# PRELIMINARY DESIGN OF TWIN-CYLINDER ENGINES FOR HYBRID ELECTRIC VEHICLE APPLICATIONS

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## Abstract

Most of HEV are built using existing engines. These engines have been developed for conventional car and are not especially tailored for HEV. The objectives of this work are to establish general requirements placed on an ICE for a HEV application, to draw the main characteristics of the engine and to find the best configuration of twin-cylinder ICE to motorize an HEV. The twin-cylinder engine offer interesting perspectives in the field of HEV thanks to its small size, low weight and low cost.

## Introduction

Background

Facing environmental and energy challenges, automotive industry has to improve the fuel economy of new vehicles and reduce their polluting emissions. One short to mid-term solution is the hybrid electric vehicle (HEV) using in combination a conventional internal combustion engine (ICE) and an electric motorization. Many car manufacturers are developing their own hybrid systems to motorize new or existing models of vehicle. Currently, most of hybrid electric vehicles are built using existing engines. These engines have been developed for conventional car and are not especially designed for HEV, for which the engine has a specific duty, working differently than in a conventional vehicle[1].

Objectives

The objectives of this study are to establish general requirements placed on an ICE for a HEV application, to draw the main characteristics of different arrangements of twin-cylinder engine and to find the best configurations of twin-cylinder engine to motorize a HEV.

## Methods

ICE requirements for HEV applications

In order to design an ideal engine for hybrid applications, a list of general requirements is established[1]:

1. Work most of the time at full load, may it be in a narrow operating range as in series hybrid or a broader one for the other configurations.

2. Work in an intermittent manner thus requiring rapid and reliable starting and rapid rise in temperature in order to avoid emissions due to cold start.

3. Be as light as possible in order not to increase too much the overall weight of the vehicle.

4. Be as easy as possible to pack under the hood in order to ease the integration of the other components (electric machines, electronic and so on) and also to lower the hood in order to improve the aerodynamic profile of the vehicle.

5. Have the lowest fuel consumption and emission level as possible which places a requirement on the overall efficiency of the ICE.

6. Be as cheap as possible in order to fight against the increase of the total cost of the powertrain.

Several requirements (1, 2 and 5) are synonyms of engine downsizing i.e. replacing the engine by a smaller one of the same power. It means that the HEV has to be propelled by a small displacement engine with a high specific power. Other constraints (3 and 4) set conditions on the size and weight of the engine. The ideal ICE should be light and compact; in particular, the height of the engine has to be limited. The last requirement imposed to limit the engine cost, so it implies to prefer a configuration with few cylinders (two or three) than a configuration with a lot of small cylinders (four or more) that would be more expensive due to the higher number of parts[2].

Simulation steps

It appears that twin-cylinder engines are potentially interesting for hybrid applications but the difficulty with this kind of engines comes from the balancing of the inertia forces and moments inside the engine. A perfectly balanced engine is one in which the relative motion of the component parts do not generate an accumulation of forces that tends to make the engine shake and rock[3].

Several arrangements of twin-cylinder engine (boxer or in-line engine, in-phase or out-of-phase motion of the pistons), shown in table 1, are simulated here in order to emphasize the advantages and drawbacks of each one and to find the best configurations for hybrid vehicles[4].

In the in-line engine, cylinders are arranged consecutively in one row. This is by far the most common configuration used for engines mounted in small and medium-size cars. The boxer engine, also known as flat engine, has its two cylinder banks horizontal and opposed. These two types of engine represent two particular cases of vee-engine (vee-angle equal to zero degree for the in-line engine and vee-angle equal to 180 degrees for the boxer engine).

