BALANCING AND COUNTERWEIGHTING OF CRANKS

Balancing and counterweighting of cranks

The engine frame is stressed by a system of actions (forces and moments) which are transmitted to the mounting structures. The time variation of such a field of actions can cause engine vibrations that locally can give rise to resonance.

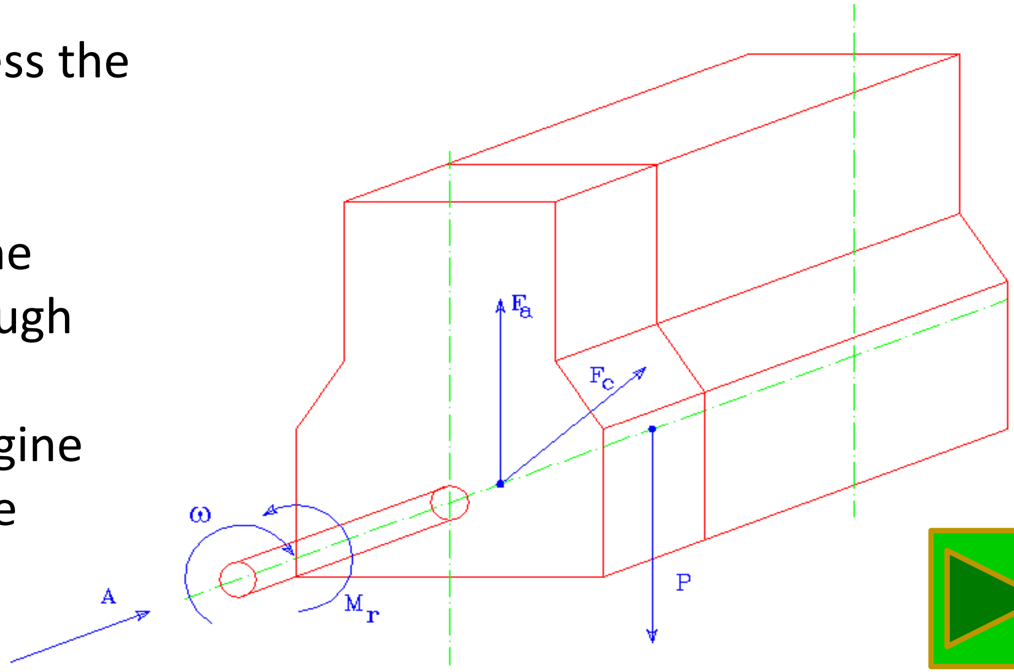
The aim of the crank balancing is that of either attenuate or cancel such induced vibrations:

→ the objective is then to regularize the forces acting upon the mounting structures so that the deformations of the mounts do not vary in the curse of time.

Balancing and counterweighting of cranks

Forces and moments that stress the engine frame:

- Surface forces that the engine exerts on the structure through the mounts;
- Torque exchanged by the engine with the outside through the crankshaft



- Forces exchanged by the engine with the accessories: induced forces, that are not considered in this dissertation
- Forces acting on the engine when the engine is fitted on the vehicle: force exerted by the clutch, co-axial with the crankshaft axis (neglected in this dissertation)

Balancing and counterweighting of cranks

- Mass forces

  - Weight $\overline{P} = m_m \overline{g}$; is a constant action that is not considered in the balancing calculations.
  - Inertia forces $\overline{F}_a$: are due to the reciprocating movement of the piston (piston pin or hinge-pin, piston rings, etc.) in the cylinder. These forces act along the cylinder axis only.
  - Centrifugal inertia forces $\overline{F}_c$: are due to the centripetal accelerations of the rotating masses (crank webs, pin shaft, big end of the connecting-rod, etc.)
  - Inertia torques: $\overline{M}_{i,m} = -I_m \dot{\omega}$. This torque varies with crank angle as a result of the inertia of the piston and connecting rod.

Balancing and counterweighting of cranks

- Mass forces

- I_m is the equivalent mass moment of inertia on the crankshaft axis while $\overline{M}_{i,m}$ is generated by the irregularity of the instantaneous velocity $\omega(t)$.
- $\sum \overline{M}_{i,b} = -I_0 \ddot{\gamma}$ This torque represents the correction term arising from the fact that the moment of inertia of the assumed rod is not equal to the moment of inertia of the actual rod.

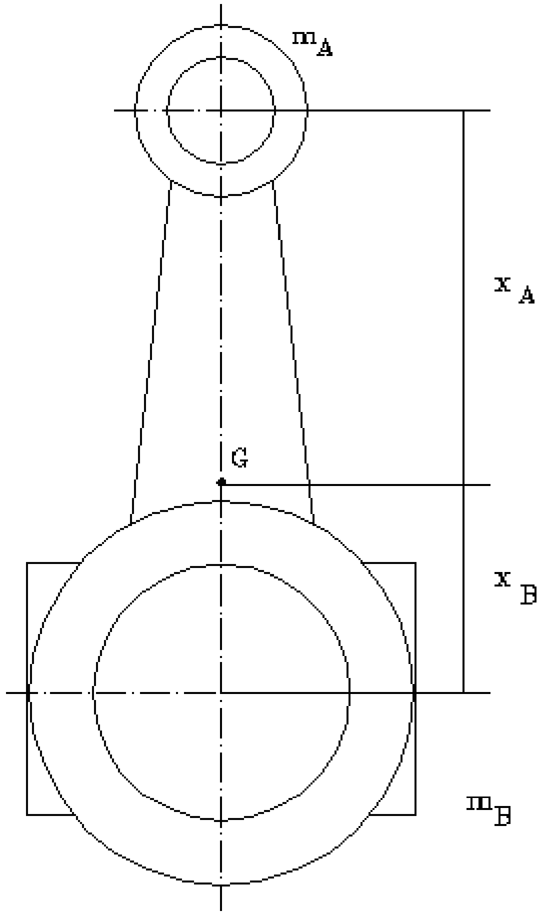
Balancing and counterweighting of cranks

The roto-translatory motion of the connecting rod should be accounted for. It is convenient to consider the connecting rod as equivalent to two masses concentrated at its ends, such that the sum of the masses is equal to the mass of the rod,

$$m_b = m_A + m_B$$

and the center of gravity of the two masses has to match that of the actual conrod.

$$m_A x_A = m_B x_B$$



Balancing and counterweighting of cranks

Since the mass substituted for the connecting rod is equal to the actual mass of the rod, and the center of gravity is at the same point, the magnitude and direction of the resultant forces given by this assumption are equal to the magnitude and direction of the resultant of the actual forces on the connecting rod.

Couples which may exist will be considered later. This substitution of a hypothetical connecting rod of two masse results in great simplification, since

- the substitute mass at the piston end of the rod reciprocates with the piston and may be considered a part of the piston assembly,
- the substitute mass at the crank end of the rod simply rotates at constant speed with the crankpin.

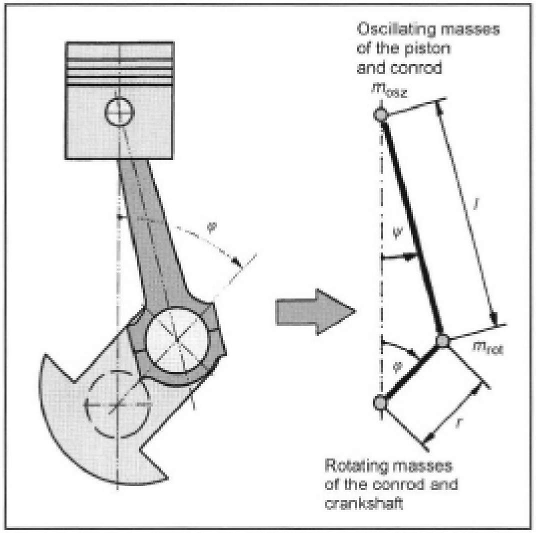


Fig. 6-30 Reduction of the crankshaft drive to two mass points.

Balancing and counterweighting of cranks

We have taken the connecting rod to be dynamically equivalent to two masses concentrated at its ends and having the same total mass and center of gravity as the actual rod.

While this assumption introduces no error in computing forces and moments, it is not correct in the case of inertia torque, because the moment of inertia of the actual rod referred to its center of gravity will not be quite as large as the moment of inertia of the assumed equivalent.

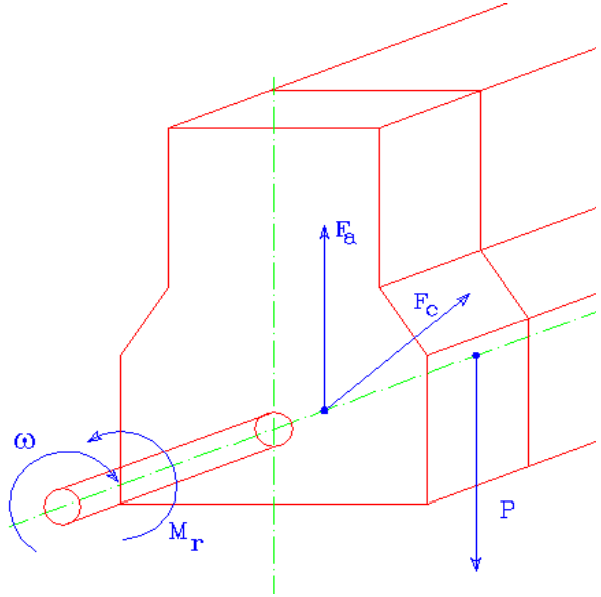
Then the following relation must be satisfied:

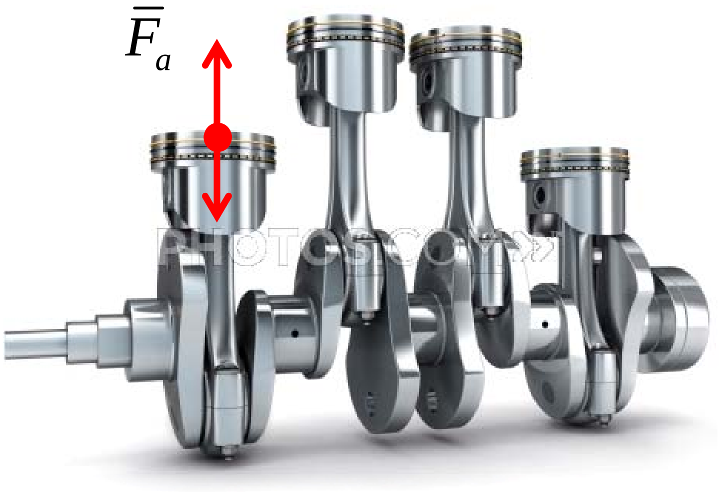
$$I_b = m_A x_A^2 + m_B x_B^2 + I_0$$

and I_0 is the “on frame correction for rod inertia” mass moment of inertia

Balancing and counterweighting of cranks

- Moment perpendicular to the crankshaft axis: the moments caused by the reciprocating forces and the centrifugal forces are both in a plane normal to the crankshaft axis.

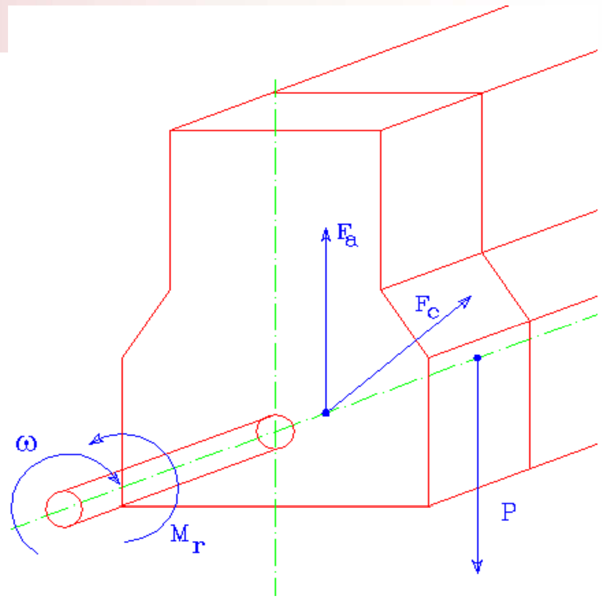


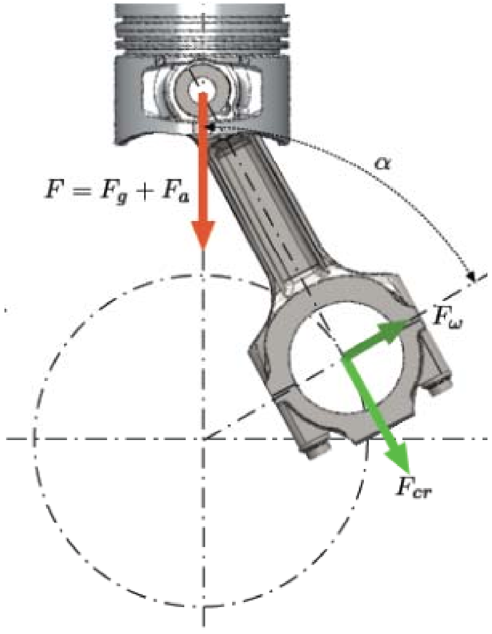


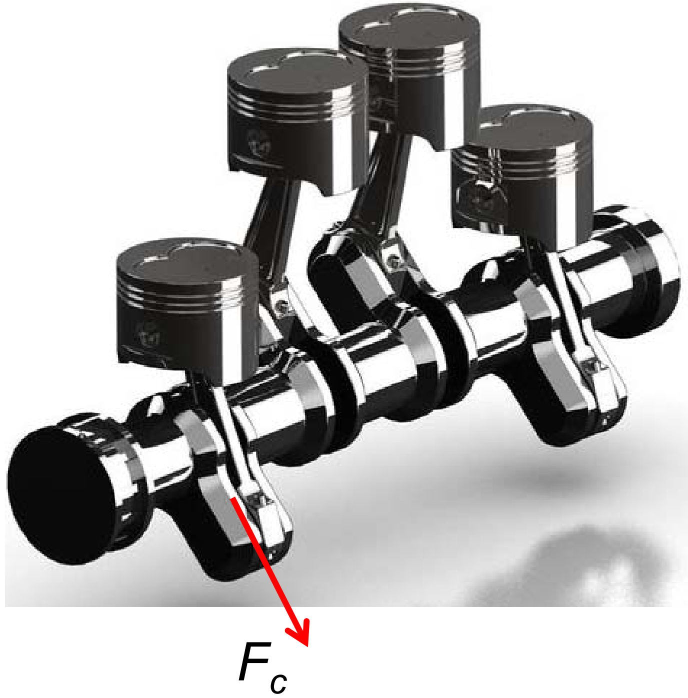
The inertia forces $\overline{F}_a$, acting along a constant direction (the cylinder axis) with variable amplitude, generate a moment $\overline{M}_{F_a}$ with constant direction and variable amplitude.

Balancing and counterweighting of cranks

For a given crankshaft angular velocity ω, the radial inertia forces on the crankpin have constant amplitude and variable direction and give rise to a momentum $\bar{M}_{F_c}$ which rotate with the crankshaft in a plane normal to it.







Balancing and counterweighting of cranks

As said before, in this dissertation we don't consider the axial action due to the clutch. Usually, the mount of the crankshaft on clutch side has a mechanical solution that balances the axial forces.

In general, the resultant of the forces on the mounts will be:

$$\overline{R}_s = \Sigma\overline{F}_a + \Sigma\overline{F}_c + m_m\overline{g} = \overline{R}_{F_a} + \overline{R}_{F_a} + m_m\overline{g}$$

To obtain $\overline{R}_s$ constant such as to avoid vibration means:

$$\overline{R}_{F_c} = 0 \qquad \overline{R}_{F_a} = 0$$

Furthermore, considering the moments acting in the plane normal to the crankshaft axis, we must have:

$$\overline{M}_{F_a} = 0 \qquad \overline{M}_{F_c} = 0$$

Balancing and counterweighting of cranks

In the same way, the reaction torque on the crankshaft should be constant:

$$\bar{M}_{reaction} = \bar{M}_r + \bar{M}_{i,m} + \Sigma\bar{M}_{i,b}$$

(this is the torque that the engine, throughout the mounts, exerts on the fixed structure around the crankshaft axis).

To study in depth the question, let us write the equation of the crankshaft dynamic equilibrium:

$$\bar{M}_r + \bar{M}_{eng} + \bar{M}_{i,m} + \Sigma\bar{M}_{i,b} = 0$$

that is

$$\bar{M}_{reaction} = -\bar{M}_{eng}$$

Balancing and counterweighting of cranks

and then, to obtain $\bar{M}_{reaction} = const$ means $\bar{M}_{eng} = const$.

To summarize, the balance of the crankshaft can be obtained with:

$$\bar{M}_{eng} = const$$

$$\bar{R}_{F_a} = 0 \qquad \bar{M}_{F_a} = 0$$

$$\bar{R}_{F_c} = 0 \qquad \bar{M}_{F_c} = 0$$

The first expression means to aim for the maximum uniformity of the engine torque, while the other expressions require an automatic balance of the inertia forces (reciprocating and centrifugal).

Balancing and counterweighting of cranks

Analysis of F_a and F_c forces

Centrifugal forces $$F_c = -m_c \omega^2 r$$

Reciprocating forces $$F_a = -m_a \ddot{x} \cong -m_a \omega^2 r \left[\cos\theta + \lambda \cos 2\theta + .. \right]$$

where $\lambda = r/l$ is the crank-connecting-rod mechanism (conrod ratio), the value of which is usually between 0,25÷0,33.

Usually in the development of the acceleration $\ddot{x}$ is sufficient to consider only the first two terms because the higher terms can be considered negligible.

Balancing and counterweighting of cranks

The coefficient of θ in the expression of F_a will be referred hereafter as the *order* of the harmonic.

Thus, the first term of the expression represents the first-order force, or the one that varies periodically synchronal to the shaft speed.

$$F_{a'} = -m_a \omega^2 r \cos\theta$$

The second term represents the second-order force and varies periodically twice per shaft revolution.

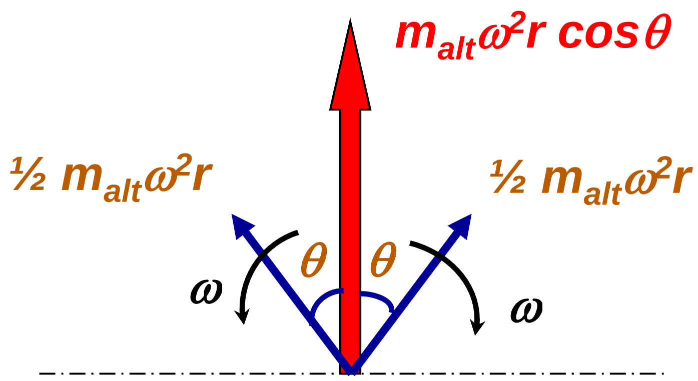
$$F_{a''} = -m_a \omega^2 r \lambda \cos 2\theta$$

Balancing and counterweighting of cranks

The first- and second-order forces display variable modulus and constant direction. It is hence possible to reduce these forces to an equivalent field made up of two vectors half the value of the force itself:

$$\frac{1}{2}\left|F_{a,\max}\right|$$

The sum of the two vectors fully corresponds to the actual force amplitude as they rotate clockwise and counterclockwise, respectively.

