = \frac{\theta_{ciclu}}{i} = \frac{720^o}{6} = 120^o \quad [RAC] \qquad (1.30)$$

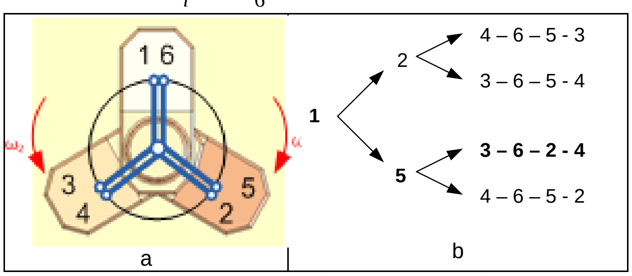

Figure 1.11.a. - Determination of crank configuration.
Figure 1.11.b. - The optimal ignition order according to the bearings load.

It is found that the optimum ignition sequence shown in the figure above, excludes the same bearing adjacent consecutive ignition.
Depending on the specific requirements for engine use, ignition order and crankshaft configuration is determined taking into account the reduction of stresses made by transversal vibrations of the engine.
In the figure below, choosing the ignition order „1 - 5 – 3 – 6 – 2 – 4” corresponds to the minimum bearing load but we note that it is not the best option in terms of torsional vibration of the crankshaft.
The optimal solution, in terms of reducing the torsional vibration of the crankshaft, is the ignition order „1 - 2 – 4 – 6 – 5 – 3”, even if we encounter successive ignitions between 1 - 2 and 6 – 5 cylinders.

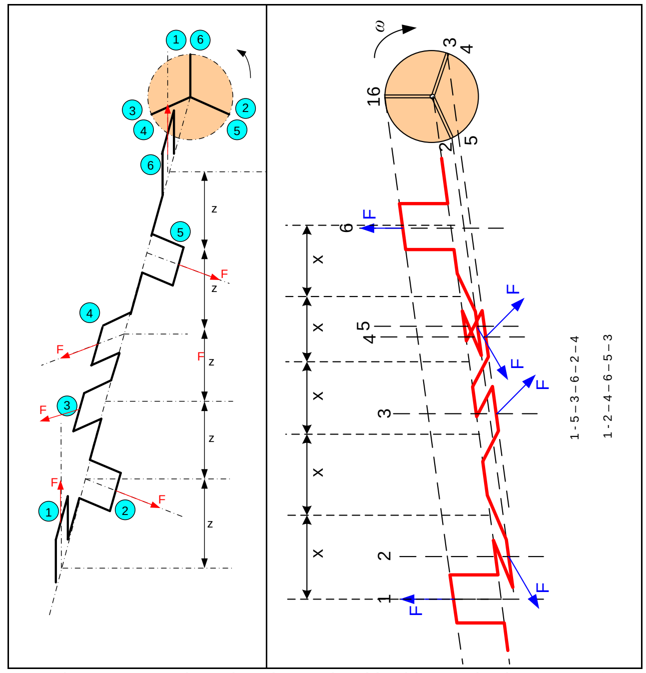

Figure 1.12. Choosing the optimal ignition order in terms of reducing the torsional vibration of the crankshaft for a 6-cylinder engine.

As a conclusion, a lower crankshaft loading is obtained if successive ignitions not occur in the adjacent cylinders, these criteria ensuring the engine mechanism durability; however, in some cases, choosing the ignition order is made by the following primary criteria:

- reducing the torsional vibration stresses;
- reducing the stresses under the action of the overturning moment;
- increasing the fresh load admission of the cylinders; etcetera.

Crank configurations influence on the boxer engine equilibration

Boxer engine architecture defines a piston that moves in a horizontal plane, on both sides of the crankshaft.

