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Study of an Oscillating Sliding Mechanism

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Abstract: A mechanism with an oscillating slider is a mechanism quite often used in the technique. It has some advantages over other similar mechanisms being able to work with accelerations and reduced forces under the same conditions with another similar mechanism. For example, an internal combustion engine could be built more efficiently with this mechanism, resulting in a motor that would successfully replace Otto's classic one. In the paper, the kinematics, forces and their distribution will be presented within such a mechanism.

Keywords: Sliding Mechanism, Robots, Manipulators, Automation, Engines, Mechanical Transmissions, Kinematics, Forces, Dynamics, Dynamic Kinematics, Dynamic Forces

Introduction

Figure 1 shows an oscillating sliding mechanism.

A mechanism with an oscillating slider is a mechanism quite often used in the technique. It has some advantages over other similar mechanisms being able to work with accelerations and reduced forces under the same conditions with another similar mechanism. For example, an internal combustion engine could be built more efficiently with this mechanism, resulting in a motor that would successfully replace Otto's classic one.

Fig. 1: An oscillating sliding mechanism

In the paper, the kinematics, forces and their distribution will be presented within such a mechanism (Frăţilă *et al*., 2011; Pelecudi, 1967; Antonescu, 2000; Amoresano *et al*., 2013; Comănescu *et al*., 2010; Aversa *et al*., 2016a; 2016b; 2016c; 2016d; 2017a; 2017b; 2017c; 2017d; 2017e; Mirsayar *et al*., 2017; Cao *et al*., 2013; Dong *et al*., 2013; De Melo *et al*., 2012; Garcia *et al*., 2007; Garcia-Murillo *et al*., 2013; He *et al*., 2013; Lee, 2013; Lin *et al*., 2013; Liu *et al*., 2013; Padula and Perdereau, 2013; Perumaal and Jawahar, 2013; Petrescu and Petrescu, 1995a; 1995b; 1997a; 1997b; 1997c; 2000a; 2000b; 2002a; 2002b; 2003; 2005a; 2005b; 2005c; 2005d; 2005e; 2016a; 2016b; 2016c; 2016d; 2016e; 2013; 2012a; 2012b; 2011; Petrescu *et al*., 2009; 2016a; 2016b; 2016c; 2016d; 2016e; 2017a; 2017b; 2017c; 2017d; 2017e; 2017f; 2017g; 2017h; 2017i; 2017j; 2017k; 2017l; 2017m; 2017n; 2017o; 2017p; 2017q; 2017r; 2017s; 2017t; 2017u; 2017v; 2017w; 2017x; 2017y; 2017z; 2017aa; 2017ab; 2017ac; 2017ad; 2017ae; Petrescu and Calautit, 2016a; 2016b; Reddy *et al*., 2012; Tabaković *et al*., 2013; Tang *et al*., 2013; Tong *et al*., 2013; Wang *et al*., 2013; Wen *et al*., 2012; Antonescu and Petrescu, 1985; 1989; Antonescu *et al*., 1985a; 1985b; 1986; 1987; 1988; 1994; 1997; 2000a; 2000b; 2001; List the first flights, From Wikipedia; Chen and Patton, 1999; Fernandez *et al*., 2005; Fonod *et al*., 2015; Lu *et al*., 2015; 2016; Murray *et al*., 2010; Palumbo *et al*., 2012; Patre and Joshi, 2011; Sevil and Dogan, 2015; Sun and Joshi, 2009; Crickmore, 1997;

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Goodall, 2003; Graham, 2002; Jenkins, 2001; Landis and Dennis, 2005; Clément, Wikipedia; Cayley, Wikipedia; Coandă-1910, Wikipedia; Gunston, 2010; Laming, 2000; Norris, 2010; Goddard, 1916; Kaufman, 1959; Oberth, 1955; Cataldo, 2006; Gruener, 2006; Sherson *et al*., 2006; Williams, 1995; Venkataraman, 1992; Oppenheimer and Volkoff, 1939; Michell, 1784; Droste, 1915; Finkelstein, 1958; Gorder, 2015; Hewish, 1970).

