American Journal of Engineering and Applied Sciences

Review

Dynamic Models of Rigid Memory Mechanisms

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Article history
Received: 21-10-2018
Revised: 29-10-2018
Accepted: 19-11-2018

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Abstract: Rigid memory mechanisms have played an important role in the history of mankind, contributing greatly to the industrial, economic, social changes in society, thus leading to a real evolution of mankind. Used in automated tissue wars, in cars as distribution mechanisms, automated machines, mechanical transmissions, robots and mechatronics, precision devices and medical devices, these mechanisms have been real support for mankind along the time. For this reason, I considered useful this paper, which presents some dynamic models that played an essential role in designing rigid memory mechanisms.

Keywords: Cam Gears, Rigid Memory Mechanisms, Gearboxes, Dynamic Models

Introduction

The development and diversification of road vehicles and vehicles, especially of cars, together with thermal engines, especially internal combustion engines (being more compact, robust, more independent, more reliable, stronger, more dynamic etc.)., has also forced the development of devices, mechanisms and component assemblies at an alert pace. The most studied are power and transmission trains.

The four-stroke internal combustion engine (four-stroke, Otto or Diesel) comprises in most cases (with the exception of rotary motors) and one or more camshafts, valves, valves and so on.

The classical distribution mechanisms are robust, reliable, dynamic, fast-response and although they functioned with very low mechanical efficiency, taking much of the engine power and effectively causing additional pollution and increased fuel consumption, they could not be abandoned until the present. Another problem was the low speed from which these mechanisms begin to produce vibrations and very high noises.

Regarding the situation realistically, the mechanisms of cam casting and sticking are those that could have produced more industrial, economic, social revolutions in the development of mankind. They have contributed substantially to the development of internal combustion engines and their spreading to the detriment of external combustion (Steam or Stirling) combustion engines.

The problem of very low yields, high emissions and very high power and fuel consumption has been greatly improved and regulated over the past 20-30 years by developing and introducing modern distribution mechanisms that, besides higher yields immediately deliver a high fuel economy) also performs optimal noise-free, vibration-free, no-smoky operation, as the maximum possible engine speed has increased from 6000 to 30000 [rpm].

The paper tries to provide additional support to the development of distribution mechanisms so that their performance and the engines they will be able to further enhance.

Particular performance is the further increase in the mechanical efficiency of distribution systems, up to unprecedented quotas so far, which will bring a major fuel economy.

The current oil and energy reserves of mankind are limited. Until the implementation of new energy sources (to take real control over fossil fuels), a real alternative source of energy and fuel is even "the reduction in fuel consumption of a motor vehicle", whether we burn oil, gas and petroleum derivatives, whether we will implement biofuels first and later hydrogen (extracted from water).

The drop in fuel consumption for a given vehicle type over a hundred kilometers traveled has been consistently since 1980 and has continued to continue in the future.

Even if hybrids and electric motor cars are to be multiplied, let us not forget that they have to be charged with electricity, which is generally obtained by burning fossil fuels, especially oil and gas, in a current planetary proportion of about 60%. We burn oil in large heat plants to warm up, have domestic hot water and electricity to consume and some of that energy is extra and we add it to electric cars (electric vehicles), but the global energy problem is not resolved, the crisis even deepens. This was the case when we electrified the railroad for trains, when we generalized trams, trolleybuses and subways, consuming more electric power produced mainly from oil; oil consumption has grown a lot, its price has had a huge leap and we look at how the reserves disappear quickly.

Generally generalizing electric cars (though we are not really ready for this), we will give a new blow to oil and gas reserves.

Fortunately, biofuels, biomass and nuclear power have developed very much lately (currently based on the nuclear fission reaction). These together with the hydroelectric power plants have managed to produce about 40% of the total energy consumed globally. Only about 2-3% of global energy resources are produced by various other alternative methods (despite the efforts made so far).

This should not disarm us and abandon the implementation of solar, wind, etc.

However, as a first necessity to further reduce the share of global energy from oil and gas, the first vigorous measures that will need to be pursued will be to increase biomass and biofuels production along with the widening of the number of nuclear power plants (despite some undesirable events, which only show that nuclear fission power plants must be built with a high degree of safety and in no way eliminated from now on and they are still the one that has been so far "a bad evil ").

Alternative sources will take them on an unprecedented scale, but we expect the energy they provide to be more consistent in global percentages so that we can rely on them in a real way (otherwise, we risk that all these alternative energies remain a sort of "fairy tale").