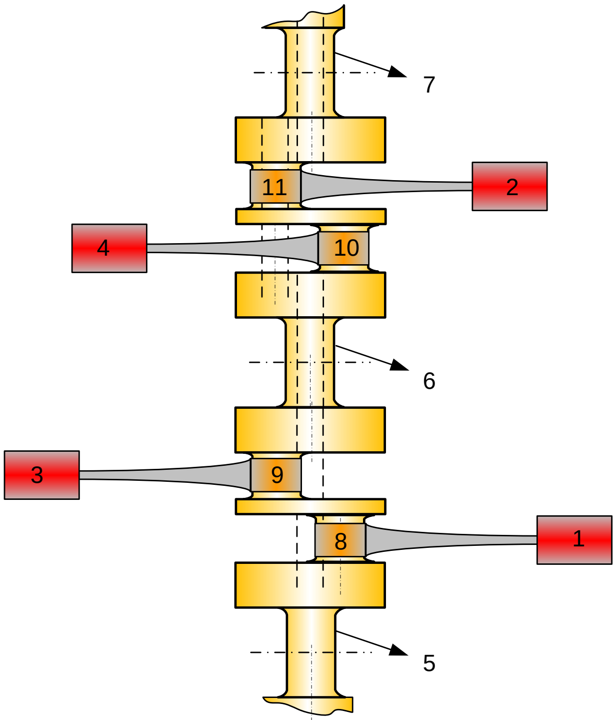

Figure 1.13. Graphical representation of the "boxer" type engine mechanism: 1,2,3,4 – pistons; 5,6,7 – main bearings; 8,9,10,11 - crankpins

One of the main advantages of this configuration is a lower height of engine, and thus a lower center of gravity. Another advantage is the perfect driveline symmetry and centered body engine. Also, boxer engines have a very low vibration level.

The difference between the boxer engine and the horizontally 180^0 V engine, is that the boxer engine pistons are driven individually by one crankpin, compared to horizontally 180^0 V engine, where the pistons share in pairs same crankpin.

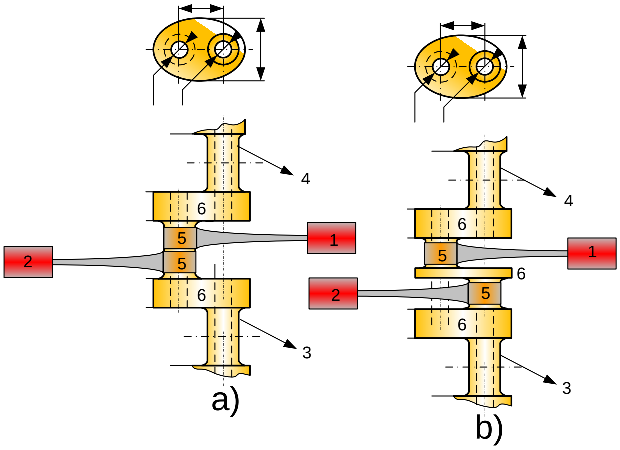

Figure 1.14. Graphical representation of the engine mechanism for:

a) the horizontally 180^0 cylinders layout V engine;
b) boxer engine;
1,2 – pistons; 3,4 – main bearings; 5 – crankpins; 6 – crankshaft webs.

Also, boxer engines should not be confused with opposed piston engines, which have a single combustion chamber.

Boxer engine configuration is the only configuration that has no unbalanced forces in terms of engine dynamics, no need for counterweights on the crankshaft to balance the forces of the engine mechanism. This aspect gives a lower weight of engine mechanism and thus faster acceleration, so, the possibility to use higher engine speeds.

Almost all inline diesel engines are equipped with balance shafts to reduce noise and vibrations. Boxer engine cancel inertial forces that cause vibrations and noise, this is due to the horizontally opposed pistons.

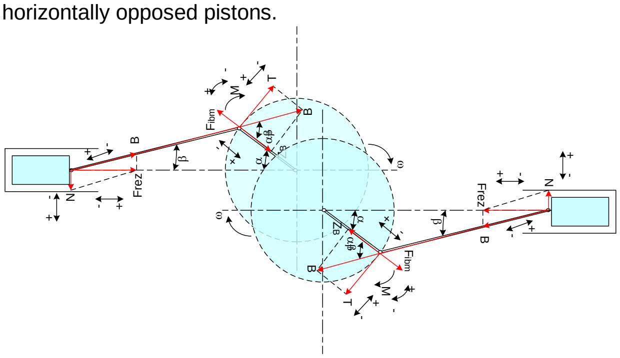

Figure 1.15. The arrangement of forces and moments in the Boxer internal combustion engine mechanism

BOXER engine does not require balance shafts and has a rigid configuration, structurally speaking.

