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Machine Motion Equations Presented in a New General Format

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Abstract: Considering the increased importance of robots nowadays, when no large factory or factory can work without robots, we want to present in the work the motion equations of the machine in an original form, both in terms of aspect and their deduction. The machine's motion equations can be used in dynamic calculations at any type of machine, whether it be a motor, a compressor, a lucrative machine, a robot, a system, a mechanism, a vehicle, a mechanical transmission, or any other type of car. The dynamics of systems is their real movement, the dynamic movement, in which the influences of three main factors interfere, which modify the kinematics of the mechanism when it moves really, dynamic. The first dynamic factor is the forces of inertia or the effect of inertial masses. The second important dynamic factor is that of the couplings, of the linkages within the respective machine mechanisms. The latter and the third dynamic factor represents the influence of system elasticity on its dynamic functioning. Only dynamic coefficient of inertia and the influence of kinematic couplings in the system were used in the analyzed sample. The dynamic coefficient due to elasticity and deformation in the system has not been taken into account since the overwhelming influence of the inertial forces is impacted by additional dynamic changes and also by the kinematic couplings in the system and the elastic deformations do not greatly influence the dynamics of the system in the case of the example remembered. If a robot was being discussed, things were similar, as in the case of various vehicles and various mechanisms and machines. However, for rigid memory transmissions, elastic deformations are important, which is why they should be considered in such systems.

Keywords: Robots, Mechatronic Systems, Structure, Dynamics, Dynamics Systems, Machines, Machine Motion Equations, Dynamic Factors

Introduction

Today, robots are increasingly present in the machine building industry, sometimes even in some sections, to replace workers altogether due to the high quality of their work, repetitive, without stopping or interrupting, without manufacturing and assembly.

In addition, robots do not get sick, do not need medical leave or rest, work faster and better than people and support dyers, general assemblies, etc.

Generally, robots have increased the quality and productivity of work and have not even created a union to defend their claims, demanding higher wages for them and larger holidays.

Interestingly, a robot works without pause, but unpaid, without breaks, without complaining about factory conditions.

Robots can work on three shifts, that is, permanently, but not by moving them as humans, but they always remain the same robots deployed on a day without interruptions, without pauses, without rest, without problems.

Robots are today highly valued by major carmakers which even build complete sections where only robots work because they do not have a trade union, they do not require increased salaries (they actually work without any salary), they do not have to leave on holidays, do not want free days and can even work on Saturdays and Sundays, without breaks, if necessary, on three exchanges, including in toxic, dangerous environments, or even in hard-to-reach areas. The importance of implementing robots can no longer be challenged. They have increased the quality of work and the production of an enterprise so that they can no longer give up their help.

Workers reclassified and worked only in more friendly jobs or other jobs, such as supermarkets, in better conditions, with higher wages, more days off and are satisfied with the production and sales gains due to robot work in large factories.

One can clearly state that our robots have considerably improved our lives. Thanks to them, a new free day for almost all working people was introduced on Friday, in addition to Saturday and we will soon be able to enter another free day, but we have to choose Monday or Thursday.

People were initially trained by trade union leaders to track and sabotage robots, destroy them and not accept them. Today things are clear and the robots work quietly in the big companies and factories for the good of all, so now we can accept the silence of automation, robotics, electronics, without letting us be fooled by the trade union leaders who slowly slow down and calm down.

Whether we like it or not, the robots have already stolen all their heavy jobs.

Certain anthropomorphic robots are, as we have already said, in most of the most widespread and widely used works around the world, due to their ability to adapt quickly to forced labor, working without breaks or 24 hours, air or salary. Anthropomorphic robots are thin, elegant, easy to configure and adapted to virtually any location, being the most flexible, useful, more penetrating, easier to install and maintain. For the first time, these robots affirmed themselves in the automotive industry and especially in the automotive industry, today they have penetrated almost all industrial fields, being easily adaptable, flexible, dynamic, resilient, cheaper than other models, occupying a workspace important. They can also work in toxic or hazardous environments used in dyeing, chemical cleaning, chemical or nuclear environments, dealing with explosive objects or military missions in land or sea mines, even if they are forbidden to use them. countries around the world that use them, such as Afghanistan.

The most used industrial robots today are built. The importance of studying anthropomorphic robots has also been signaled, being today the most widespread robots around the world, thanks to its simple design, construction, implementation, operation and maintenance. In addition, anthropomorphic systems are simpler and cheaper from a technological point of view, with consistent, demanding and repetitive work, with no major maintenance problems.

Considering the increased importance of robots nowadays, when no large factory or factory can work without robots, one wants to present in the work the motion equations of the machine in an original form, both in terms of aspect and their deduction. The machine's motion equations can be used in dynamic calculations at any type of machine, whether it be a motor, a compressor, a lucrative machine, a robot, a system, a mechanism, a vehicle, a mechanical transmission, or any other type of car. The dynamics of systems is their real movement, the dynamic movement, in which the influences of three main factors interfere, which modify the kinematics of the mechanism when it moves really, dynamic. The first dynamic factor is the forces of inertia or the effect of inertial masses. The second important dynamic factor is that of the couplings, of the linkages within the respective machine mechanisms. The latter and the third dynamic factor represents the influence of system elasticity on its dynamic functioning (Antonescu and Petrescu, 1985; 1989; Antonescu *et al*., 1985a; 1985b; 1986; 1987; 1988; 1994; 1997; 2000a; 2000b; 2001; Aversa *et al*., 2017a; 2017b; 2017c; 2017d; 2017e; 2016a; 2016b; 2016c; 2016d; 2016e; 2016f; 2016g; 2016h; 2016i; 2016j; 2016k; 2016l; 2016m; 2016n; 2016o; Cao *et al*., 2013; Dong *et al*., 2013; Comanescu, 2010; Franklin, 1930; He *et al*., 2013; Lee, 2013; Lin *et al*., 2013; Liu *et al*., 2013; Padula and Perdereau, 2013; Perumaal and Jawahar, 2013; Petrescu, 2011; 2015a; 2015b; Petrescu and Petrescu, 1995a; 1995b; 1997a; 1997b; 1997c; 2000a; 2000b; 2002a; 2002b; 2003; 2005a; 2005b; 2005c; 2005d; 2005e; 2011a; 2011b; 2012a; 2012b; 2013a; 2013b; 2013c; 2013d; 2013e; 2016a; 2016b; 2016c; Petrescu *et al*., 2009; 2016; 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; 2018a; 2018b; 2018c; 2018d; 2018e; 2018f; 2018g; 2018h; 2018i; 2018j; 2018k; 2018l; 2018m; 2018n).

Materials and Methods

The angular velocity of the driving element ω^* is considered to be constant because it is given by a motor that generally operates in a constant mode with a constant angular velocity ω_n, which is considered the input speed in the system. However, due to the dynamic influences given by the whole system, the angular velocity of the driving element ω^* permanently undergoes changes of value, dynamic changes depending on the position occupied by the leading element at that time. One can, therefore, consider that the angular velocity of the leading element is a function of the φ input angle. Dynamically, variable input angular velocity ω^* is a function of constant input speed ω_n and three essential dynamic factors D_i, D_c, D_e (relationship 1, which is the first original machine motion equation):

$$\omega^* = \omega_n \cdot D_i \cdot D_c \cdot D_e \tag{1}$$

If it gets up to the square one obtains the form (2), which by derivation generates the equation of the angular acceleration (3), which is the second original machine motion equation.

$$\omega^{*2} = \omega_n^2 \cdot D_i^2 \cdot D_c^2 \cdot D_e^2 \tag{2}$$

$$\begin{cases} \varepsilon^* = \omega_n^2 \cdot \left(D_i \cdot D_i' \cdot D_c^2 \cdot D_e^2\right. \\ \left. + D_i^2 \cdot D_e^2 \cdot D_c \cdot D_c' + D_i^2 \cdot D_c^2 \cdot D_e \cdot D_e' \right) \end{cases} \tag{3}$$

The three main dynamic coefficients represent the dynamic coefficient of the inertial forces of the whole system D_i, the coefficient of forces imposed by the system couplings D_c and the coefficient of the elastic deformations in the system D_e. If want to eliminate the influence of one of the three coefficients, it is enough to equate that coefficient with the value of 1.

The inertial coefficient (the most important) is given by the relationships from the system (4), (Petrescu, 2015b):

$$\begin{cases} D_i = \sqrt{\dfrac{J_m^*}{J^*}};\ D_i^2 = \dfrac{J_m^*}{J^*}; \\ D_i \cdot D_i' = -\dfrac{1}{2}\dfrac{J_m^* \cdot J^{*\prime}}{J^{*2}} \Rightarrow D_i' = -\dfrac{1}{2}\dfrac{J_m^* \cdot J^{*\prime}}{D_i \cdot J^{*2}} \end{cases} \tag{4}$$

where, J^* is the mass inertia moment of the system and J_m^* represents its average value. The derivatives used (denoted by ') are those based on the angle φ.

D_c and D_e are variable depending on the dynamic system in question. D_c can have a considerable influence, whereas D_e it is generally of minor importance and can be disregarded by theory by its matching to the value 1. The method is original and has a general character.

Results

An example of calculation is applied to internal combustion engines (Fig. 1; Petrescu, 2015b), where the dynamic inertial coefficient has the values given by systems relations (5-6), (system 5 when the mechanism works as a complessor and the system 6 when the mechanism works as a motor):

$$\begin{cases} D_c = \sin^2\psi \\ \dot{D}_c = \sin 2\psi \cdot \dot{\psi} \\ D_c' = \sin 2\psi \cdot \psi' \\ \psi' = \dfrac{\dot{\psi}}{\omega} \end{cases} \tag{5}$$

$$\begin{cases} D_c = \sin^2(\psi - \phi) \\ \dot{D}_c = \sin 2(\psi - \phi) \cdot (\dot{\psi} - \omega) \\ D_c' = \sin\left[2(\psi - \phi)\right] \cdot (\psi' - 1) \end{cases} \tag{6}$$

To determine dynamics at an Otto engine must (first at all) to set the formula of reduced moment of inertia (7); Fig. 1 (it takes $a_1 = 0$, as the crank is already balanced):

$$\begin{cases} J^* = J_{G_1} + J_{G_2} \cdot \left(\dfrac{\dot{\psi}}{\omega}\right)^2 + m_3 \cdot \left(\dfrac{v_{G_3}}{\omega}\right)^2 + m_2 \cdot \left(\dfrac{v_{G_2}}{\omega}\right) \\ = J_{G_1} + J_{G_2} \cdot \lambda^2 \cdot \dfrac{\sin^2\varphi}{\sin^2\psi} + m_3 \cdot r^2 \cdot \dfrac{\sin^2(\psi - \varphi)}{\sin^2\psi} \\ + m_2 \cdot \left(r^2 + a^2 \cdot \psi'^2 - 2ar \cdot \psi' \cdot \cos(\psi - \varphi)\right) \\ = J_{G_1} + J_{G_2} \cdot \lambda^2 \cdot \dfrac{\sin^2\varphi}{\sin^2\psi} + m_3 \cdot r^2 \cdot \dfrac{\sin^2(\psi - \varphi)}{\sin^2\psi} \\ + m_2 \cdot \left(r^2 + a^2 \cdot \lambda^2 \cdot \dfrac{\sin^2\varphi}{\sin^2\psi} - 2a \cdot r \cdot \lambda \cdot \dfrac{\sin\varphi}{\sin\psi} \cdot \cos(\psi - \varphi)\right) \\ J^* = J_{G_1} + m_2 \cdot r^2 + \left(J_{G_2} + m_2 \cdot a^2\right) \cdot \lambda^2 \cdot \dfrac{\sin^2\varphi}{\sin^2\psi} \\ + m_3 \cdot r^2 \cdot \dfrac{\sin^2(\psi - \varphi)}{\sin^2\psi} - 2m_2 \cdot a \cdot r \cdot \lambda \cdot \dfrac{\sin\varphi \cdot \cos(\psi - \varphi)}{\sin\psi} \end{cases} \tag{7}$$

