A COMPARATIVE KINEMATIC STUDY OF SOME VARIABLE COMPRESSION RATIO MECHANISMS

Stelian MIHALCEA[1], Adrian CLENCI[1], Ionel VIERU[1]
[1]University of Pitesti, Romania,

Abstract: *The variable compression ratio (VCR) offers the possibility to run the combustion process efficiency optimal under all load and speed conditions, especially in case of high boosted engines with small displacements.*
Out of the diversity of VCR engines, two different solutions are presented in this paper: PSA and Nissan solutions, which at a glance seem to be pretty close but, as it'll be demonstrated, present some important differences.
In this paper the accent is put only on the kinematics parameters supplied by a full calculus with the analytic method. Microsoft Excel is used for the calculus and for the bi-dimensional graphics. Also, the simulation of the mechanisms in 3D is performed with CATIA V5R17.

Keywords: Variable Compression Ratio, Kinematic Simulation

1. INTRODUCTION

There are many specialists that share the same opinion: Variable Compression Ratio is one of the most promising solutions to reduce gasoline engines' fuel consumption, while opening the way to some other strategies for the future (CAI/HCCI[1], aggressive boosting + downsizing etc).

In order to name some of the recent and famous achievements in this field, here it is, for instance, Saab which unveiled its VCR prototype engine in 1999 (1.6 L supercharged called SVC - Saab Variable Compression). The SVC engine delivers 168 kW of power and 305 Nm of torque, and provides more than 30% fuel consumption reduction when compared to a conventional naturally aspirated engine of equivalent power [3, 8]. A bit later, in 2000, FEV Motorentechnik also showed its own interpretation of VCR through an A6 Audi, powered by a 1.8 L VCR engine. Thanks to VCR, the FEV engine presents the same performance than that of a 3.0 L engine while reducing fuel consumption by 27% [12]. PSA and Nissan presented also their design of VCR mechanism in 2001 and 2002 [1, 2]. A pretty much complete comparative presentation of the VCR solutions is given in [7]. Here, one can see also the solution currently in development at University of Pitesti. It is about the Hara VCR mechanism [4, 5].

Out of this diversity of VCR engines, two different solutions are to be presented in this paper: PSA and Nissan solutions, which at a glance seem to be pretty close but, as it'll be demonstrated hereafter, present some important differences [6].

2. THE KINEMATICS ANALYSIS OF THE PSA AND NISSAN ENGINE MECHANISMS

Both mechanisms – PSA (fig. 2.1, a) and Nissan (fig. 2.2, a) – have between crankshaft and con rod a lever which allows to modify the distance between the bolt axis and the crankshaft axis, generating in this way a variable compression ratio.

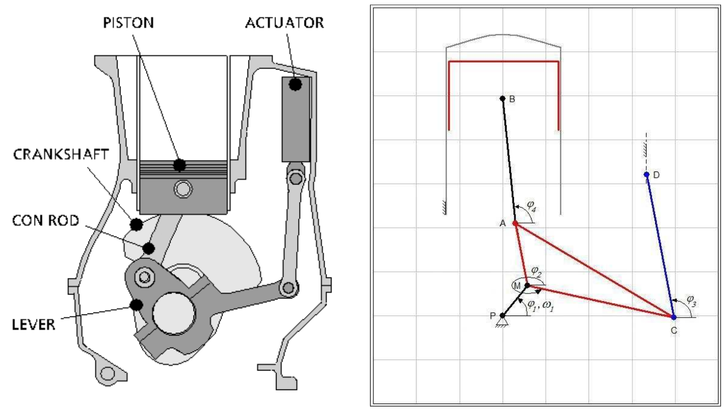
This lever is linked with a straight course actuator for the PSA mechanism and eccentric actuator for the Nissan mechanism.