It follows, therefore, compared with inline engines, Boxer engine, has low vibrations, a low center of gravity and a high degree of rigidity.

Configuration solutions for boxer engines

In this chapter are presented some configuration solutions for boxer engines, some ideas for construction possibilities.

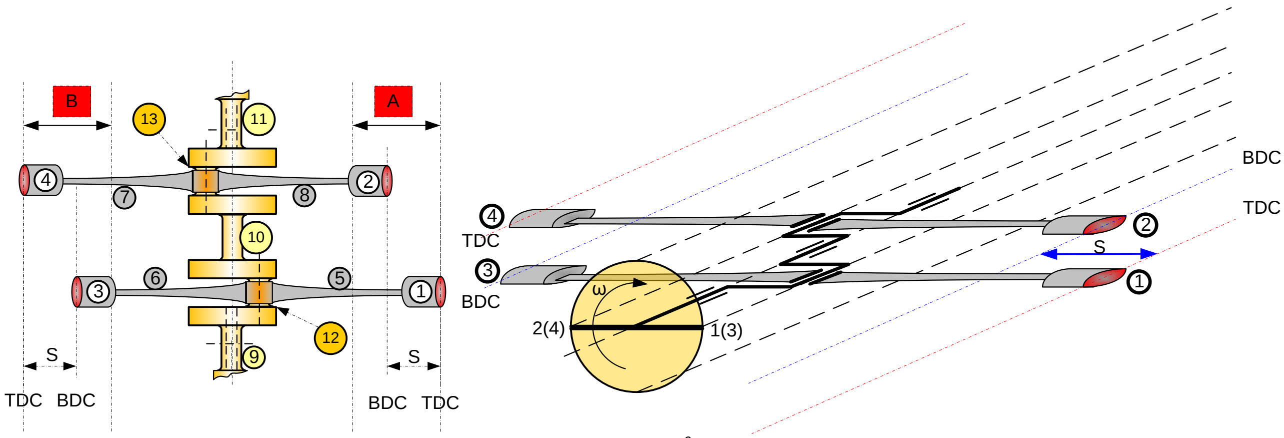

Figure 1.16. 4 cylinders, 4 stroke horizontally 180^0 cylinders layout V engine configuration.
1,2,3,4 – pistons; 5,6,7,8 – connecting rods; 9,10,11 – main bearings; 12,13 – crankpins; S – piston stroke.

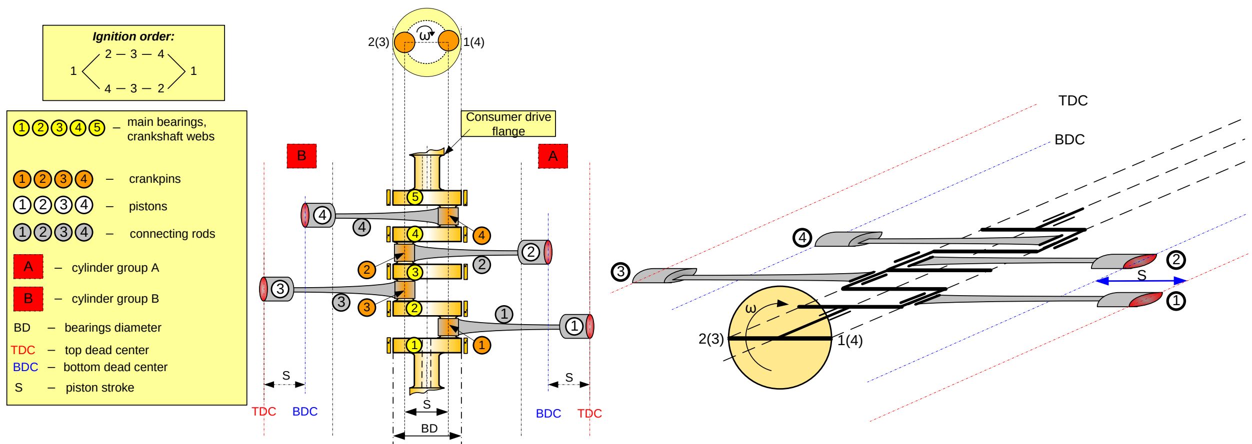

Figure 1.16'. 4 cylinders, 4 stroke boxer engine configuration, constructive version.