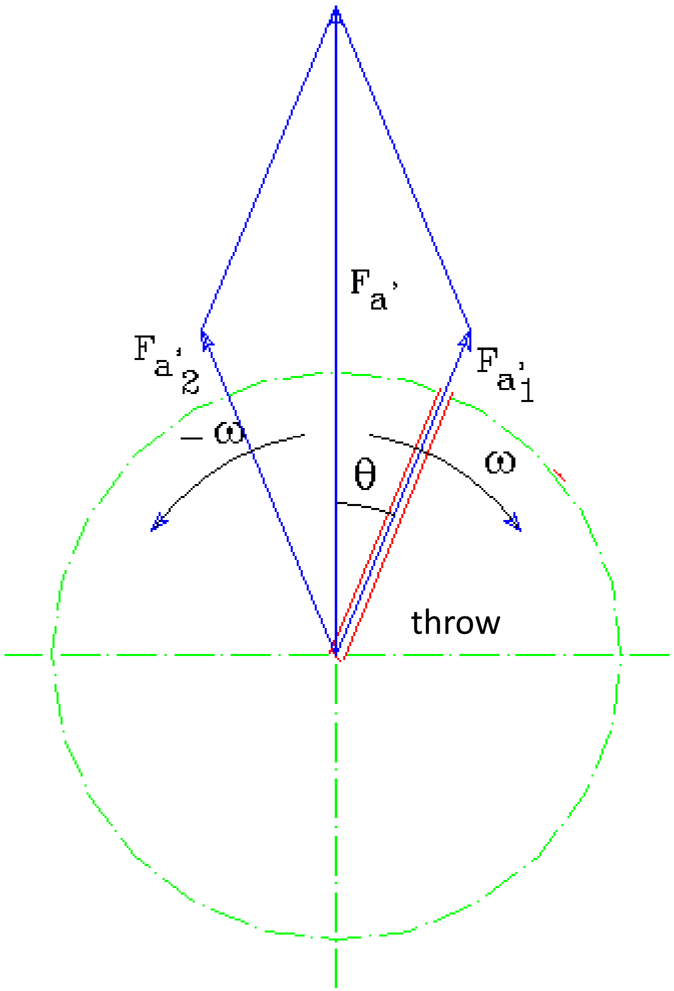


Balancing and counterweighting of cranks

I° order forces

At each crank angle step, the force F_a is the resultant of two symmetrical forces with constant modulus that rotate with ω and – ω respectively.

The two force fields turn out to be equivalent to the centrifugal forces, rotating at crankshaft speed in the two versus (crank direction and opposite crank direction).



Balancing and counterweighting of cranks

I° order forces

Thus we define:

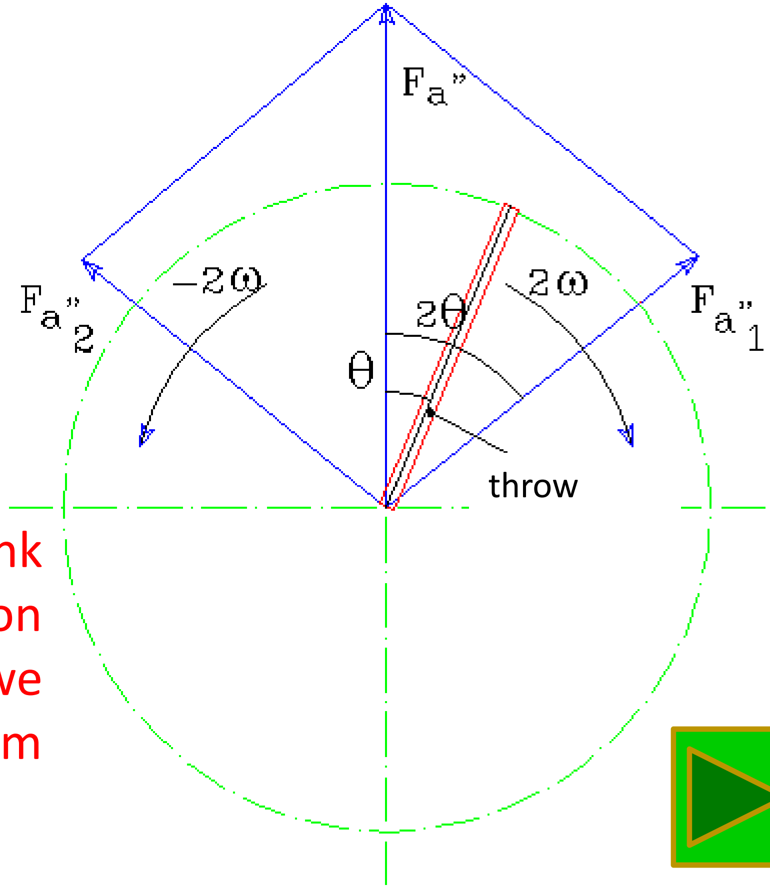
- F'_{a_1} the first order reciprocating force rotating in the crank direction;
- F'_{a_2} the first order reciprocating force rotating in the opposite crank direction;

The system of first order forces F'_{a_2} rotating in the opposite crank direction for *in-line engines* is such that if the system F'_{a_1} of forces will be cancelled out, so will the system of forces F'_{a_2} .

(This is not true for V-engines)

Balancing and counterweighting of cranks

II° order forces



F_{a_1}'' is the second order reciprocating force rotating in the crank direction;

F_{a_2}'' is the second order reciprocating force rotating in the opposite crank direction;

The system of forces rotating in the crank direction or in the opposite direction constitutes a constant rigid system: then we can consider the balance of the rigid system of forces in any time instant.

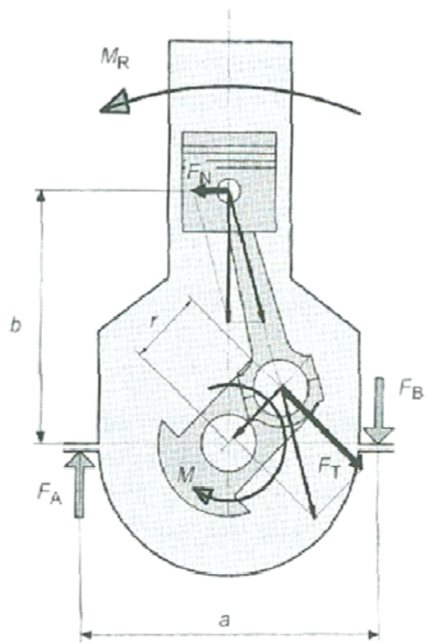
So, once we have determined the resultant of the system of forces in a time instant, in the following instants the amplitude of the resultant will be the same and the vector will rotate with 2ω (or ω, 3ω, etc.).

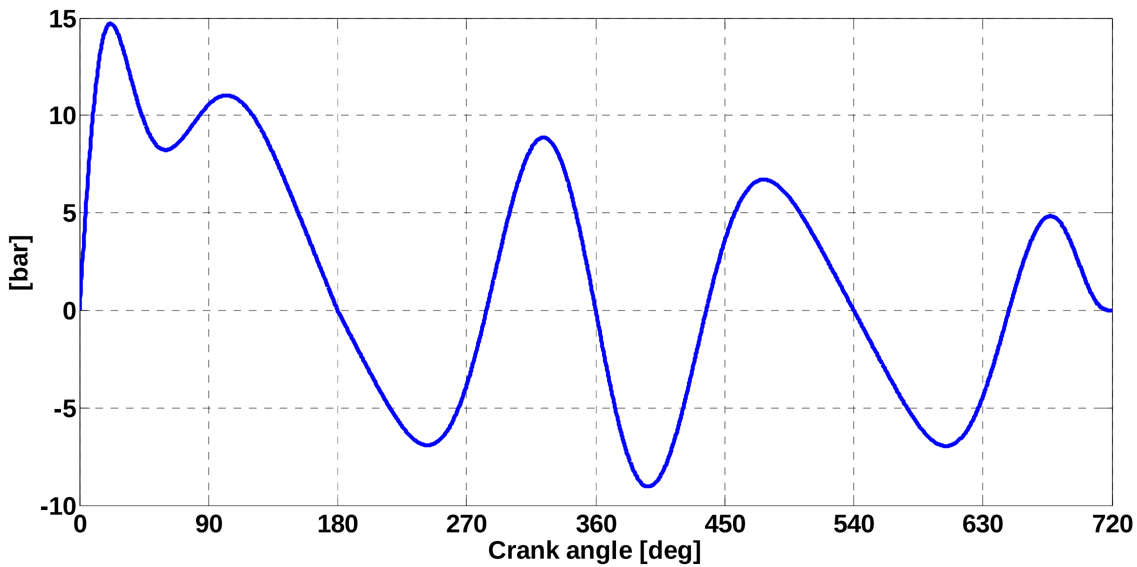
Balancing and counterweighting of cranks

Engine Torque.

The engine releases a torque through the shaft and hence undergoes an equivalent action from the mounts (M_R). In order to reduce such action, the engine torque needs to be as regular as possible.

The single cylinder engine torque turns out to be highly variable throughout the engine cycle





Balancing and counterweighting of cranks

We can express the engine torque [1] for a single-cylinder engine with the Fourier series:

$$\bar{M}_{\text{single}} = M_0 + \sum_k M_k \sin\left(k\omega t + \psi_k\right)$$

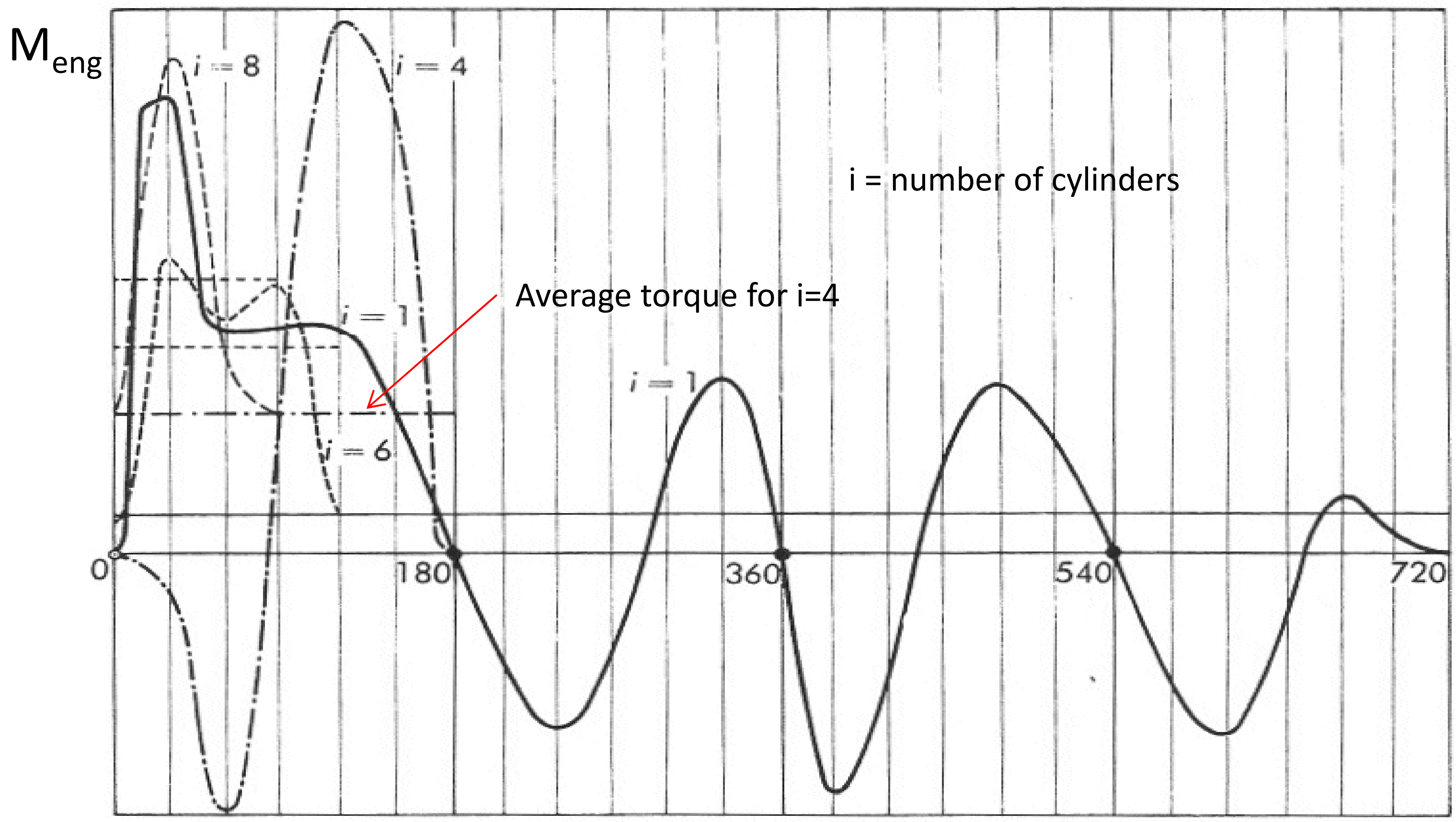
where:

M_0 = constant medium torque
M_k = magnitude of k-th harmonic
k = harmonic order; $k = \frac{n}{m}$ with n=1, 2, 3, 4, the order number of the harmonic
m=1 for 2T (two-stroke engines)
m=2 for 4T (four-stroke engines) (for 4T we have half-order coefficients)

ψ_k = phase of k-harmonic

(1) The torque, due to torsional forces, is a function of the load (specific work). It cannot be described by a closed function and is therefore subject to a Fourier analysis; this is composed of a static component (nominal load torque) and a dynamic component (a basic vibration and overlapping harmonics). The exciting frequencies are, hence, the basic frequency (number of work cycles per unit time) and their integral multiples. They are proportional to the crankshaft speed.

Balancing and counterweighting of cranks: engine Torque



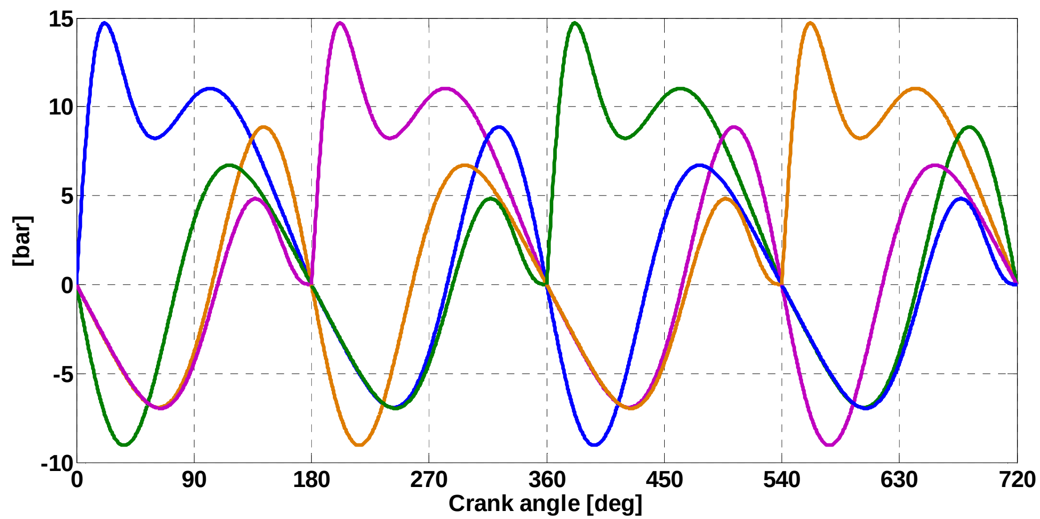
Balancing and counterweighting of cranks: engine Torque

To obtain a more uniform (or smooth) engine torque in a multicylinder engine the usual arrangement is with the cranks evenly spaced.

Then the angular disposition of the cranks will have a phase shift $\Delta\varphi$ given by:

$$\Delta\varphi = 2\pi \frac{m}{i_{tot}}$$

where i_{tot} is the number of cylinders of the engine.

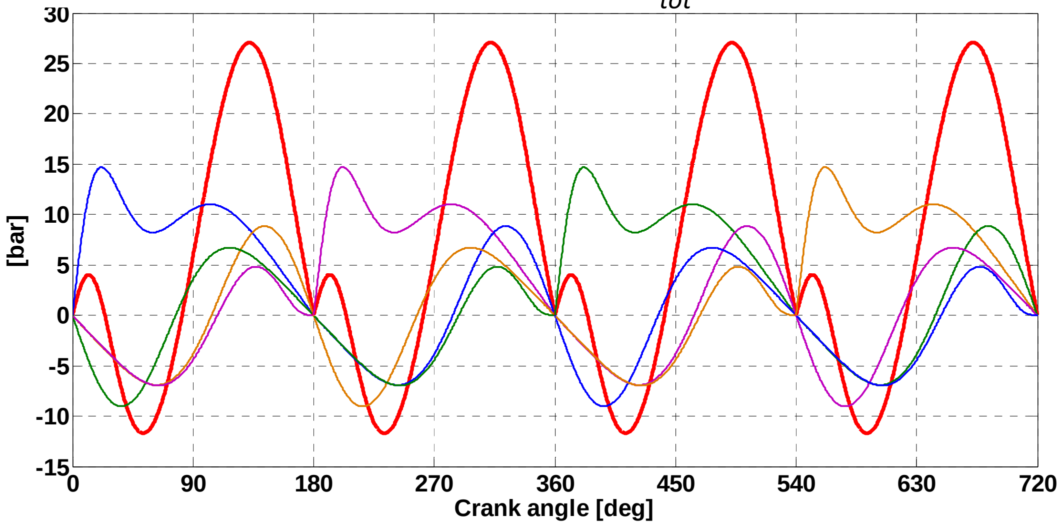


Balancing and counterweighting of cranks: engine Torque

The resulting moment is obtained by properly summing together the moments related to each of the cylinders and by properly accounting for the phase shift.

Taking as reference the first crank, the phase shift of the crank number *i* is:

$$\varphi_i = \Delta\varphi(i-1) = 2\pi\frac{m}{i_{tot}}(i-1)$$



Balancing and counterweighting of cranks: engine Torque

Then the resulting moment for the multicylinder engine is express by:

$$M_{multi} = \sum_{i=1}^{i_{tot}} M_i = i_{tot} M_0 + \sum_{i=1}^{i_{tot}} \sum_{k} M_k \sin\left(k\omega t + \psi_k - \delta_{i_k}\right)$$

with the hypothesis that M_0 is the same for all cylinders.

δ_{ik} is the phase shift of the harmonic of order *k* for the crank number *i*

The *i-th* crank will travel through the reference position after a time interval $t_i = \varphi_i / \omega$ and then the phase shift of *k-th* harmonic will be:

$$\delta_{ik} = (k\omega) t_i = k\varphi_i = k 2\pi \frac{m}{i_{tot}} (i-1) = k\Delta\varphi (i-1)$$

and then

$$\Delta\left(\delta_{i_k}\right) = k\Delta\varphi$$

Balancing and counterweighting of cranks

For example, we consider a 8-cylinder four-stroke engine, with the hypothesis of cycle uniformity.

➔ Then the magnitude of k-th harmonic will be the same for all cylinders.