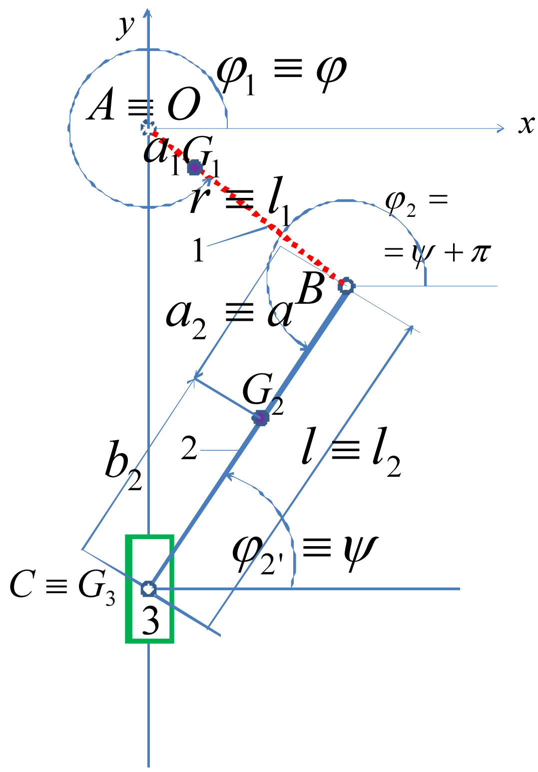


Fig. 1: The geometry of an Otto engine mechanism

One determines:

$$J^*_{Max}, J^*_{\min}, J^*_{med} \equiv J^*_m = \frac{J^*_{Max} + J^*_{\min}}{2} \tag{8}$$

and $J^{*\prime}$, with relations (8, 9):

$$\begin{cases} J^{*\prime} = \dfrac{1}{\sin^4\psi} \cdot \left\{ \left(J_{G_2} + m_2 \cdot a^2\right) \cdot \lambda^2 \right. \\ .\left(\sin 2\varphi \cdot \sin^2\psi - 2\lambda \cdot \sin^3\varphi \cdot \cos\psi\right) \\ +m_3 \cdot r^2 \cdot \left[\sin 2(\psi - \varphi) \cdot \left(\lambda \sin\varphi \cdot \sin\psi - \sin^2\psi\right)\right. \\ \left. -2\lambda \cdot \sin^2(\psi - \varphi) \cdot \cos\psi \cdot \sin\varphi \right] \\ +2m_2 a r \lambda \cdot \left[\sin\varphi \cdot \sin\psi \cdot \sin(\psi - \varphi)\right. \\ \cdot\left(\lambda \sin\varphi \cdot \sin\psi - \sin^2\psi\right) \\ +\lambda \cdot \sin^2\varphi \cdot \sin\psi \cdot \cos(\psi - \phi) \cdot \cos\psi \\ \left.\left. -\sin^3\psi \cdot \cos\varphi \cdot \cos(\psi - \varphi)\right]\right\} \end{cases} \tag{9}$$

Method is applied separately for two distinct situations: When the engine is working on a compressor and into the motor system. Calculations should be made for an engine with a single cylinder.

Dynamic Kinematics Analysis for the Otto Engine in Compressor System

Now, one can see the engine main mechanism in compressor system (when the motor mechanism is acting from the crank). It is determining now, the velocities and the accelerations of the piston and motor shaft, normal and dynamic (Fig. 2-5).

Dynamic Kinematics Analysis for the Otto Engine in Motor System

Now, one can see the engine main mechanism in motor system (when the motor mechanism is acting from the piston). It is determining now, the velocities and the accelerations of the piston and motor shaft, normal and dynamic (Fig. 6-9).

Dynamic Kinematics Analysis for an Otto Engine, Mono-cylinder

Now, one can see the engine main mechanism in motor and compressor system (when the motor mechanism is acting from the piston and from the crank). It is determining now, the velocities and the accelerations of the piston and motor shaft, normal and dynamic (Fig. 10-14).

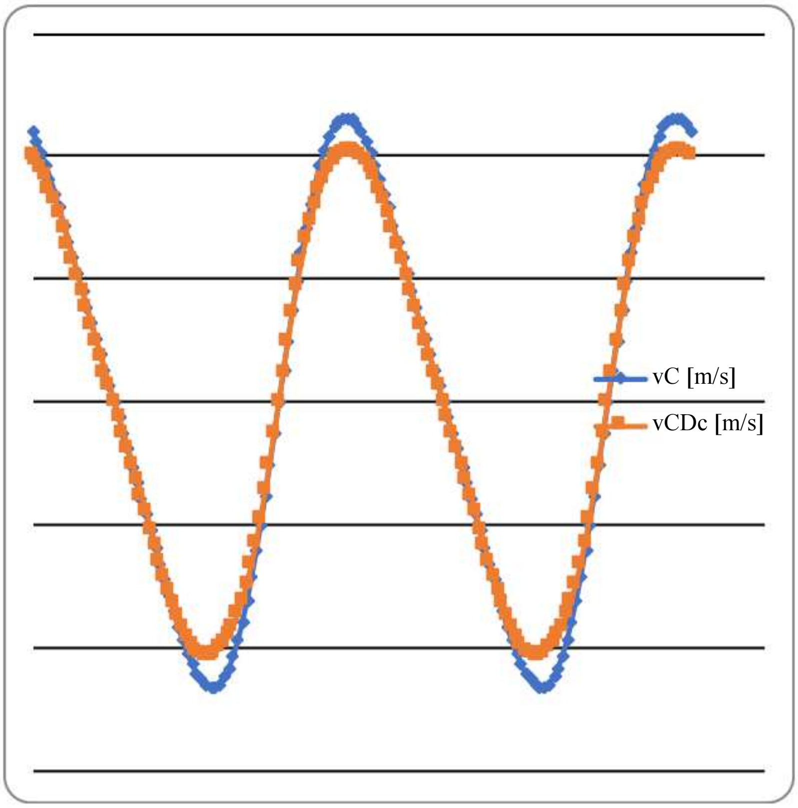


Fig. 2: The velocities of the piston, when the engine works in the compressor system

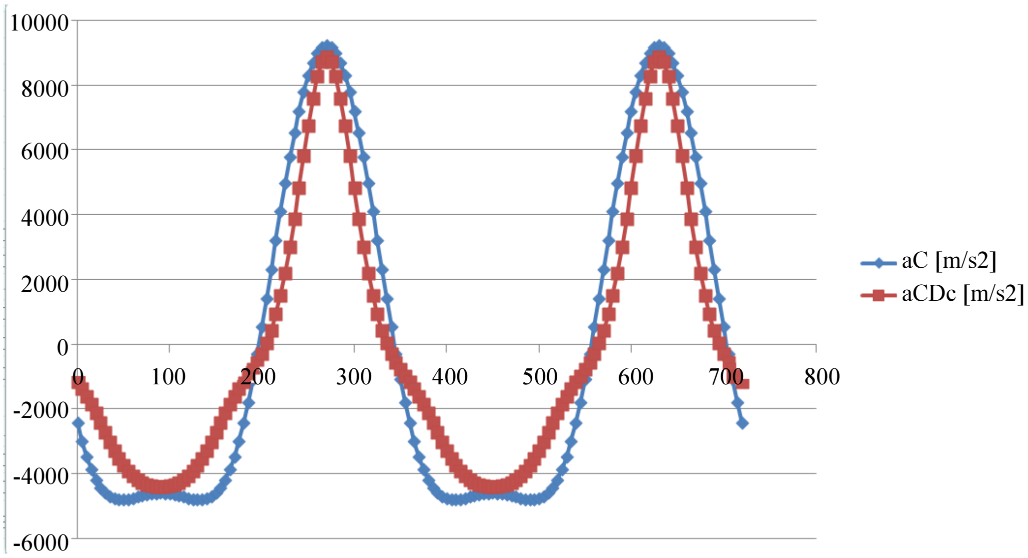


Fig. 3: The accelerations of the piston, when the engine works in the compressor system

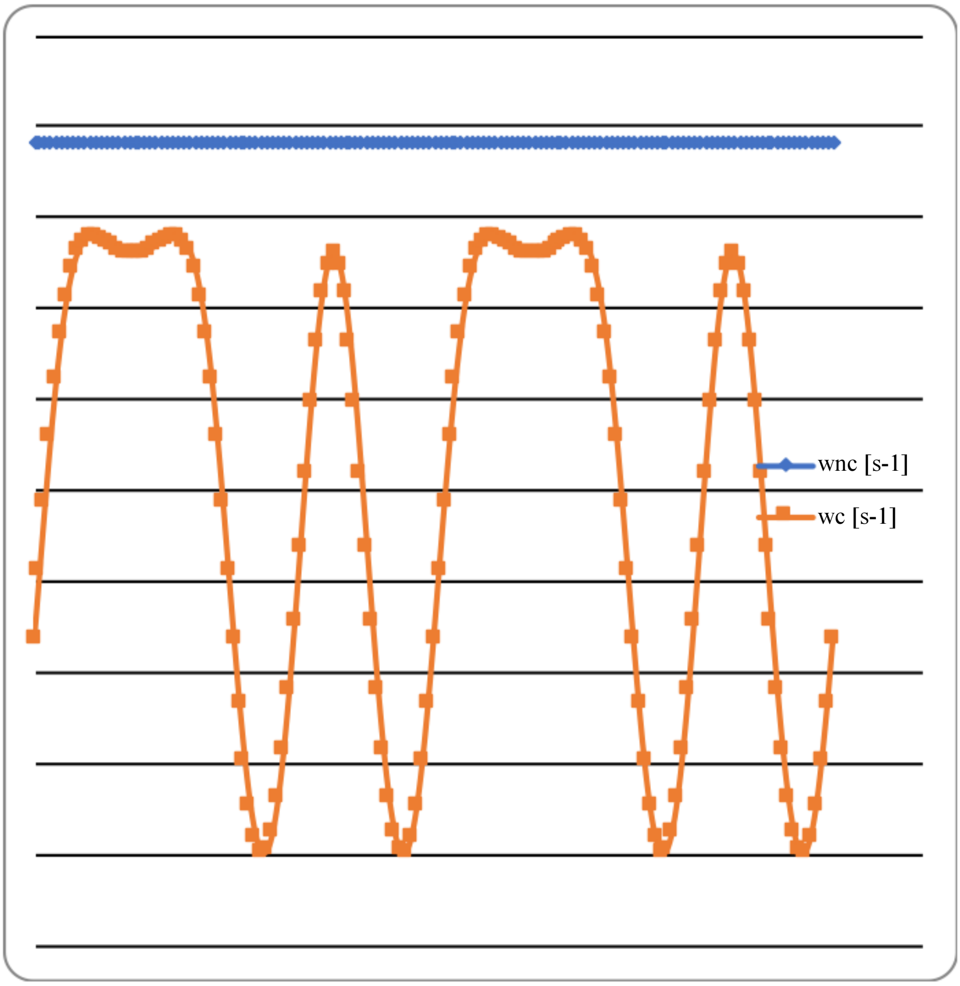


Fig. 4: The angular velocities of the motor shaft, when the engine works in the compressor system

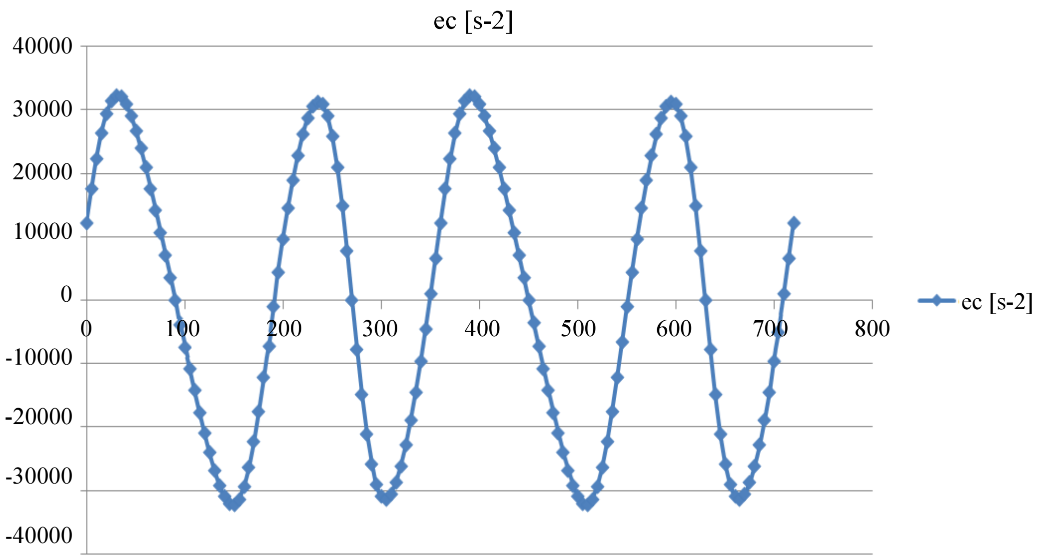


Fig. 5: The angular accelerations of the motor shaft, when the engine works in the compressor system