Hydrogen fuel energy "when it starts when it stops" so there is no real time now to save energy through them, so they can no longer be priority, but the trucks and buses could even be implemented now that the storage problems have been partially solved. The bigger problem with hydrogen is no longer the safe storage, but the high amount of energy needed to extract it and especially for its bottling. The huge amount of electricity consumed for bottling hydrogen will have to be obtained entirely through alternative energy sources, otherwise hydrogen programs will not be profitable for humanity at least for the time being. Personally, I think the immediate use of hydrogen extracted from the water with alternative energies would be more appropriate for seagoing vessels.

Maybe just to say that due to his energy crisis (and not just energy, from 1970 until today), the production of cars and cars has increased at an alert pace (but naturally) instead of falling and they have and were marketed and used. The world's energy crisis (in the 1970s) began to rise from around 200 million vehicles worldwide, to about 350 million in 1980 (when the world's energy and global fuel crisis was declared), about 500 million vehicles worldwide and in 1997 the number of world-registered vehicles exceeded 600 million.

In 2010, more than 800 million vehicles circulate across the planet (Frățilă *et al*., 2011; Pelecudi, 1967; Antonescu, 2000; 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; Donald, 2003; Goodall, 2003; Graham, 2002; Jenkins, 2001; Landis and Dennis, 2005; Clément, Wikipedia; Cayley, Wikipedia; Coandă, 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

Dynamic Model with a Degree of Freedom with Double Internal Damping

In the paper (Wiederrich and Roth, 1974), there is presented a basic single-degree model with two springs and double internal damping to simulate the movement of the cam and punch mechanism (Fig. 1) and the relationships (1-2):

$$\ddot{x} + 2\xi_2\omega_2\dot{x} + \omega_2^2 x = \omega_1^2 y + 2\xi_1\omega_1\dot{y} \quad (1)$$

$$\omega_1 = \frac{K_1}{M};\omega_2 = \frac{(K_1 + K_2)}{M};$$
$$2\xi_1\omega_1 = \frac{c_1}{M};2\xi_2\omega_2 = \frac{(c_1 + c_2)}{M} \quad (2)$$