Let us evaluate the first 8 harmonics of the engine torque M_{single} that will determine M_{multi}:

All the torques act about the same axis and are applied to the same body and can hence be directly summed up.

$$\Delta\varphi = 2\pi \frac{m}{i_{tot}} = \pi / 2$$

Balancing and counterweighting of cranks

Example: 8-cylinder four-stroke engine, displaying a uniform torque amongst the cylinders.

$$\Delta\varphi = 360° \cdot \frac{m}{i_{tot}} = 360° \cdot \frac{2}{8} = 90°$$

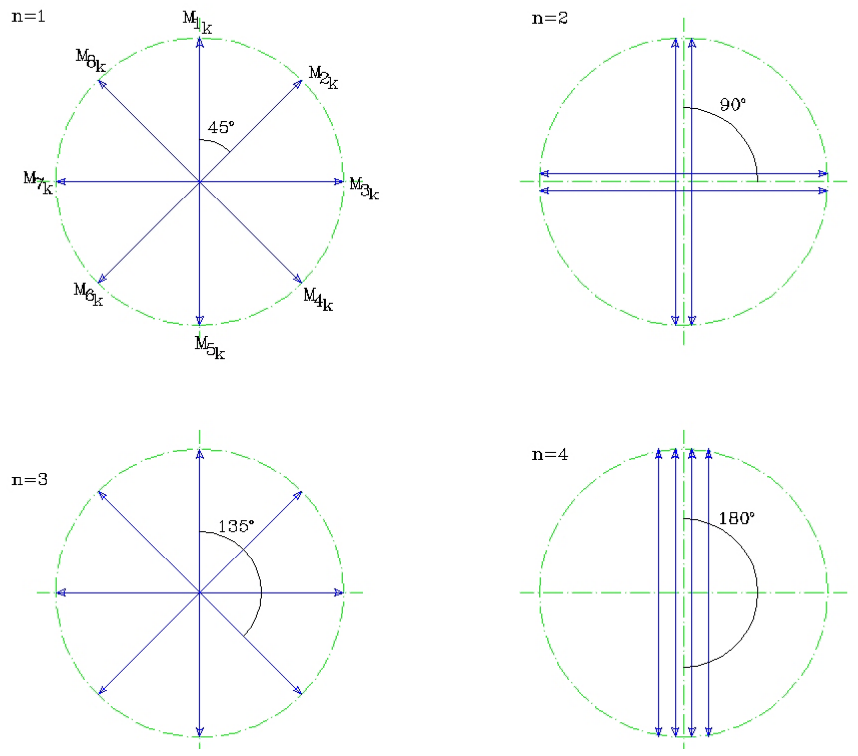
$$\Delta\left(\delta_{i_k}\right) = k \cdot 90°$$

$$n = 1; \quad k = 1/2; \quad \Delta\left(\delta_{i_k}\right) = \frac{1}{2} \cdot 90° = 45°$$

$$n = 2; \quad k = 1; \quad \Delta\left(\delta_{i_k}\right) = 90°$$

$$n = 3; \quad k = 3/2; \quad \Delta\left(\delta_{i_k}\right) = \frac{3}{2} \cdot 90° = 135°$$

$$n = 4; \quad k = 2; \quad \Delta\left(\delta_{i_k}\right) = 180°$$



Balancing and counterweighting of cranks

$$\Delta\varphi = 360^\circ \cdot \frac{m}{i_{tot}} = 360^\circ \cdot \frac{2}{8} = 90^\circ$$

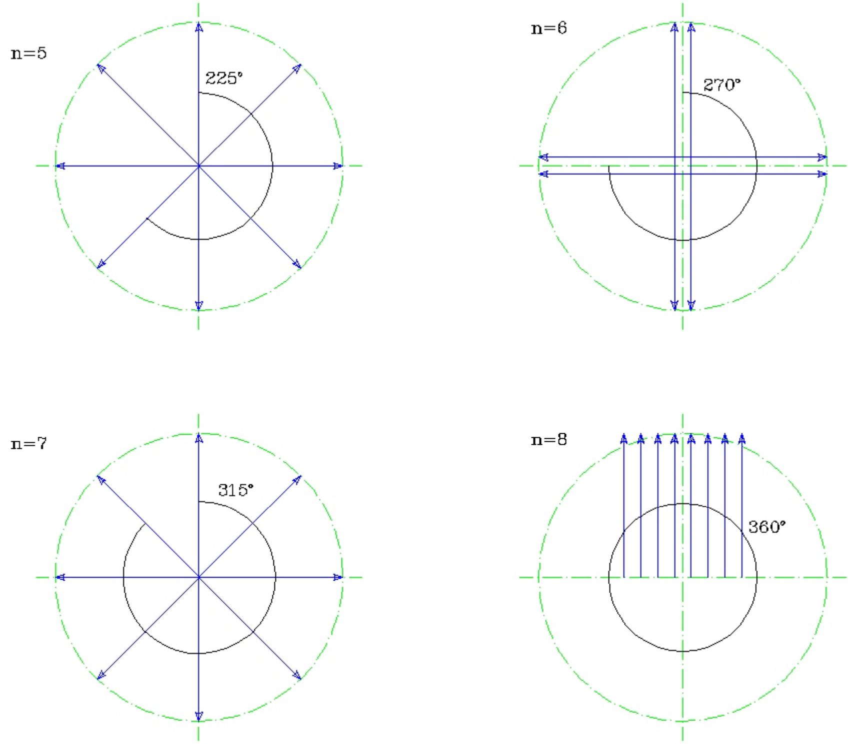
$$\Delta\left(\delta_{i_k}\right) = k \cdot 90^\circ$$

$$n = 5; \quad k = 5/2; \quad \Delta\left(\delta_{i_k}\right) = \frac{5}{2} \cdot 90^\circ = 225^\circ$$

$$n = 6; \quad k = 3; \quad \Delta\left(\delta_{i_k}\right) = 270^\circ$$

$$n = 7; \quad k = 7/2; \quad \Delta\left(\delta_{i_k}\right) = \frac{7}{2} \cdot 90^\circ = 315^\circ$$

$$n = 8; \quad k = 4; \quad \Delta\left(\delta_{i_k}\right) = 360^\circ$$



Balancing and counterweighting of cranks

From the figure we can see that for k=1/2, 1, 3/2, 2, 5/2, 3, 7/2 the torque vectors have a regular star layout and so the resultant is cancelled out, while for k=4 we add the eight vectors M_{i_k}:

→ we have identified the principal harmonic.

This example illustrates the general principle that, with evenly spaced firing, the torque vectors for 4-stroke engine add up when k is a multiple of $i_{tot}/2$ and are cancelled out for other values of k.

Balancing and counterweighting of cranks

In general, the principal harmonics that appear in multicylinder engines with even firing intervals and uniform pressure-crank-angle diagrams are characterized by the conditions:

$$\Delta\left(\delta_{i_k}\right) = 2\pi n \quad \left(n = 1, 2, ..\right)$$

$$\Delta\varphi \cdot k = 2\pi \frac{m}{i_{tot}} k = 2\pi n \quad => \quad k = n \cdot \frac{i_{tot}}{m}$$

then for an 8-cylinder four-stroke engine the principal harmonics have the order 4, 8, 12, ...

If we consider only the principal harmonics, the torque for the multicylinder engine can be written as:

$$M_{multi} = i_{tot} \left[M_0 + \sum_{p = n \cdot i_{tot}/m} M_p \sin\left(p\omega t + \psi_p\right) \right]$$

with M_{i_p} constant for all cylinders.

Balancing and counterweighting of cranks

It should be remembered that the preceding discussion assumed identical indicator diagrams for all cylinders of a given engine.

In practice, of course, the diagrams vary to an appreciable extent between strokes of a given cylinder and also between cylinders during a given revolution of the crankshaft.

On account of this non-uniformity, all engines will show some degree of torque at all harmonics of the cyclic variation as well as of the crank speed.

However, since only small differences are involved, the magnitude of the torque variations in orders that theoretically cancel out will usually be small compared with those that theoretically add up in the lower (<8) orders.

Balancing and counterweighting of cranks

Engines with non-uniform firing: in practice, a good many engines are made with non-uniform firing intervals.

The reason for such an arrangement may be for mechanical simplicity (for example, to reduce the number of cranks), for engine compactness (for example, small V angle) or to avoid a serious vibration resonance at a critical frequency.

For such cases, all normal orders are present in the torque diagram even assuming cyclic uniformity.

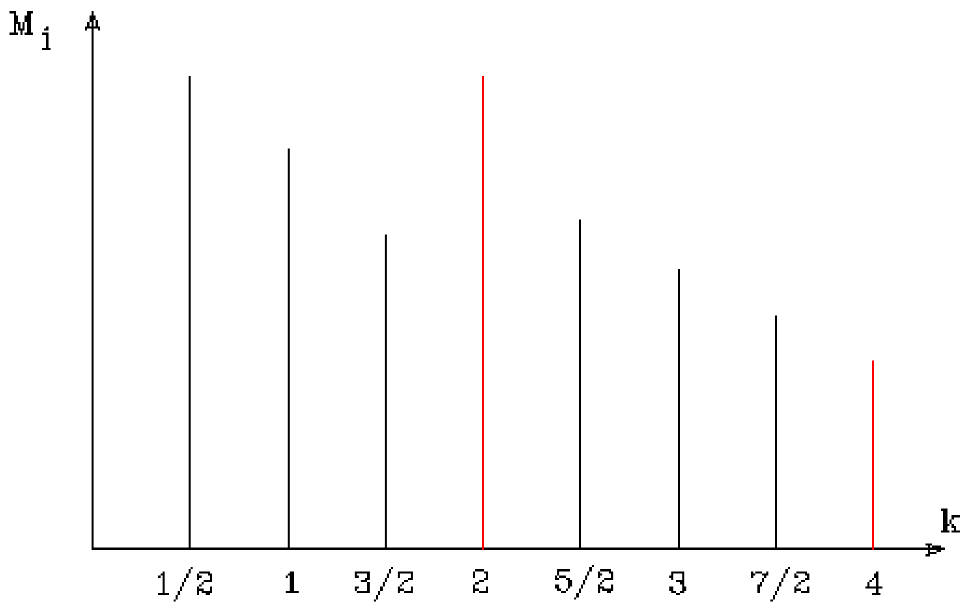
The magnitude and phase of each order can be determined by the methods previously described.

Even though exact prediction of engine-torque coefficients is impossible, the coefficients based on cycle uniformity are generally the most important ones, and analysis based on such data is valuable.

Balancing and counterweighting of cranks

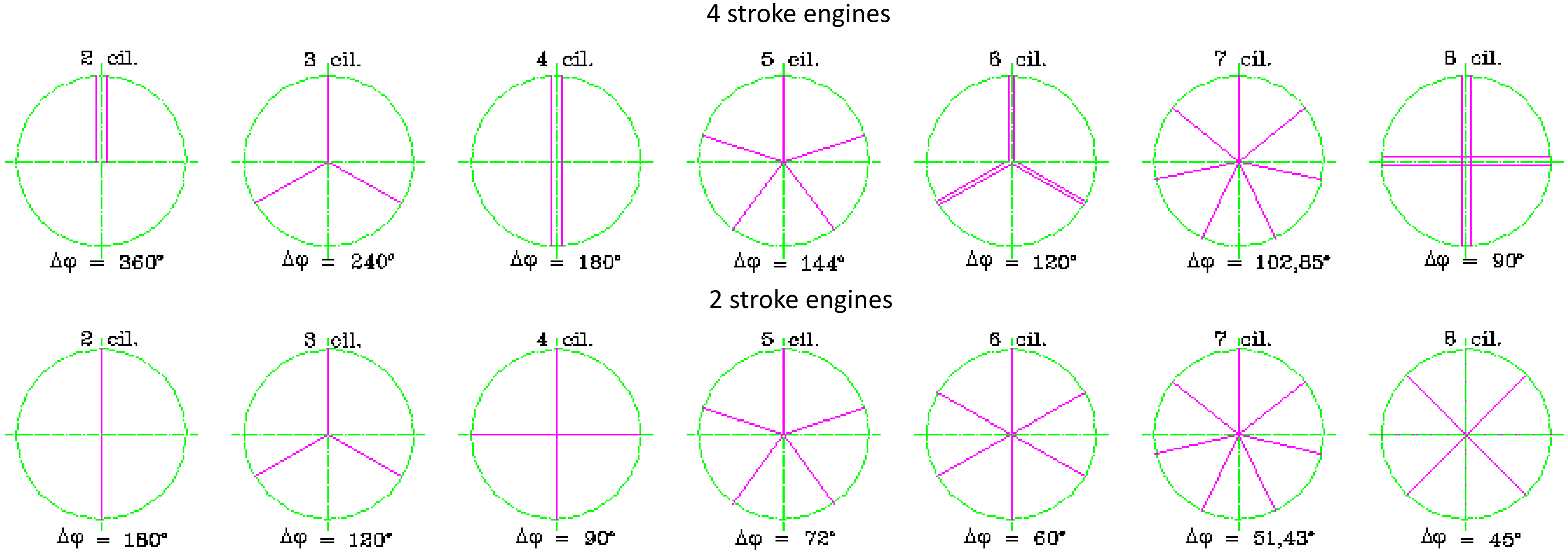
The Figure shows possible magnitudes of the first eight harmonics for a 4-stroke engine.

It can be seen that from the harmonic number 4 (order 2) the amplitudes are decreasing monotonic functions: for p≥2 (four-cylinder engine) we can smooth the engine torque progressively increasing the number of cylinders.



Balancing and counterweighting of cranks

Let us consider now the field of centrifugal forces for multicylinder engines evaluating the angular layout of the cranks in engines with crank evenly spaced.



In general, for an odd number of cylinders, 2-stroke and 4-stroke engines have the same angular layout of cranks.

Balancing and counterweighting of cranks

Lengthwise layout of cranks

Let us see if it is possible to cancel $\bar{R}_{F_c}$ and $\bar{M}_{F_c}$ making sure of the lengthwise layout of the cranks in the three cases:

i_{tot} even (4-stroke engines)

i_{tot} odd (4-stroke and 2-stroke engines)

i_{tot} even (2-stroke engines)

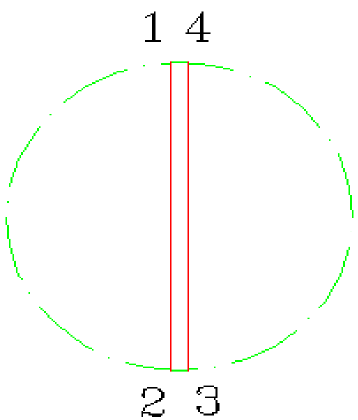
We can consider before the **4-stroke engines**

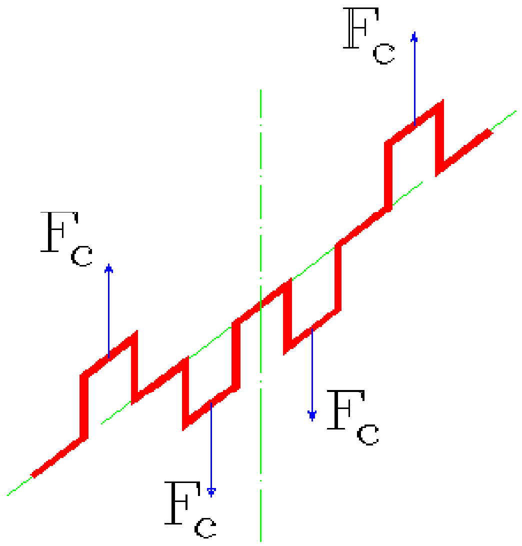
Balancing and counterweighting of cranks

4-stroke engines

To cancel $\overline{R}_{F_c}$ and $\overline{M}_{F_c}$ we must adopt a symmetrical lengthwise layout relative to the middle of the crankshaft.

For example, for a 4 cylinder : $$\Delta\varphi = \frac{360° m}{i_{tot}} = 180°$$





$R_{F_c} = 0$

$M_{F_c} = 0$

Balancing and counterweighting of cranks

4-stroke engines

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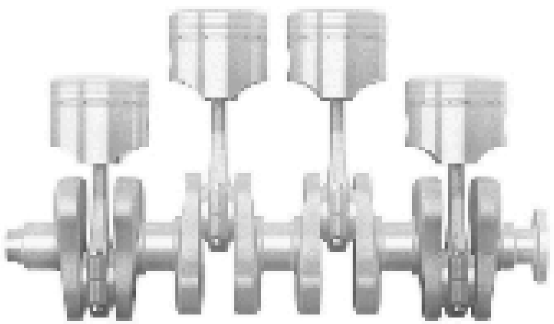
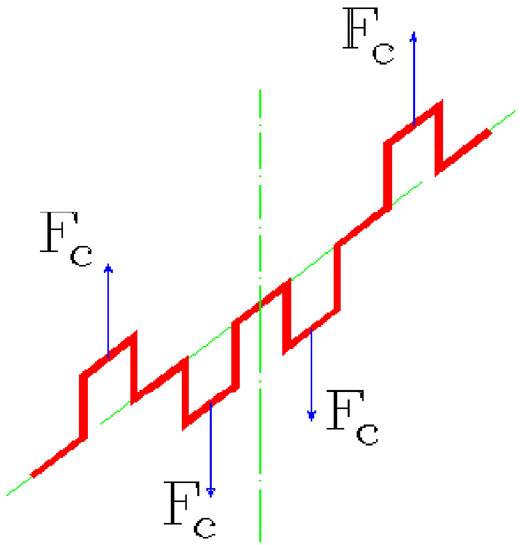


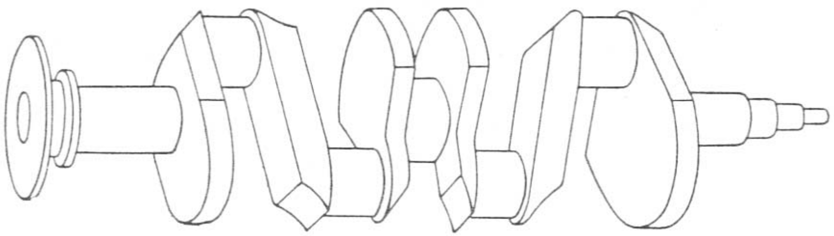


$$R_{F_c} = 0$$

$$M_{F_c} = 0$$

Balancing and counterweighting of cranks

4-stroke engines



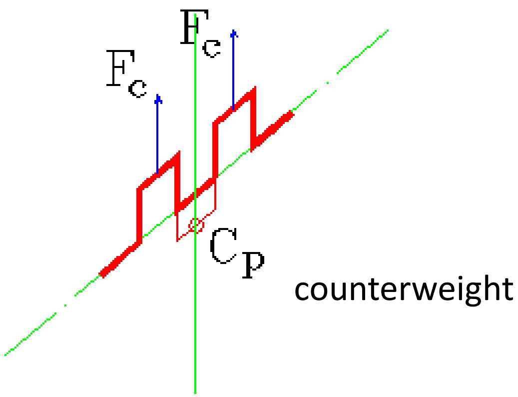
Crankshaft with 4 throws and three main bearing journals with counterweights

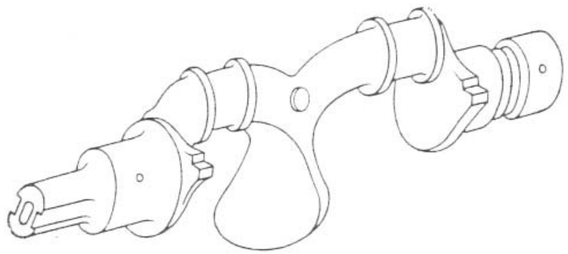
Balancing and counterweighting of cranks

4-stroke engines

With the exception of 2 cylinder 4-stroke engines for which, adopting the criteria of a uniform phase displacement of cranks, we obtain

$$\Delta\varphi = \frac{360° m}{i_{tot}} = 360°$$





In this case, to cancel $\overline{R}_{F_c}$ we must add a counterweight (C_p in the figure). Anyhow, $\overline{M}_{F_c}$ is cancelled out.

Balancing and counterweighting of cranks

4-stroke engines

Summing up:

i_{tot} even ➔ $\bar{M}_{F_c} = 0$ adopting a symmetrical lengthwise layout of cranks in relation to the middle of the crankshaft.

The same thing goes for $M_{F_{a'}} = 0$ $\left(M_{F_{a'_1}} = 0, \quad M_{F_{a'_2}} = 0 \right)$

i_{tot} odd ➔ $\bar{M}_{F_c} => 0$ with an *anti-metric lengthwise layout of cranks*

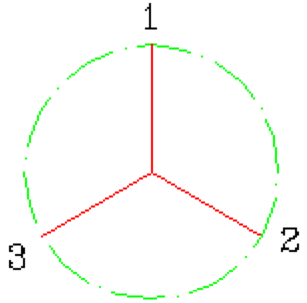
We can cancel $\bar{M}_{F_c}$ only if we don't adopt an evenly spaced cranks.

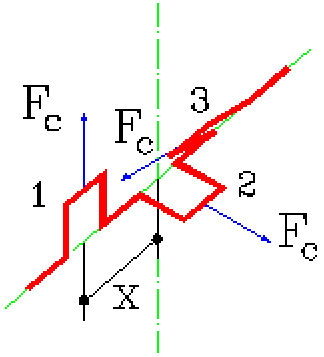
In this dissertation we consider only evenly spaced cranks. Then, the best lengthwise layout is the anti-metric, for which $\bar{M}_{F_c} => 0$.