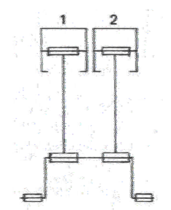
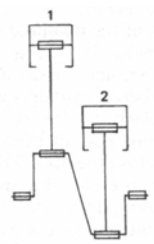
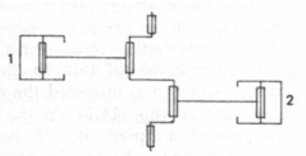
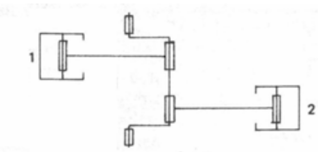
| | In-phase | Out-of-phase |
|---|---|---|
| In-line |  |  |
| Boxer |  |  |

**Table 1: Four different configurations of twin-cylinder engine simulated in this study**

The basic notions for the calculation of inertia forces and for the balancing of engines are well-known for a long time but the subject is regaining an interest, in particular, the more specific arrangements (boxer engine, vee-engine with unusual angle between cylinder rows). For instance, Grigoryev, Vasilyev and Dolgov[5, 6] developed a method to determine which arrangement (vee-angle, shape of the crankshaft…) of a given engine has the minimum mass and vibrations of the engine.

The determination of the best configuration of engine for a given application is done following several steps:

1. Computing the inertia forces and moments for each arrangement of engine. The calculations are based on simplified models developed from the motion equations of uncoupled rigid pistons (see the equations section).

2. Balancing the engines. This is made by using different methods alone or a combination of them. These methods are namely the optimization of the crankshaft counterweights, the addition of one or two first or second order balance shafts.

3. Introducing in the simulation the effect of the gas pressure inside the cylinder. This allows calculating the forces and moments due to the combustion, their effect on the balancing of the engine and the engine torque (instantaneous and average).

4. Comparing the different twin-cylinder configurations with a classical four-cylinder engine in order to determine which configuration offers an equivalent comfort in terms of vibration but remains simple enough from a design point of view.

Equations

The equations of the inertia forces for one cylinder (equations 1 and 2) are obtained by linearization of the piston motion equations[7]. The oscillating mass is the mass of the piston and the connecting rod (see figure 1) while the rotating mass is the mass of the connecting rod and the crankshaft. We made the assumption that one third of the connecting rod mass (small end) is part of the reciprocating mass and two thirds (big end) is part of the rotating mass (equations 4 and 5)[8].

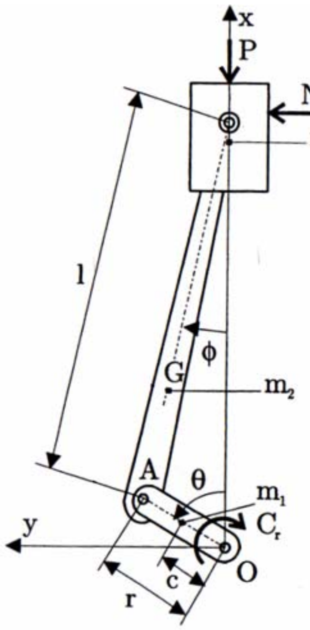



**Figure 1: Piston motion, inertia forces, oscillating and rotating masses[9]**

$$F_y = r \cdot \omega^2 \cdot m_r \cdot \sin\theta \quad \textbf{(1)}$$

$$F_x = r \cdot \omega^2 \cdot [m_r \cdot \cos\theta + m_o \cdot (\cos\theta + A_2 \cdot \cos 2\theta + A_4 \cdot \cos 4\theta + A_6 \cdot \cos 6\theta + ...)] \quad \textbf{(2)}$$

$$F_{res} = \sqrt{F_x^2 + F_y^2} \quad \textbf{(3)}$$

$m_r$ = rotating mass = $m_1$ + 2/3 $m_2$ **(4)**

$m_o$ = oscillating mass= $m_3$ + 1/3 $m_2$ **(5)**

$$A_2 = \lambda + \frac{1}{4} \cdot \lambda^3 + \frac{15}{128} \cdot \lambda^5 + ... \quad \textbf{(6)}$$

$$A_4 = -\frac{1}{4}\cdot\lambda^3 - \frac{3}{16}\cdot\lambda^5 + \ldots \qquad (7)$$

$$A_6 = \frac{9}{128}\lambda^5 + \ldots \qquad (8)$$

The force generated by a balance shaft in the two main directions (X and Y) is calculated with the equations 9 and 10.

$$F_x = r_{bs}\cdot m_{bs}\cdot\omega^2\cdot\cos\theta \qquad (9)$$

$$F_y = r_{bs}\cdot m_{bs}\cdot\omega^2\cdot\sin\theta \qquad (10)$$

$r_{bs}$*$m_{bs}$ = unbalance mass of the balance shaft

The forces and moments produced by each cylinder and each balance shaft are calculated separately and added together to evaluate the total inertia forces and moments of the engine.