DOI: 10.3844/ajeassp.2018.1242.1257

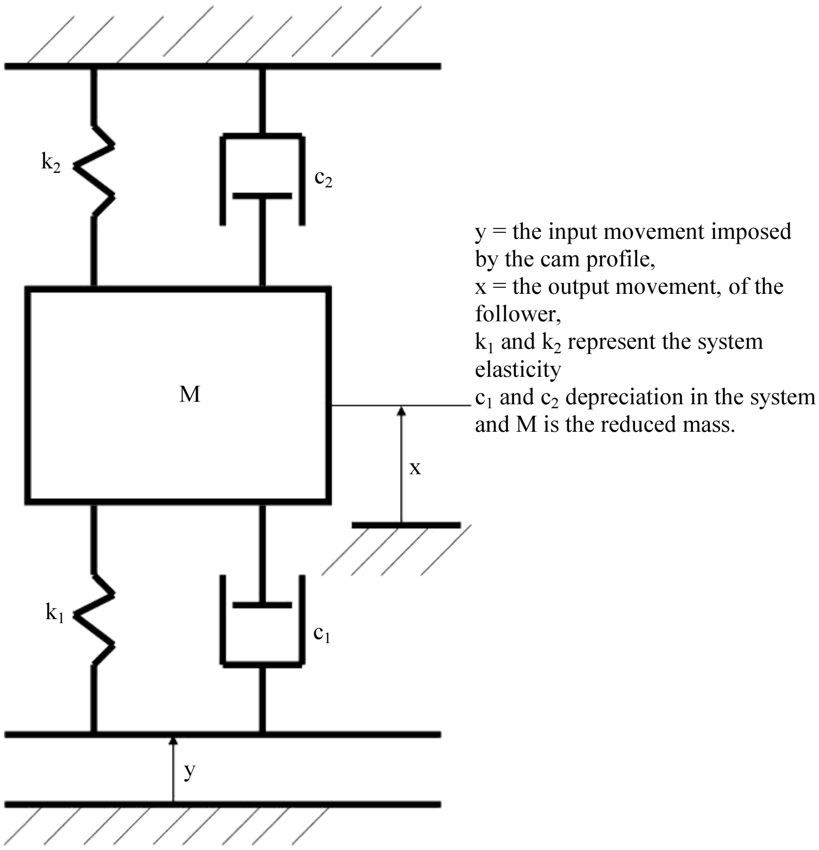

Fig. 1: Dynamic model with a degree of freedom with double internal damping

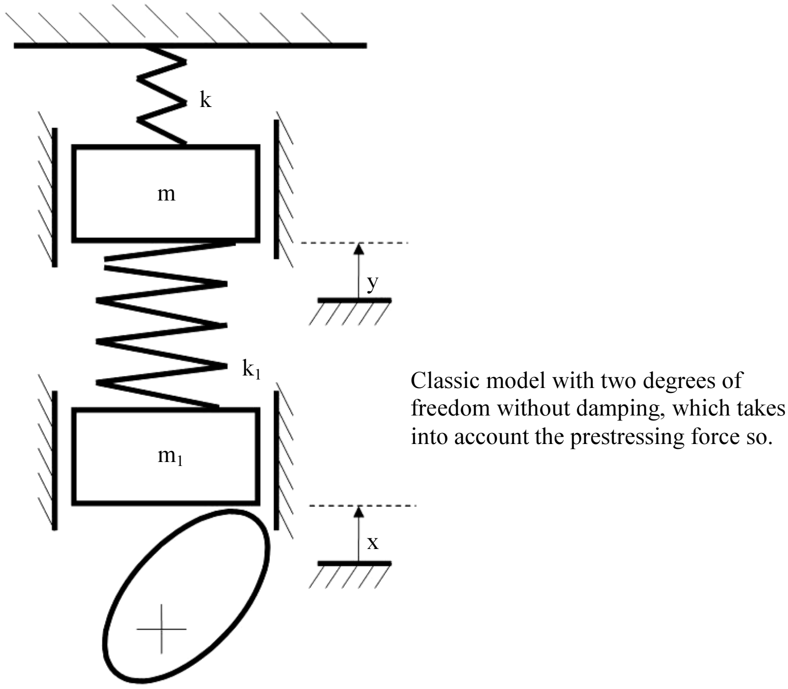

Fig. 2: Dynamic model with two degrees of freedom without internal damping

The motion equation of the proposed system (1) uses the notations (relations) in the system (2); ω_1 and ω_2 represents the system's own pulses and is calculated from the relationship system (2), depending on the elasticities K_1 and K_2 of the system in Fig. 1 and the reduced mass M of the system.

Dynamic Model with Two Degrees of Freedom without Internal Damping

In the paper (Fawcett and Fawcett, 1974), the basic dynamic model of a mechanism with cam, barrel and valve, with two degrees of freedom, without internal damping (Fig. 2, Equation 3-5) is presented:

$$y = x + z \tag{3}$$

$$m\frac{d^2y}{dt^2} + (K_1 + K)y = K_1x - s_0 \tag{4}$$

$$F_n = m_1\ddot{x} - K_1(y - x) = m_1\ddot{x} - k_1z \tag{5}$$

Dynamic Model with a Degree of Freedom with Internal and External Damping

A dynamic model with both system damping, external (spring valve) and internal damping is the one presented in the paper (Jones and Reeve, 1974), (Fig. 3).

Dynamic Model with a Degree of Freedom, Taking into Account the Internal Damping of the Valve Spring

A dynamic model with a generalized degree of freedom is presented in (Tesar and Matthew, 1974), (Fig. 4).

The motion equation is written as (6):

$$\frac{M}{K}\frac{d^2y}{dt^2} + \frac{C_r}{K}\frac{dy}{dt} + \frac{(K + K_r)}{K}y = S \tag{6}$$

Using the known relation (7), Equation (6) takes the form (8):

$$\frac{d^Ky}{dt^K} = y^{(K)}\omega^K \tag{7}$$

$$S = \mu_M y'' + \mu_C y' + \mu_K y \tag{8}$$

where the coefficients μ have the form (9):

$$\mu_M = \frac{M}{K}\omega^2; \mu_C = \frac{C_r}{K}\omega; \mu_K = \frac{(K + K_r)}{K} \cong 1, with K_r << K \tag{9}$$

The vertical reaction has the form (10):

$$F_K = K(S - y) + P = M\omega^2 y'' + C_r\omega y' + K_r y + P \tag{10}$$

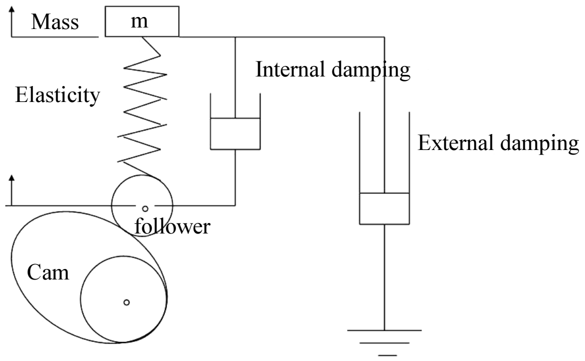

Fig. 3: Dynamic model with a degree of freedom with internal and external damping