Materials and Methods

After removing the mechanism crank, the structural group dyad RTR is isolated, a known and often used a mechatronic module (Fig. 2). The coupling parameters *C* and *B* are known and the kinematic parameters *s* and φ_3 with their derivatives must be determined, which is done by means of the calculation relations of the system (1):

$$
\begin{cases}
s^2 = \left(x_B - x_C\right)^2 + \left(y_B - y_C\right)^2 \Rightarrow s = \sqrt{\left(x_B - x_C\right)^2 + \left(y_B - y_C\right)^2} \\
\begin{cases} x_B = x_C + s \cdot \cos\varphi_3 \\ y_B = y_C + s \cdot \sin\varphi_3 \end{cases}
\begin{cases} \cos\varphi_3 = \dfrac{x_B - x_C}{s} \\ \sin\varphi_3 = \dfrac{y_B - y_C}{s} \end{cases}
\Rightarrow \varphi_3 = semn\left(\sin\varphi_3\right) \cdot \cos^{-1}\left(\cos\varphi_3\right) \\
2 \cdot s \cdot \dot{s} = 2 \cdot \left(x_B - x_C\right) \cdot \left(\dot{x}_B - \dot{x}_C\right) + 2 \cdot \left(y_B - y_C\right) \cdot \left(\dot{y}_B - \dot{y}_C\right) \Rightarrow \\
\dot{s} = \dfrac{\left(x_B - x_C\right) \cdot \left(\dot{x}_B - \dot{x}_C\right) + \left(y_B - y_C\right) \cdot \left(\dot{y}_B - \dot{y}_C\right)}{s} \\
\ddot{s} = \dfrac{\left(\dot{x}_B - \dot{x}_C\right)^2 \cdot \left(\dot{y}_B - \dot{y}_C\right)^2 - \dot{s}^2}{s} + \dfrac{\left(x_B - x_C\right) \cdot \left(\ddot{x}_B - \ddot{x}_C\right) + \left(y_B - y_C\right) \cdot \left(\ddot{y}_B - \ddot{y}_C\right)}{s} \\
\begin{cases} \dot{x}_B - \dot{x}_C = \dot{s} \cdot \cos\varphi_3 - s \cdot \sin\varphi_3 \cdot \dot{\varphi}_3 \mid \cdot \left(-\sin\varphi_3\right) \\ \dot{y}_B - \dot{y}_C = \dot{s} \cdot \sin\varphi_3 + s \cdot \cos\varphi_3 \cdot \dot{\varphi}_3 \mid \cdot \left(\cos\varphi_3\right) \end{cases}
\Rightarrow \dot{\varphi}_3 = \dfrac{\left(\dot{y}_B - \dot{y}_C\right)\cos\varphi_3 - \left(\dot{x}_B - \dot{x}_C\right) \cdot \sin\varphi_3}{s} \\
\begin{cases} \ddot{x}_B - \ddot{x}_C = \ddot{s} \cdot \cos\varphi_3 - 2 \cdot \dot{s} \cdot \sin\varphi_3 \cdot \dot{\varphi}_3 \\ -s \cdot \cos\varphi_3 \cdot \dot{\varphi}_3^2 - s \cdot \sin\varphi_3 \cdot \ddot{\varphi}_3 \mid \cdot \left(-\sin\varphi_3\right) \\ \ddot{y}_B - \ddot{y}_C = \ddot{s} \cdot \sin\varphi_3 - 2 \cdot \dot{s} \cdot \cos\varphi_3 \cdot \dot{\varphi}_3 \\ -s \cdot \sin\varphi_3 \cdot \dot{\varphi}_3^2 + s \cdot \cos\varphi_3 \cdot \ddot{\varphi}_3 \mid \cdot \left(\cos\varphi_3\right) \end{cases}
\Rightarrow \ddot{\varphi}_3 = \dfrac{\left(\ddot{y}_B - \ddot{y}_C\right) \cdot \cos\varphi_3 - \left(\ddot{x}_B - \ddot{x}_C\right) \cdot \sin\varphi_3 - 2 \cdot \dot{s} \cdot \dot{\varphi}_3}{s}
\end{cases}
\tag{1}
$$