In *Fig. 2.1, b* and *Fig. 2.2, b* are represented the kinematics schemes for the mechanisms in which the *AB* segment is the con rod, the *ACM* element represents the lever and the *CD* segment is the secondary con rod which links the actuator with the lever. The lever *ACM* is articulated in its three points, as follows: in point *A* with the bigger head of the con rod, in the point *M* with the crankshaft

[1] ***C****ontrolled* ***A****uto* ***I****gnition/****H****omogeneous* ***C****harge* ***C****ontrolled* ***I****gnition*

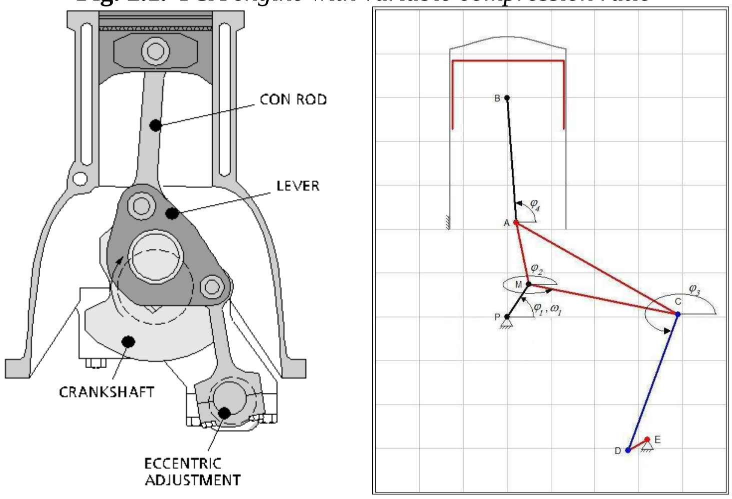
and in the point *C* with the secondary con rod *CD*. The points *P* (the main axis of the crankshaft) and *D* (the link axis with the actuator) are considered stationary during the spin of the crankshaft. However, the displacement of the point *D* realizes the variation of the compression ratio, independently with the crankshaft rotation.

The con rod's larger head (point *A*) kinematics depends by the position of the actuator (point *D*). On a complete mechanism cycle, the point *A* describes an unregulated trajectory (in fact is a six degree equation), relative to the classic engine mechanism in which the point *A* describes a circle. The course of the piston varies depending by the position of the actuator and the result of this fact is that those mechanisms realize a variable engine capacity. On the other part, when the piston is at TDC, the position of the crankshaft varies.



a. *Structural scheme* ***b.*** *Kinetic scheme*

Fig. 2.1. *PSA engine with variable compression ratio*



a. *Structural scheme* ***b.*** *Kinetic scheme*

Fig. 2.2. *Nissan engine with variable compression ratio*

To analyze the kinematics of the two mechanisms, some specifications require:

- The mechanism has a unique movement;
- The dimension of all elements are known, including the coordinate of the points $P(X_P;Y_P)$ and $D(X_D;Y_D)$; for the Nissan mechanism is known the position of the point $E(X_E Y_E)$, point D have a spin movement around the point E;
- Considering that the motor element is the crankshaft (PE segment) which rotates around the fixed point P, the angular speed is ω_1;
- The study of mechanism movement is made without considering the forces and torques which produce the movement;
- The kinematics analysis is made during an entire cycle, i.e. the period after which the kinematics parameters will repeat themselves.

The kinematics calculus is the same for the both engines.

The positions of the points, which define the mechanism configuration depend by α ≡ the angle between crankshaft and Oy axis.

Settle of point M ($X_M;Y_M$) coordinates.

$$\begin{cases} X_M = X_P + PM\cos\alpha \\ Y_M = Y_P + PM\sin\alpha \end{cases} \tag{2.1}$$

It's easy to observe that the M trajectory it is a circle with the center point in $P(X_P;Y_P)$ and the radius r = PM. In the diagram, this circle it is obtained for α = 0÷360°.

Settle of point C($X_C;Y_C$) coordinates.

$$\begin{cases} X_C = X_D - CD\cos\left(\arccos\dfrac{DM^2 + CD^2 - CM^2}{2\cdot DM\cdot CD} + arctg\,\dfrac{Y_M - Y_D}{X_M - X_D}\right) \\ Y_C = Y_D - CD\sin\left(\arccos\dfrac{DM^2 + CD^2 - CM^2}{2\cdot DM\cdot CD} + arctg\,\dfrac{Y_M - Y_D}{X_M - X_D}\right) \end{cases} \quad (2.2)$$

where $DM = \sqrt{(X_D - X_M)^2 + (Y_D - Y_M)^2}$.