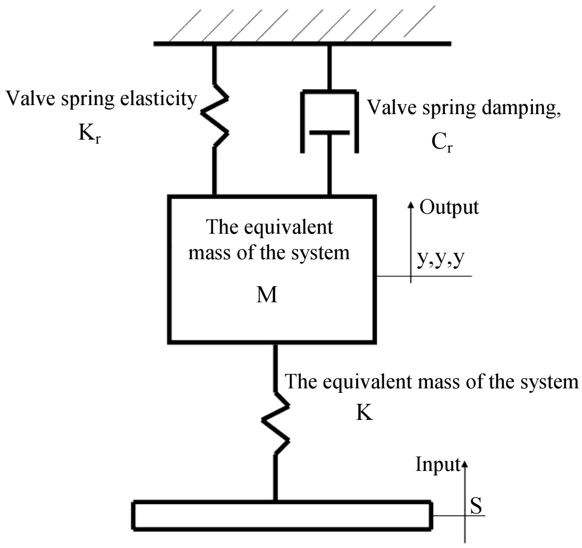

Fig. 4: Dynamic model with a degree of freedom, taking into account the internal damping of the valve spring

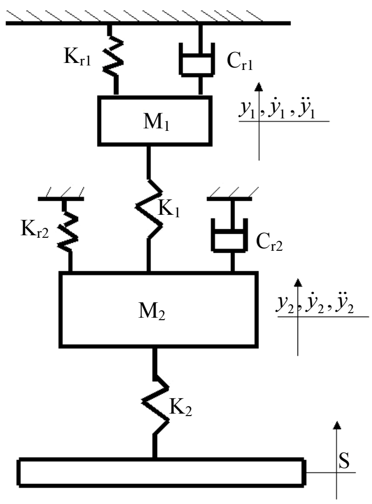

Fig. 5: Dynamic two-degree, dual damping model

Dynamic Two-Degree, Dual Damping Model

Also in the paper (Tesar and Matthew, 1974) is presented the model with two degrees of freedom (Fig. 5) with double damping.

The calculation relationships used are (11-16):

$$S = P_4 y_1'''' + P_3 y_1''' + P_2 y_1'' + P_1 y_1' + P_0 y_1 \tag{11}$$

$$P_4 = \frac{M_1 M_2}{K_1 K_2}\omega^4 \tag{12}$$

$$P_3 = \frac{\left(M_2 C_{r1} + M_1 C_{r2}\right)}{K_1 K_2}\omega^3 \tag{13}$$

$$P_2 = \frac{\left[M_2\left(K_1 + K_{r1}\right) + M_1\left(K_1 + K_2 + K_{r2}\right) + C_{r1}C_{r2}\right]}{K_1 K_2}\omega^2 \tag{14}$$

$$P_1 = \frac{\left[C_{r2}\left(K_1 + K_{r1}\right) + C_{r1}\left(K_1 + K_2 + K_{r2}\right)\right]}{K_1 K_2}\omega \tag{15}$$

$$P_0 = \frac{\left(K_1 K_{r1} + K_1 K_2 + K_2 K_{r1} + K_1 K_{r2} + K_{r1} K_{r2}\right)}{K_1 K_2} \tag{16}$$

Dynamic Model with Four Degrees of Freedom, with Torsional Vibrations

In the paper (Sava, 1970) a dynamic model with 4 degrees of freedom is proposed, obtained as follows: The model has two moving masses; these by vertical vibration each impose a degree of freedom; one mass is thought to vibrate and transverse, generating yet another degree of freedom; and the last degree of freedom is generated by torsional torsion of the camshaft (Fig. 6).

The calculation relationships are (17-20).