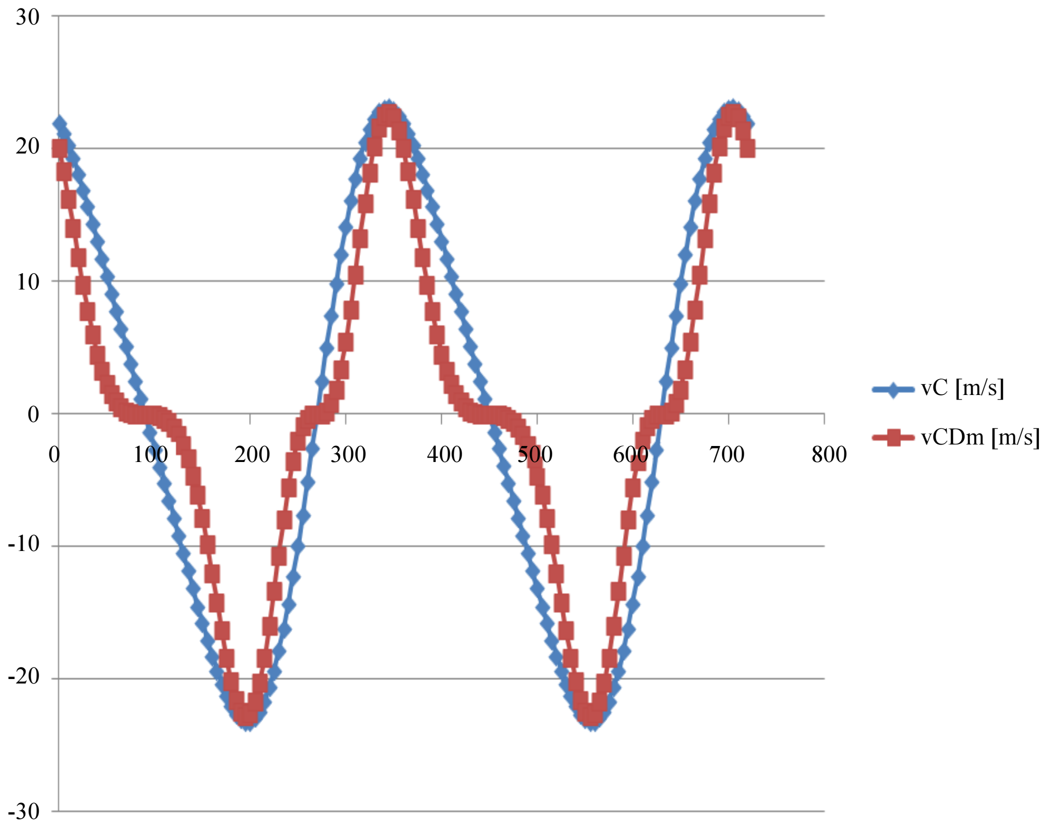


Fig. 6: The velocities of the piston, when the engine works in the motor system

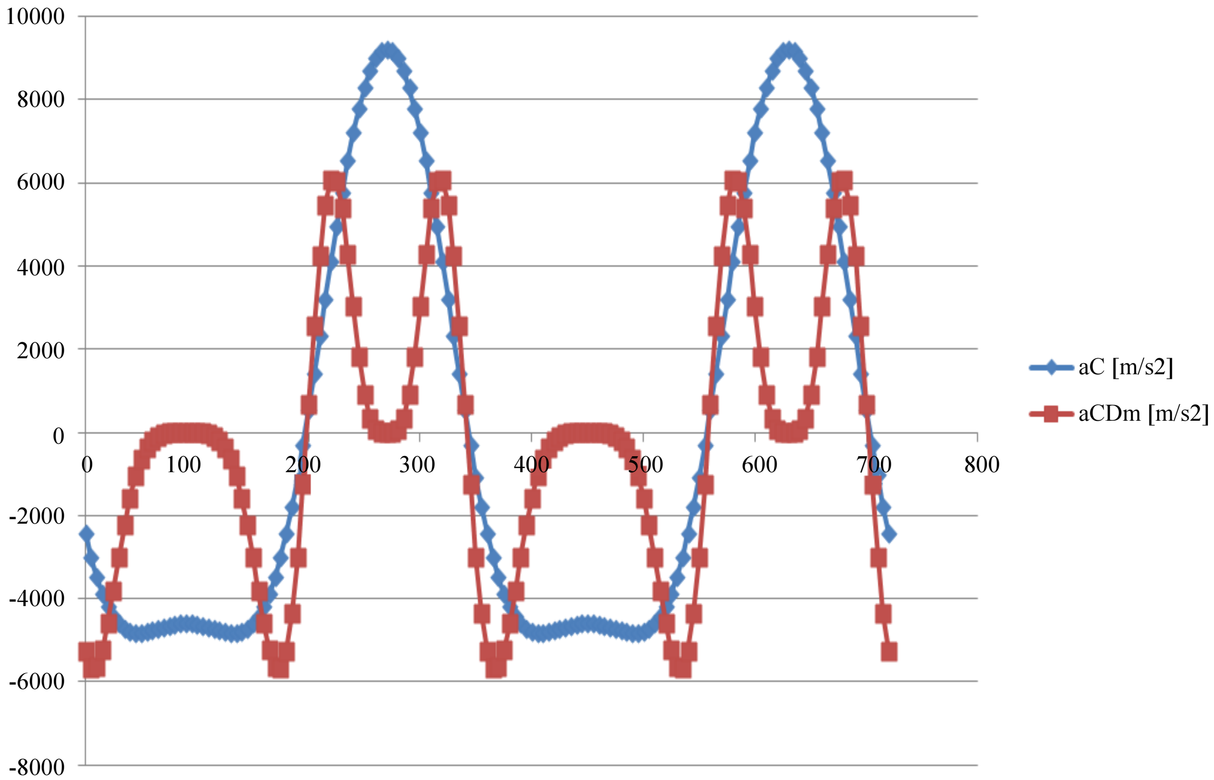


Fig. 7: The accelerations of the piston, when the engine works in the motor system

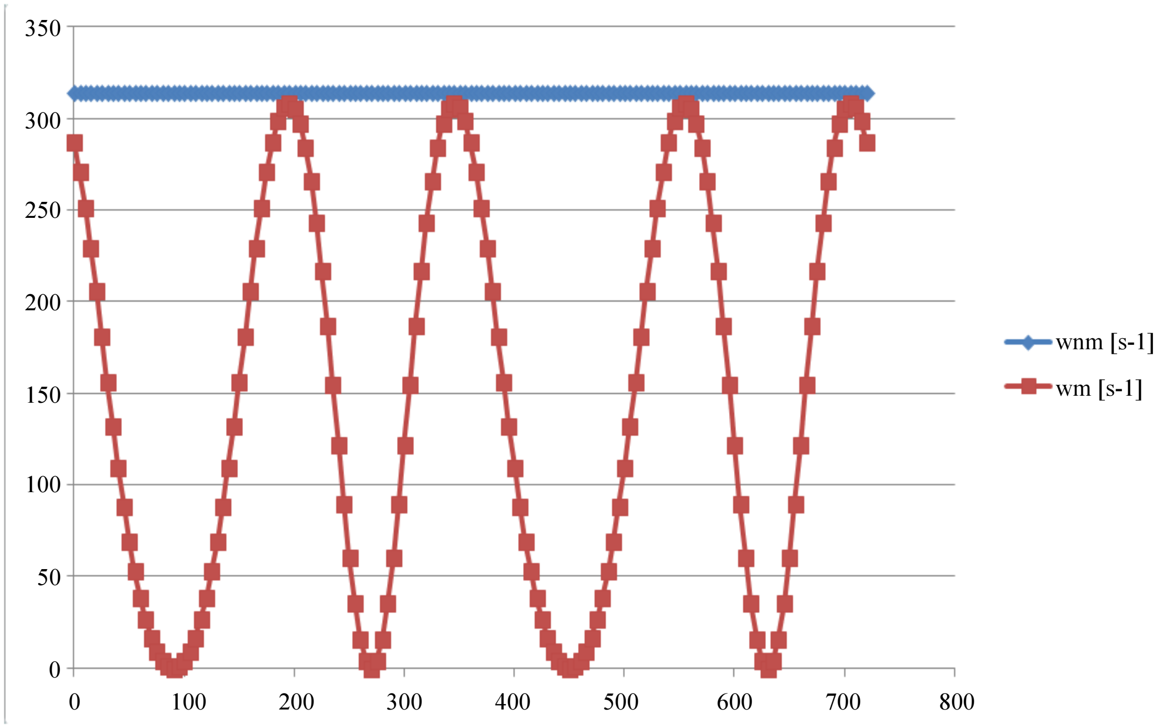


Fig. 8: The angular velocities of the motor shaft, when the engine works in the motor system

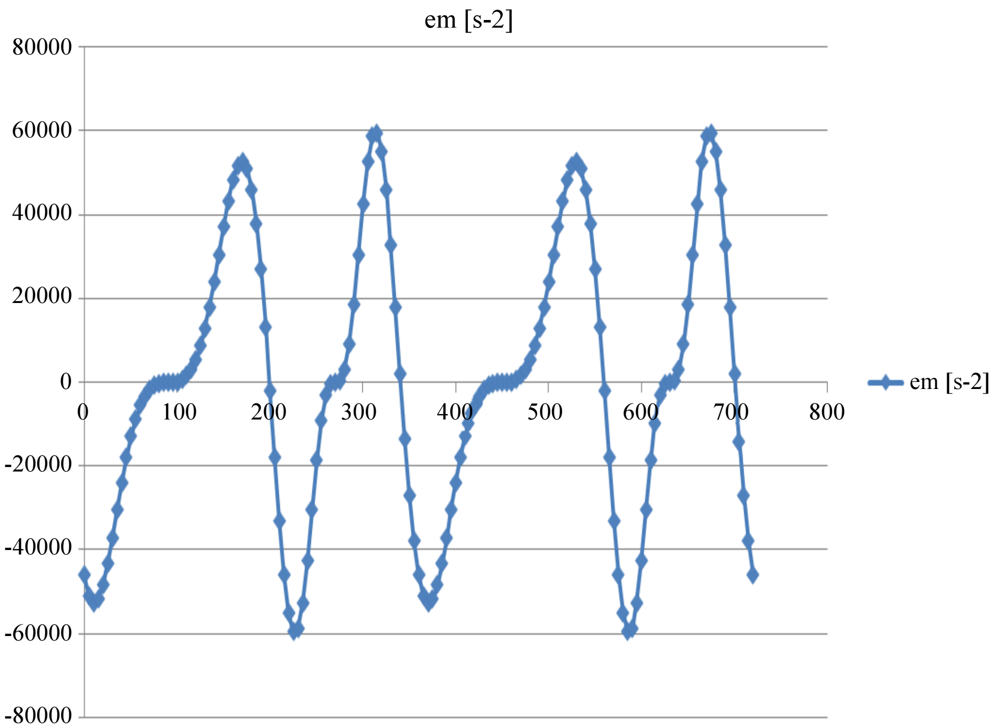


Fig. 9: The angular accelerations of the motor shaft, when the engine works in the motor system