Settle of point $A(X_A;Y_A)$ coordinates.

$$\begin{cases} X_A = X_C - AC\cos\left(\arccos\dfrac{AC^2 + CM^2 - AM^2}{2\cdot AC\cdot CM} - arctg\,\dfrac{Y_M - Y_C}{X_M - X_C}\right) \\ Y_A = Y_C - AC\sin\left(\arccos\dfrac{AC^2 + CM^2 - AM^2}{2\cdot AC\cdot CM} - arctg\,\dfrac{Y_M - Y_C}{X_M - X_C}\right) \end{cases} \quad (2.3)$$

Settle of point $B(X_B;Y_B)$ coordinates.

Considering that the cylinder axis is the same with the Oy axis, it result the point B abscissa is the same with point P abscissa. Accordingly with that:

$$\begin{cases} X_B = X_P \\ Y_B = Y_A + \sqrt{AB^2 - (X_A - X_B)^2} \end{cases} \quad (2.4)$$

To establish the linear and angular speed and acceleration of the elements the calculus will contain the derivation of the point's coordinates. The angles which define the element's position are φ_i, i = 1,…,4 (*Fig. 2.1, b* and *Fig. 2.2, b*):

$$\varphi_1 = 0 \div 360^\circ,\ \varphi_2 = \arcsin\frac{Y_C - Y_M}{MC},\ \varphi_3 = \arccos\frac{X_D - X_C}{CD},\ \varphi_4 = \arccos\frac{Y_B - Y_A}{AB} \quad (2.5)$$

3. NUMERICAL ANALYSIS OF THE MECHANISMS

After the entire kinematics calculus was made, a numerical analysis of the mechanisms is required. To obtain that, the dimensions of the mechanisms presented in table 1 are used:

Table 1. *Dimensional characteristics of the PSA and Nissan engine*

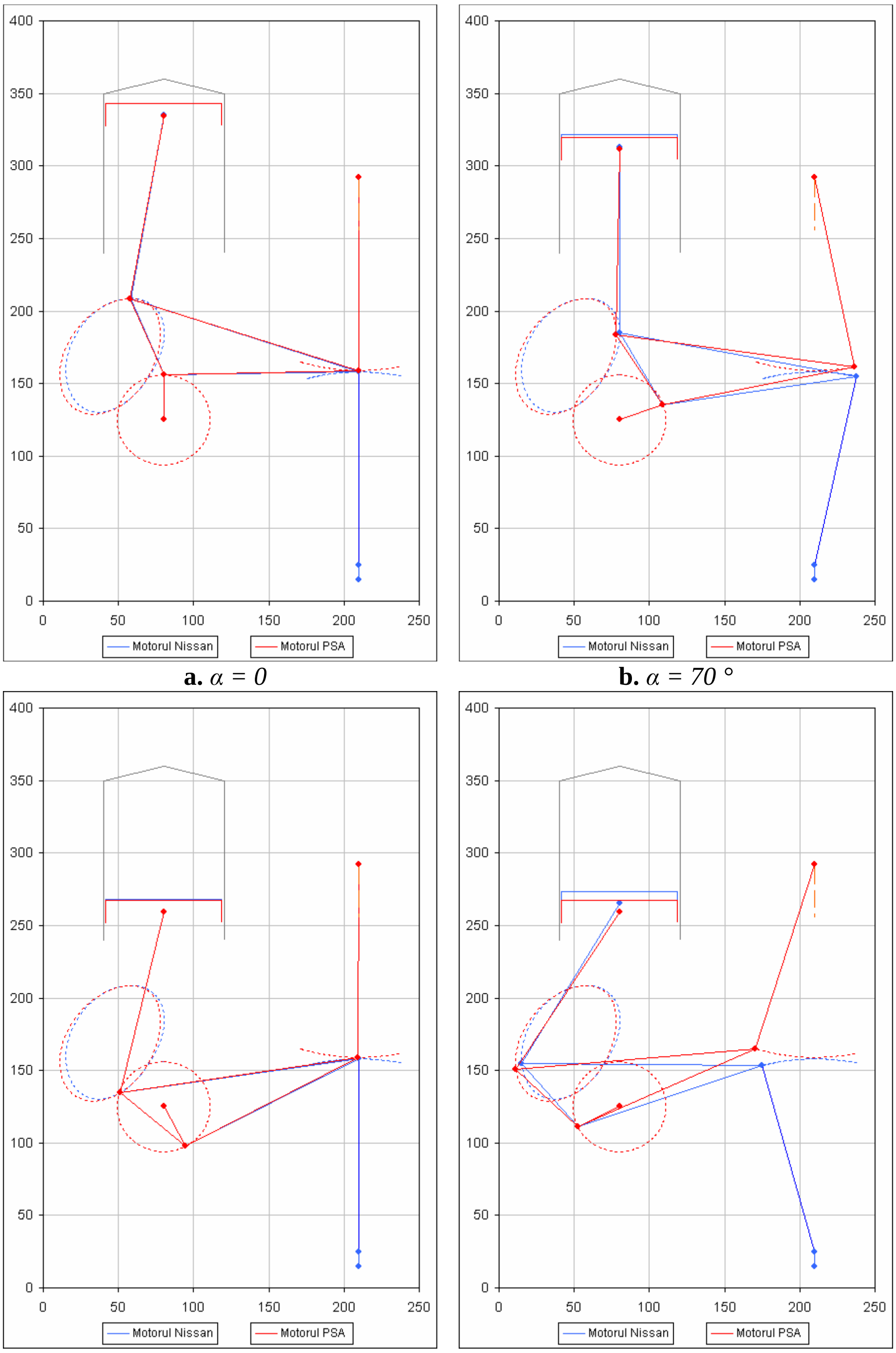
Parameter	***Engine PSA***	***Engine Nissan***
Crank radius, PM [mm]	31	31
Cylinder diameter, D [mm]	79,5	79,5
Con rod's length, AB [mm]	128	128
AC segment [mm]	160	160
MC segment [mm]	130	130
AM segment [mm]	57	57
Secondary con rod's length, CD [mm]	133	133
Eccentric's value, e [mm]	–	10, with 180° rotation
Linear displacement, d [mm]	21	–
Compression ratio, ε_v	9.7 ÷ 14.0	9.7 ÷ 14.0
$P(x_P;y_P)$	(80;75)	(70;125)
$D(x_D;y_D)$ / $E(x_E;y_E)$	(210; 221÷242)	(200;15)