The first two equations resolve normal vertical vibrations, the third equation takes into account the camshaft torsional vibration and the last equation (independent of the others), the fourth, deals only with the transverse vibration of the system:

$$M\ddot{x}_1 + 2c\dot{x}_1 + \left(k + K\right)x_1 - c\dot{x}_2 - Kx_2 = -P(t) \tag{17}$$

$$\begin{aligned} &m\ddot{x}_2 + 2c\dot{x}_2 + \left(K + k_{ac}\right)x_2 \\ &-c\dot{x}_1 - Kx_1 = F_v + c\dot{s} + k_{ac}s \end{aligned} \tag{18}$$

$$\begin{aligned} &J\ddot{q} + c_r\dot{q} + k_r q - s' k_{ac} x_2 - cs'\dot{x}_2 = \\ &-s'\left(k_{ac}s + cs'\right) \end{aligned} \tag{19}$$

$$m\ddot{u} + k_t u = F_h \tag{20}$$

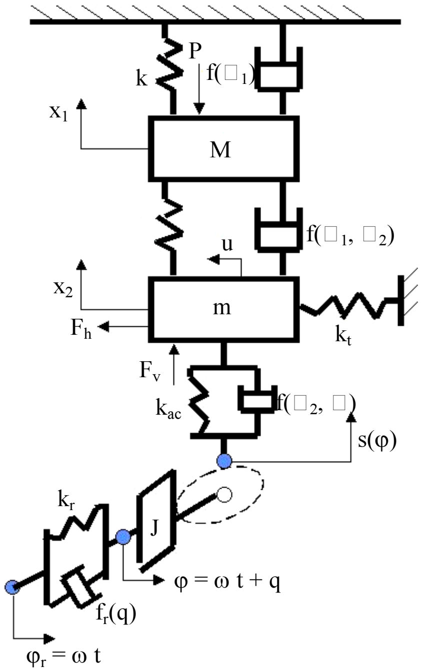

Fig. 6: Dynamic model with four degrees of freedom, with torsional vibrations

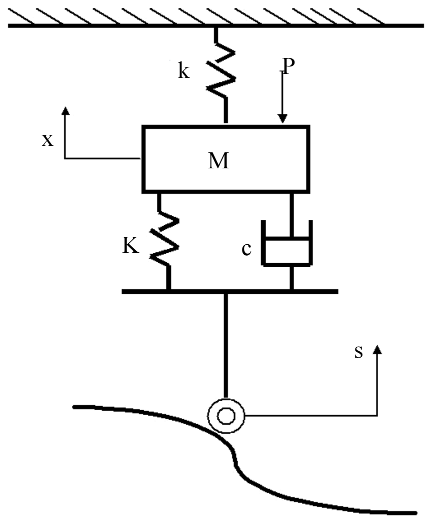

Fig. 7: Mono-dynamic damped dynamic model

Mono-Dynamic Damped Dynamic Model

Also in the paper (Sava, 1970) is presented a simplified dynamic model, amortized monomass (Fig. 7).

The motion equation used has the form (21):

$$M\ddot{x} + c\dot{x} + \left(k + K\right)x = c\dot{s} + Ks - P \tag{21}$$

Which can be written more conveniently, (22):

$$x'' = A_1(y' - x') + \omega_1^2(y - x) - F \tag{22}$$

Where the coefficients A_1, ω_1^2 and F are calculated with the expressions given in relation (23):

$$A_1 = \frac{ct_0}{M}; \omega_1^2 = \frac{(2K + k)t_0^2}{M}; F = \frac{Pt_0^2}{Ms_0} \tag{23}$$

Dynamic Damped Two-Mass Model

In Fig. 8 the bimass model proposed in the paper (Sava, 1970) is presented.

The mathematical model is written (24, 25):

$$M\ddot{x}_1 + 2c\dot{x}_1 + (k + K)x_1 - c\dot{x}_2 - Kx_2 = -P(t) \tag{24}$$

$$\begin{aligned} &m\ddot{x}_2 + 2c\dot{x}_2 + (K + k_{ac})x_2 - c\dot{x}_1 - Kx_1 \\ &= F_v + c\dot{s} + k_{ac}s \end{aligned} \tag{25}$$

Equations (24-25) can be written as:

$$x_1'' = A_1(x_2' - 2x_1') + \omega_1^2(x_2 - x_1) - F \tag{26}$$

$$\begin{aligned} &x_2'' = A_1(y' - 2x_2' + x_1') + \omega_2^2(y - x_2) + \mu\omega_1^2 x_1 \\ &+ [\mu F + (1 + \mu)y''](B_1 + B_2 y' + B_3 y) \end{aligned} \tag{27}$$

where the notations (28) were used.