The gas pressure in the cylinder has been determined by experiments on an existing similar diesel engine by Breuer Technical Development[10]. The gas pressure at full throttle is known for every position of the crankshaft angle between 0 and 720 degrees and for different engine speeds (see figure 2). The force created by this pressure (p) is calculated thanks to the equation (11).

$$F_{x,gp} = p\cdot\pi\cdot\left(\frac{b}{2}\right)^2 \qquad (11)$$

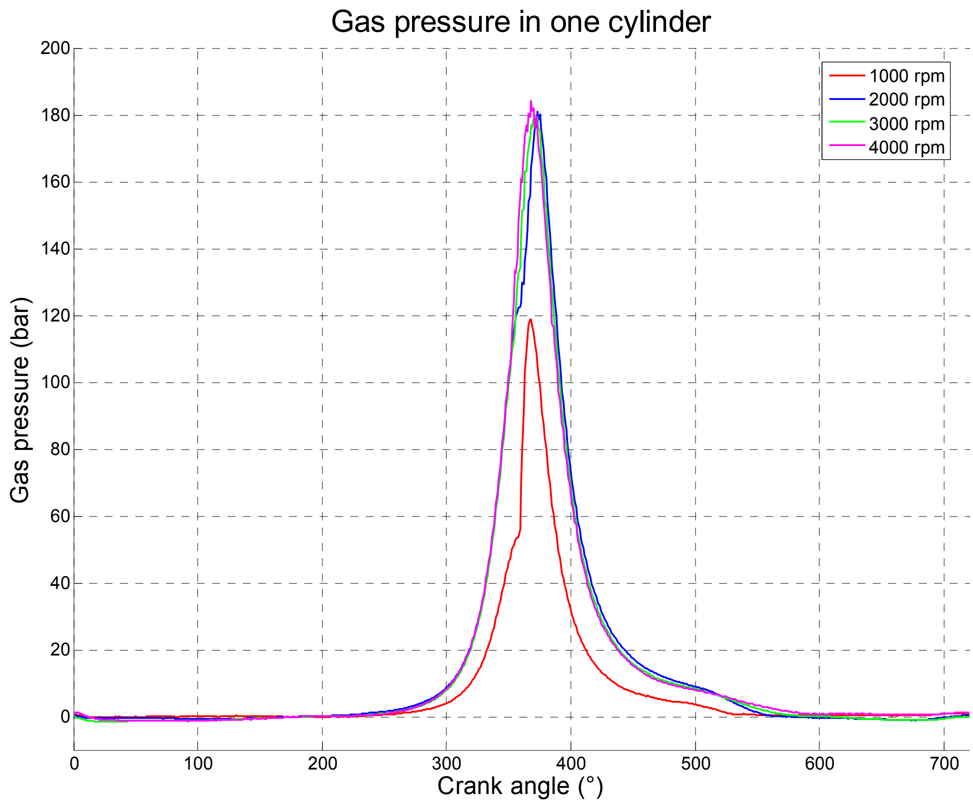



**Figure 2: Gas pressure inside the cylinder for different rotation speeds**

Used data

The values of the important engine data used in this work are given in the table 2.

| | |
|---|---|
| Cylinder bore b (mm) | 79,5 |
| Distance between piston centers a (mm) | 88 |
| Stroke 2r (mm) | 95,5 |
| Crank arm length r (mm) | 47,75 |
| Connecting rod l (mm) | 144 |
| Stroke to connecting rod ratio λ=r/l | 0,3316 |
| Rotation speed (rpm) | 4000 |
| Piston (pin included) weight (kg) | 0,7754 |
| Connecting rod weight (kg) | 0,6287 |
| Unbalance mass of one half crank (kg*m) | 0,02001 |

**Table 2: Reference data of the engine**

## Results and discussion

Calculation of the inertia forces and moments

First, the inertia forces generated inside one cylinder are considered, in the case of an engine with no balance shaft and a well-balanced crankshaft (figure 3). This means that the crankshaft and the related part of the connecting rod do not produce inertia forces. We focus mainly on the forces in the direction of the piston motion because there are no resulting forces in the Y direction (the crankshaft is well-balanced).