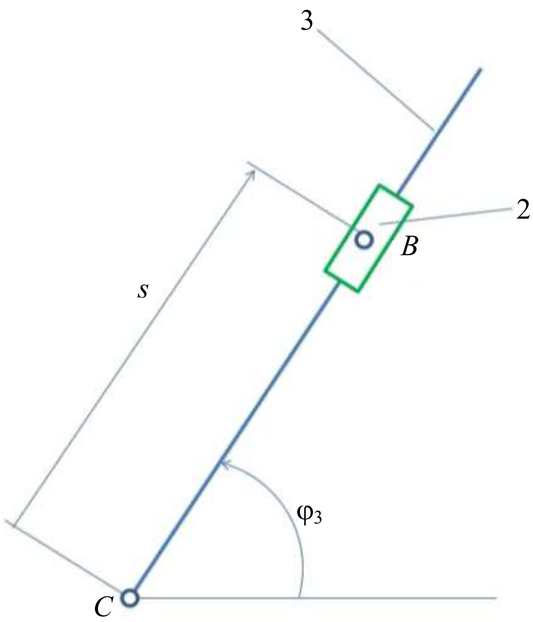


Fig. 2: The structural group, a dyad RTR

The forces from the dyad of the third RTR aspect can be traced in Fig. 3 and the calculations in the relational system (2):

$$\begin{cases} \begin{cases} x_{G_3} = x_C + s_3 \cdot \cos\varphi_3 \\ y_{G_3} = y_C + s_3 \cdot \sin\varphi_3 \end{cases} \begin{cases} \dot{x}_{G_3} = \dot{x}_C - s_3 \cdot \sin\varphi_3 \cdot \dot{\varphi}_3 \\ \dot{y}_{G_3} = \dot{y}_C + s_3 \cdot \cos\varphi_3 \cdot \dot{\varphi}_3 \end{cases} \\ \Rightarrow \begin{cases} \ddot{x}_{G_3} = \ddot{x}_C - s_3 \cdot \cos\varphi_3 \cdot \dot{\varphi}_3^2 - s_3 \sin\varphi_3 \cdot \ddot{\varphi}_3 \\ \ddot{y}_{G_3} = \ddot{y}_C - s_3 \cdot \sin\varphi_3 \cdot \dot{\varphi}_3^2 + s_3 \cos\varphi_3 \cdot \ddot{\varphi}_3 \end{cases} \\ \begin{cases} F^{ix}_{G_3} = -m_3 \cdot \ddot{x}_{G_3} \\ F^{iy}_{G_3} = -m_3 \cdot \ddot{y}_{G_3} \\ M^i_3 = -J_{G_3} \cdot \ddot{\varphi}_3 \end{cases} \begin{cases} F^{ix}_{G_2} = -m_2 \cdot \ddot{x}_{G_3} = -m_2 \cdot \ddot{x}_B \\ F^{iy}_{G_2} = -m_2 \cdot \ddot{y}_{G_2} = -m_2 \cdot \ddot{y}_B \\ M^i_2 = -J_{G_2} \cdot \ddot{\varphi}_2 = -J_{G_2} \cdot \ddot{\varphi}_3 \end{cases} \\ \sum M^{(2)}_B = 0 \Rightarrow M_{32} + M^i_2 = 0 \Rightarrow M_{32} = -M^i_2 \Rightarrow M_{23} = M^i_2 \\ \sum M^{(3)}_C = 0 \Rightarrow R_{23} \cdot s + M_{23} + M^i_3 - F^{ix}_{G_3} \cdot \left(y_{G_3} - y_C\right) \\ +F^{iy}_{G_3} \cdot \left(x_{G_3} - x_C\right) = 0 \\ \Rightarrow R_{23} = \dfrac{F^{ix}_{G_3} \cdot \left(y_{G_3} - y_C\right) + F^{iy}_{G_3} \cdot \left(x_C - x_{G_3}\right) - M_{23} = M^i_3}{s} \\ R_{32} = -R_{23} \Rightarrow \begin{cases} R^x_{32} = R_{32} \cdot \cos\left(\varphi_3 + \dfrac{\pi}{2}\right) = -R_{32} \cdot \sin\varphi_3 \\ R^y_{32} = R_{32} \cdot \sin\left(\varphi_3 + \dfrac{\pi}{2}\right) = R_{32} \cdot \cos\varphi_3 \end{cases} \\ \sum F^{(2)}_x = 0 \Rightarrow R^x_{12} + R^x_{32} + F^{ix}_{G_2} = 0 \Rightarrow R^x_{12} = -R^x_{32} - F^{ix}_{G_2} \\ \sum F^{(2)}_y = 0 \Rightarrow R^y_{12} + R^y_{32} + F^{iy}_{G_2} = 0 \Rightarrow R^y_{12} = -R^y_{32} - F^{iy}_{G_2} \\ \Rightarrow R_{12} = \sqrt{\left(R^x_{12}\right)^2 + \left(R^y_{12}\right)^2} \\ \sum F^{(3)}_x = 0 \Rightarrow R^x_{03} + F^{ix}_{G_3} + R^x_{23} = 0 \Rightarrow R^x_{03} = -F^{ix}_{G_3} + R^x_{32} \\ \sum F^{(3)}_y = 0 \Rightarrow R^y_{03} + F^{iy}_{G_3} + R^y_{23} = 0 \Rightarrow R^y_{03} = -F^{iy}_{G_3} + R^y_{32} \\ \Rightarrow R_{03} = \sqrt{\left(R^x_{03}\right)^2 + \left(R^y_{03}\right)^2} \end{cases} \tag{2}$$