$\mu = \frac{M}{m} \Rightarrow$ the ratio of the two masses, $\omega_2^2 = \frac{(k_{ac} + K)t_0^2}{m} \cong \frac{k_{ac}t_0^2}{m} \Rightarrow$ the self dimensional pulse of the mass m:

$$B_1 = \mu_1; B_2 = \frac{\mu_2 s_0}{\phi_0}; B_3 = \mu_3 s_0 \tag{28}$$

A Dynamic Model with a Single Mass with Torsional Vibrations

In Fig. 9 we can see a dynamic monosomic model that also takes into account the torsional vibrations of the camshaft (Sava, 1970).

The study points out that camshaft torsional vibrations have a negligible influence and can, therefore, be excluded from dynamic calculation models.

The same conclusion results from the work (Sava, 1971) where the torsion model is studied in more detail.

Influence of Transverse Vibrations

Tappet elasticity, variable length of the camshaft during cam operation, pressure angle variations, camshaft eccentricity, kinetic coupler friction, translation wear, technological and manufacturing errors, system gaming and other factors are factors that favor the presence of a transverse vibration of the rod weight (Sava, 1970).

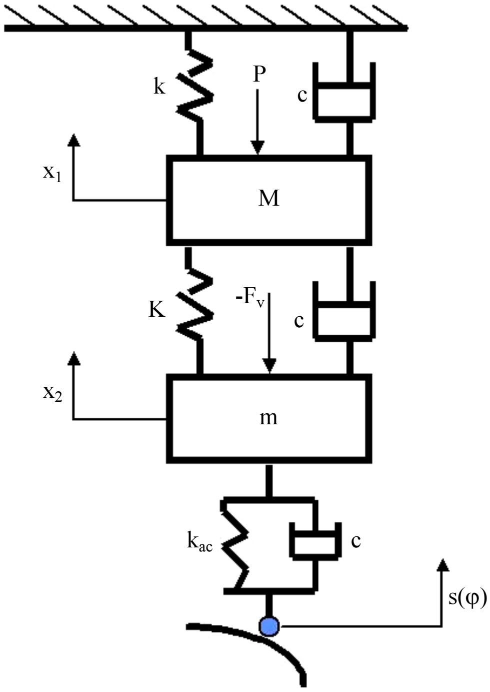

Fig. 8: Dynamic damped two-mass model

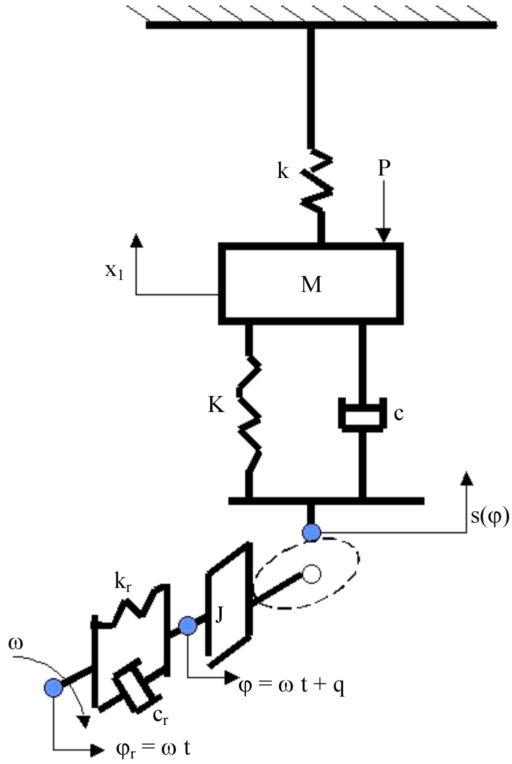

Fig. 9: A dynamic model with a single mass with torsional vibrations

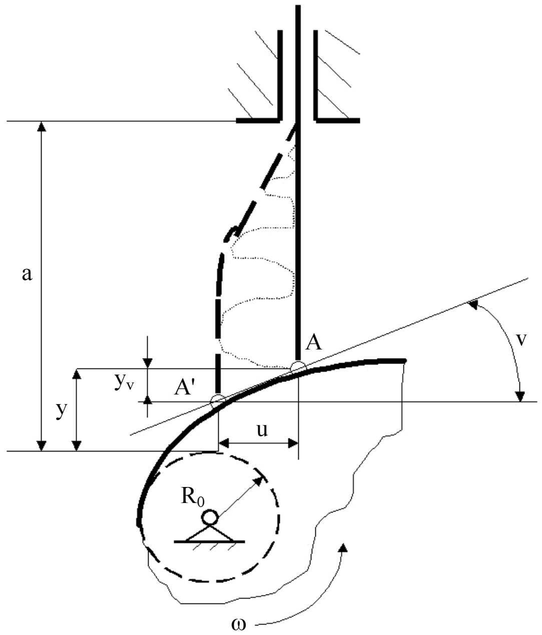

Fig. 10: Influence of transverse vibrations

In the case of high amplitude vibrations, the response parameters to the last element of the tracking system will be influenced. Following Fig. 10, it can be seen that if the curve a is the trajectory of the tip A, the point A will periodically reach point *A*', in which case the actual stroke of the y_r bar will change according to the law: $y_r = y\text{-}y_v = y\text{-}u.tgv$, where y is the longitudinal displacement of the tappet, u represents the transverse displacement of the mass m, of the tappet and v is the pressure angle. The actual stroke, y_r, will change after the law (29):

$$y_r = y - y_v = y - utg(v) \tag{29}$$

The motion equation (dimensional) is written (30):

$$u'' + \frac{A_1 u}{(1 - A_2 y)^3} = [F + (1+\mu)y''](B_{11} + B_{21}y' + B_{31}y) \tag{30}$$

where were denoted by (31) the non-dimensional constants:

$$A_1 = \frac{3EIt_0^2}{ma^3}; A_2 = \frac{s_0}{a}; \\ B_{11} = f_1 B_1; B_{21} = f_1 B_2; B_{31} = f_1 B_3 \tag{31}$$

Also in the work (Sava, 1970), the influence of the diameter of the rod, the lifting interval, the maximum length outside the tiller guides, the maximum lifting stroke and the various cam profiles on the A trajectory are analyzed.

Some conclusions:

It is noted that the reduction of the diameter of the rod of the tappet leads to the increase of the amplitude and the decrease of the average frequency of the transverse vibrations. Reducing the diameter of 1.35 times, leads to an increase in amplitude of almost three times and the average frequency decreases sensitively. Initial amplitudes are higher at the beginning of the interval, decreasing to the midpoint of the lifting interval, oscillation becoming insignificant and towards the end of the rise due to the reduction of the length a by decreasing the y stroke the frequency increases and consequently the amplitude decreases from double to simple the beginning of the interval. Increasing the stick length beyond its 2.2 to 3 cm guides leads to an increase in vibration amplitude of about 25 times.

The law of motion without leaps in the input acceleration curve reduces the amplitude of the transverse vibration of the tappet. The author of the paper (Sava, 1970) mentions that whatever the influence of the listed parameters is, for the cases considered, the amplitude values remain fairly small and in case of reduced friction in the upper coupler, they can decrease even more. Consequently, the author of the paper (Sava, 1970) concludes that the transverse vibrations of the tachet exist and must draw the attention of the constructor only in the case of exaggerated values of the constants that characterize these vibrations. Regarding the distribution of internal combustion engines, the transverse vibration can be neglected without affecting the response parameters made at the valve.

Results

Dynamic Model with Four Degrees of Freedom, with Bending Vibrations

In the paper (Koster, 1974), a four-degree dynamic model with a single oscillating motion mass is presented, representing one of four degrees of freedom. The other three freedoms result from a torsional deformation of the camshaft, a vertical bending (z), camshaft and a bending strain of the same shaft, horizontally (y), all three deformations, in a plane perpendicular to the axis of rotation (Fig. 11). The sum of the momentary efficiency and the momentary losing coefficient is 1.

The work (Koster, 1974) is extremely interesting by the model it proposes (all types of deformations are being studied), but especially by the hypothesis it advances, namely: The cams speed is not constant but variable, the angular velocity of the cam $\omega = f(\beta)$ being a function of the position of the cam (the cam angle of rotation β).

The angular velocity of the cam is a function of the position angle β (which we usually mark with φ) and its variation is caused by the three deformations (torsion and two bends) of the shaft, as well as by the angular gaps existing between the source motors (drive) and camshaft.