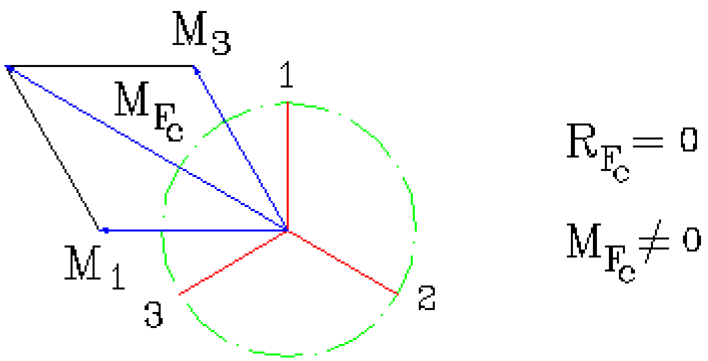
Balancing and counterweighting of cranks

4-stroke engines

We consider now the case of 3 cylinders:







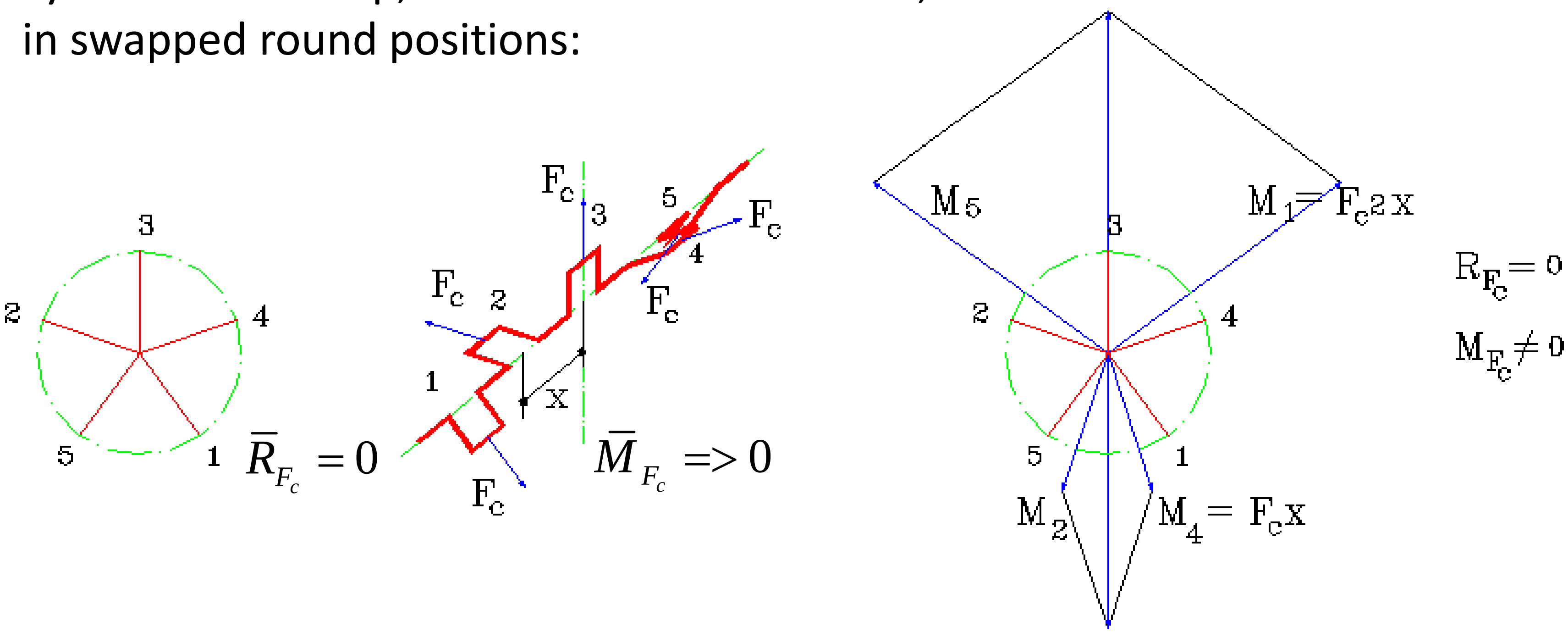
We have $\bar{R}_{F_c} = 0$; to obtain $\bar{M}_{F_c} = 0$ we must add the counterweights.

Balancing and counterweighting of cranks

5 cylinders

The phase shift of the cranks is $\Delta\varphi = 360° \cdot \frac{2}{5} = 144°$

Anti-metric lengthwise layout of cranks: we put the crank of the central cylinder at the top, then the two that follow, and then the outermost ones in swapped round positions:



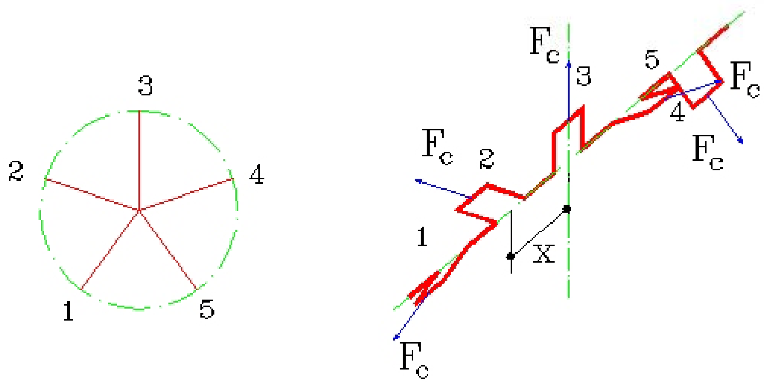
The resultant is not cancelled out, but we obtain a certain balance.

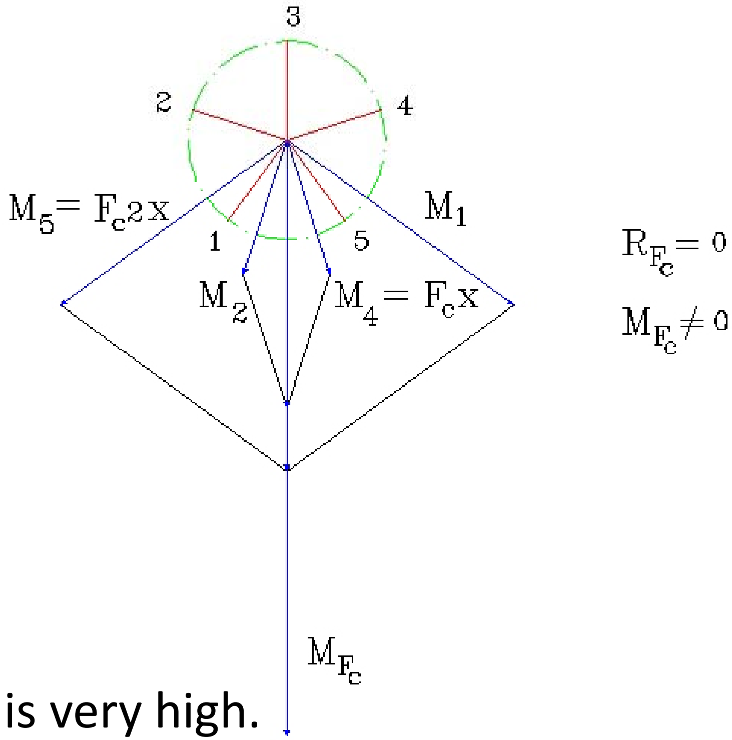
Balancing and counterweighting of cranks

<u>5 cylinders</u>

The phase shift of the cranks is $\Delta\varphi = 360° \cdot \frac{2}{5} = 144°$

If we don't adopt the *anti-metric lengthwise layout of cranks* we have





The resultant is not cancelled out, and is very high.

Balancing and counterweighting of cranks

2-stroke engines

We must distinguish two conditions:

i_{tot} odd ➔ We adopt the same lengthwise layout of cranks determined for the 4-stroke engines (*anti-metric lengthwise layout of cranks)*:

$$\bar{M}_{F_c} => 0 \qquad \bar{M}_{F_{a'}} => 0$$

i_{tot} even: in this case we have to distinguish between the two situations:

$\frac{i_{tot}}{2}$ odd ➔ $\bar{M}_{F_c} = 0$

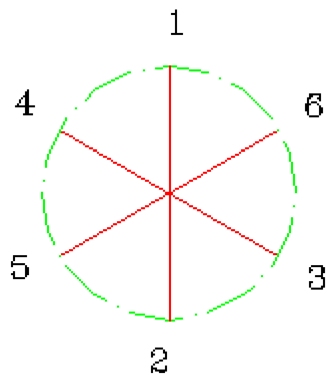
$\frac{i_{tot}}{2}$ even ➔ the complete balance is not possible;

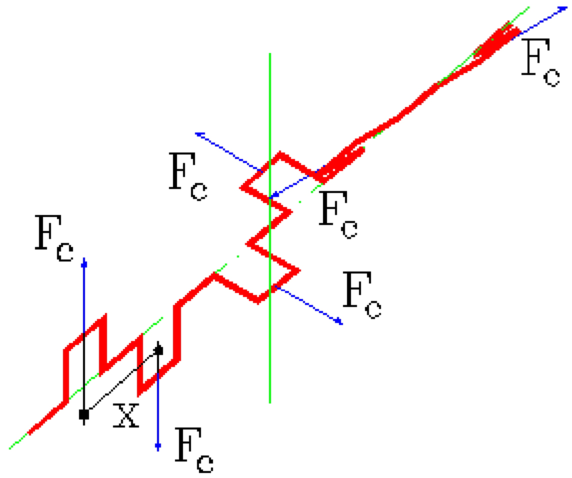
$$\bar{M}_{F_c} => 0 \qquad \bar{M}_{F_{a'}} => 0$$

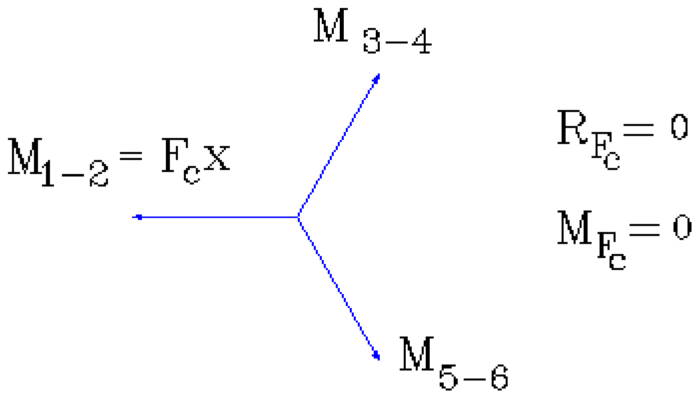
Balancing and counterweighting of cranks

2T: 6 cylinders

The phase shift of the cranks is $\Delta\varphi = 360° \cdot \frac{1}{6} = 60°$

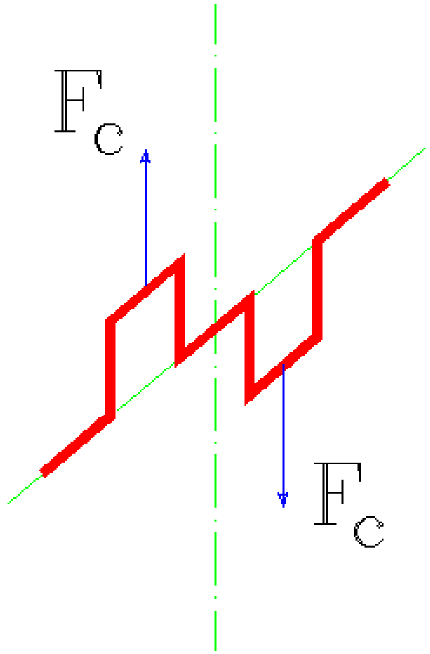






Balancing and counterweighting of cranks

the exception is the case of the **2 cylinders**, for which $\Delta\varphi = 180°$



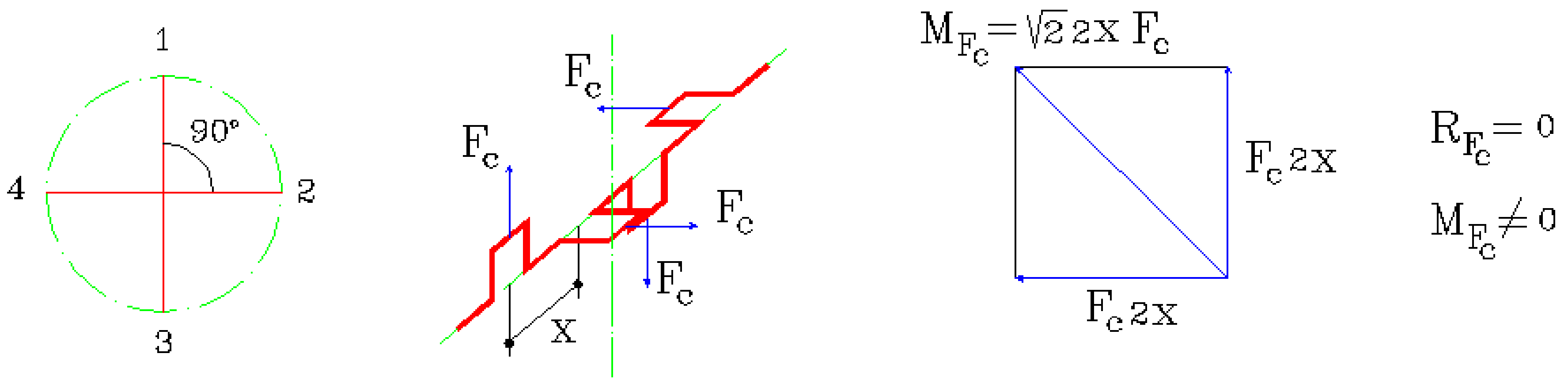
$$R_{F_C} = 0$$

$$M_{F_C} \neq 0$$

Balancing and counterweighting of cranks

2T: 4 cylinders

The phase shift of the cranks is $\Delta\varphi = 90°$

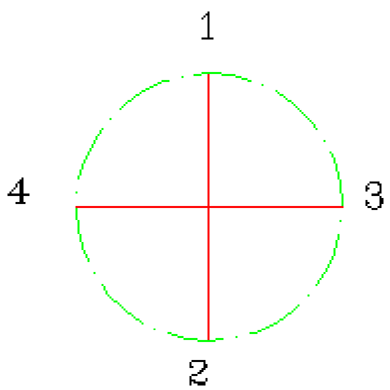


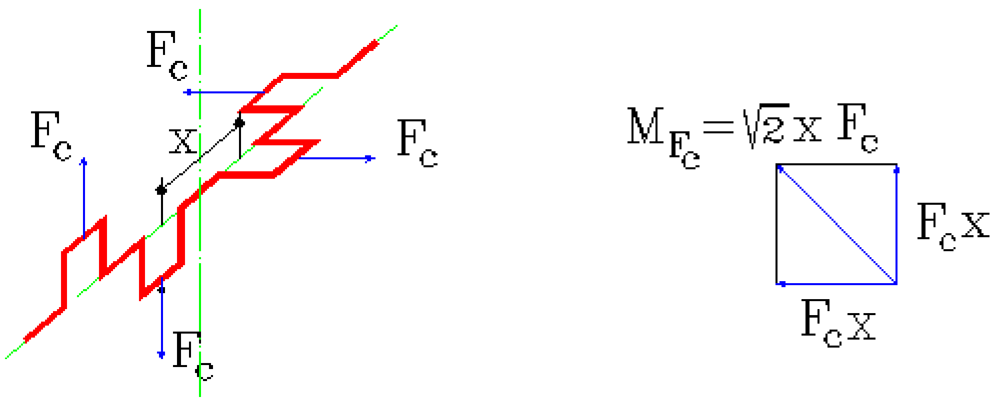
This is **not** an optimal configuration;

Balancing and counterweighting of cranks

2T: 4 cylinders

the configuration below is better:





$M_{F_c} = \sqrt{2}\, x\, F_c$

$R_{F_c} = 0$

$M_{F_c} \neq 0$

because it has a moment resultant lower than the previous one.

Balancing and counterweighting of cranks

Therefore, the resultant is not cancelled out for this example: for the balancing of the first order <u>residual reciprocating forces</u> we can use:

for the $F_{a_1'}$ ➔ masses rotating with ω in the crank direction;

for the $F_{a_2'}$ ➔ masses rotating with -ω in the opposite crank direction

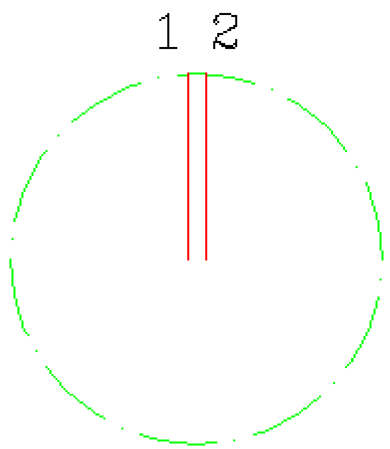
Balancing and counterweighting of cranks

To determine the lengthwise layout of the cranks sometimes the criteria of even spaced of cranks is not adopted: in fact, we can consider now the case of the *4-stroke 2 cylinder engine*:

- with the even spaced crank configuration, the phase shift of the cranks is

$$\Delta\varphi = 360° \cdot \frac{2}{2} = 360°$$

4T 2 cylinders



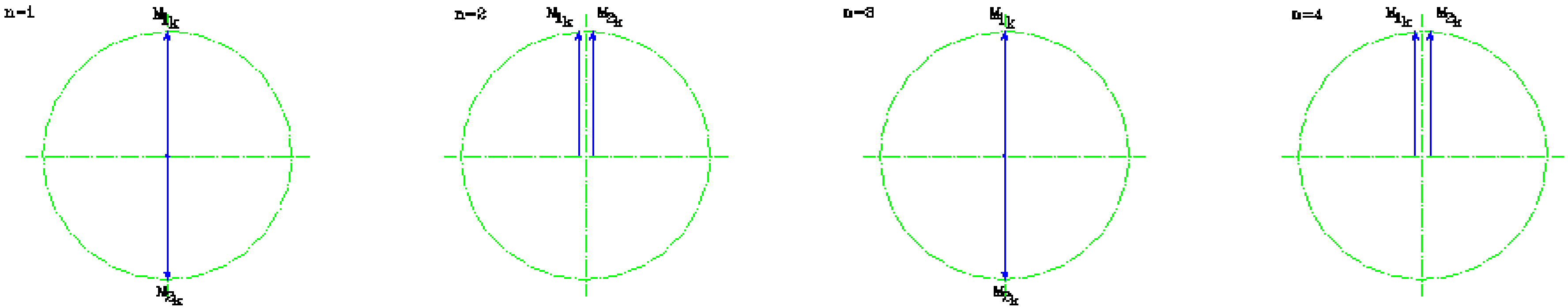
$$R_{F_c} \neq 0 \qquad R_{F_a'} \neq 0$$

$$M_{F_c} = 0 \qquad M_{F_a'} = 0$$

- The resultants $\overline{R}_{F_c}$ and $\overline{R}_{F_{a'}}$ are not cancelled out

Balancing and counterweighting of cranks

From the point of view of the torque uniformity, we consider now the first 4 harmonics:



$$\Delta\left(\delta_{i_k}\right)=k\Delta\varphi=k\cdot 360° \quad \text{con} \quad k=\frac{n}{2} \quad (n=1,2,3,..)$$

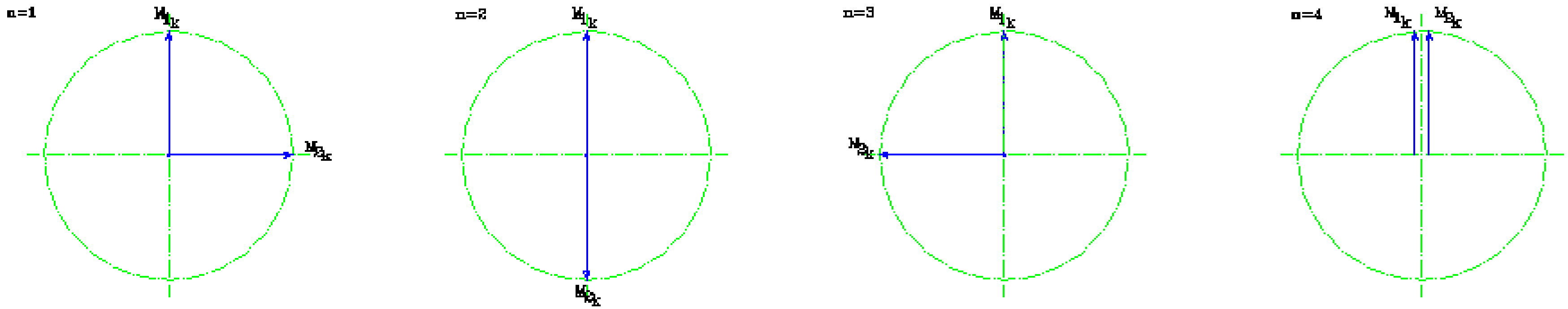
We have cancelled out the harmonics with k=1/2 and k=3/2, but this doesn't mean that we don't have important roughness (in fact, the harmonics k=1 and k=2 add up and are not much lower than the ones with k=1/2 and k=3/2

Balancing and counterweighting of cranks

Vice versa, the configuration with a phase shift of the cranks $\Delta\varphi = 180°$

$$\overline{R}_{F_c} = 0; \quad \overline{M}_{F_c} \neq 0$$

while for the harmonics $\Delta\left(\delta_{i_k}\right) = k\Delta\varphi = k \cdot 180°$ con $k = \dfrac{n}{2}$ $(n = 1, 2, 3, ..)$



we have cancelled out the harmonic of the first order while the others are present.