The distribution of forces to the mechanism having an oscillating slide is made for the compressor mode according to Fig. 4. The calculation relations are given by the system (3). Force distribution shows how the forces are distributed within the module elements and takes into account the kinematic couplers existing within the module. It is totally different from the forces in that mechanism that show the forces that load the module onto the elements and the couplings. Forces distribution also shows how forces are transmitted from one element to the next element of the module within the link couple between the two mobile elements:

$$\begin{cases} \cos\gamma = \cos\left(\dfrac{3\pi}{2} + \varphi_3 - \varphi_1\right) = \cos\left(\varphi_1 - \varphi_3 - \dfrac{3\pi}{2}\right) \\ = \sin\left(2\pi - \varphi_1 - \varphi_3\right) = \sin\left(\varphi_3 - \varphi_1\right) \\ \begin{cases} F_u = F_m \cdot \cos\gamma F_m \cdot \sin\left(\varphi_3 - \varphi_1\right) \\ v_m = v_B = l_1 \cdot \omega_1 \\ v_u \equiv \dot{s} = v_m \cdot \sin\left(\varphi_3 - \varphi_1\right) \end{cases} \\ \Rightarrow \eta^c_i = \dfrac{F_u \cdot \dot{s}}{F_m \cdot v_m} \\ \Rightarrow \eta^c_i = \dfrac{F_m \cdot \sin\left(\varphi_3 - \varphi_1\right) \cdot v_m \cdot \sin\left(\varphi_3 - \varphi_1\right)}{F_m \cdot v_m} = \sin^2\left(\varphi_3 - \varphi_1\right) \\ \eta^{Dc}_i = \dfrac{F_u \cdot \dot{s}}{F_m \cdot v_m} = \dfrac{F_m \sin\left(\varphi_3 - \varphi_1\right) v_m \sin\left(\varphi_3 - \varphi_1\right)}{F_m \cdot v_m} = \sin^2\left(\varphi_3 - \varphi_1\right) \\ \Rightarrow \begin{cases} \eta^{Dc}_i = \eta^c_i \\ \eta^{Dc}_i = D^c \cdot \eta^c_i \end{cases} \Rightarrow D^c = 1; \\ \eta^c_i \sin^2\left(\varphi_3 - \varphi_1\right) \\ \Rightarrow \eta^c_i = \dfrac{l_0^2 \cdot \cos^2\varphi_1}{l_0^2 + l_1^2 + 2 \cdot l_0 \cdot l_1 \cdot \sin\varphi_1} = \dfrac{\cos^2\varphi_1}{1 + \lambda^2 + 2 \cdot \lambda \cdot \sin\varphi_1} \quad \lambda = \dfrac{l_1}{l_0} \end{cases} \tag{3}$$

For the motor regime, the force distribution can be traced in Fig. 5 and the corresponding calculation relationships are given by the relational system (4).

Here, it should be made clear that the two mechanisms of the mechanism considered as a motor mechanism are appropriate to the two separate phases. When the mechanism is operated from the crank we have a compressor regime in which the engine mechanism works.