The aim that the dimensions were equally chosen is to realize a right analysis between the two mechanisms. Also, the compression ratio is the same for both engines. After this analysis the evolution of the compression ratio, the piston stroke and the engine capacity for these engines may be established.

a) Positions analysis

Both engines have compression ratios between 9.7 and 14.0. In *Fig. 3.1* are designed a few overlapped positions in the case ε_v = 9.7.

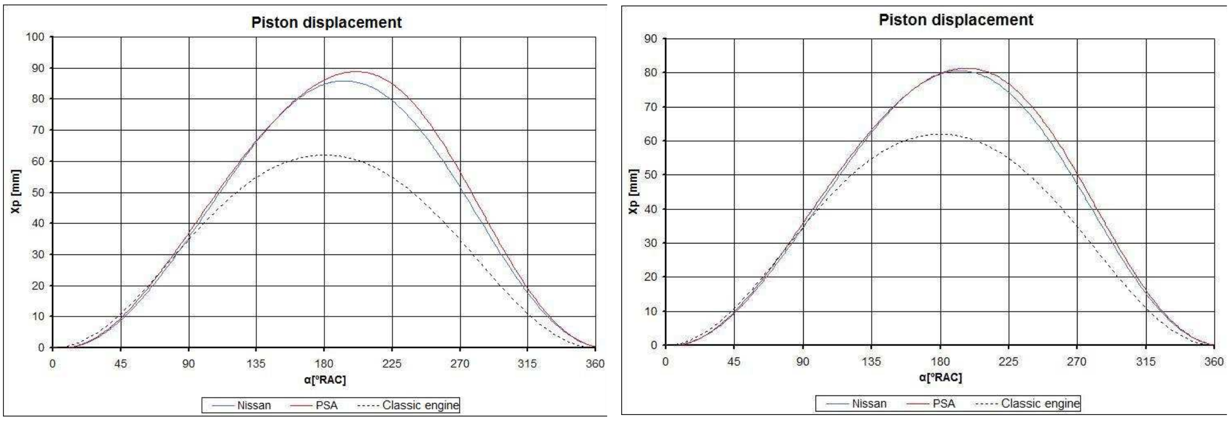
From *Fig. 3.1*, for the same position of the crank PM, it can be easily seen that the displacement of the two pistons (PSA in red color and Nissan in blue color) it is not identically. The reason for that is happening is the different trajectories of the point C, concave for PSA and convex for Nissan mechanism. In *Fig. 3.2* is designed the evolution of the displacement for the two pistons during a kinematics cycle, and in *Fig. 3.3* it is designed the difference between PSA and Nissan pistons displacements, considering that the engines works with extreme compression ratios, i.e. 9.7 and 14.0.