The figure 3 shows that the total force, in the X direction, is the sum of different order forces. The influence of the four order force is already very small and the higher order forces (>4) are negligible. The aspect of the total force curve depends mainly of the interaction between the first and second order forces. At the top dead center (0° or 360°), the first and second order forces have the same direction, so their values add to each other to reach the maximum value of the total force (10987 N). At the bottom dead center (180°), the second order force is positive whereas the first order force is negative, thus the second order force reduces the peak value of the first order force. Therefore, the total force curve is not symmetric.

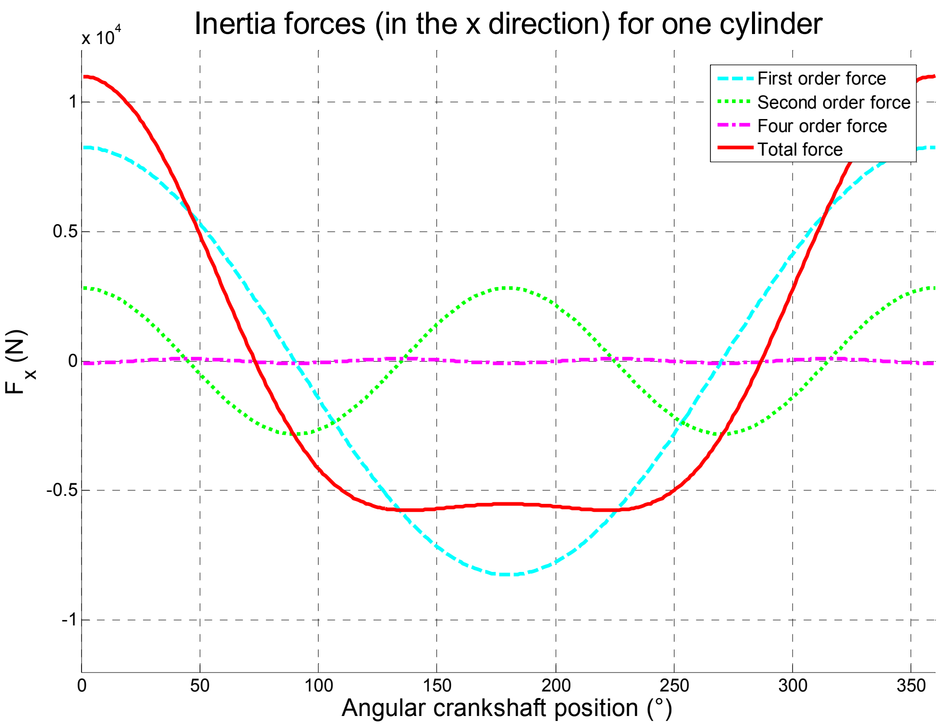



**Figure 3: Inertia forces (X direction) for one cylinder (rotation speed = 4000 rpm)**

The forces calculated in each cylinder are added up, taking care of the position and phase shift between the two cylinders, to obtain the inertia forces and moments for the different configurations of twin-cylinder engine.

<u>Balancing the engines</u>

Each configuration of twin-cylinder is subjected to two types of inertia loadings (first or second order forces or moments). In this section, we look for reducing these forces, first, by a better choice of crankshaft counterweights and then by adding balance shafts to the engine. We know that modifying the unbalance mass of the crank will increase the loads in the Y direction; so to avoid increasing the total loadings, we search the value of the unbalance mass of the crankshaft that minimizes the maximum resulting force or moment. This is carried out by making a parametric study of the unbalance mass with respect to the resulting force or moment (illustrated in figure 4 in the case of a single cylinder and a minimization of the resulting force). The red point indicates the reference value of the crankshaft. This means that an optimal choice of counterweight can nearly cut by two the maximum resulting force or moment.