When actuating from the piston, the mechanism actually switches to the engine, which is the only one that gives power to the engine directly. And in the compressor mode, power is transmitted to the mechanism either due to inertial forces or due to the action of other engine regimes from other cylinders in work, knowing that an internal combustion engine generally has more cylinders and not just one:

$$\begin{cases} \cos\gamma = \cos\left(\dfrac{3\pi}{2} + \varphi_3 - \varphi_1\right) = \cos\left(\varphi_1 - \varphi_3 - \dfrac{3\pi}{2}\right) \\ = \sin\left(2\pi - \varphi_1 - \varphi_3\right) = \sin\left(\varphi_3 - \varphi_1\right) \\ \begin{cases} F_u = F_m \cdot \cos\gamma = F_m \cdot \sin\left(\varphi_3 - \varphi_1\right) \\ v_u = v_B = l_1 \cdot \omega_1 \; v^D_u = v_m \cdot \sin\left(\varphi_3 - \varphi_1\right) \\ v_m \equiv \dot{s} = v_B \cdot \sin\left(\varphi_3 - \varphi_1\right) \end{cases} \\ \Rightarrow \eta^M_i = \dfrac{F_u \cdot v_u}{F_m \cdot \dot{s}} = \dfrac{F_m \cdot \sin\left(\varphi_3 - \varphi_1\right) \cdot v_B}{F_m \cdot v_B \cdot \sin\left(\varphi_3 - \varphi_1\right)} = 1 \\ \eta^{DM}_i = \dfrac{F_u \cdot v^D_u}{F_m \cdot v_m} = \dfrac{F_m \sin\left(\varphi_3 - \varphi_1\right) v_m \sin\left(\varphi_3 - \varphi_1\right)}{F_m \cdot v_m} = \sin^2\left(\varphi_3 - \varphi_1\right) \\ \Rightarrow \begin{cases} \eta^{DM}_i = \sin^2\left(\varphi_3 - \varphi_1\right) \\ \eta^{DM}_i = D^M \cdot \eta^M_i = D^M \end{cases} \Rightarrow D^M \sin^2\left(\varphi_3 - \varphi_1\right); \eta^M_i = 1 \end{cases} \tag{4}$$

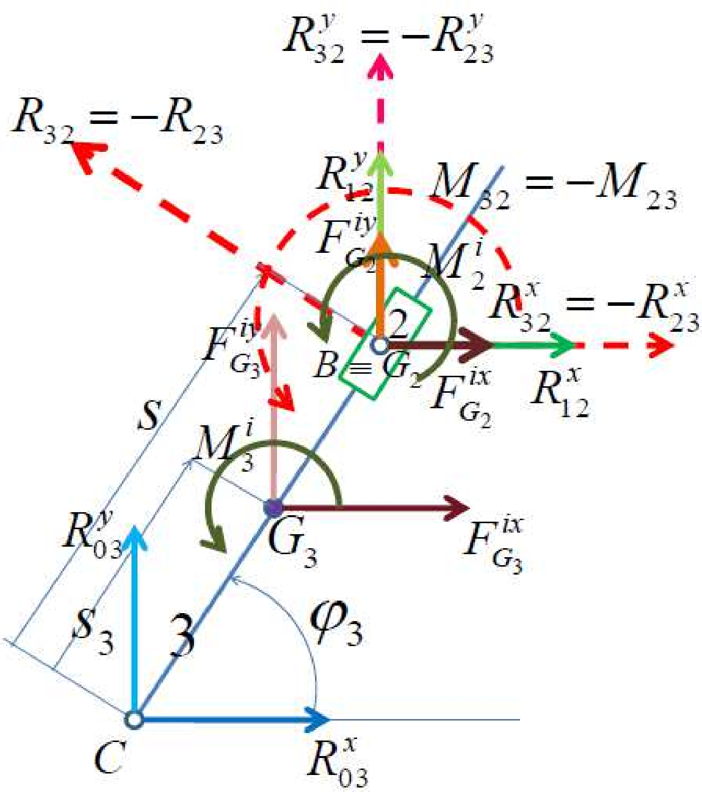
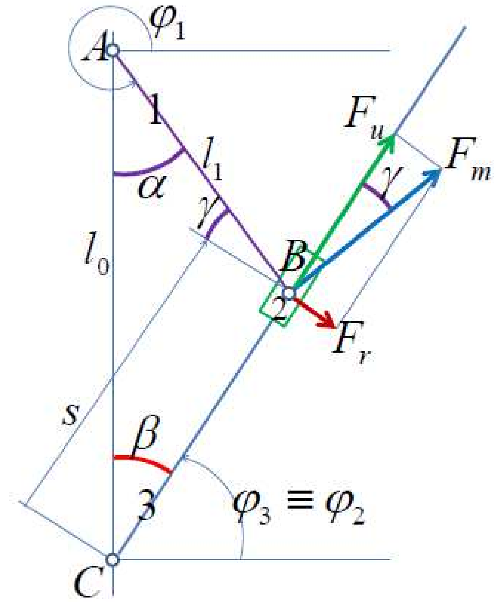
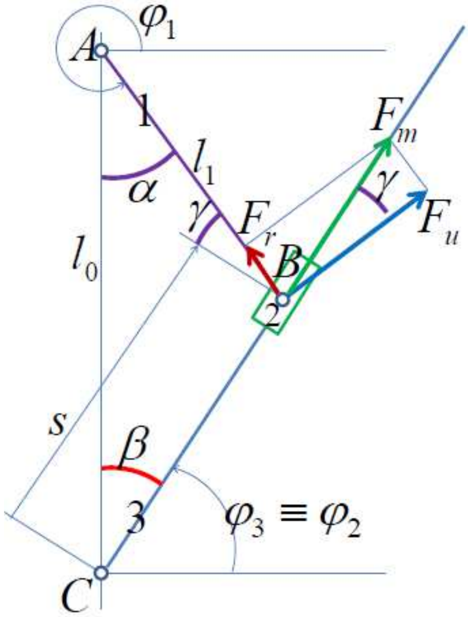