Balancing and counterweighting of cranks

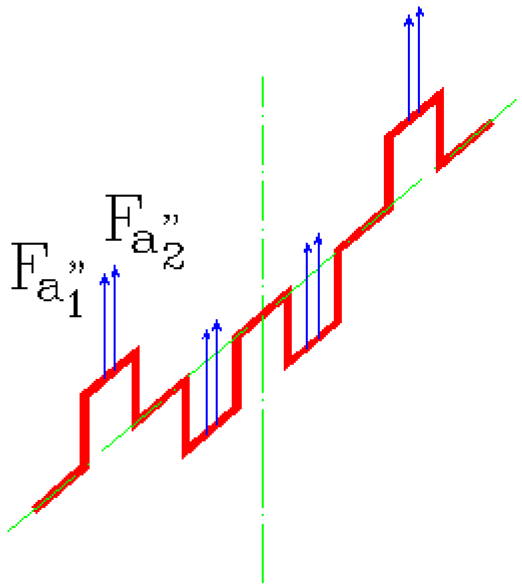
We must take into account that a crankshaft, with a determined lengthwise layout, even if it may have worse conditions from the moment point of view, it can be utilized because it has some advantages in the balancing of second order reciprocating forces.

Second order reciprocating forces

With the scheme adopted previously, also the second order reciprocating forces have been led back to two systems rotating in the crank and opposite directions but with the angular speed of 2ω.

Balancing and counterweighting of cranks

4-stroke 4 cylinder engine:



$$R_{F_{a_1}"} = 4\, F_{a_1}"$$

$$R_{F_{a_2}"} = 4\, F_{a_2}"$$

$$M_{F_a"} = 0$$

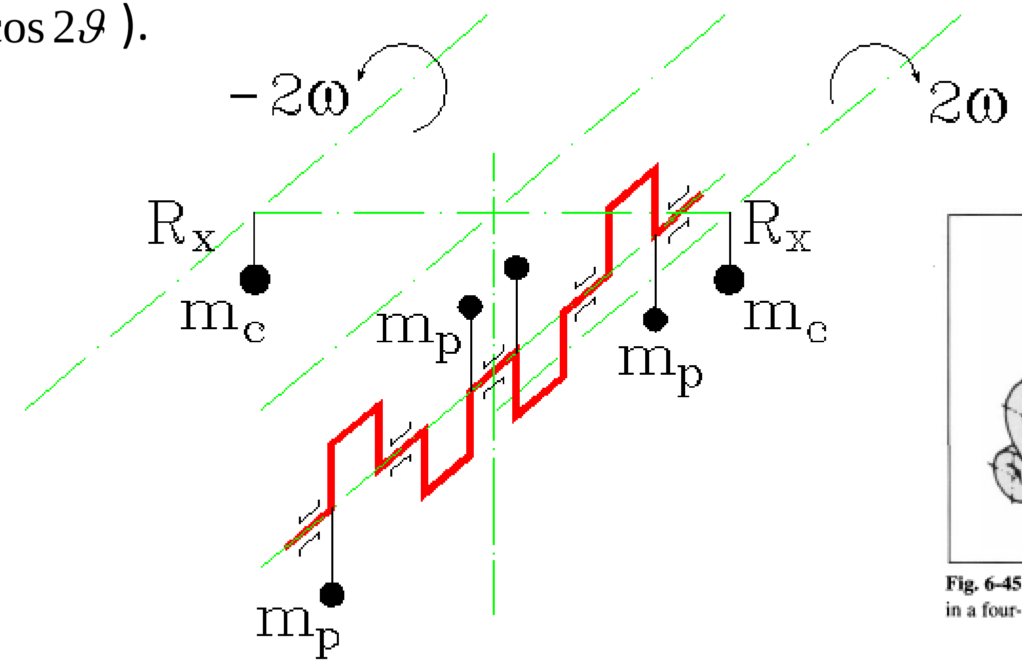
The considerations relative to the field of forces $F_{a_1^{"}}$ are also valid for the field of forces $F_{a_2^{"}}$, specular to the former.

From the figure we can see that $R_{F_{a"}} \neq 0$, while $M_{F_{a"}} = 0$.

Balancing and counterweighting of cranks

We can balance $R_{F_{a''}}$ using two shafts rotating in opposite directions with off-centre masses. The former, rotating with 2ω, balances $R_{F_{a''_1}}$, while the latter, rotating with -2ω, balances $R_{F_{a''_2}}$.

The two shafts are parallel and symmetric respect to the crankshaft axis and have off-centre masses in a normal plane passing through the centre line ($F_{a''} = -m_a \omega^2 r \lambda \cos 2\vartheta$).



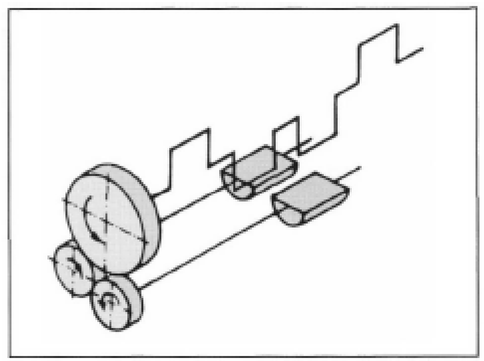


Fig. 6-45 Diagram of mass balancing of the second order in a four-stroke crank gear.

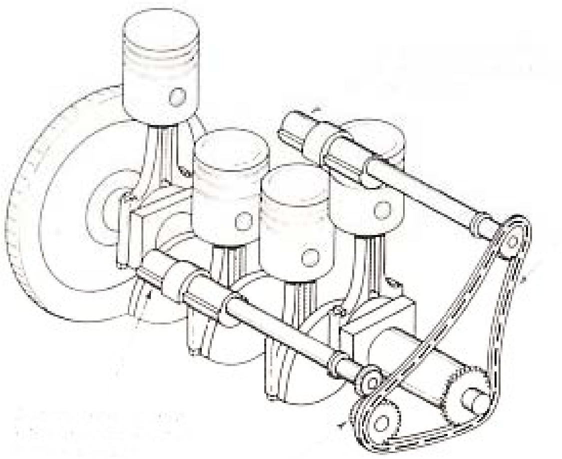
Balancing and counterweighting of cranks

$$R_{F_{a"_1}} = 4F_{a"_1} = 4 \cdot \frac{1}{2} m_a \omega^2 r \lambda$$

this force must be balanced by $m_c (2\omega)^2 R_x$ and then

$$2m_a \omega^2 r \lambda = m_c 4\omega^2 R_x \qquad => \qquad m_c = \frac{1}{2} m_a \frac{r}{R_x} \lambda$$

Then the mass m_c can be determined taking into account the design requirements of the engine (weight of cylinder block and room available).



Balancing and counterweighting of cranks

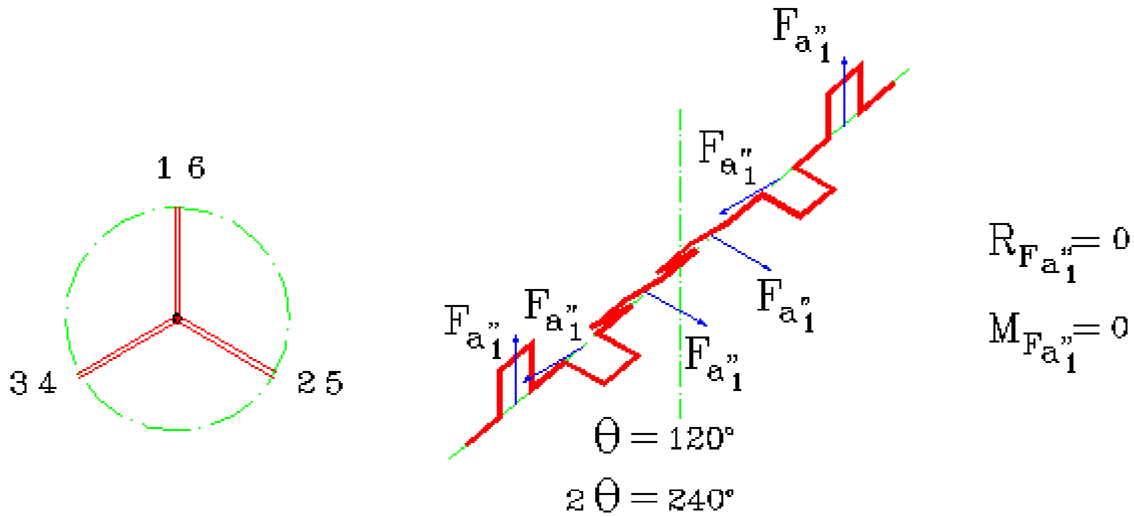
The engine crankshaft must then be counterweighted as in the figure above. In fact, the engine crankshaft is supported by five main-bearing caps (n_{cyl} +1) and the counterweighting is intended to reduce the stresses on the single support.

An increase in the number of supports, as is the current tendency, makes the structure strongly redundant and it is therefore appropriate to discharge every single support.

Nevertheless it is important that the counterbalance produces a field of forces in itself balanced.

Balancing and counterweighting of cranks

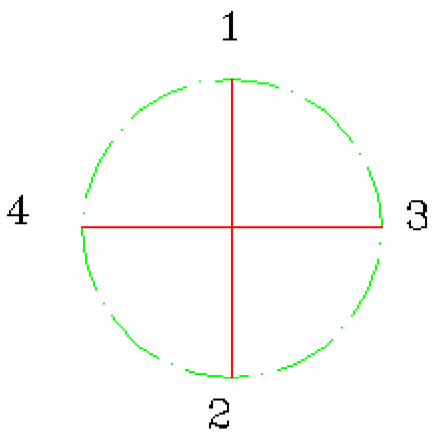
4-stroke 6 cylinders engine:

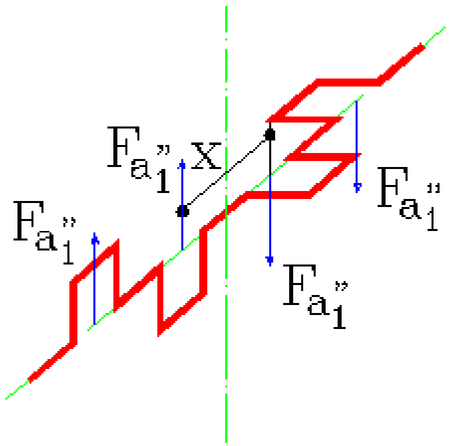


The 6 cylinder engine is "naturally" balanced. But the crankshaft has a considerable lengthwise extent and so it must be strengthened: it appears then to be particularly indicated for Diesel engines (these engines are in general stronger and have higher stresses).

Balancing and counterweighting of cranks

2-stroke 4 cylinders engine:





$$R_{F_c} = 0 \qquad R_{F_a'} = 0$$

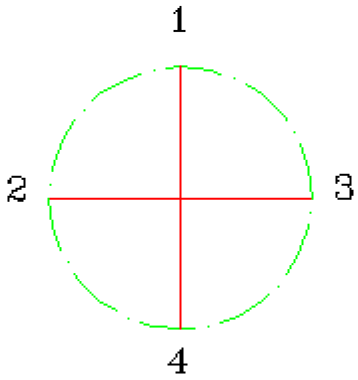
$$M_{F_c} \neq 0 \quad \left(M_{F_c} = \sqrt{2}\, x\, F_c \right)$$

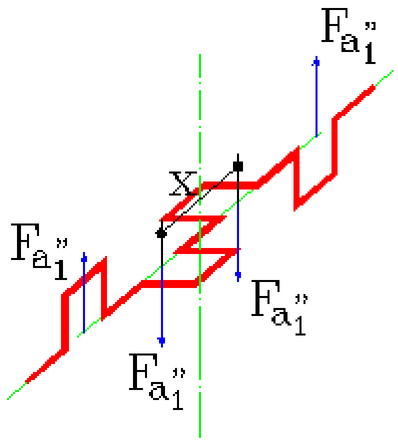
$$R_{F_a''} = 0$$

$$M_{F_{a_1}''} = F_{a_1}'' \cdot 3x + F_{a_1}'' \cdot x = F_{a_1}'' \cdot 4x$$

Balancing and counterweighting of cranks

instead, with the layout





$R_{F_c} = 0 \qquad R_{F_{a'}} = 0$

$M_{F_c} \neq 0 \quad (M_{F_c} = \sqrt{10}\ x\ F_c)$

$R_{F_{a''}} = 0$

$M_{F_{a_1''}} = 0 \qquad M_{F_{a_2''}} = 0$

the second order forces are balanced.

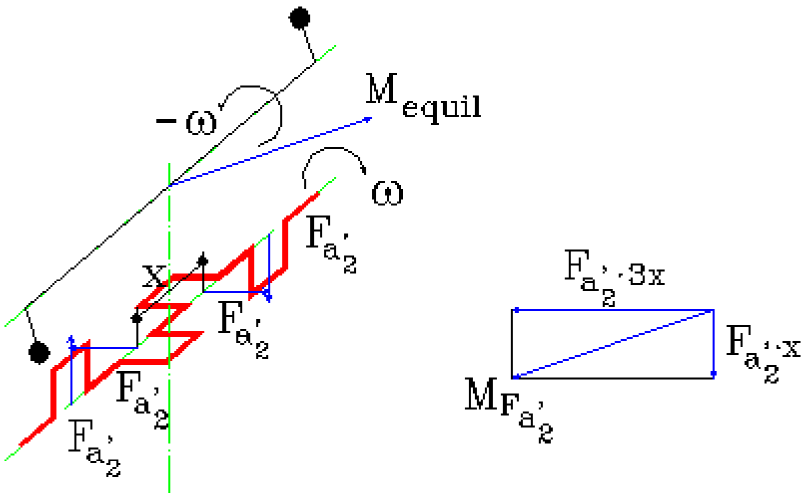
Counterweights must be insert on the crankshaft to balance $F_{a'}$ and F_c . (the counterweights rotate with ω and then $\bar{M}_{F_{a'_2}}$ is left out).

Balancing and counterweighting of cranks

For the forces $F_{a'_2}$ rotating in the opposite crank direction only the resultant is null.

To balance the resultant moment $M_{F_{a'_2}}$ we can insert a little shaft, rotating in the opposite crank direction, with the axis in a plane passing through the crankshaft axis, that gives rise to a moment equal to :

$$M_{F_{a'_2}} = \sqrt{10} F_{a'_2} x$$



Balancing and counterweighting of cranks

V engine

Bank of cylinders is a group of cylinders located on the same side of the shaft with their axes in a plane passing through the crankshaft axis.

Row of cylinders is a group of radial cylinders lying in a plane at right angles to the crankshaft axis and operating on a single crank.

Balancing and counterweighting of cranks

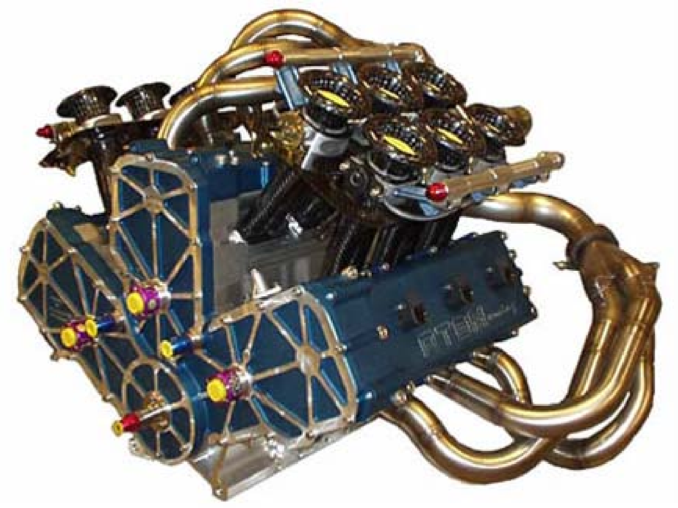
V engine

V engine is an engine with two banks of cylinders, corresponding cylinders in each bank forming a 2-cylinder row.

V angle is the angle between the cylinder planes of a V engine.

W engine is similar to a V engine but with three banks of cylinders. The two V angles are usually equal.

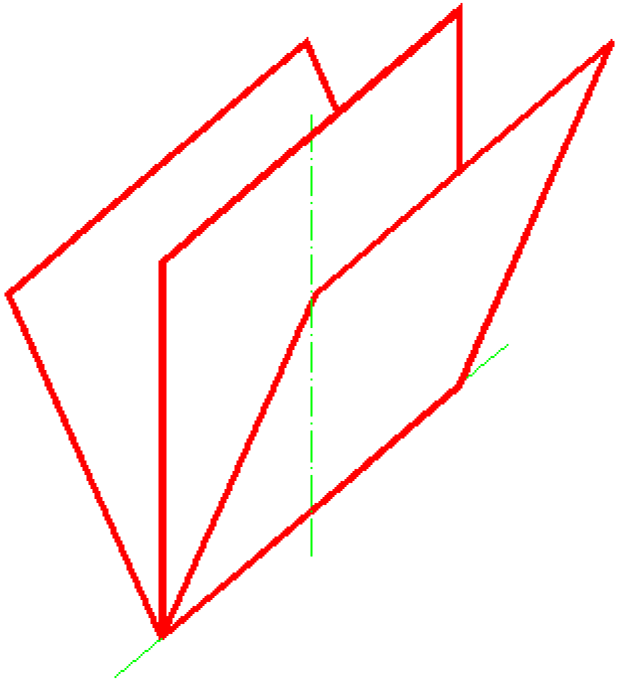
Opposed engine is a V engine with a 180° V angle.