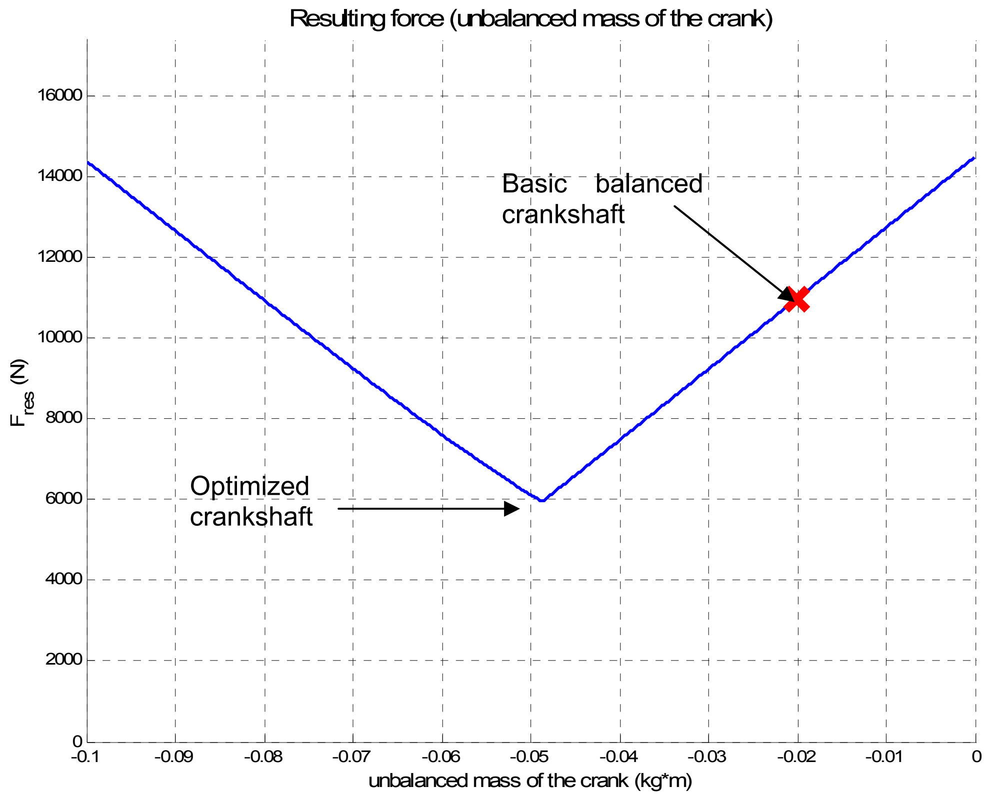



**Figure 4: Minimization of the resulting force for one cylinder by variation of the unbalance mass of the crank (parametric study)**

Afterwards, different associations of balance shafts (normal or double balance shafts, illustrated in table 3, first or second order balance shafts) are simulated and compared to find the best configuration.

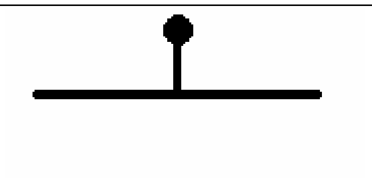
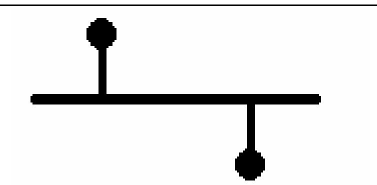
| Normal balance shafts | Double balance shafts |
|---|---|
| Produce inertia forces | Produce inertia moments |
|  |  |

**Table 3: Normal and double balance shafts**

Gas pressure effect

The gas pressure force acting on the piston is the force that produces the useful torque. It is transmitted from the piston to the crankshaft through the connecting rod and from the crankshaft to the engine block through the bearings. The gas pressure creates exactly the same force on the cylinder head but in the other direction. This force is transmitted from the cylinder head to the engine block by the cylinder head bolts. Therefore, there are no resulting forces because the two forces counteract each other. The forces due to the gas pressure are inner forces; they are only responsible for the torque and for tensile stresses in the engine[11]. They do not need to be balanced.