Fig. 3: The forces from the structural group, the dyad RTR



$$\begin{cases} \alpha = \varphi_1 - \frac{3\pi}{2}; \quad \beta = \frac{\pi}{2} - \varphi_3; \quad \gamma = \pi - \frac{\pi}{2} - \alpha - \beta \\ \gamma = \pi - \varphi_1 + \frac{3\pi}{2} - \frac{\pi}{2} + \varphi_3 - \frac{\pi}{2} = \frac{3\pi}{2} + \varphi_3 - \varphi_1 \end{cases}$$

Fig. 4: The distribution of forces to the mechanism having an oscillating slide, in a compresor regime



$$\begin{cases} \alpha = \varphi_1 - \frac{3\pi}{2}; \quad \beta = \frac{\pi}{2} - \varphi_3; \quad \gamma = \pi - \frac{\pi}{2} - \alpha - \beta \\ \gamma = \pi - \varphi_1 + \frac{3\pi}{2} - \frac{\pi}{2} + \varphi_3 - \frac{\pi}{2} = \frac{3\pi}{2} + \varphi_3 - \varphi_1 \end{cases}$$

Fig. 5: The distribution of forces to the mechanism having an oscillating slide, working in a engine regime

Results and Discussion

The dynamic calculation requires the determination of the variable angular velocity of the steering crank 1 and the corresponding angular acceleration. The angular velocity is determined with known relationships (5):

$$\begin{cases} \omega^D = D \cdot \omega \\ D^C = 1 \\ D^M = \sin^2\left(\varphi_3 - \varphi_1\right) = \dfrac{l_0^2 \cdot \cos^2 \varphi_1}{l_0^2 + l_1^2 + 2 \cdot l_0 \cdot l_1 \cdot \sin \varphi_1} \\ \omega^{DC} = D^C \cdot \omega = \omega; \omega^{DM} = D^M \cdot \omega = \sin^2\left(\varphi_3 - \varphi_1\right) \cdot \omega \end{cases} \tag{5}$$

Angular acceleration is calculated with relations (6):

$$\begin{cases} D^C = 1 \Rightarrow \varepsilon^C = 0 \\ D^M = \sin^2\left(\varphi_3 - \varphi_1\right) = \dfrac{l_0^2 \cdot \cos^2 \varphi_1}{l_0^2 + l_1^2 + 2 \cdot l_0 \cdot l_1 \cdot \sin \varphi_1} \\ \Rightarrow \varepsilon^M \equiv \varepsilon_1 = \left(\dot{\omega}^{DM}\right) = \dfrac{d\left(D^M \cdot \omega\right)}{dt} = D^{M\prime} \cdot \omega^2 \\ D^{M\prime} = \sin\left[2 \cdot \left(\varphi_3 - \varphi_1\right)\right] \cdot \left(\varphi_3' - 1\right) \\ D^{M\prime} = \sin\left[2 \cdot \left(\varphi_3 - \varphi_1\right)\right] \cdot \dfrac{l_1 \cdot \cos\left(\varphi_3 - \varphi_1\right) - s}{s} \\ \varepsilon^M = \sin\left[2 \cdot \left(\varphi_3 - \varphi_1\right)\right] \cdot \dfrac{l_1 \cdot \cos\left(\varphi_3 - \varphi_1\right) - s}{s} \cdot \omega^2 \end{cases} \tag{6}$$