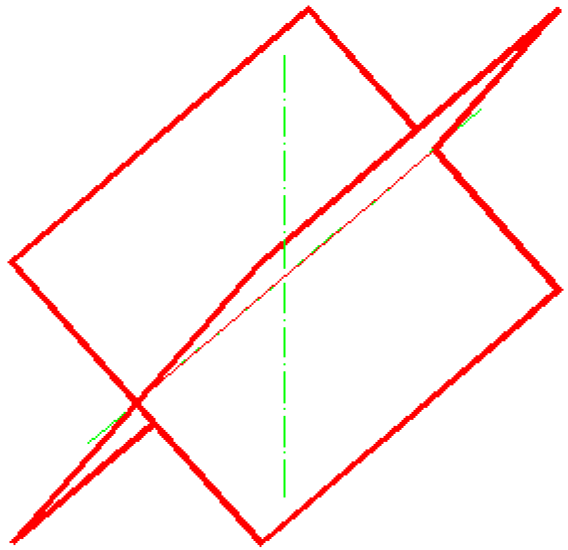


W-type engine

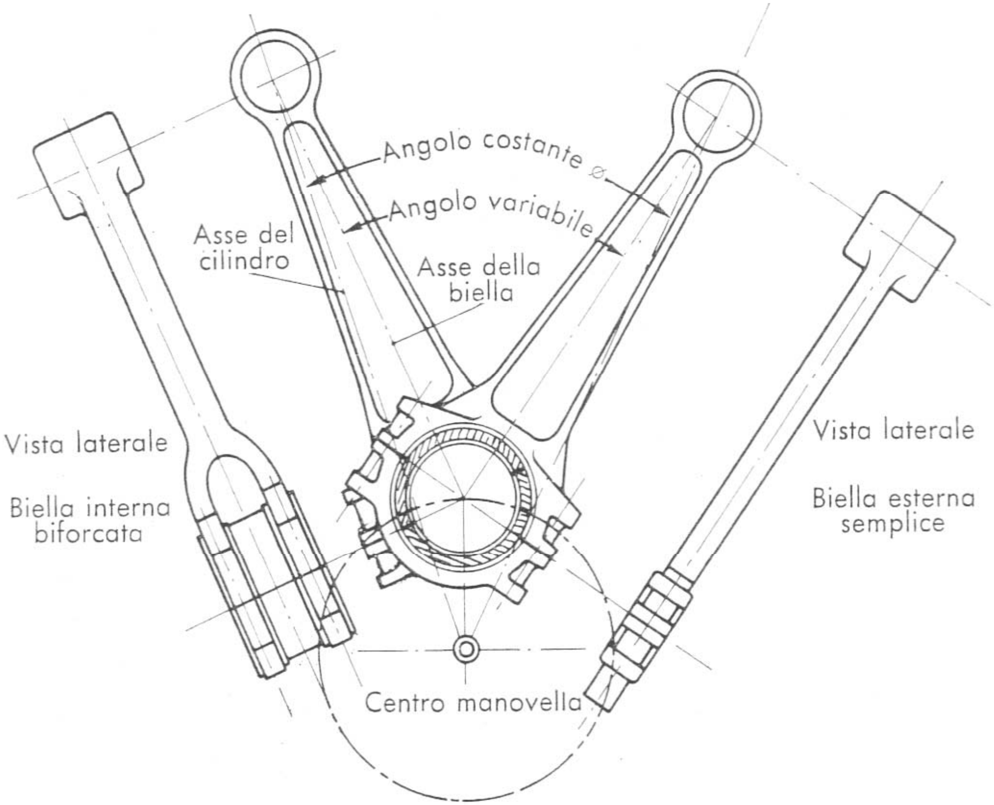
Boxer engine

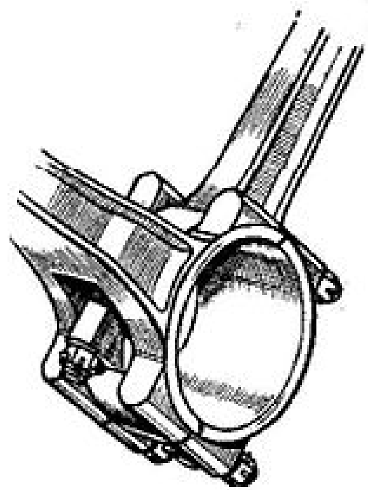
Balancing and counterweighting of cranks





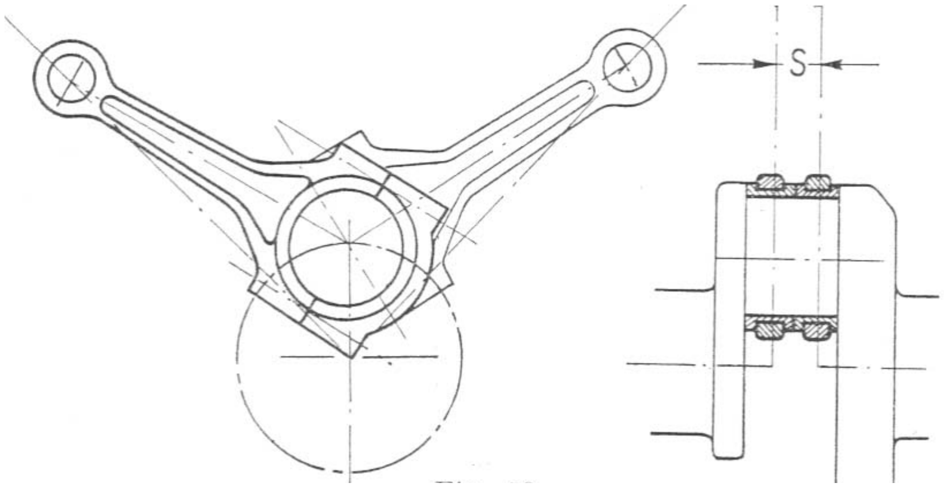
Balancing and counterweighting of cranks





Connecting rods for engines with proper V.

Balancing and counterweighting of cranks



Two connecting rods on the same crankpin for a V engines.

Balancing and counterweighting of cranks

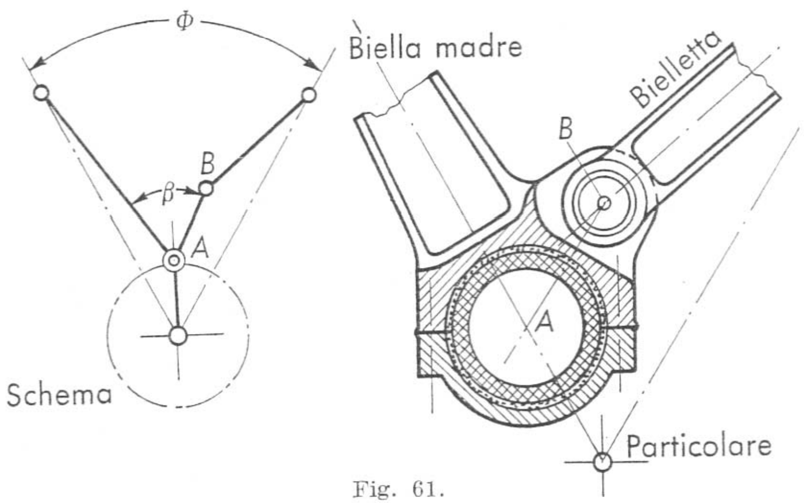
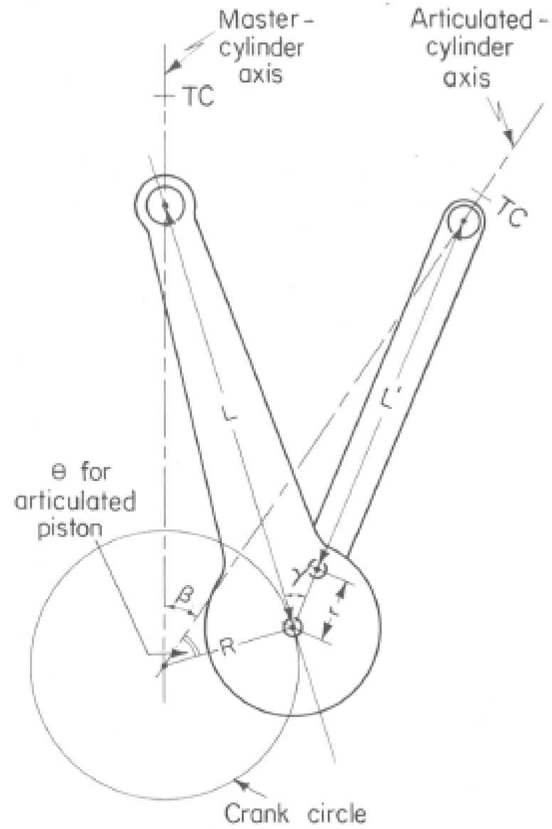


Fig. 61.



Master-articulated connecting rods for engines with proper V.

Balancing and counterweighting of cranks

Let us see how to determine the V angle of the engine.

Engines with proper V.

We use only one connecting rod journal for two or more cylinders [2-cylinder row] (connecting rods concentric in their big end; another solution is the articulated connecting rod).

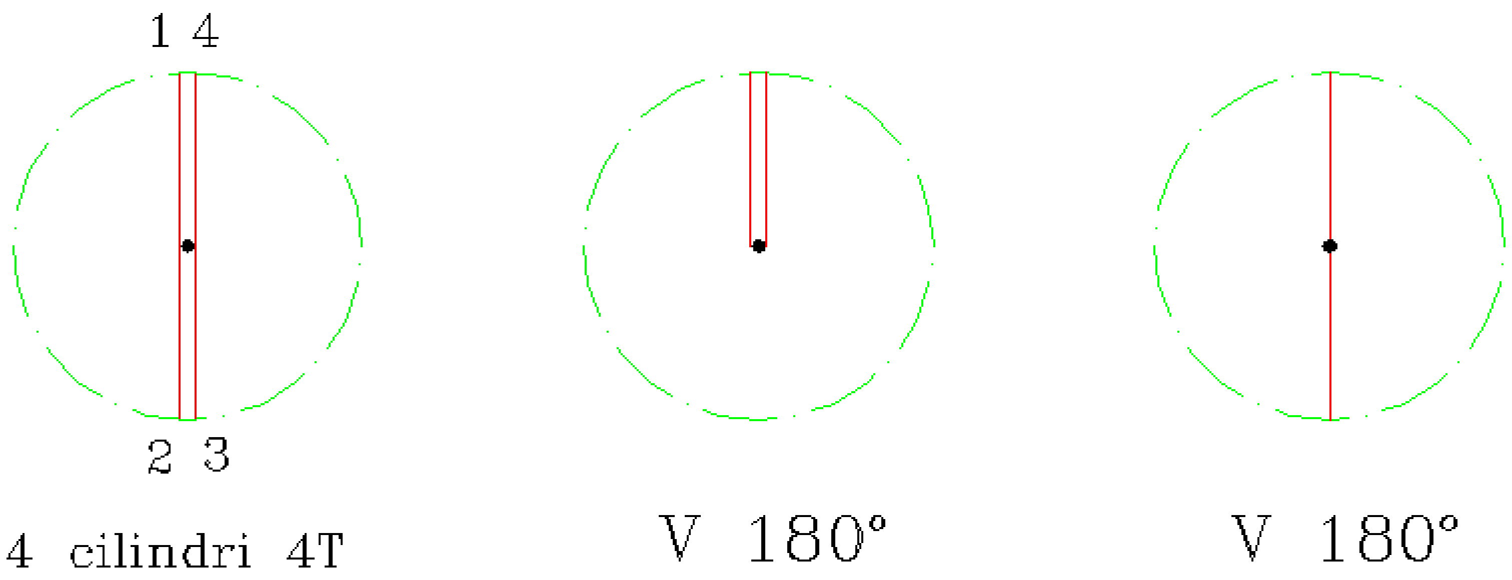
We would like to maintain the uniform firing of the cycles: we start from the in-line layout and then we apply the <u>partial rotation principle</u>:

> <u>*if we rotate the cylinder and its crank rigidly, the phase displacement of their cycle doesn't change compared to the other cycles.*</u>

We shall now consider the case of a V engine with two banks of cylinders.

Balancing and counterweighting of cranks

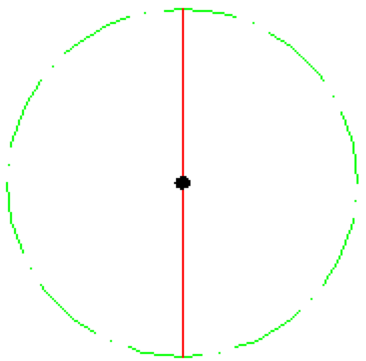
For the 4 cylinder 4-stroke engine we have only two cranks:



The solution that we may adopt depends on whether the reciprocating inertia forces are balanced or not.

Balancing and counterweighting of cranks

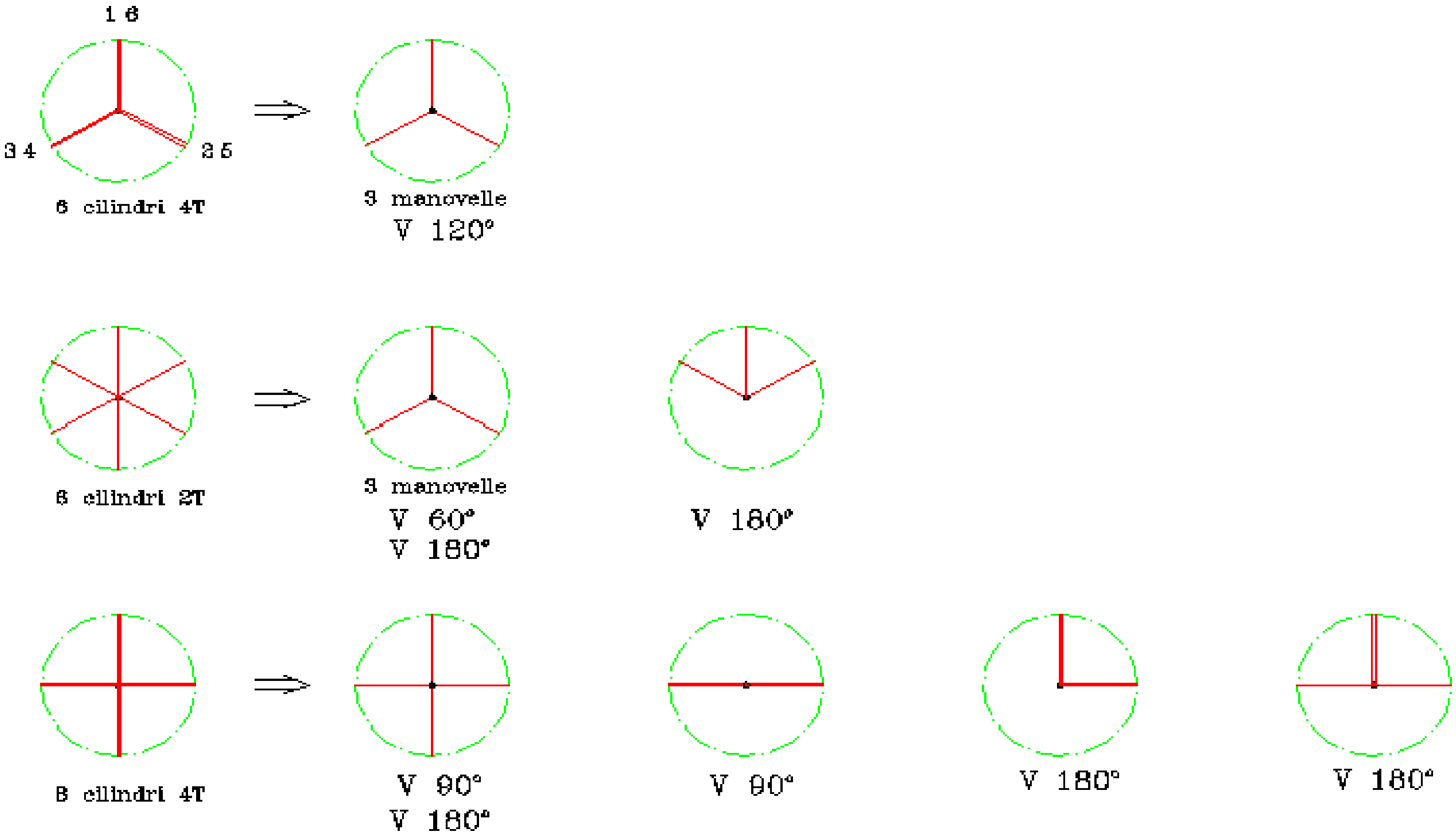
For the 4 cylinder 4-stroke engine we have only two cranks:



V 180°

The solution that we may adopt depends on whether the reciprocating inertia forces are balanced or not.

Balancing and counterweighting of cranks



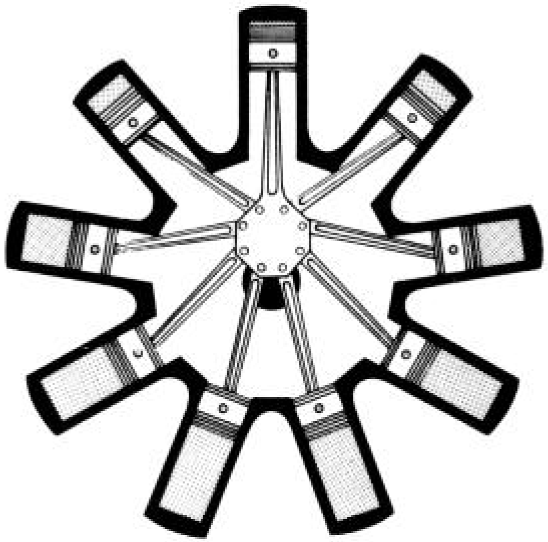
Considerations concerning the forces F_c and F_a will lead us to the best solution.

Balancing and counterweighting of cranks

Radial engines

Radial engine is an engine having three or more cylinders in one row with equal angles between their axes.

Multirow radial engine is a radial engine with more than one row of cylinders.

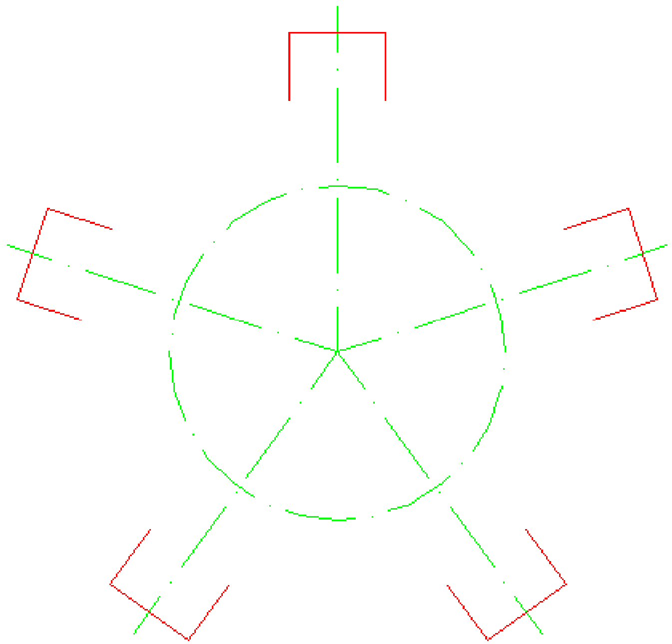


Balancing and counterweighting of cranks

Radial engines

Starting from the radial layout for the cranks and a row of cylinders we can obtain a radial layout for the cylinders, with just one crank.

4-stroke engines displaying an even number of cylinders cannot be designed with a single row of cylinders. The rotation of the cylinders would in fact lead to coincident axes.



Balancing and counterweighting of cranks

Radial engines

The multi-row radial engine architecture fits better the odd cylinder numbers. Radial engines are normally air cooled and the cylinder of the second row must fit in the space between the two consecutive cylinders of the first row.

Balancing and counterweighting of cranks

In many V engines the forces due to piston acceleration and the resulting moments are balanced for each bank of cylinders.

This is the case in conventional 4-stroke 12- and 16-cylinder engines.

In some cases, however, notably in the case of 90° V engines, it is possible to obtain excellent balance, even though the separate banks of cylinders may not in themselves be balanced.

The best method for analyzing the inertia forces of V engines is to consider a row of 2-cylinders that acts on a crank.

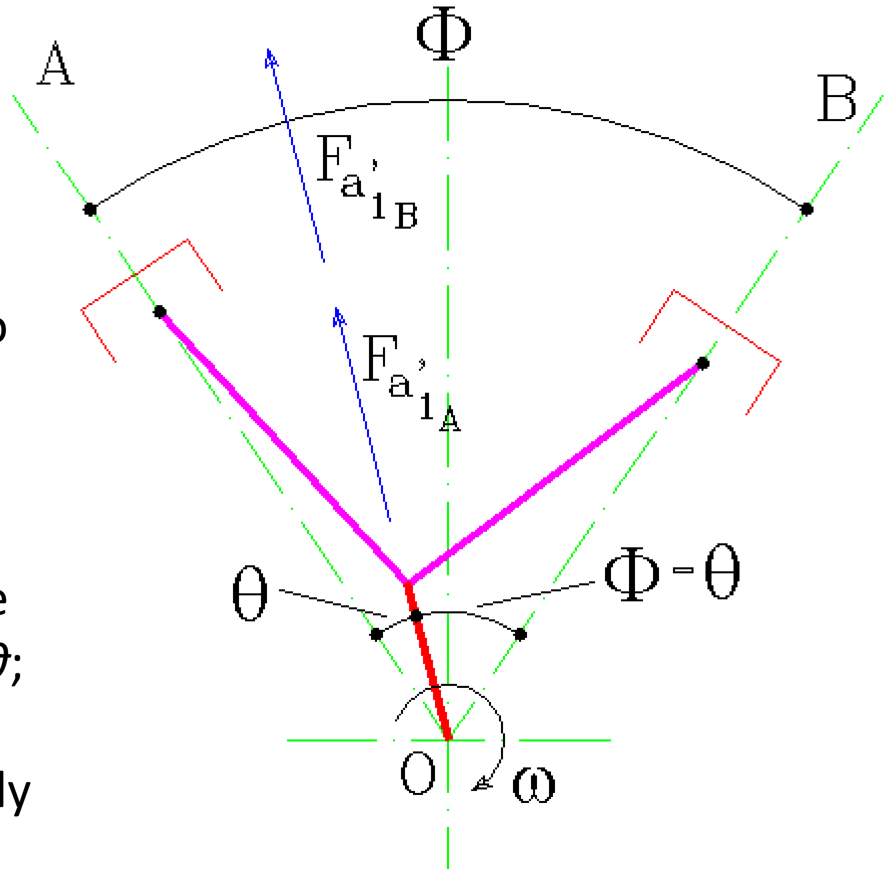
Balancing and counterweighting of cranks

Balancing of proper V

OA and OB are the directions of the V engine banks.