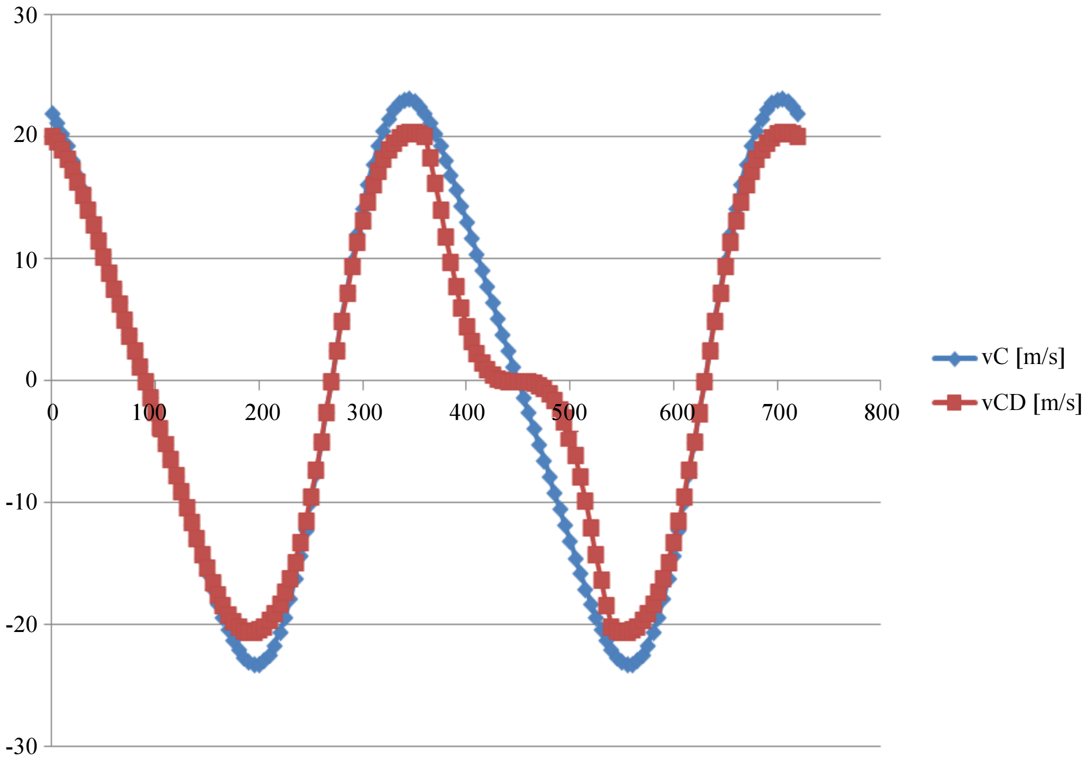


Fig. 10: The velocities of the piston, for a mono cylinder engine

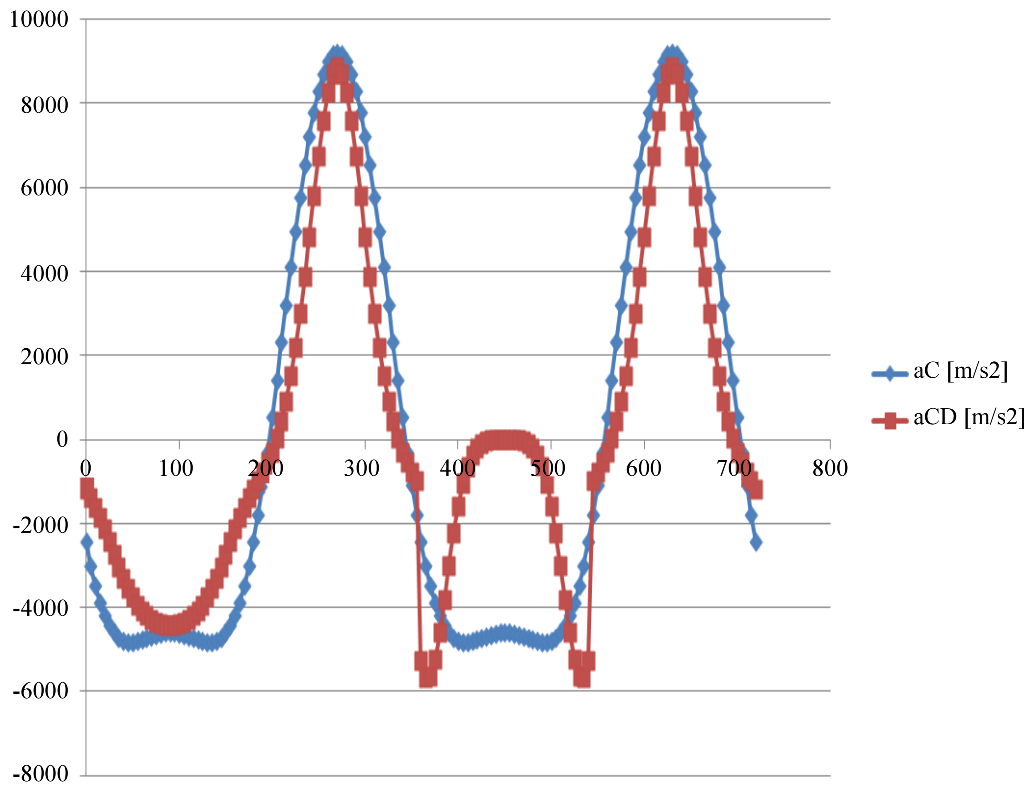


Fig. 11: The accelerations of the piston, for a mono cylinder engine, oriented upside down

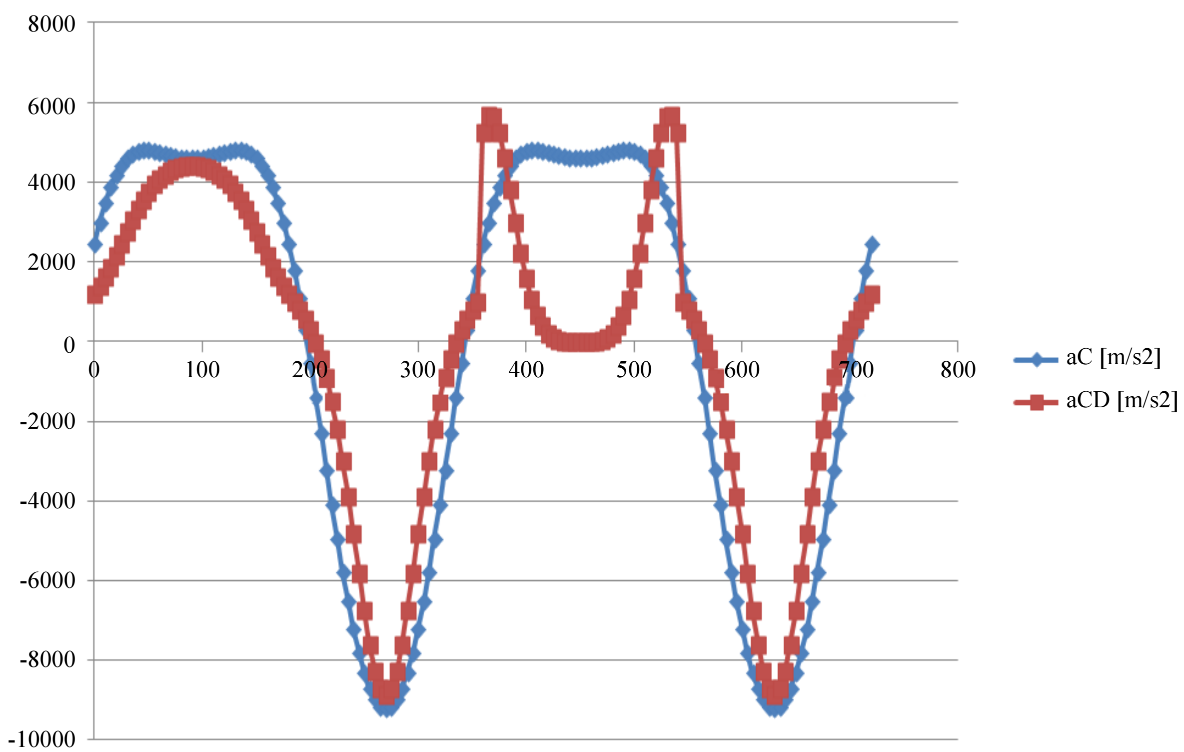


Fig. 12: The accelerations of the piston, for a mono cylinder engine, oriented head upward

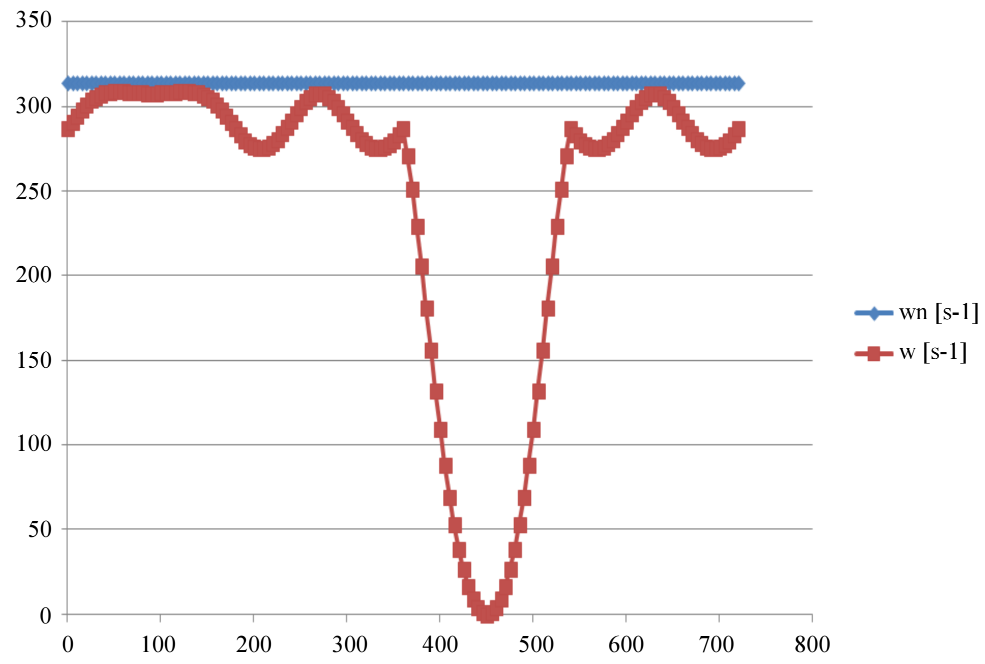


Fig. 13: The angular velocities of the motor shaft, for a mono cylinder engine

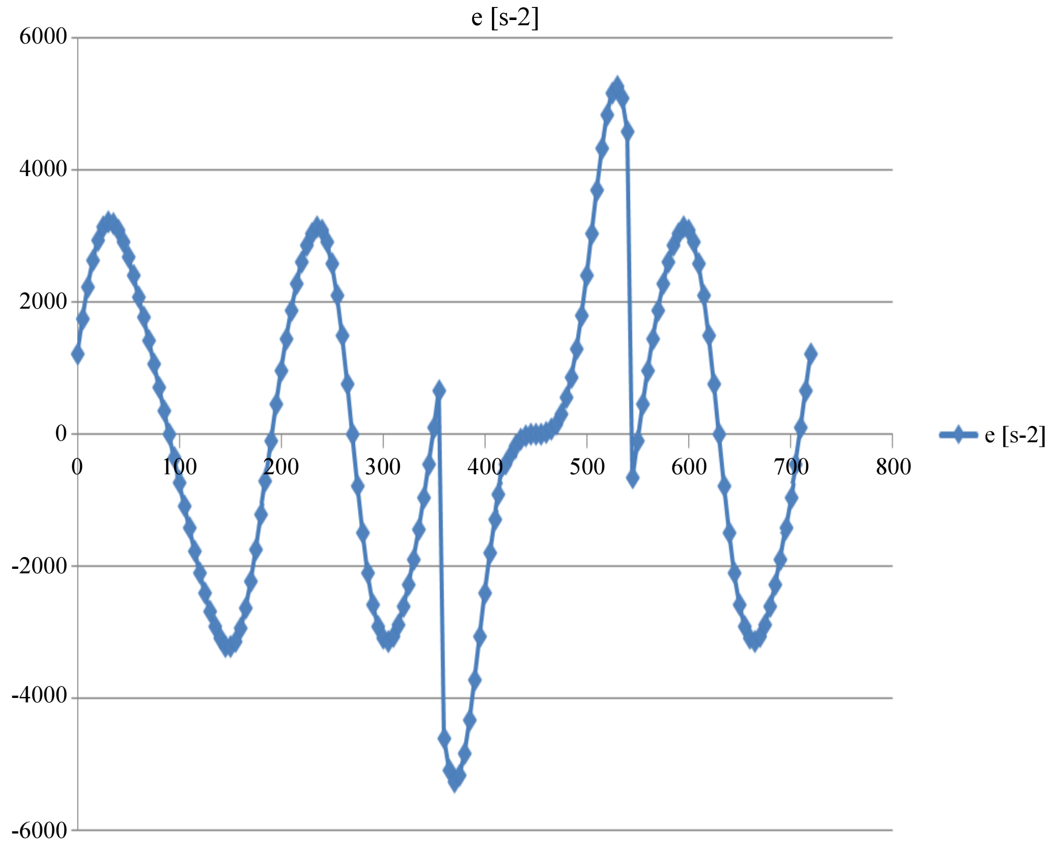


Fig. 14: The angular accelerations of the motor shaft, for a mono cylinder engine

Discussion

Only dynamic coefficient of inertia and the influence of kinematic couplings in the system were used in the analyzed sample.