The moment of mechanical or mass inertia (of the whole mechanism) reduced to the crank is determined by the relation (7):

$$\begin{aligned} & J^* = J_{G_1} + \left(J_{G_2} + J_{G_3}\right) \cdot \left(\frac{\omega_3}{\omega_1}\right)^2 \\ & + m_2 \cdot \left(x_{G_2}^{'2} + y_{G_2}^{'2}\right) + m_3 \cdot \left(x_{G_3}^{'2} + y_{G_3}^{'2}\right) \Rightarrow \\ & J^* = J_{G_1} + \left(J_{G_2} + J_{G_3}\right) \cdot \left(\frac{\omega_3}{\omega_1}\right)^2 \\ & + m_2 \cdot \left(x_B^{'2} + y_B^{'2}\right) + m_3 \cdot \left(x_{G_3}^{'2} + y_{G_3}^{'2}\right) \end{aligned} \tag{7}$$

The positioning of the weight centers of the mechanism is made according to the kinematic scheme shown in Fig. 6 so that the center of gravity of the movable element 2 coincides with the joint B and the center of gravity of the element 1 (already balanced) coincides with the fixed joint A.

Such a mechanism used as a motor mechanism can bring great advantages due to its better functioning, with much smaller accelerations and shocks compared to the classic Otto engines, thus achieving a smooth operation without shocks and vibrations so virtually no noise or interruptions, lower smoke, low fuel consumption at a higher installed power, with a superior dynamics even with classic Otto engines.

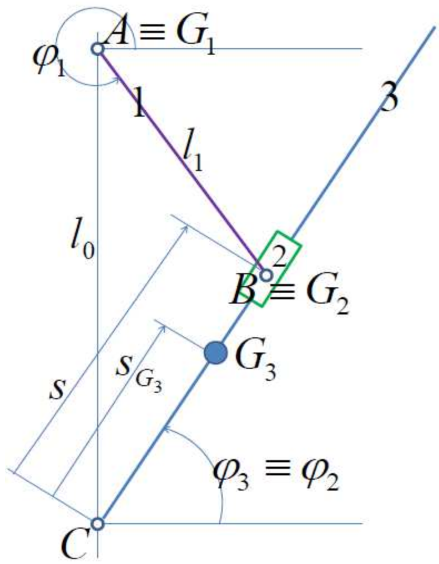


Fig. 6: The weight centers of the swing slide mechanism

The big advantages of such an engine are today translated by their possible use of gasoline and gas, to the detriment of diesel and diesel engines, as we can extend the life of gasoline engines for about 200 years and those for gas for at least 1000 years, given that we will have enough gas for stoves in the kitchens of the housewives or restaurants, but by clearly eliminating the consumption of gas to obtain electric or thermal energy, which are quickly replaced by nuclear power plants and regenerative energies, headed by the sun and the wind.

Conclusion

A mechanism with an oscillating slider is a mechanism quite often used in the technique. It has some advantages over other similar mechanisms being able to work with accelerations and reduced forces under the same conditions with another similar mechanism.

For example, an internal combustion engine could be built more efficiently with this mechanism, resulting in a motor that would successfully replace Otto's classic one. In the paper, the kinematics, forces and their distribution will be presented within such a mechanism.

To the presented motor mechanism the dynamic-kinematics is different from the classical-kinematics known, but when the constructive parameters are setting on normal values, the dynamic motor velocities and accelerations take the same values as the classical motor speeds and accelerations known.

In the presented article was showed a new model of an internal combustion engine, able to running with reduced exhaust emissions.

The new mechanism was designed and intended for industrial production.

As long as we produce electricity and heat by burning fossil fuels is pointless to try to replace all thermal engines with electric motors, as loss of energy and pollution will be even larger. However, it is well to continuously improve the thermal engines, to reduce thus fuel consumption.

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Author's Contributions

All the authors contributed equally to prepare, develop and carry out this manuscript.

Ethics

This article is original and contains unpublished material. Authors declare that are not ethical issues and no conflict of interest that may arise after the publication of this manuscript.

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