a. *α = 0* **b.** *α = 70 °*

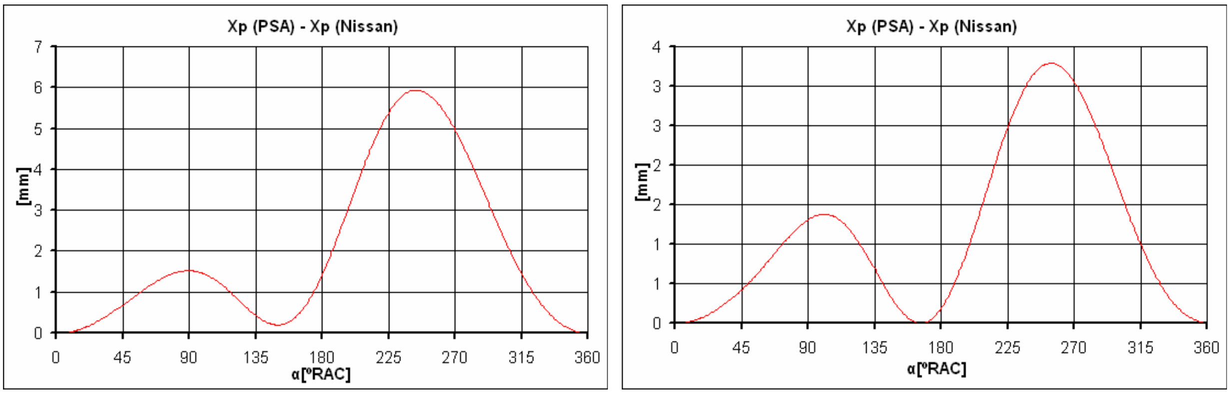
c. *α = 152 °* **d.** *α = 244 °*

Fig. 3.1. *Superposition of the two mechanisms, for ε_v = 9.7*



ε_v = 9.7 ε_v = 14.0

Fig. 3.2. *The variation of piston displacement*



ε_v = 9.7 ε_v = 14.0

Fig. 3.3. *The variation of difference between pistons displacements*

The maximum postponement is obtained for *α = 244 °* and the value is 5.937 mm (*Fig. 3.1, d*).

At the PSA mechanism, the actuator pushes downward the secondary con rod along 21 mm, increasing the volumetric compression ratio from 9.7 to 14.0 (*Fig. 3.4*). In the Nissan case, the eccentric realize 180° rotation pushing downward the secondary con rod along a distance equal to double of eccentricity, i.e. 20 mm (e=10 mm), resulting ε_v=14.0 (*Fig. 2.4*).