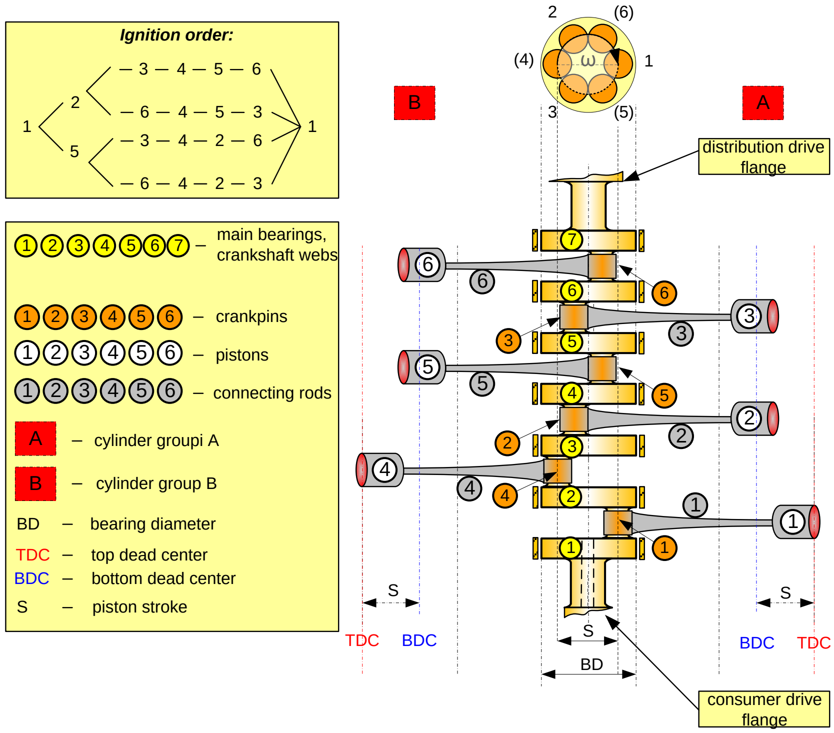

Figure 1.17. 6 cylinders, 4 stroke boxer engine configuration, constructive version.

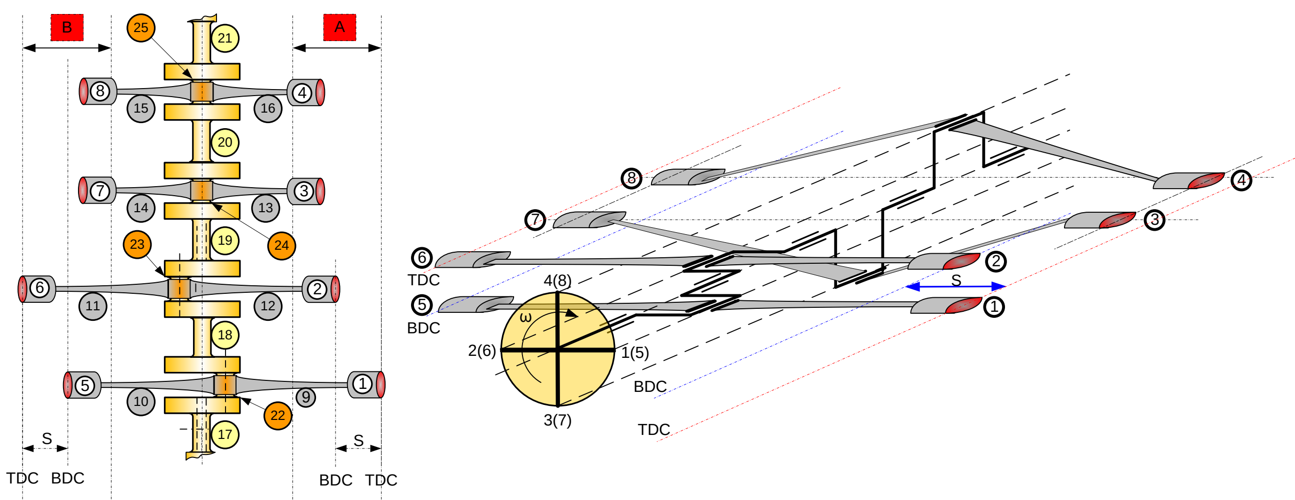

Figure 1.18. 8 cylinders, 4 stroke boxer engine configuration:
1,2,3,4,5,6,7,8 – pistons; 9,10,11,12,13,14,15,16 – connecting rods; 17,18,19,20,21 – main bearings; 22,23,24,25 – crankpins; S – piston stroke.