The dynamic coefficient due to elasticity and deformation in the system has not been taken into account since the overwhelming influence of the inertial forces is impacted by additional dynamic changes and also by the kinematic couplings in the system and the elastic deformations do not greatly influence the dynamics of the system in the case of the example remembered.

If a robot was being discussed, things were similar, as in the case of various vehicles and various mechanisms and machines. However, for rigid memory transmissions, elastic deformations are important, which is why they should be considered in such systems.

During the 1970s energy crisis, the production and sale of cars equipped with internal combustion engines increased from several million to over sixty million a year and the world fleet ranged from tens of millions to billions.

As long as we produce electricity and heat by burning fossil fuels, it is useless to try to replace all electric motors as electricity and the pollution will be even greater. However, it is good to continually improve thermal motors to reduce fuel consumption.

Otto and diesel engines are today the best solution for transporting our daily work together with electric and reaction engines. For these reasons, it is imperative that we can accurately calculate the efficiency of the engine so that it can be increased continuously. However, it is good to continually improve thermal motors to reduce fuel consumption.

Since today the moments of mass inertia, i.e., the masses of rotation motion elements considered around the axis of rotation, are less known, one reintroduce in this paper some main cases (Fig. 15-27), (Petrescu *et al.*, 2016).

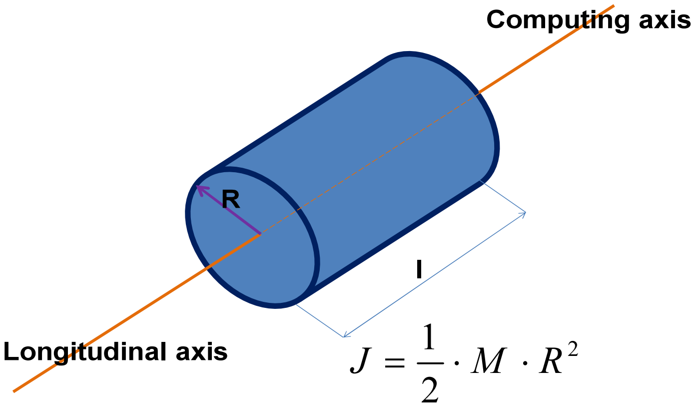


Fig. 15: Mass moment of inertia of a cylinder or a disc, fixed to the longitudinal axis of the cylinder or disc

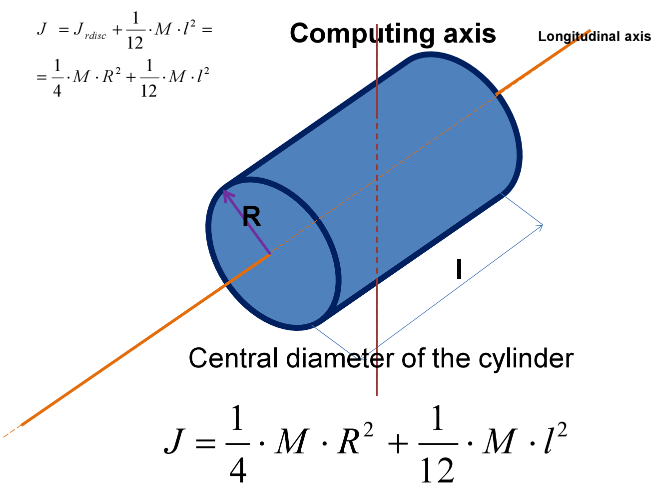
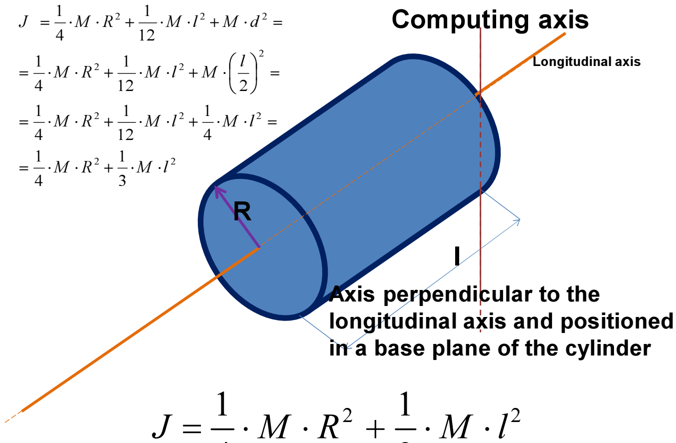


Fig. 16: Mass moment of inertia of a cylinder, determined in about an axis central diametral

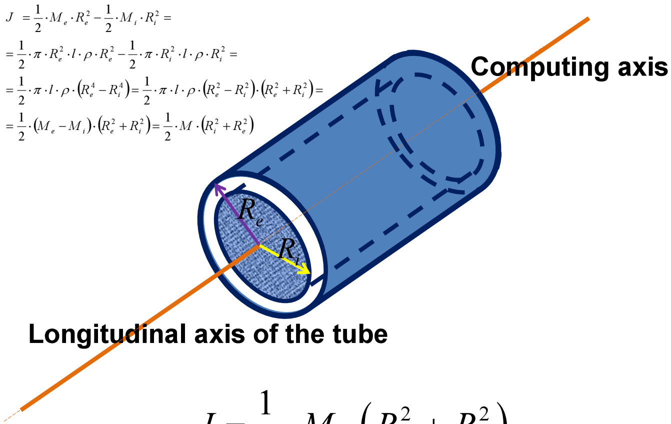
$$
\begin{aligned}
J &= \frac{1}{4}\cdot M\cdot R^2+\frac{1}{12}\cdot M\cdot l^2+M\cdot d^2=\\
&=\frac{1}{4}\cdot M\cdot R^2+\frac{1}{12}\cdot M\cdot l^2+M\cdot\left(\frac{l}{2}\right)^2=\\
&=\frac{1}{4}\cdot M\cdot R^2+\frac{1}{12}\cdot M\cdot l^2+\frac{1}{4}\cdot M\cdot l^2=\\
&=\frac{1}{4}\cdot M\cdot R^2+\frac{1}{3}\cdot M\cdot l^2
\end{aligned}
$$



$$J=\frac{1}{4}\cdot M\cdot R^2+\frac{1}{3}\cdot M\cdot l^2$$

Fig. 17: Mass moment of inertia of the cylinder, caused about an axis lying in the plane of the end of the cylinder perpendicular to the longitudinal axis

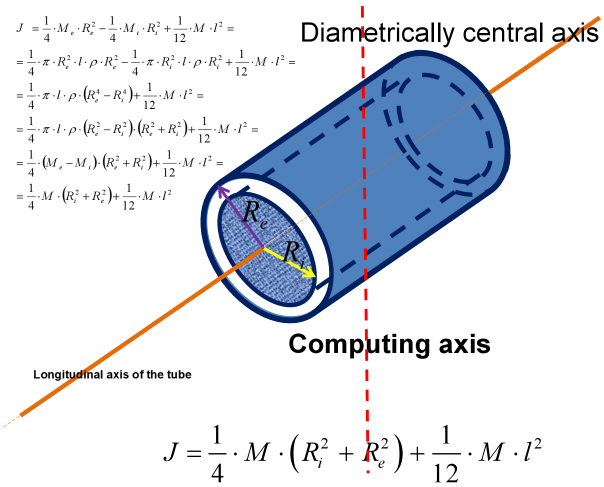
$$
\begin{aligned}
J &= \frac{1}{2}\cdot M_e\cdot R_e^2-\frac{1}{2}\cdot M_i\cdot R_i^2=\\
&=\frac{1}{2}\cdot\pi\cdot R_e^2\cdot l\cdot\rho\cdot R_e^2-\frac{1}{2}\cdot\pi\cdot R_i^2\cdot l\cdot\rho\cdot R_i^2=\\
&=\frac{1}{2}\cdot\pi\cdot l\cdot\rho\cdot\left(R_e^4-R_i^4\right)=\frac{1}{2}\cdot\pi\cdot l\cdot\rho\cdot\left(R_e^2-R_i^2\right)\cdot\left(R_e^2+R_i^2\right)=\\
&=\frac{1}{2}\cdot\left(M_e-M_i\right)\cdot\left(R_e^2+R_i^2\right)=\frac{1}{2}\cdot M\cdot\left(R_i^2+R_e^2\right)
\end{aligned}
$$



$$J=\frac{1}{2}\cdot M\cdot\left(R_i^2+R_e^2\right)$$

Fig. 18: Mass moment of inertia to a tube (pipe or annulus) determined about the longitudinal axis

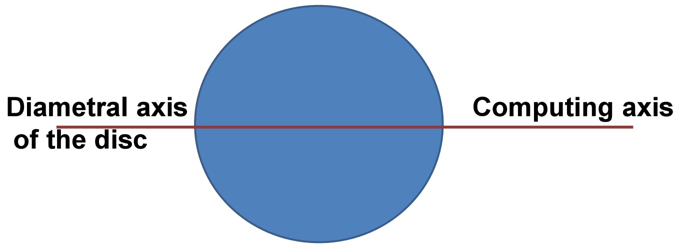
$$
\begin{aligned}
J &= \frac{1}{4}\cdot M_e\cdot R_e^2-\frac{1}{4}\cdot M_i\cdot R_i^2+\frac{1}{12}\cdot M\cdot l^2=\\
&=\frac{1}{4}\cdot\pi\cdot R_e^2\cdot l\cdot\rho\cdot R_e^2-\frac{1}{4}\cdot\pi\cdot R_i^2\cdot l\cdot\rho\cdot R_i^2+\frac{1}{12}\cdot M\cdot l^2=\\
&=\frac{1}{4}\cdot\pi\cdot l\cdot\rho\cdot\left(R_e^4-R_i^4\right)+\frac{1}{12}\cdot M\cdot l^2=\\
&=\frac{1}{4}\cdot\pi\cdot l\cdot\rho\cdot\left(R_e^2-R_i^2\right)\cdot\left(R_e^2+R_i^2\right)+\frac{1}{12}\cdot M\cdot l^2=\\
&=\frac{1}{4}\cdot\left(M_e-M_i\right)\cdot\left(R_e^2+R_i^2\right)+\frac{1}{12}\cdot M\cdot l^2=\\
&=\frac{1}{4}\cdot M\cdot\left(R_i^2+R_e^2\right)+\frac{1}{12}\cdot M\cdot l^2
\end{aligned}
$$



$$J=\frac{1}{4}\cdot M\cdot\left(R_i^2+R_e^2\right)+\frac{1}{12}\cdot M\cdot l^2$$