The instant torque produced by internal combustion engines is not constant because the pressure inside the cylinders changes during the cycle and the numbers of cylinder is finite. In particular with few cylinders, the variation of torque is important (see figure 5) and a large flywheel is required to smooth the speed variation of the crankshaft.

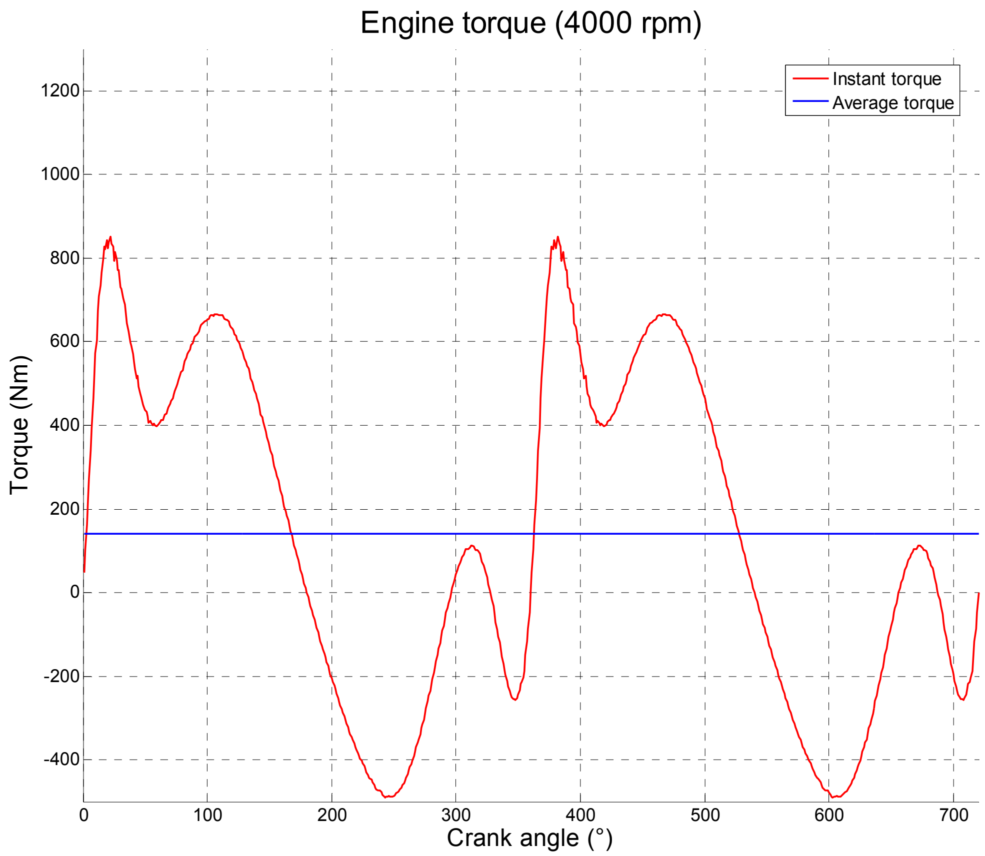



**Figure 5: Variation of torque in an in-phase (boxer or in-line) twin-cylinder engine (4000 rpm)**

Comparison with a four-cylinder engine

Figure 6 shows the maximum value of inertia forces and moments for different configurations of twin-cylinder engine. The two horizontal red lines show the maximum values of forces and moments of an equivalent (same power) four-cylinder engine in its basic configuration (without balance shaft) that serves as a reference.