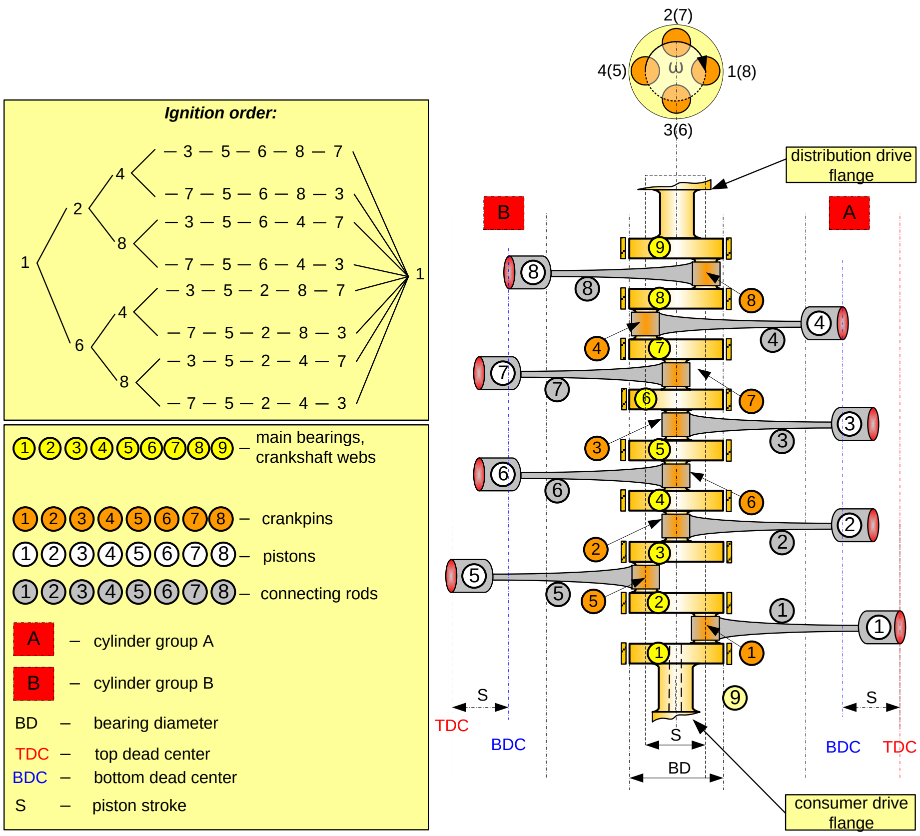

Figure 1.18'. 8 cylinders, 4 stroke boxer engine configuration, constructive version.