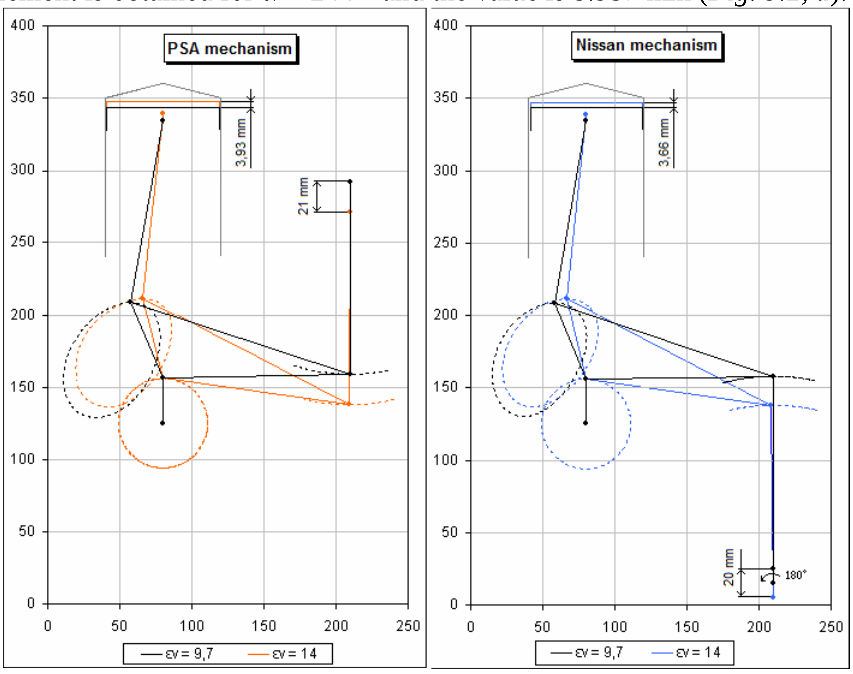


Fig. 3.4.

Comparing the two engines working, result the difference between the pistons displacement is smaller in ε_v = 14.0 than ε_v = 9.7, the effective maxim value being 3.287 mm, for α = 254°. Also, once increased the compression ratio for both engines, the courses of the pistons decrease, (see *Table 2*). The position of TDC and BDC varies with compression ratio, without maintaining a constant distance between them. According with that, the pistons courses varies with compression ratio.

Table 2. *Influence of compression ratio variation on piston course.*

	Nissan engine	**PSA engine**
ε_V = 9.7	S = 85.854 mm	S = 88.845 mm
ε_V = 14.0	S = 80.606 mm	S = 81.376 mm

According with (3.1), in this case results the cylinder capacity varies (see *Table 3*).

$$V_S = \frac{\pi \cdot D^2}{4} S \tag{3.1}$$

Table 3. *Influence of compression ratio variation on cylinder capacity.*

	Nissan engine	**PSA engine**
ε_V = 9.7	V_S = 85.854 cm^3	V_S = 88.845 cm^3
ε_V = 14.0	V_S = 80.606 cm^3	V_S = 81.376 cm^3

Considering that the engine has 4 cylinders, results the engine capacity (see *Table 4*)

Table 4. *The influence of compression ratio variation on engine capacity.*

	Nissan engine	**PSA engine**
ε_V = 9.7	V_t = 1704.687 cm^3	V_t = 1764.083 cm^3
ε_V = 14.0	V_t = 1600.475 cm^3	V_t = 1615.775 cm^3

b) Definition to the burn chamber capacity.

The compression ratio can be calculated with:

$$\varepsilon_V = \frac{V_{\max}}{V_{\min}} = 1 + \frac{V_S}{V_k} = 1 + \frac{\frac{\pi D^2}{4} S}{V_1 + V_2} \tag{3.2}$$

Like any internal combustion engine, the VCR engine's burning chamber is composed by two sub-volumes:

1. The volume of the burning chamber from cylinder head, V_1, which is constant both on classic engines and PSA and Nissan engines;
2. The volume of the burning chamber from cylinder, V_2, which is variable for VCR engines in distinction with classic engines where this volume is constant too.

From relation 3.2 result that the compression ratio depends simultaneous by two variables: the variable piston course S and the variable volume V_2.

The variation of burning chamber from the cylinder, for PSA and Nissan engines, it is explained by variation of TDC with compression ratio.

Another determination is that the position TDC and BDC are variables with compression ratio (*Fig. 3.5*).