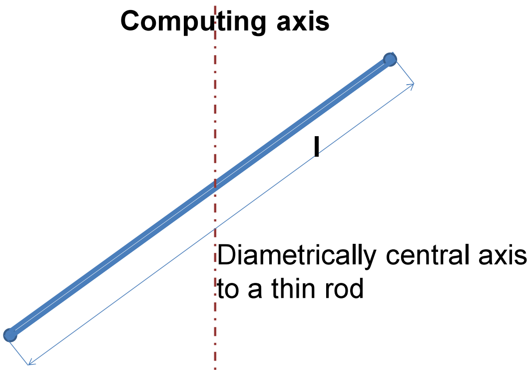
Fig. 19: Mass moment of inertia to a tube (pipe or annulus) determined about the diametrically central axis

$$J_r = \frac{1}{2} \cdot J_a = \frac{1}{2} \cdot \frac{1}{2} \cdot M \cdot R^2 =$$
$$= \frac{1}{4} \cdot M \cdot R^2$$



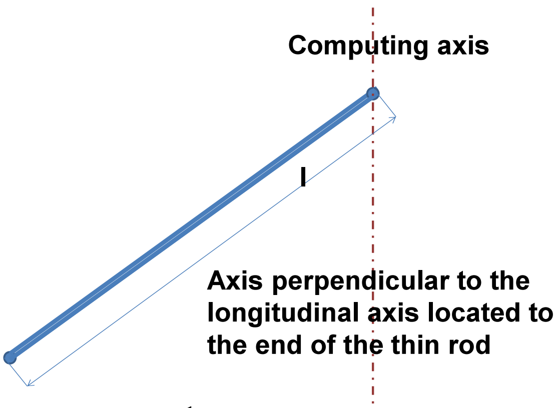
$$J = \frac{1}{4} \cdot M \cdot R^2$$

Fig. 20: Mass moment of inertia to a disc determined to the radial (diametral) axis of the disc



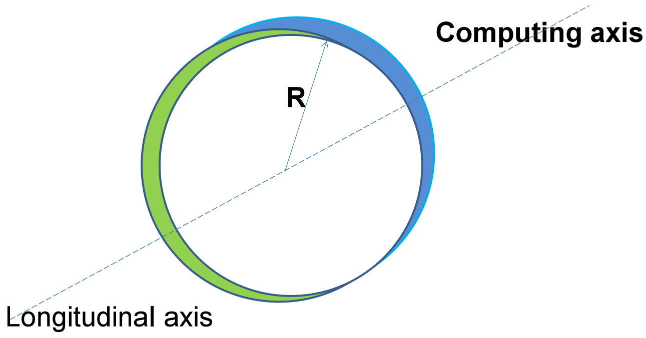
$$J = \frac{1}{12} \cdot M \cdot l^2$$

Fig. 21: Mass moment of inertia to a thin rod, led around an axis passing through a central diameter of the rod



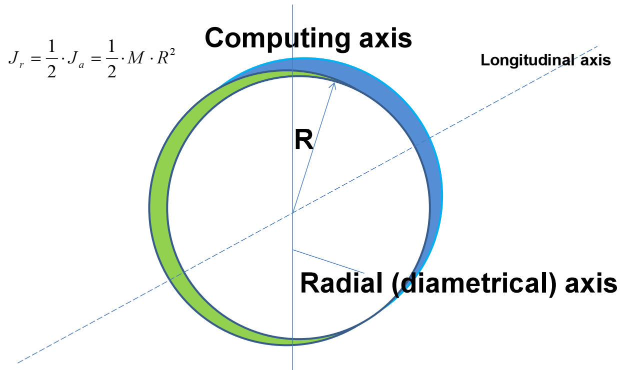
$$J = \frac{1}{3} \cdot M \cdot l^2$$

Fig. 22: Mass moment of inertia of a thin rod, determined about an axis located at one of the rod ends, perpendicular to the longitudinal axis of the stem



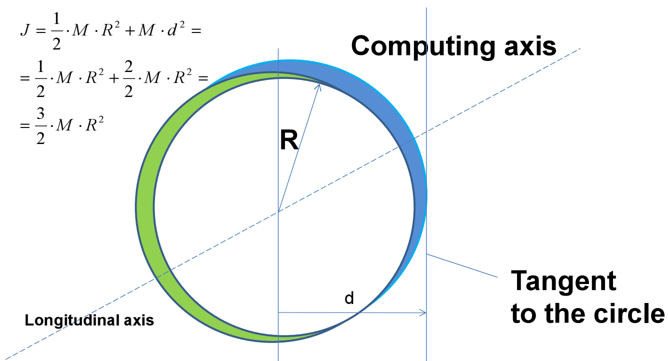
$$J = M \cdot R^2$$

Fig. 23: Mass moment of inertia of a ring fixed (calculated) around the longitudinal axis of the ring



$$J = \frac{1}{2} \cdot M \cdot R^2$$

Fig. 24: Mass moment of inertia of a ring fixed around a radial (diametrical) axis of the ring



$$J = \frac{3}{2} \cdot M \cdot R^2$$

Fig. 25: Mass moment of inertia of a ring fixed around an axis tangent to the circle of the ring

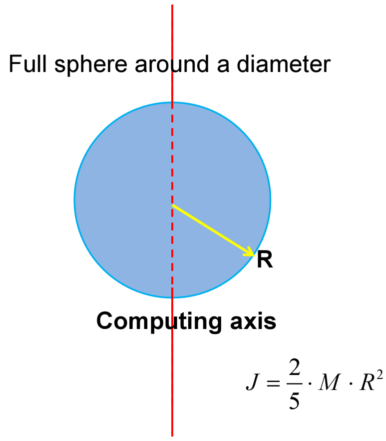


Fig. 26: Mass moment of inertia to a full sphere determined around a diameter

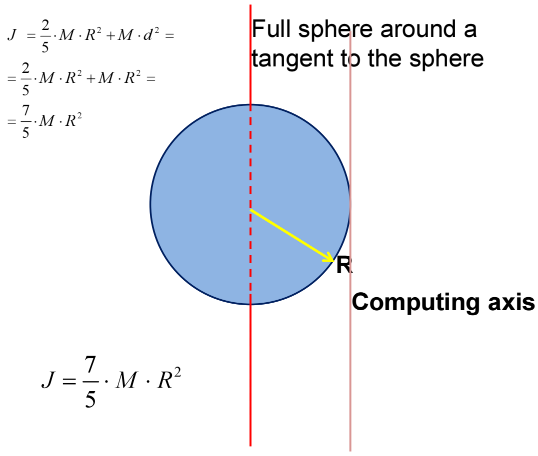


Fig. 27: Mass moment of inertia to a full sphere determined around a tangent to the sphere

New materials presented in this paper open a new perspective in the study of general mechanical dynamics (Rulkov *et al*., 2016; Agarwala, 2016; Babayemi, 2016; Gusti and Semin, 2016; Mohamed *et al*., 2016; Wessels and Raad, 2016; Maraveas *et al*., 2015; Khalil, 2015; Rhode-Barbarigos *et al*., 2015; Takeuchi *et al*., 2015; Li *et al*., 2015; Vernardos and Gantes, 2015; Bourahla and Blakeborough, 2015; Stavridou *et al*., 2015; Ong *et al*., 2015; Dixit and Pal, 2015; Rajput *et al*., 2016; Rea and Ottaviano, 2016; Zurfi and Zhang, 2016 a-b; Zheng and Li, 2016; Buonomano *et al*., 2016a; 2016b; Faizal *et al*., 2016; Ascione *et al*., 2016; Elmeddahi *et al*., 2016; Calise *et al*., 2016; Morse *et al*., 2016; Abouobaida, 2016; Rohit and Dixit, 2016; Kazakov *et al*., 2016; Alwetaishi, 2016; Riccio *et al*., 2016a; 2016b; Iqbal, 2016; Hasan and El-Naas, 2016; Al-Hasan and Al-Ghamdi, 2016; Jiang *et al*., 2016; Sepúlveda, 2016; Martins *et al*., 2016; Pisello *et al*., 2016; Jarahi, 2016; Mondal *et al*., 2016; Mansour, 2016; Al Qadi *et al*., 2016b; Campo *et al*., 2016; Samantaray *et al*., 2016; Malomar *et al*., 2016; Rich and Badar, 2016; Hirun, 2016; Bucinell, 2016; Nabilou, 2016b; Barone *et al*., 2016; Chisari and Bedon, 2016; Bedon and Louter, 2016; Santos and Bedon, 2016; Minghini *et al*., 2016; Bedon, 2016; Jafari *et al*., 2016; Chiozzi *et al*., 2016; Orlando and Benvenuti, 2016; Wang and Yagi, 2016; Obaiys *et al*., 2016; Ahmed *et al*., 2016; Jauhari *et al*., 2016; Syahrullah and Sinaga, 2016; Shanmugam, 2016; Jaber and Bicker, 2016; Wang *et al*., 2016; Moubarek and Gharsallah, 2016; Amani, 2016; Shruti, 2016; Pérez-de León *et al*., 2016; Mohseni and Tsavdaridis, 2016; Abu-Lebdeh *et al*., 2016; Serebrennikov *et al*., 2016; Budak *et al*., 2016; Augustine *et al*., 2016; Jarahi and Seifilaleh, 2016; Nabilou, 2016a; You *et al*., 2016; AL Qadi *et al*., 2016a; Rama *et al*., 2016; Sallami *et al*., 2016; Huang *et al*., 2016; Ali *et al*., 2016; Kamble and Kumar, 2016; Saikia and Karak, 2016; Zeferino *et al*., 2016; Pravettoni *et al*., 2016; Bedon and Amadio, 2016;

Chen and Xu, 2016; Mavukkandy *et al*., 2016; Yeargin *et al*., 2016; Madani and Dababneh, 2016; Alhasanat *et al*., 2016; Elliott *et al*., 2016; Suarez *et al*., 2016; Kuli *et al*., 2016; Waters *et al*., 2016; Montgomery *et al*., 2016; Lamarre *et al*., 2016; Daud *et al*., 2008; Taher *et al*., 2008; Zulkifli *et al*., 2008; Pourmahmoud, 2008; Pannirselvam *et al*., 2008; Ng *et al*., 2008; El-Tous, 2008; Akhesmeh *et al*., 2008; Nachiengtai *et al*., 2008; Moezi *et al*., 2008; Boucetta, 2008; Darabi *et al*., 2008; Semin and Bakar, 2008; Al-Abbas, 2009; Abdullah *et al*., 2009; Abu-Ein, 2009; Opafunso *et al*., 2009; Semin *et al*., 2009a; 2009b; 2009c; Zulkifli *et al*., 2009; Marzuki *et al*., 2015; Bier and Mostafavi, 2015; Momta *et al*., 2015; Farokhi and Gordini, 2015; Khalifa *et al*., 2015; Yang and Lin, 2015; Chang *et al*., 2015; Demetriou *et al*., 2015; Rajupillai *et al*., 2015; Sylvester *et al*., 2015; Ab-Rahman *et al*., 2009; Abdullah and Halim, 2009; Zotos and Costopoulos, 2009; Feraga *et al*., 2009; Bakar *et al*., 2009; Cardu *et al*., 2009; Bolonkin, 2009a; 2009b; Nandhakumar *et al*., 2009; Odeh *et al*., 2009; Lubis *et al*., 2009; Fathallah and Bakar, 2009; Marghany and Hashim, 2009; Kwon *et al*., 2010; Aly and Abuelnasr, 2010; Farahani *et al*., 2010; Ahmed *et al*., 2010; Kunanoppadon, 2010; Helmy and El-Taweel, 2010; Qutbodin, 2010; Pattanasethanon, 2010; Fen *et al*., 2011; Thongwan *et al*., 2011; Theansuwan and Triratanasirichai, 2011; Al Smadi, 2011; Tourab *et al*., 2011; Raptis *et al*., 2011; Momani *et al*., 2011; Ismail *et al*., 2011; Anizan *et al*., 2011; Tsolakis and Raptis, 2011; Abdullah *et al*., 2011; Kechiche *et al*., 2011; Ho *et al*., 2011; Rajbhandari *et al*., 2011; Aleksic and Lovric, 2011; Kaewnai and Wongwises, 2011; Idarwazeh, 2011; Ebrahim *et al*., 2012; Abdelkrim *et al*., 2012; Mohan *et al*., 2012; Abam *et al*., 2012; Hassan *et al*., 2012; Jalil and Sampe, 2013; Jaoude and El-Tawil, 2013; Ali and Shumaker, 2013; Zhao, 2013; El-Labban *et al*., 2013; Djalel *et al*., 2013; Nahas and Kozaitis, 2013; Petrescu and Petrescu, 2014a; 2014b; 2014c; 2014d; 2014e; 2014f; 2014g; 2014h; 2014i; 2015a; 2015b; 2015c; 2015d; 2015e; 2016a; 2016b; 2016c; 2016d; Fu *et al*., 2015; Al-Nasra *et al*., 2015; Amer *et al*., 2015; Sylvester *et al*., 2015b; Kumar *et al*., 2015; Gupta *et al*., 2015; Stavridou *et al*., 2015b; Casadei, 2015; Ge and Xu, 2015; Moretti, 2015; Wang *et al*., 2015).