The most interesting configuration of engine is the in-phase boxer engine because it does not need to be equipped with balance shafts to reach a low level of vibrations. Some modification of the crankshaft counterweights is sufficient to have a well-balanced engine. Another solution to reduce the loadings is to reduce the distance between bore centers which is also an advantage for the size of the engine. The most interesting solutions for the other configurations of engine are: the out-of-phase in-line engine with a modified crankshaft and one first order double balance shaft, the out-of-phase boxer engine with two first order balance shafts (it is also possible in this case to reduce the distance between bore centers) and the in-phase in-line engine with two first order balance shafts.

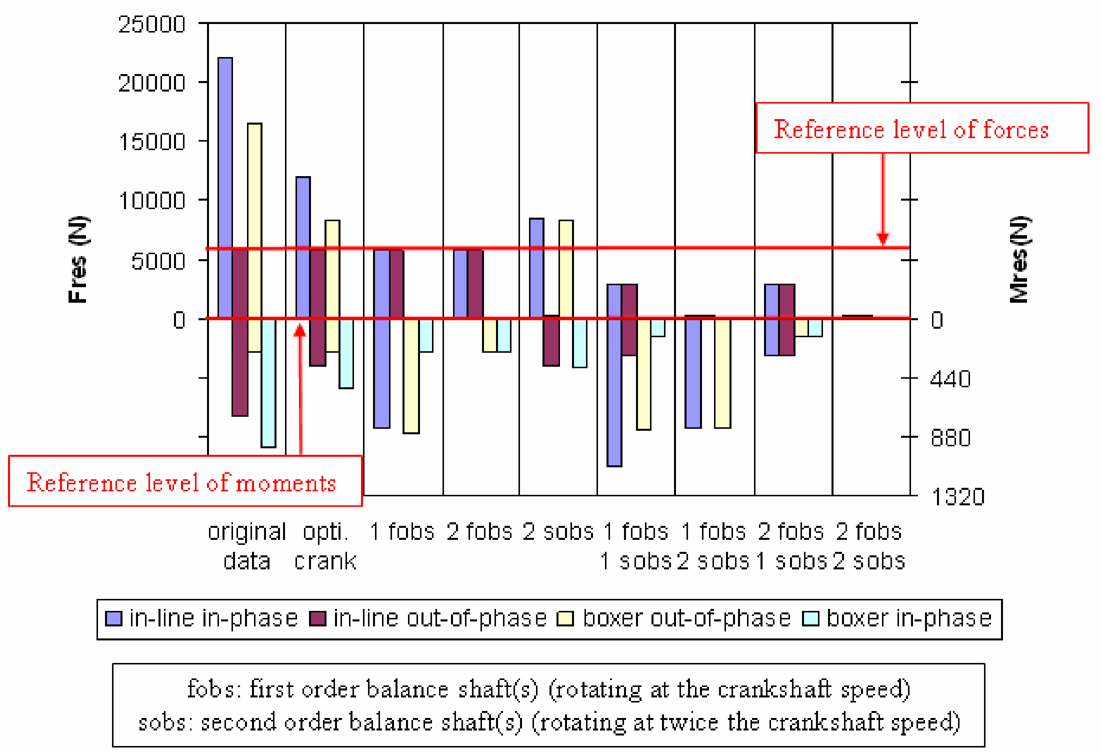



**Figure 6: Resulting forces and moments for different balancing systems of twin-cylinder engine**

## Conclusion

Each configuration of engine has its own characteristics in terms of inertia forces and moments (see table 4). For all these engines, we try different combinations of balance shafts and crankshaft in order to minimize the forces and moments.

| | First order forces | High order forces (>1) | First order moments | High order moments (>1) |
|---|---|---|---|---|
| In-line, in-phase | Yes | Yes | No | No |
| In-line, out-of-phase | No | Yes | Yes | No |
| Boxer, in-phase | No | No | Yes | Yes |
| Boxer, out-of-phase | Yes | No | No | Yes |

**Table 4: Forces and moments in the different configurations of twin-cylinder engine**

The twin-cylinder engines offer interesting perspectives in the field of HEV thanks to their small sizes, low weights and low costs. Among the different configurations, the boxer in-phase engine seems very promising because its inertia forces are naturally balanced and it has a very small height that make the packaging of the engine and the hybrid system under the hood easier.

## Acknowledgements

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