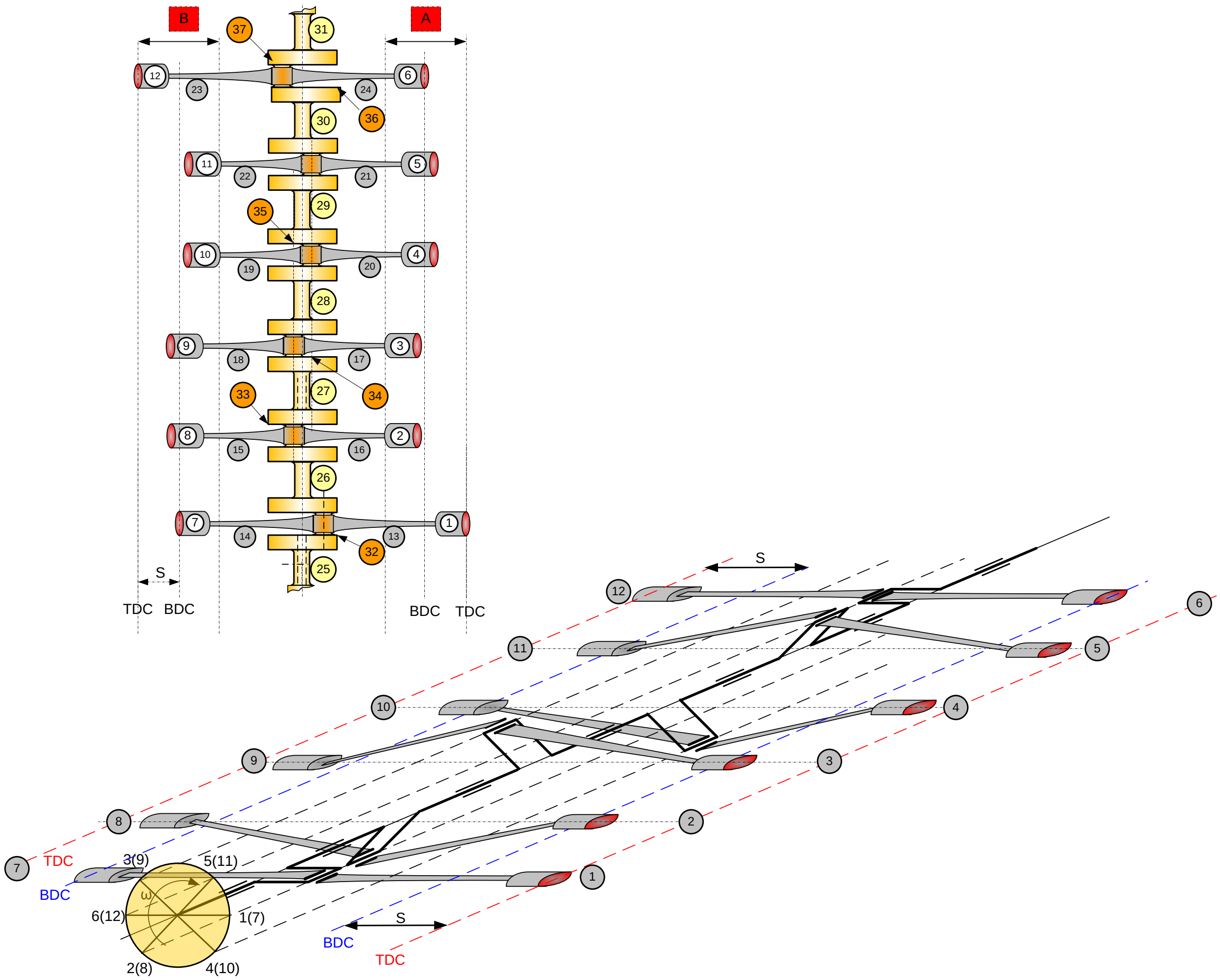

Figure 1.19. 12 cylinders, 4 stroke boxer engine configuration, constructive version:
1,2,3,4,5,6,7,8,9,10,11,12 – pistons; 13,14,15,16,17,18,19,20,21,22,23,24 – connecting rods; 25,26,27,28,29,30,31 – main bearings; 32,33,34,35,36,37 – crankpins; S – piston stroke.

Conclusions

There is a big difference between boxer engines and opposed piston engines. In the opposed piston engine, each cylinder has two pistons but no cylinder head. The boxer engine crankpins control only one piston while the 180^0 engines share crankpins as you saw in figure 1.14.

Boxer engines, flat engines in general, cancel out unwanted vibration where inline and v-type engines require additional components to accomplish this, they do not require a balance shaft or counterweights on the crankshaft to balance the weight of the reciprocating parts.

Boxers are one of only three cylinder layouts that have a natural dynamic balance; the others are the straight-6 and the V12. These engines are very smoothly and free of unbalanced forces with a four-stroke cycle.

A straight engine is considerably easier to build than an otherwise equivalent boxer engine. The cylinder bank and crankshaft can be made from a single part, and it requires fewer cylinder heads and camshafts. In-line engines are also smaller in overall physical dimensions than designs such as the radial, and can be mounted in any direction.

The increased mechanical complication and more difficult packaging is the downfall of most horizontally opposed engines, but the advantages of boxer engines must not be neglected, and the possibilities to use this kind of engines in naval propulsion should not be less important.

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