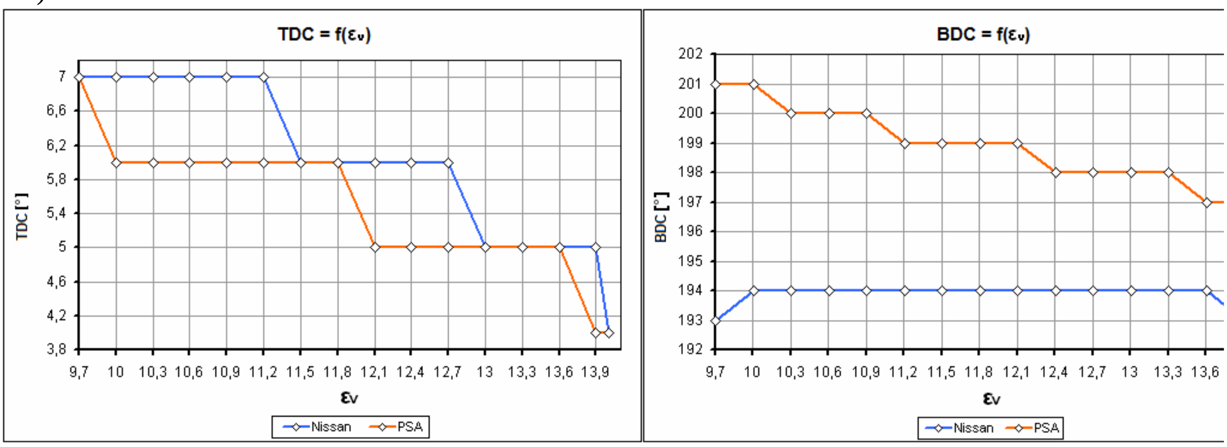


Fig. 3.5. *The variation of TDC and BDC with compression ratio for Nissan and PSA engines*

Analyzing the two evolutions results:

- Both TDC and BDC positions are strongly moved comparing with classic engine (in which the TDC position is 0°, and the BDC position is 180°);
- For PSA engine it can be notice a large variation of BDC position, unlike the Nissan engine where this position is almost constant;

It is well known the fact that in the dead center positions, the piston speed is zero. This fact can be verified by analyzing the piston speed graphics. Also, the postponement of the dead center positions of piston results from *Fig. 3.2* for ε_v = 9.7 and ε_v = 14.0.

4. ENGINES KINEMATICS STUDY USING CATIA V5 SOFTWARE

The component parts of the engines were schematic designed with *Part Design* module abiding by base dimensions. The assembly was made in *Assembly Design* module and the graphics were made in *DMU Kinematics* module (*Fig. 4.1* and *Fig. 4.2*)

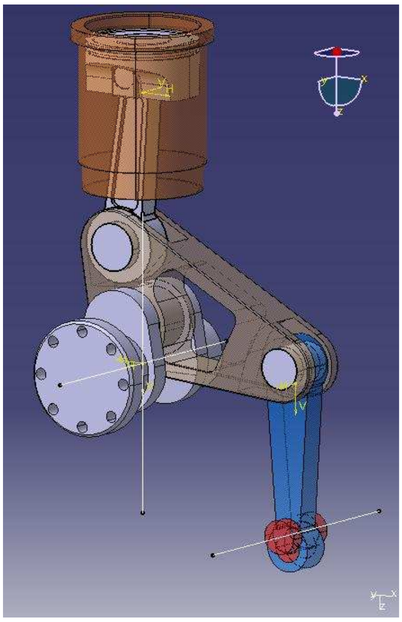


Fig. 4.1. *Nissan engine – 3D view*

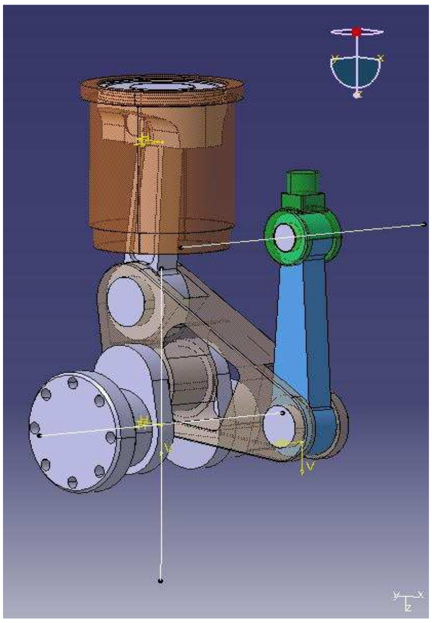


Fig. 4.2. *PSA engine – 3D view*

In *Fig. 4.3…4.8* are presented the diagrams for kinematics parameters using CATIA V5 software. There are no differences comparing to the results obtained with the analytical method.