We can project the unbalanced forces from each cylinder on the axis of symmetry and on the perpendicular to it through the crankshaft center; then we add the components to find the unbalanced forces for the row.

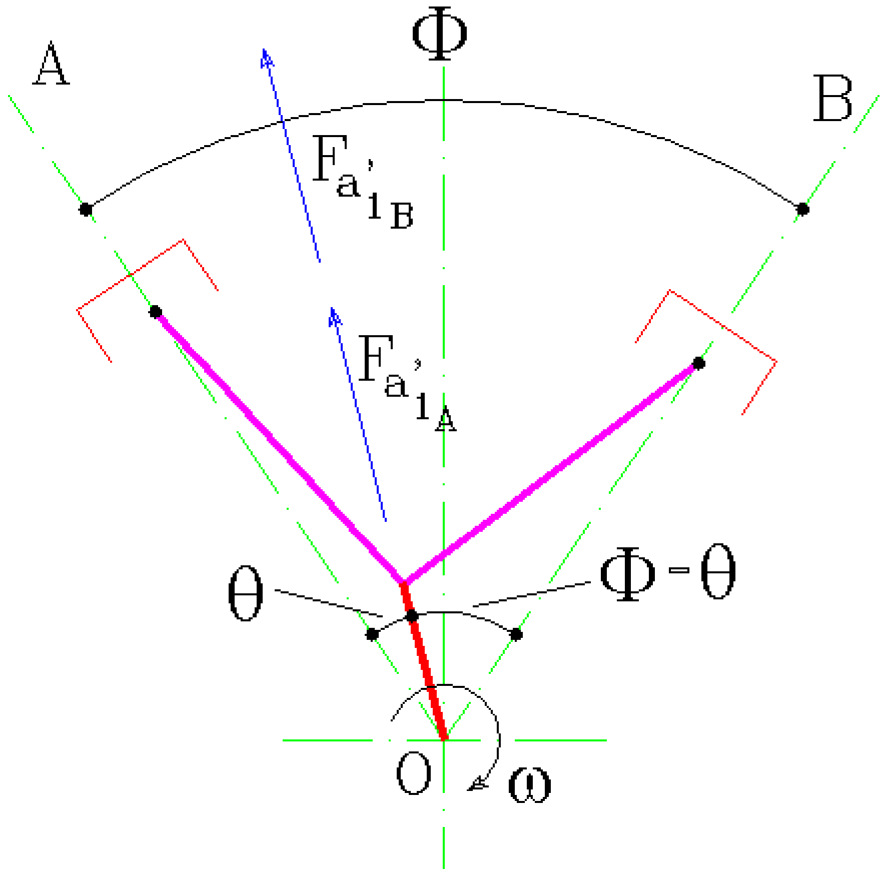
With the V angle Φ, let the crank angle for the left-bank cylinder be taken as ϑ; that for the right-bank will then be $2\pi-(\Phi-\theta)$, which is trigonometrically equivalent to $\theta-\Phi$.



Balancing and counterweighting of cranks

The centrifugal forces are added up in a simple way and as do the first order reciprocating forces $F_{a'_1}$.

They form the angle θ with the cylinder axis of bank OA (that is our reference), rotating in the direction of the crank.



Balancing and counterweighting of cranks

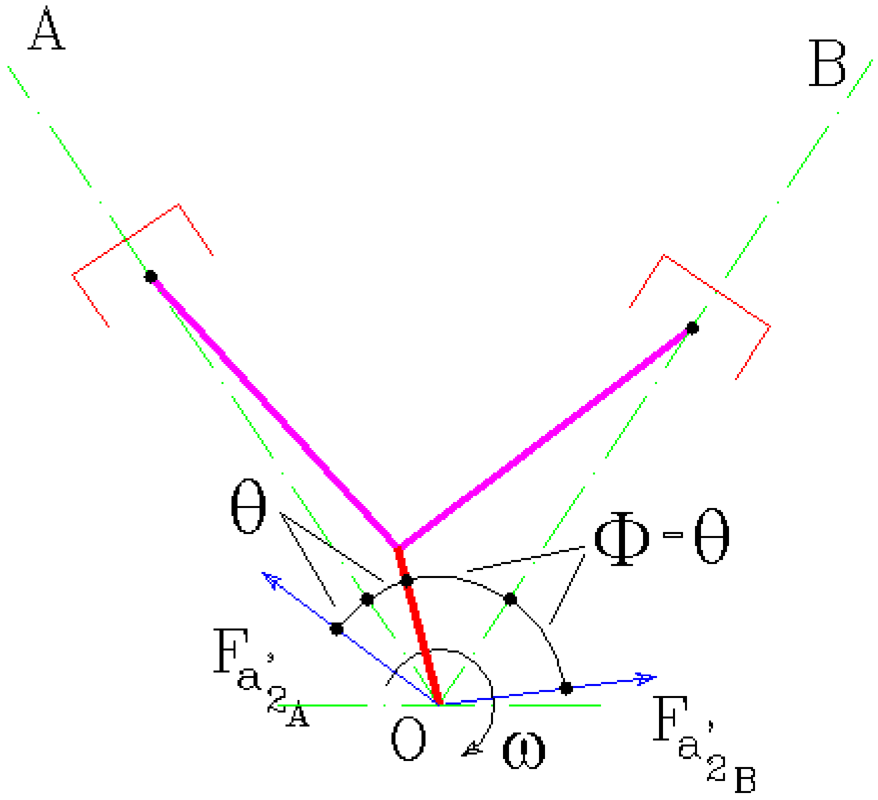
We can now consider the reciprocating forces $F_{a'_2}$

the angle ψ between the

vectors $F_{a'}$ of the two banks is

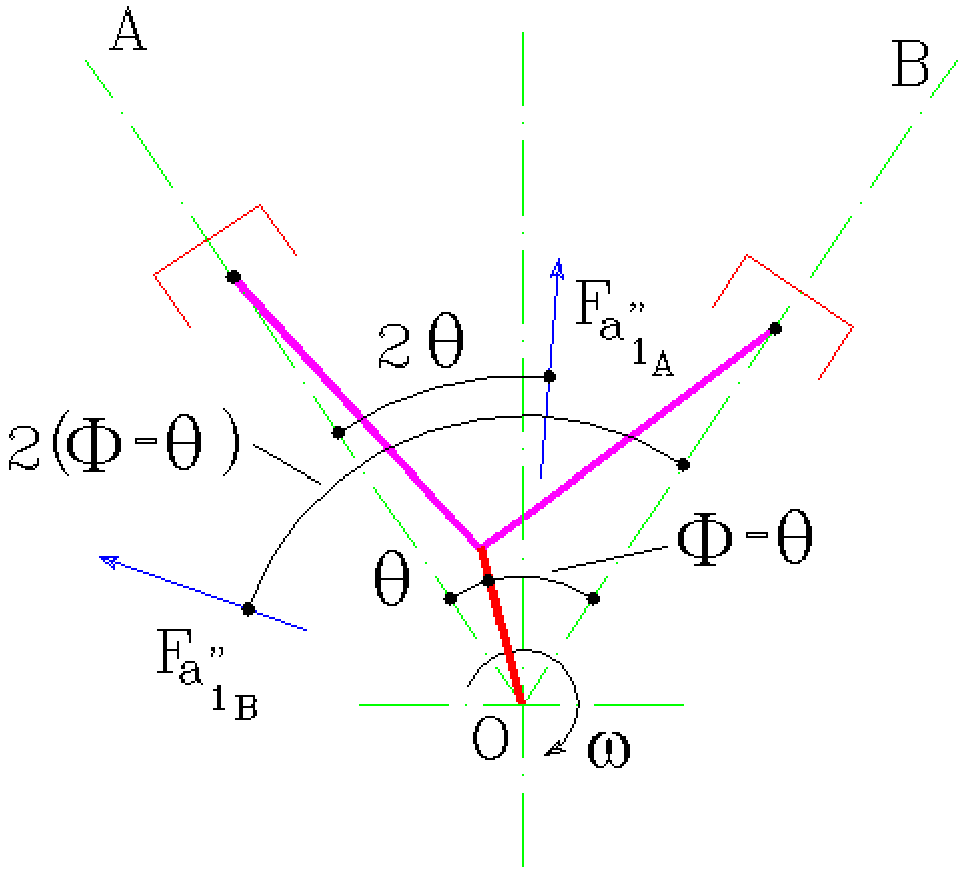
$$F_{a'_1} => \psi = 0$$

$$F_{a'_2} => \psi = 2\theta + 2(\Phi - \theta) = 2\Phi$$



Balancing and counterweighting of cranks

We shall consider now the second order reciprocating forces $F_{a"_1}$ rotating in the crank direction.

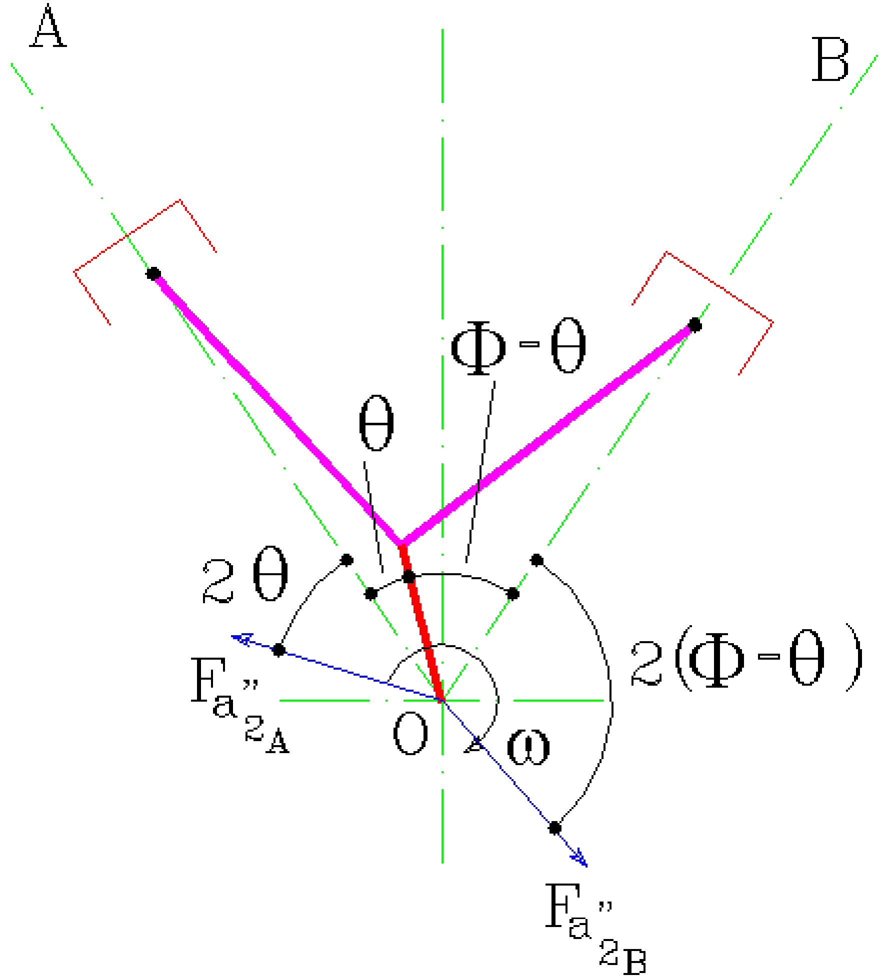


the angle ψ between the vectors $F_{a"_1}$ of the two banks is

$$\psi = 2(\Phi - \theta) - [\Phi - 2\theta] = \Phi$$

Balancing and counterweighting of cranks

… and then the second order reciprocating forces $F_{a"_2}$ rotating in the opposite crank direction.

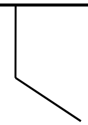
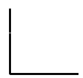
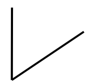
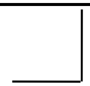


the angle ψ between the vectors $F_{a"_2}$ of the two banks is

$$\psi = 2\theta + \Phi + 2\left(\Phi - \theta\right) = 3\Phi$$

Balancing and counterweighting of cranks

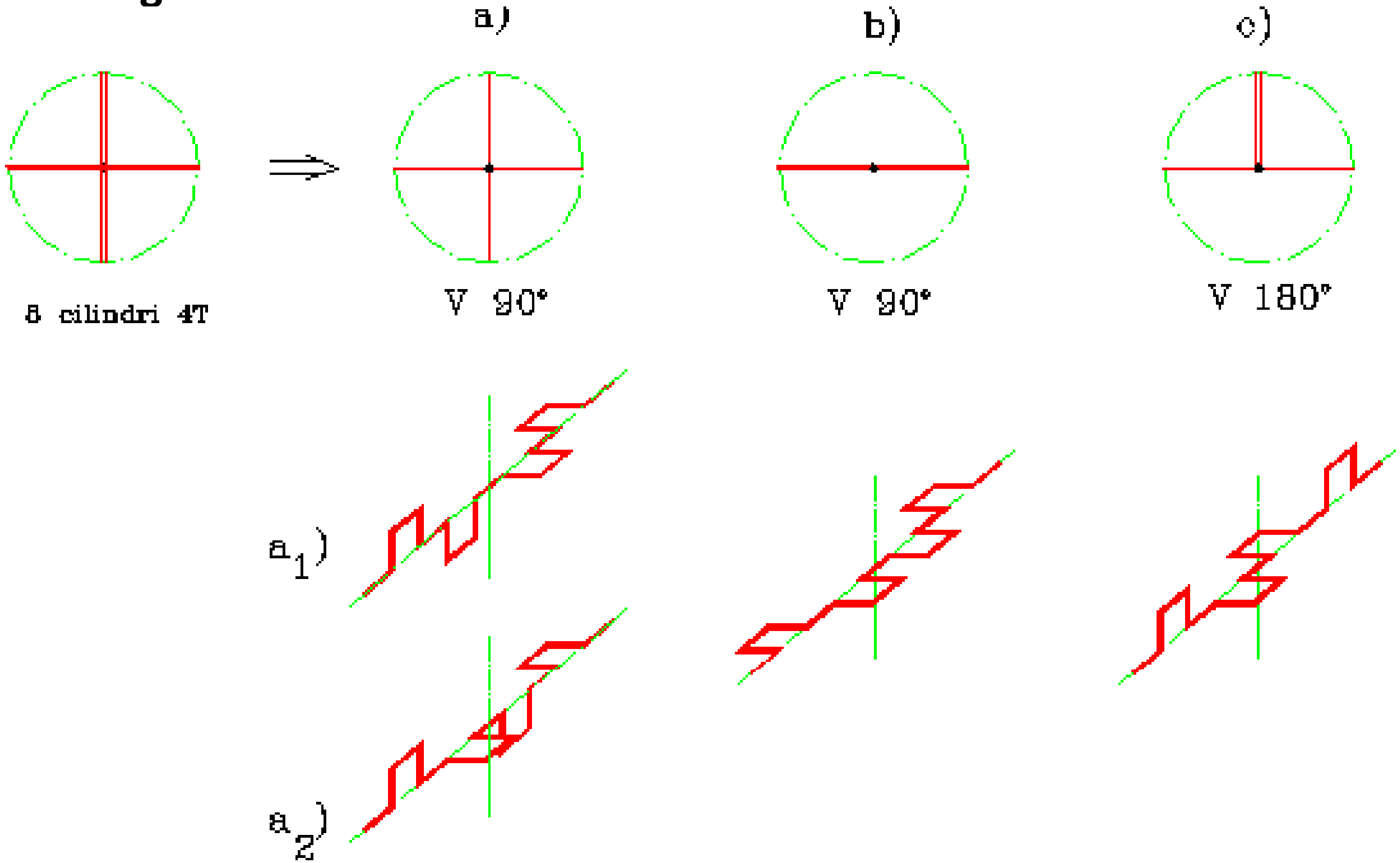
Let us sum up the obtained results in the following table:

Forces	V 90°	V 180°	V 60°	Phase shift ψ
$F_{a'_1}$	++	++	++	0
$F_{a'_2}$	X	++		2Φ
$F_{a''_1}$		X		Φ
$F_{a''_2}$		X	X	3Φ

(++ => the sum of the vectors; x => the resultant of vectors is cancelled)

Balancing and counterweighting of cranks

4 stroke engine



Balancing and counterweighting of cranks

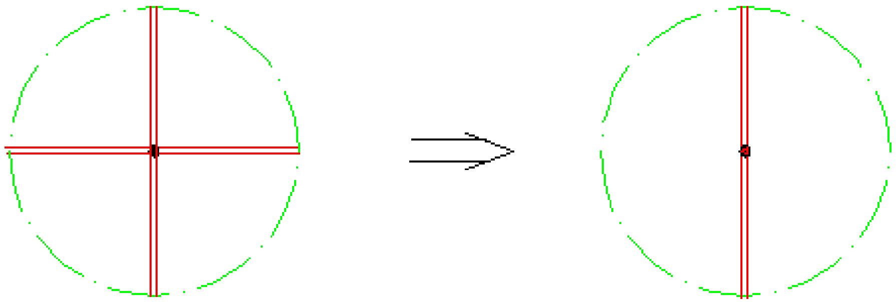
For every scheme proposed we have to consider the resultant of the forces $F_{a'_1}, F_{a'_2}, F_{a''_1}$ e $F_{a''_2}$ as a function of the corresponding V angle for every cranks and then see the balance of the crankshaft.

For the solution a) there are two possible configurations of the crankshaft , both already considered for the centrifugal forces F_c (4 cylinder 2 stroke engine).

Balancing and counterweighting of cranks

To give an example, for the configuration b) we have:

8 cilindri 4T



V 90°

$$R_{F_a'} = 0 \qquad M_{F_a'} = 0$$

$$R_{F_a''} = 4\, F_a''$$

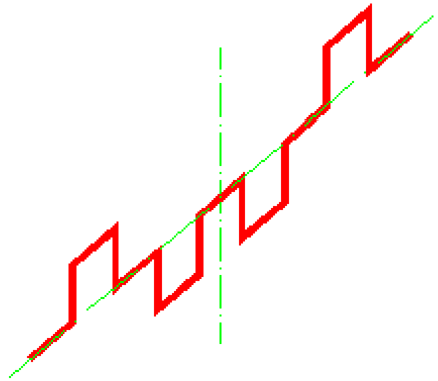
$$M_{F_a''} = 0$$

$$F'_{a_1} = F'_{a_{1_A}} + F'_{a_{1_B}}$$

$$F'_{a_2} = 0$$

$$F''_{a_1} = \sqrt{2}\, F''_{a_{1_A}}$$

$$F''_{a_2} = \sqrt{2}\, F''_{a_{2_A}}$$



Balancing and counterweighting of cranks

Appendix

Internal part from the cylinder block and head

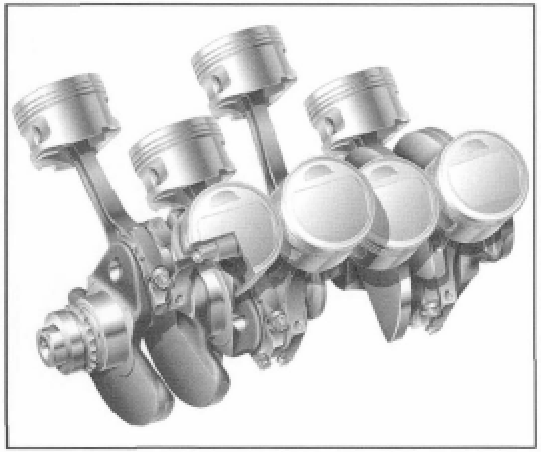


Fig. 6-1 Crank gear of a V-8 passenger car spark-ignition engine.