Dynamic Kinematics to the Rod-Crank-Piston System

The cinematics of the piston crankshaft mechanism of Fig. 28 is generally known to be solved by the relationships (II.1-13):

$$\begin{cases} r\cdot\cos\varphi + l\cdot\cos\psi = -e \\ r\cdot\sin\varphi + l\cdot\sin\psi = y_B \end{cases} \tag{II.1}$$

$$\cos\psi = -\frac{e + r\cdot\cos\varphi}{l} \tag{II.2}$$

$$s = y_B = r\cdot\sin\varphi + l\cdot\sin\psi \tag{II.3}$$

$$\begin{cases} -r\cdot\dot\varphi\cdot\sin\varphi - l\cdot\dot\psi\cdot\sin\psi = 0 \\ r\cdot\dot\varphi\cdot\cos\varphi + l\cdot\dot\psi\cdot\cos\psi = \dot y_B \end{cases} \tag{II.4}$$

$$\dot\psi = -\frac{r\cdot\sin\varphi}{l\cdot\sin\psi}\cdot\dot\varphi \tag{II.5}$$

$$\dot y_B = r\cdot\dot\varphi\cdot\cos\varphi + l\cdot\dot\psi\cdot\cos\psi \tag{II.6}$$

$$\begin{cases} -r\cdot\dot\varphi^2\cdot\cos\varphi - l\cdot\dot\psi^2\cdot\cos\psi - l\cdot\ddot\psi\cdot\sin\psi = 0 \\ -r\cdot\dot\varphi^2\cdot\sin\varphi - l\cdot\dot\psi^2\cdot\sin\psi + l\cdot\ddot\psi\cdot\cos\psi = \ddot y_B \end{cases} \tag{II.7}$$

$$\ddot\psi = -\frac{r\cdot\dot\varphi^2\cdot\cos\varphi + l\cdot\dot\psi^2\cdot\cos\psi}{l\cdot\sin\psi} \tag{II.8}$$

$$\ddot y_B = l\cdot\ddot\psi\cdot\cos\psi - r\cdot\dot\varphi^2\cdot\sin\varphi - l\cdot\dot\psi^2\cdot\sin\psi \tag{II.9}$$

$$\alpha = \psi - 90 \tag{II.10}$$

$$\begin{cases} \cos\alpha = \sin\psi \\ \sin\alpha = -\cos\psi \end{cases} \tag{II.11}$$

$$\sin\alpha = \frac{e + r\cdot\cos\varphi}{l} \tag{II.12}$$

$$\begin{cases} v_B = \dot y_B = r\cdot\dot\varphi\cdot\cos\varphi + l\cdot\dot\psi\cdot\cos\psi \\ = r\cdot\dot\varphi\cdot\cos\varphi - \dfrac{r\cdot\dot\varphi\cdot\sin\varphi\cdot\cos\psi}{\sin\psi} \\ = \dfrac{r\cdot\dot\varphi}{\sin\psi}\cdot\left(\cos\phi\cdot\sin\psi - \sin\varphi\cdot\cos\psi\right) \\ = r\cdot\dot\varphi\cdot\dfrac{\sin(\psi-\varphi)}{\sin\psi} = r\cdot\omega\cdot\dfrac{\sin(\psi-\varphi)}{\sin\psi} \\ v_B = r\cdot\omega\cdot\dfrac{\sin(\psi-\varphi)}{\sin\psi} \end{cases} \tag{II.13}$$

In dynamic kinematics, the (dynamic) velocities are aligned in the direction of the forces as is natural, so that they no longer coincide with the known classical cinematic velocities (Fig. 29). Dynamic speeds due to forces occur, speeds that represent the dynamic kinematics (not to mention the influence of inertial forces, the influence that determines the final dynamic aspect of speeds).

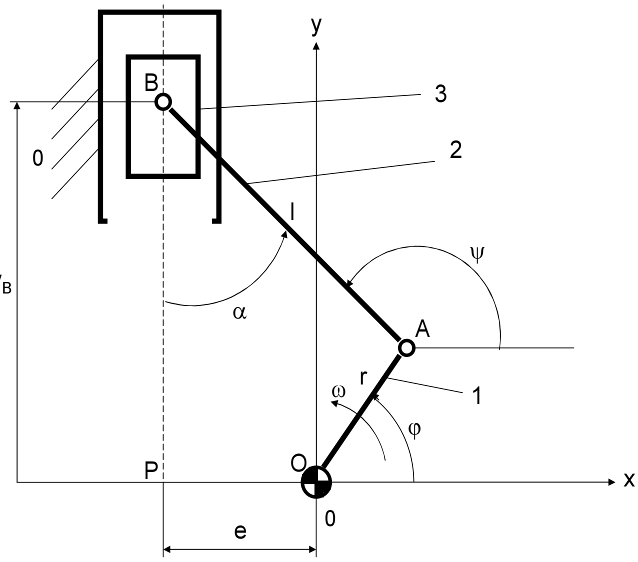


Fig. 28: The kinematic scheme of the piston crankshaft mechanism

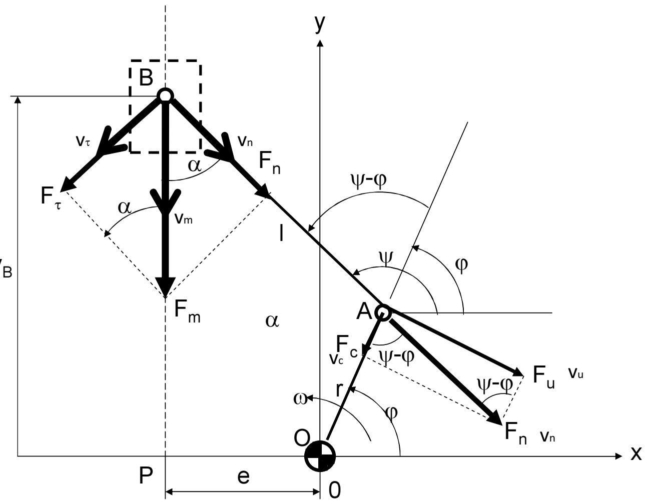


Fig. 29: Dynamic forces and speeds in the piston crank shaft mechanism, when power is transmitted from the piston to the crank

The dynamic kinematics (Petrescu, 2011) is, therefore, the kinematic study of the movements, speeds and accelerations resulting from the direction of operation of the speeds after the direction of the forces. The expressions of velocity in the dynamic kinematics are easily obtained, it is derived in relation to the time to determine the expressions of the accelerations in the dynamic kinematics and the expressions of the velocities are integrated in order to obtain the corresponding movements. Determining the movements in the dynamic cinematic becomes, therefore, more difficult.

To begin with, one will determine the speeds in the dynamic kinematics for piston cranked piston rod mechanism (Fig. 29).

One can write the relationships (II.14-16):

$$v_B = v_m \tag{II.14}$$

$$v_n = v_m \cdot \cos\alpha = v_m \cdot \sin\psi \tag{II.15}$$

$$v_u = v_n \cdot \sin(\psi - \varphi) = v_m \cdot \sin\psi \cdot \sin(\psi - \varphi) \tag{II.16}$$

One also want to find out the dynamic yield, more precisely the mechanical efficiency instantly when the mechanism has dynamic regimes and the speeds are those in the dynamic cinematic, the actuation of the mechanism being of the motor type, ie from the piston.

The useful force is determined by the relationship (II.17) presented in the previous chapter:

$$F_u = F_n \cdot \sin(\psi - \varphi) = F_m \cdot \sin\psi \cdot \sin(\psi - \varphi) \tag{II.17}$$

Puterea utilă se scrie în această situaţie sub forma II.18:

$$\begin{cases} P_u = F_u \cdot v_u \\ F_m \cdot \sin\psi \cdot \sin(\psi-\varphi) \cdot v_m \cdot \sin\psi \cdot \sin(\psi-\varphi) \\ = F_m \cdot v_m \cdot \sin^2\psi \cdot \sin^2(\psi-\varphi) \end{cases} \tag{II.18}$$

Expression of power consumed is given by relationship II.19:

$$P_c = F_m \cdot v_m \tag{II.19}$$

It can now determine the dynamic yield, more precisely instant dynamic yield (relationship II.20);

$$\eta_i^{DM} = \frac{P_u}{P_c} = \sin^2\psi \cdot \sin^2(\psi-\varphi) = \eta_i \cdot D^M \tag{II.20}$$

where, η_i is the instantaneous mechanical efficiency of the crankshaft piston actuated by the piston and D^M is a dynamic coefficient, which for the piston-driven piston rod (in Motor mode) is II.21:

$$D^M = \sin^2(\psi-\varphi) = \sin^2(\varphi-\psi) \tag{II.21}$$

In this case, let us recall that the instantaneous mechanical efficiency has the expression II.22:

$$\eta_i = \sin^2\psi \tag{II.22}$$

It should be noted that the dynamic yield is precisely the product of the known, simple (cinematic) yield and the dynamic coefficient (relation II.23):

$$\eta_i^{DM} = \eta_i \cdot D^M \tag{II.23}$$

The kinematic expression of the speed of point B (relationship II.24) is known:

$$v_m \equiv v_B = v_A \cdot \frac{\sin(\psi-\varphi)}{\sin\psi} \tag{II.24}$$

With relation II.24 introduced in formula II.16, velocity v_u takes the form II.25.

$$\begin{cases} v_u = v_n \cdot \sin(\psi-\varphi) = v_m \cdot \sin\psi \cdot \sin(\psi-\varphi) \\ = v_A \cdot \dfrac{\sin(\psi-\varphi)}{\sin\psi} \cdot \sin\psi \cdot \sin(\psi-\varphi) \\ = v_A \cdot \sin^2(\psi-\varphi) \equiv v_A^D = v_A \cdot D \end{cases} \tag{II.25}$$

$$D^M = \sin^2(\psi-\varphi) \tag{II.26}$$

It is obtained from here (from the dynamic kinematics) the expression of the dynamic coefficient D^M of the reciprocating piston piston actuated mechanism (relation II.26), noting that it is identical to the expression II.21 where the dynamic coefficient was determined based on the dynamic yield calculation immediate. This checks the uniqueness of the dynamic coefficient for the same actuated mechanism in the same way. To complete this new theory, the dynamic coefficient of the crank shaft driven by the crank shaft (in compressor mode) is to be determined further.