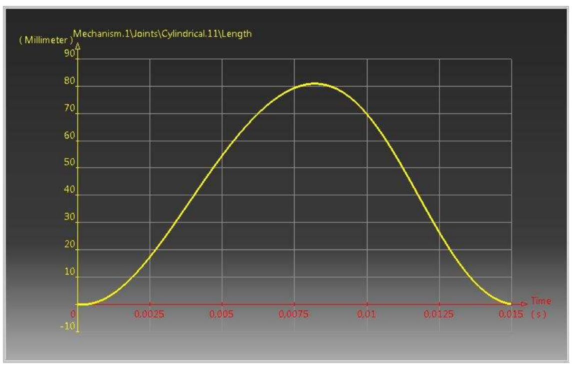


Fig. 4.3. *PSA piston displacement evolution, ε_v=14, n = 4000 rpm*

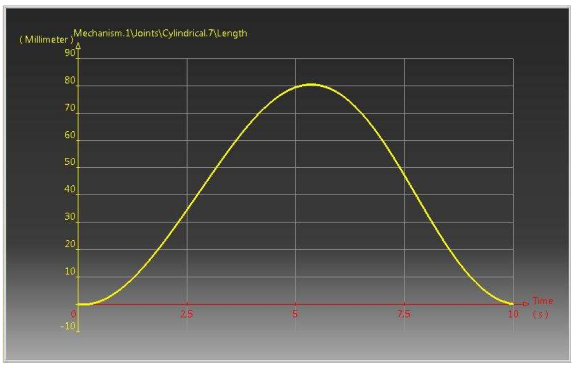


Fig. 4.4. *Nissan piston displacement evolution, ε_v=14, n = 4000 rpm*

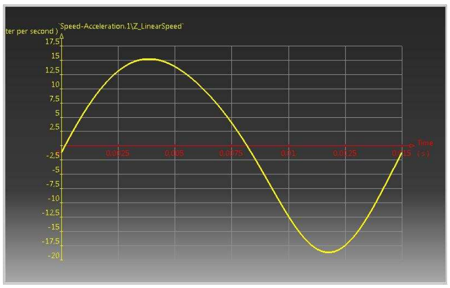


Fig. 4.5. *PSA piston speed evolution,*
ε_v=14, n = 4000 rpm

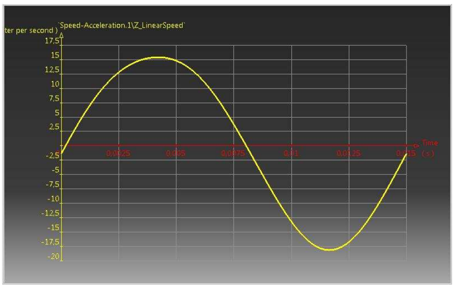


Fig. 4.6. *Nissan piston speed evolution,*
ε_v=14, n = 4000 rpm

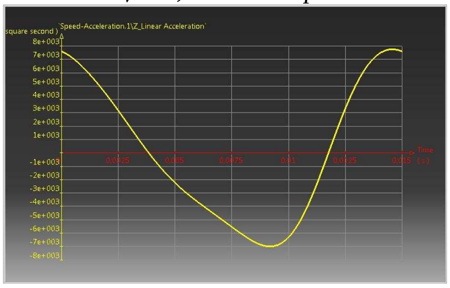


Fig. 4.7. *PSA piston acceleration evolution,*
ε_v=14, n = 4000 rot/min

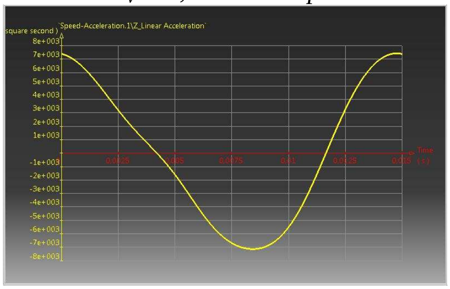


Fig. 4.8. *Nissan piston acceleration evolution,*
ε_v=14, n = 4000 rot/min

5. CONCLUSIONS

The position of TDC and BDC varies with compression ratio, without maintaining a constant distance between them. According with that, the pistons strokes varies with compression ratio. It also results a variable volume for the burning room chamber.

After the 3D simulation with CATIA, the results obtained with analytic calculus were confirmed, and so, the analytic method was verified and certified.

Although in this paper it is not broach the dynamics of the two mechanisms, we can appreciate that the PSA and Nissan engines with variable compression ratio represents inefficient solution because it use additionally parts. As a result, these new elements involve additional moving mass, inertia forces and, of course, new joints and inevitable friction. Also, the movement of these new parts is not uniform and therefore the balance of these engines is difficult to achieve.

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