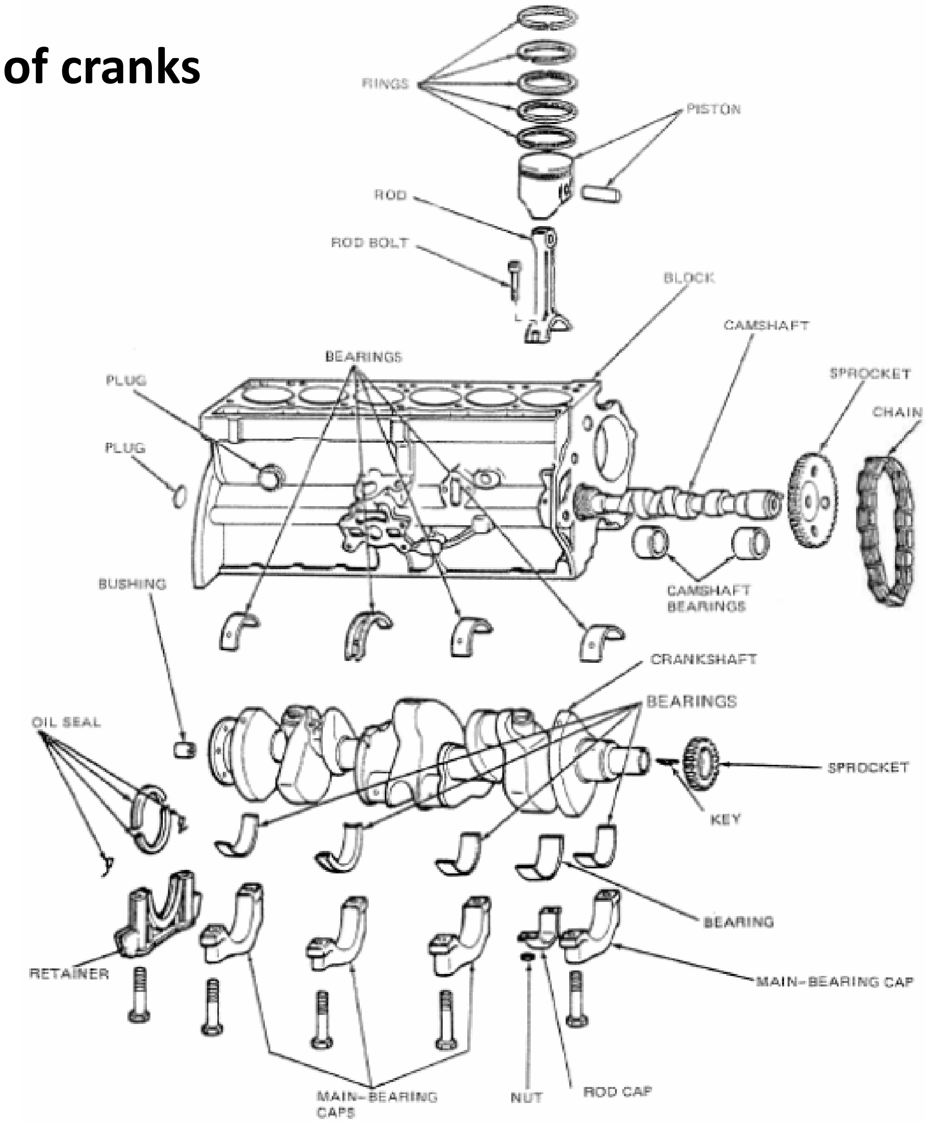


Fig. 4-6 Internal parts from the cylinder block and head of a six-cylinder engine. Only one piston, connecting rod, and set of rings are shown. (*Chrysler Corporation*)

Balancing and counterweighting of cranks

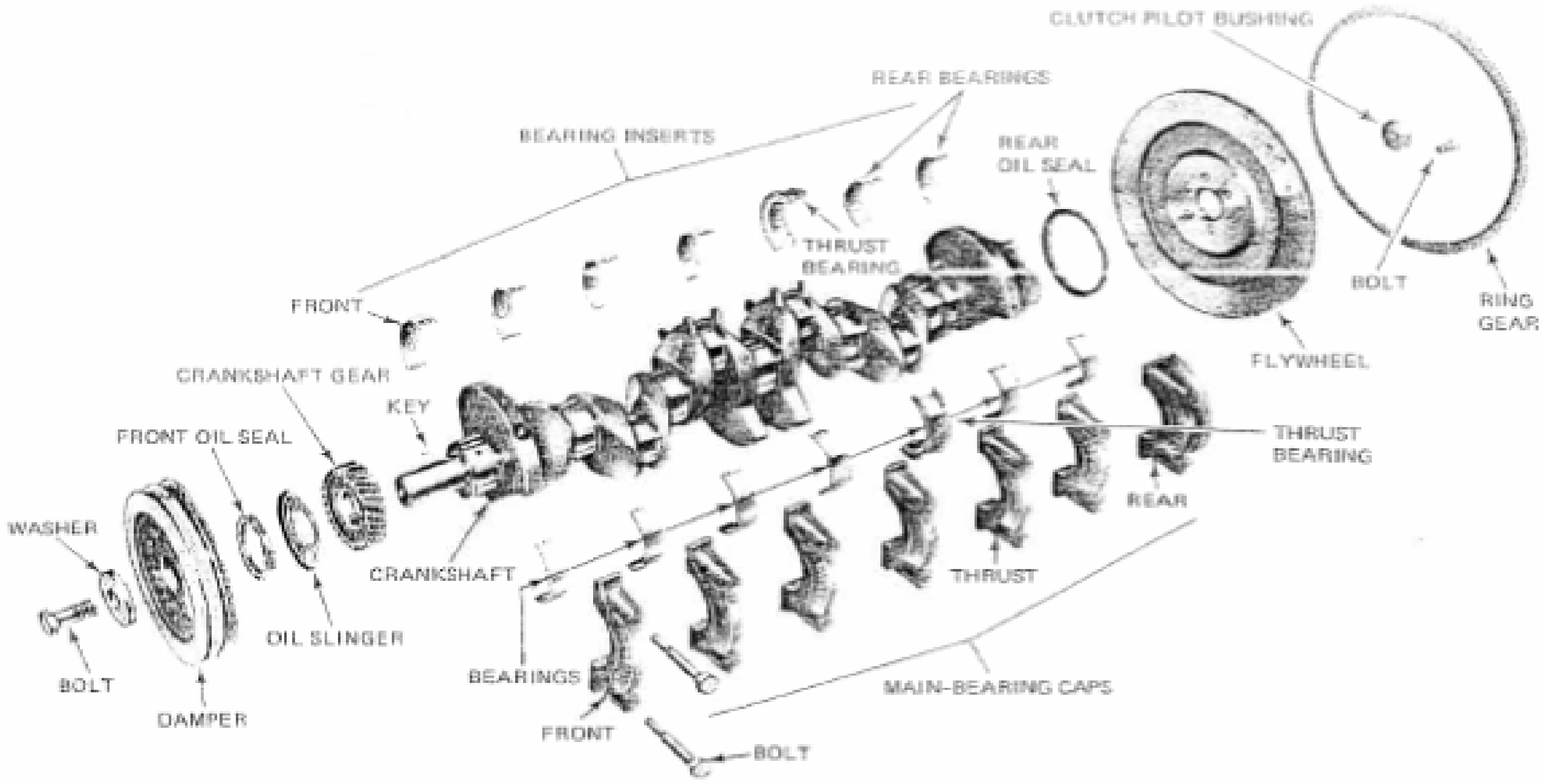
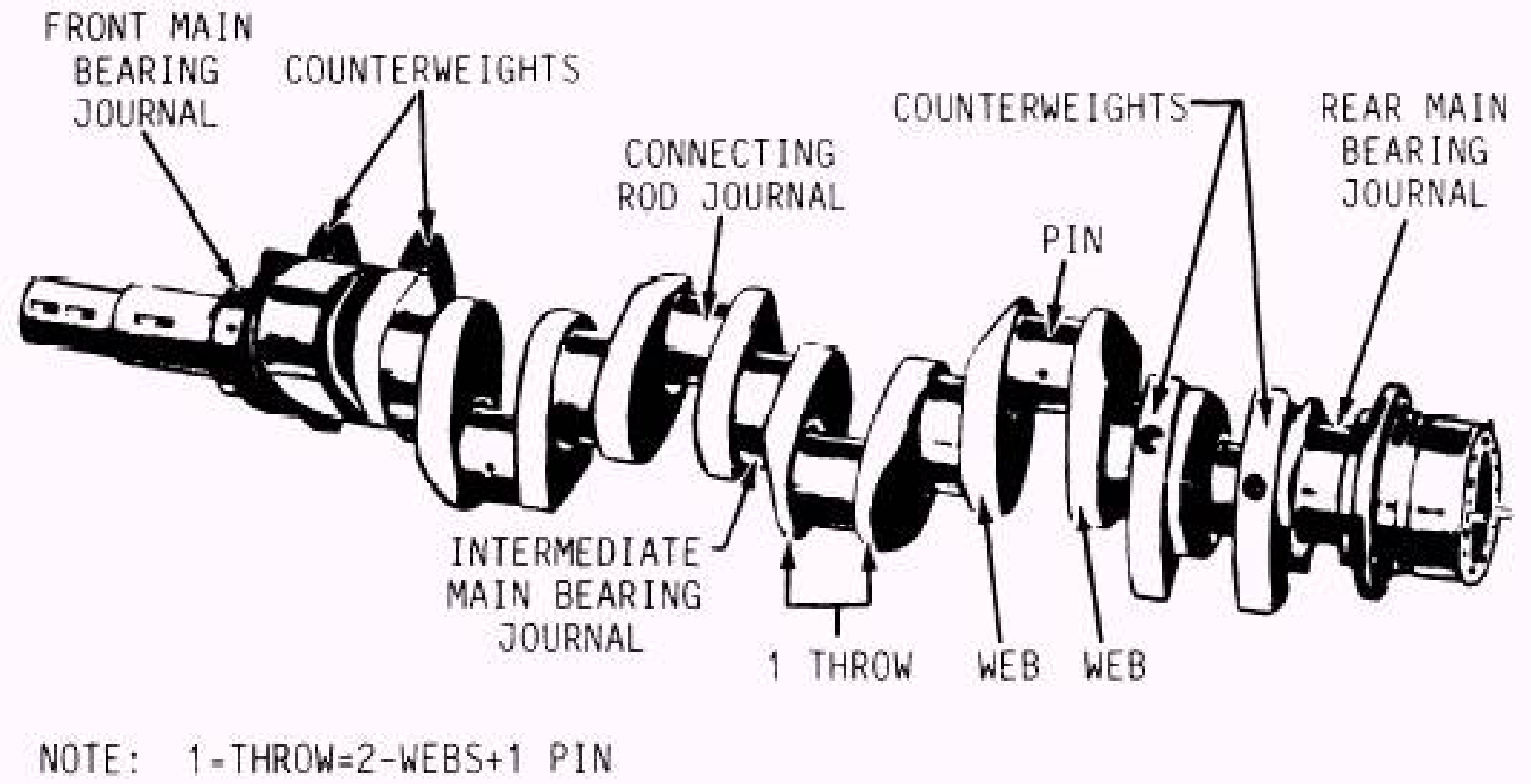
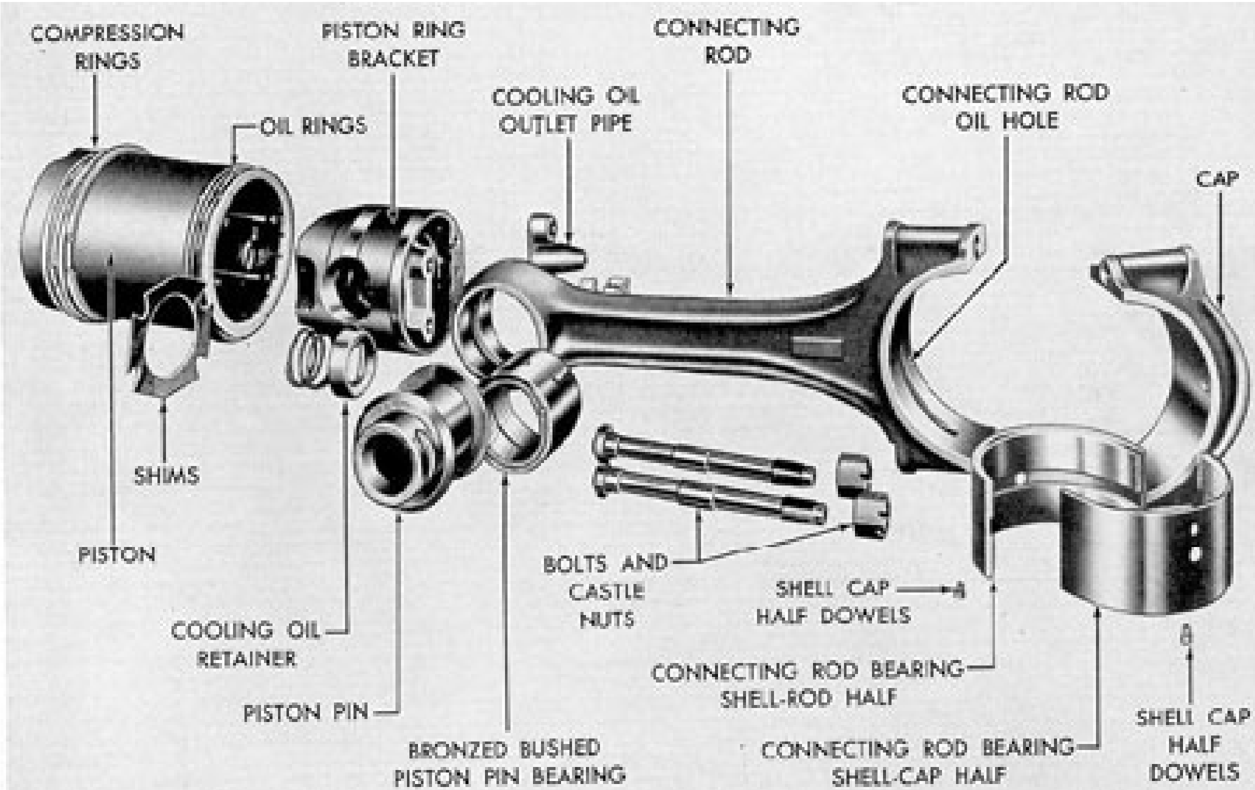


Fig. 4-7 Crankshaft and related parts for a six-cylinder engine. *(Ford Motor Company)*

Balancing and counterweighting of cranks



Balancing and counterweighting of cranks



Inertia forces

One can conceive of the oscillating inertial forces as two oppositely rotating vectors that are one-half their maximum, the vectors of the first order rotating at the same speed as the crankshaft, and those of the second order rotating at twice the crankshaft speed. The sum of the two perpendicular components of these vectors yields the momentary inertial force; the horizontal components cancel each other out (Fig. 6-33).

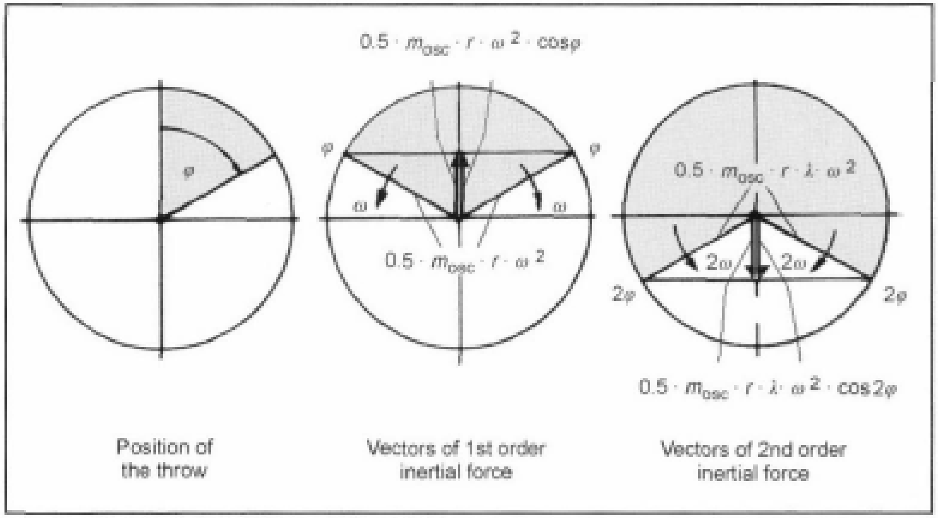


Fig. 6-33 Representation of the vectors of oscillating inertial forces.

Inertia forces

The characteristics of the oscillating inertial forces of the first order and of the second order add to form the resulting oscillating inertial force (Fig. 6-34).

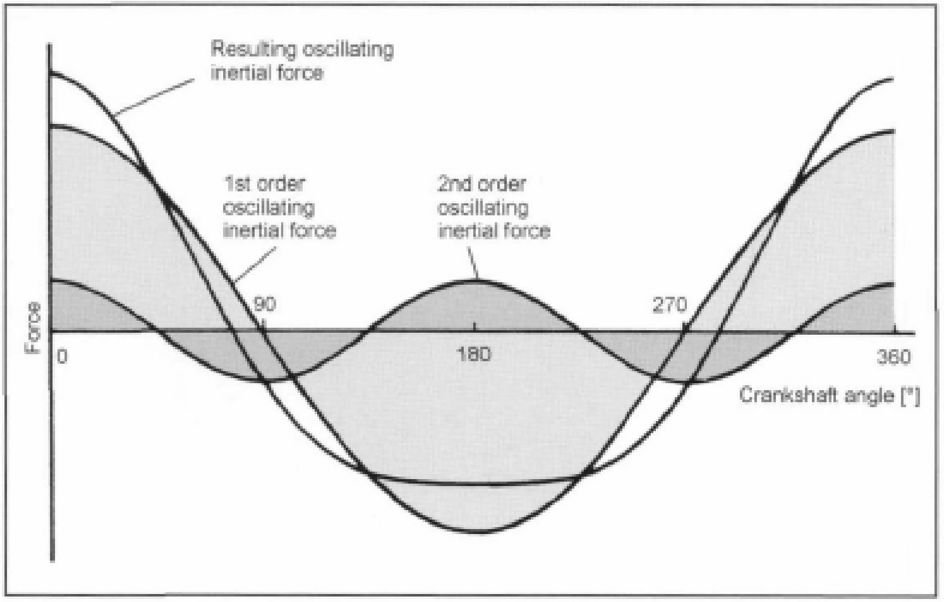


Fig. 6-34 Resulting oscillating inertial force.

Inertia forces

This overall inertial force for a cylinder results from the vectoral addition of the rotating and oscillating inertial forces of the first and second orders, and possibly the forces of a higher order (Fig. 6-35).

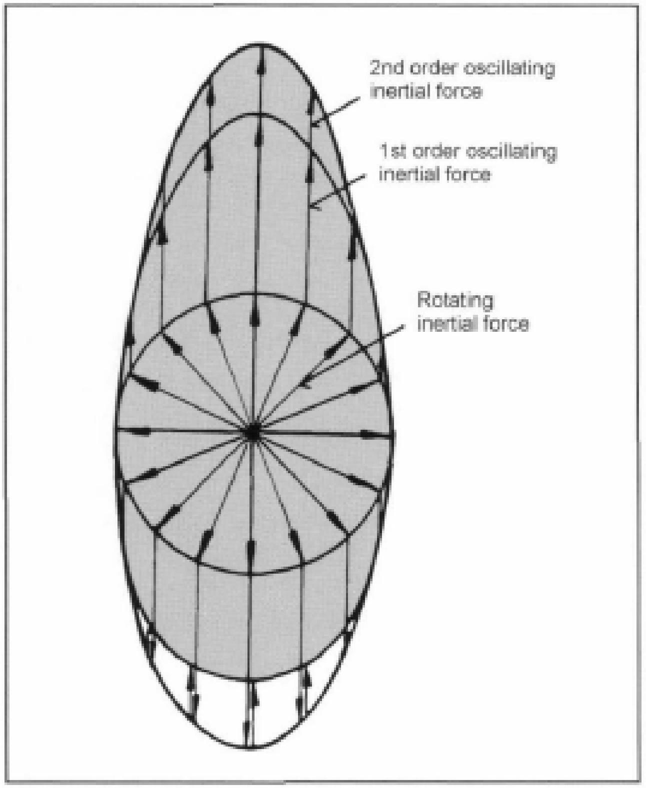


Fig. 6-35 Locus diagram of the resulting inertial force in a single-cylinder crank gear.