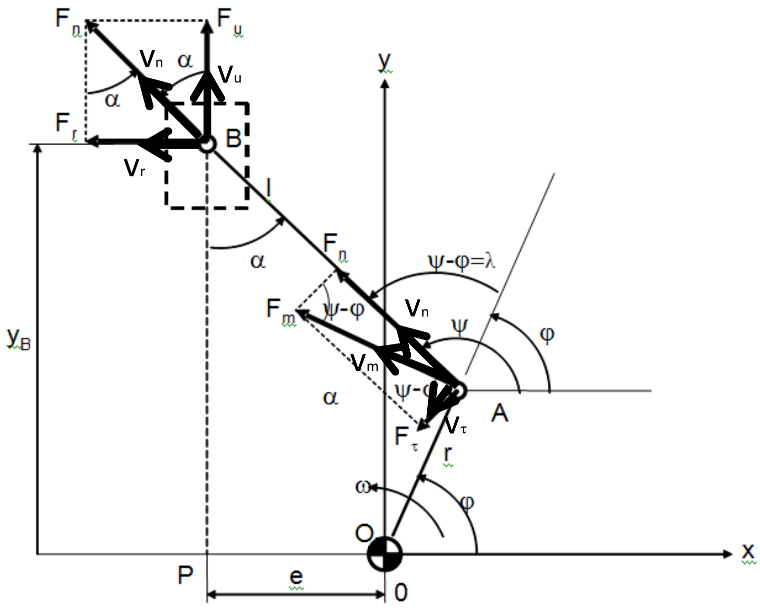


Fig. 30: Dynamic forces and speeds in a piston crankshaft system, when driven from the crank

Figure 30 shows the transmission of forces aligned with the forces, which occurs in the dynamic cinematic.

The input force F_m and the input velocity v_m decompose generating the component along the length of the rod F_n, respectively, v_n. Forces are the real forces acting on the mechanism and these cinematic-dynamic velocities are the natural ones that follow the trajectories (directions) imposed by the forces. Generally, they overlap and impose over known kinematic (static) velocities, which are calculated on the basis of the links imposed by the kinematic couple of the mechanism (depending on the kinematic chain). You can write for speeds relations II.27:

$$\begin{cases} v_B = v_A \cdot \dfrac{\sin(\psi-\varphi)}{\sin\psi}; \\ v_B^D = v_B \cdot D^C = v_A \cdot \dfrac{\sin(\psi-\varphi)}{\sin\psi} \cdot D^C \\ v_u = v_n \cdot \cos\alpha = v_n \cdot \sin\psi \\ = v_m \cdot \sin\psi \cdot \sin(\psi-\varphi) \\ = v_A \cdot \sin\psi \cdot \sin(\psi-\varphi) \\ v_u = v_B^D \Rightarrow v_A \cdot \sin\psi \cdot \sin(\psi-\varphi) \\ = v_A \cdot \dfrac{\sin(\psi-\varphi)}{\sin\psi} \cdot D^C \\ \Rightarrow D^C = \sin^2\psi \end{cases} \tag{II.27}$$

For forces, powers and yields, the following relationships are written (II.28-34):

$$\begin{cases} F_n = F_m \cdot \sin(\psi-\varphi) \\ F_\tau = F_m \cdot \cos(\psi-\varphi) \end{cases} \tag{II.28}$$

$$\begin{cases} F_u = F_n \cdot \cos\alpha = F_n \cdot \sin\psi = F_m \cdot \sin(\psi-\varphi) \cdot \sin\psi \\ F_r = F_n \cdot \sin\alpha = -F_n \cdot \cos\psi = -F_m \cdot \sin(\psi-\varphi) \cdot \cos\psi \end{cases} \tag{II.29}$$

$$\begin{cases} P_u = F_u \cdot v_B = F_m \cdot \sin(\psi-\varphi) \cdot \sin\psi \cdot \\ \dfrac{r \cdot \omega \cdot \sin(\psi-\varphi)}{\sin\psi} = F_m \cdot r \cdot \omega \cdot \sin^2(\psi-\varphi) \\ = F_m \cdot v_A \cdot \sin^2(\psi-\varphi) \end{cases} \tag{II.30}$$

$$P_c = F_m \cdot v_A = F_m \cdot r \cdot \omega \tag{II.31}$$

$$\eta_i = \frac{P_u}{P_c} = \frac{F_m \cdot v_A \cdot \sin^2(\psi-\varphi)}{F_m \cdot v_A} = \sin^2(\psi-\varphi) \tag{II.32}$$

$$\begin{cases} P_u^D = F_u \cdot v_B^D = F_m \cdot \sin(\psi-\varphi) \cdot \sin\psi \cdot v_A \cdot \sin\psi \cdot \\ \sin(\psi-\varphi) = F_m \cdot r \cdot \omega \cdot \sin^2(\psi-\varphi) \cdot \sin^2\psi \\ = F_m \cdot v_A \cdot \sin^2(\psi-\varphi) \cdot \sin^2\psi \end{cases} \tag{II.33}$$

$$\begin{cases} \eta_i^{DC} = \dfrac{P_u^D}{P_c} = \dfrac{F_m \cdot v_A \cdot \sin^2\psi \cdot \sin^2(\psi-\varphi)}{F_m \cdot v_A} \\ = \sin^2(\psi-\varphi) \cdot \sin^2\psi = \eta_i \cdot D^C \end{cases} \tag{II.34}$$

Final Discussion

The first conclusion that can be drawn is that the dynamic momentum dynamics (which is closer to the real mechanism) is less than the normal mechanical efficiency, because the dynamic yield is even the classic mechanical efficiency multiplied by the dynamic coefficient which being subunitarily results that the dynamic yield will be smaller or at most equal to the classic one.

In addition, dynamic performance is the same for crank drive and piston engine actuation and will have the same value regardless of drive type. The dynamic yield is practically uniform, but not all operating modes of the thermal motors are completely dynamic. This makes the Stirling's actual engine power or the two-stroke thermal engine (Lenoir) not much higher than the four-stroke Otto or Diesel engines. The higher the operating speeds, the operating modes become almost completely dynamic.

Today, with high and very high working speeds, four-stroke internal combustion engines achieve comparable performance to those of the Stirling engine or two-stroke engines. The higher the working speeds, the more Stirling or Lenoir benefits.

Although the dynamic mechanical performance (closest to the real) is practically calculated with the same formula regardless of the drive type, the dynamic speeds and accelerations in the couplings differ depending on the drive mode, even for the same coupling.

Thus dynamic velocities (in the dynamic kinematics) of point B are calculated with relations II.35:

$$\begin{cases} A - \textit{when action is done from the piston}: \\ D^M = \sin^2(\psi-\varphi); \eta_i = \sin^2\psi; \textit{regim Motor} \\ v_B^D = v_B \cdot D = v_A \cdot \dfrac{\sin(\psi-\varphi)}{\sin\psi} \cdot \sin^2(\psi-\varphi) \\ = v_A \cdot \dfrac{\sin^3(\psi-\varphi)}{\sin\psi} \\ v_A^D = v_A \cdot D = r \cdot \omega \cdot \sin^2(\psi-\varphi) \\ \omega^D = \omega \cdot D = \omega \cdot \sin^2(\psi-\varphi) \\ B - \textit{when action is done from the crank}: \\ D^C = \sin^2\psi; \eta_i = \sin^2(\psi-\varphi); \textit{regim Compressor} \\ v_B^D = v_B \cdot D = v_A \cdot \dfrac{\sin(\psi-\varphi)}{\sin\psi} \cdot \sin^2\psi \\ = v_A \cdot \sin(\psi-\varphi) \cdot \sin\psi \\ v_A^D = v_A \cdot D = r \cdot \omega \cdot \sin^2\psi \\ \omega^D = \omega \cdot D = \omega \cdot \sin^2\psi \end{cases} \tag{II.35}$$

Even if the dynamic output is uniform, speeds and accelerations are more linear in the crank and sharper (and vibrational) drives during the piston stroke so that four-stroke internal combustion engines are more advantageous at this point view, followed by the two-stroke (Lenoir), the last being the Stirling type engines.

Dynamic accelerations are determined with the relationships II.36, in which the dynamic velocity relation (appropriately arranged) is derived to obtain the expression of the dynamic acceleration:

$$\begin{cases} v_B^D = v_A \cdot D \cdot \dfrac{\sin(\psi-\varphi)}{\sin\psi} \Rightarrow v_B^D \cdot \sin\psi = v_A \cdot D \cdot \sin(\psi-\varphi) \\ \dot{v}_B^D \cdot \sin\psi + v_B^D \cdot \cos\psi \cdot \dot{\psi} = v_A \\ \cdot\left[\dot{D} \cdot \sin(\psi-\varphi) + D \cdot \cos(\psi-\varphi) \cdot (\dot{\psi}-\dot{\varphi})\right] \\ \Rightarrow \dot{v}_B^D = \dfrac{v_A \cdot \begin{bmatrix} \dot{D} \cdot \sin(\psi-\varphi) + D \\ \cdot\cos(\psi-\varphi)\cdot(\dot{\psi}-\dot{\varphi}) \end{bmatrix} - v_B^D \cdot \cos\psi \cdot \dot{\psi}}{\sin\psi} \\ \Rightarrow a_B^D = \dfrac{v_A}{\sin^2\psi} \\ \cdot\left[\dot{D} \cdot \sin\psi \cdot \sin(\psi-\varphi) + D \cdot \sin\psi \cdot \cos(\psi-\varphi) \cdot (\dot{\psi}-\dot{\varphi})\right. \\ \left. -D \cdot \cos\psi \cdot \sin(\psi-\varphi) \cdot \dot{\psi}\right] = \dfrac{v_A}{\sin^2\psi} \cdot \left[\dot{D} \cdot \sin\psi \cdot\right. \\ \left. \sin(\psi-\varphi) + D \cdot \dot{\psi} \cdot \sin\phi - D \cdot \dot{\varphi} \cdot \sin\psi \cdot \cos(\psi-\varphi)\right] \\ A - \text{ when action is done from the piston}: \\ D^M = \sin^2(\psi-\varphi); \dot{D}^M = 2 \\ \cdot\sin(\psi-\varphi) \cdot \cos(\psi-\varphi) \cdot (\dot{\psi}-\dot{\varphi}) \\ B - \text{ when action is done from the crank}: \\ D^C = \sin^2\psi;\ \dot{D}^C = 2 \cdot \sin\psi \cdot \cos\psi \cdot \dot{\psi} \end{cases} \quad \text{(II.36)}$$

It is seen that the dynamic disadvantages of thermal motors are in fact a contradiction. The dynamics of their mechanisms is better at crank drive (from the crankshaft), but the driving times (which have a lower dynamic kinematics) are virtually the ones needed, the only ones that produce the power (effectively) and which also generate high returns at the respective thermal engine; on the other hand, however, these engines (motor) produce not only irregular operation with shocks, vibrations and noises on the thermal engine but also generate disadvantageous features. For this reason, the four-stroke Stirling engine and two phases having each active phase show the power and load characteristic at the most disadvantageous speed.

Neither the two-stroke internal combustion engine does not have a very good feature, it also works with vibrations, vibrations and high noises, which can overcome the known beatings of four-stroke diesel engines, the traction presenting shocks (interruptions) even surpass those of the Stirling engines. The Lenoir engine does not make an engine brake, a vehicle equipped with a two-stroke thermal engine is overloaded (the brakes are getting too hot), the safety of the traffic is very low and the comfort of the passengers in the passenger compartment is much diminished.

From this point of view, the Otto or Diesel four-stroke engines are the most advantageous, the first ones actually representing the highest variant. For the Otto engines not to lose the advantage of fuel injection, many years ago it was abandoned to carburation, Otto engines being gradually fueled by the diesel fuel injection (keeps the ignition because the gasoline does not fire itself as it does diesel).

Conclusion

Only dynamic coefficient of inertia and the influence of kinematic couplings in the system were used in the analyzed sample.

The dynamic coefficient due to elasticity and deformation in the system has not been taken into account since the overwhelming influence of the inertial forces is impacted by additional dynamic changes and also by the kinematic couplings in the system and the elastic deformations do not greatly influence the dynamics of the system in the case of the example remembered.

If a robot was being discussed, things were similar, as in the case of various vehicles and various mechanisms and machines. However, for rigid memory transmissions, elastic deformations are important, which is why they should be considered in such systems.

The presented method is original and has a general character. This method replaces the older method (Petrescu, 2015b) and has a more general and unitary character, the motion equations being concise, unitary, elegant, general.

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3-Contract research. GR 69/10.05.2007: NURC in 2762; theme 8: Dynamic analysis of mechanisms and manipulators with bars and gears.

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Author's Contributions

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

Ethics

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