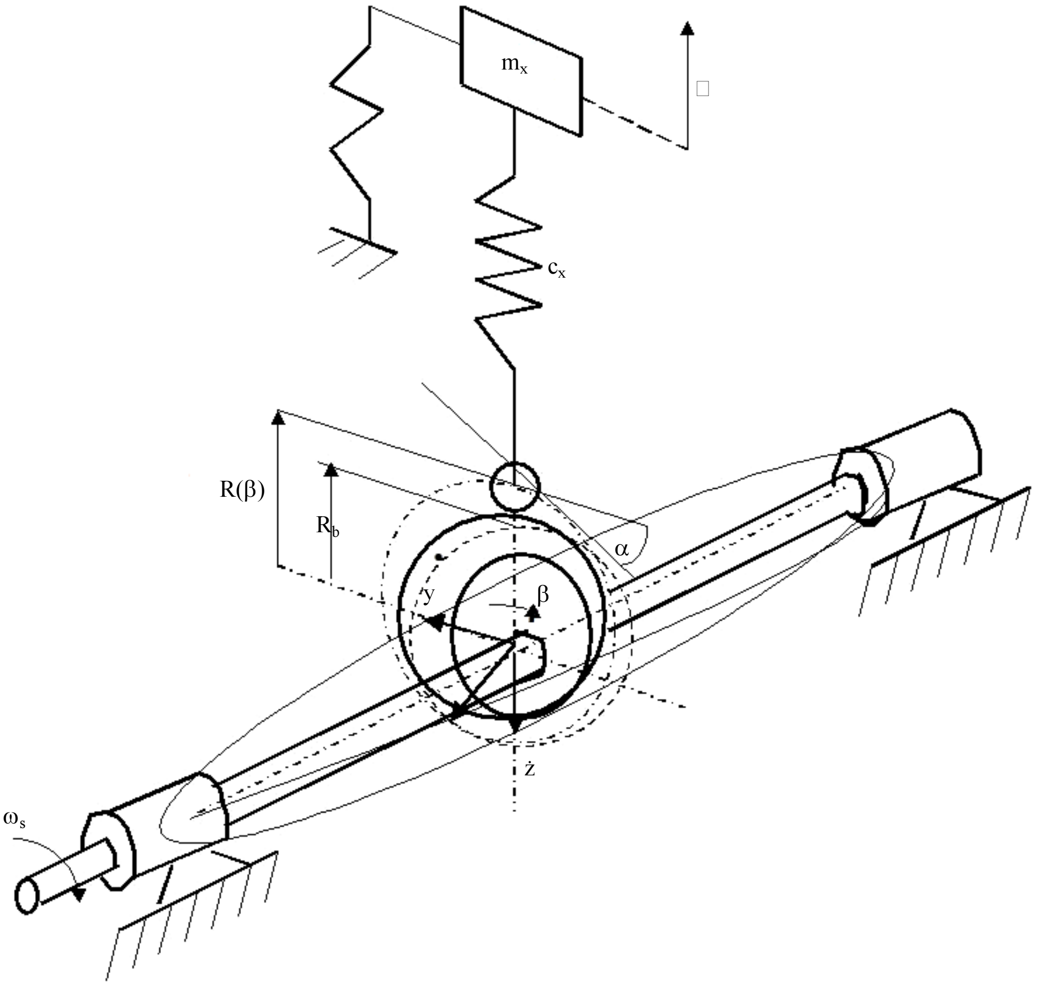

Fig. 11: Dynamic model with four degrees of freedom, with bending vibrations

The mathematical model taking into account the flexibility of the camshaft is the following; the rigidity of the cam between the cam and the cam is a function of the position β (cam angle of rotation), see the relationship (32):

$$\frac{1}{C(\beta)}=\frac{1}{C_x}+\frac{1}{C_z}+\left[\frac{1}{C_\beta(\beta)}+\frac{1}{C_y}\right]tg^2\alpha \qquad (32)$$

$$\frac{1}{C_c}=\frac{1}{C_x}+\frac{1}{C_z} \qquad (33)$$

where, $1/C_c$ see (33) is a constant rigidity given by the rigidity of the tappet (C_x) and the cam (C_z) in the direction of the tappet:

$$\frac{1}{C_{\tan}(\beta)}=\frac{1}{C_\beta(\beta)}+\frac{1}{C_y} \qquad (34)$$

And $1/C_{\tan}$ (β) see (34) represents the tangential stiffness, C_β being the torsional stiffness of the cam and C_y the flexural stiffness at the y axis of the cam, with $C_\beta(\beta)$ given by the relation (35):

$$C_\beta(\beta)=\frac{K}{\left[R(\beta)\right]^2} \qquad (35)$$

With (33) and (34) the relation (32) is rewritten in the form (36):

$$\frac{1}{C(\beta)}=\frac{1}{C_c}+\frac{tg^2\alpha}{C_{\tan}(\beta)} \qquad (36)$$

where, α is the pressure angle, which is generally a function of β and at flat tachets (used in distribution mechanisms), it has the constant value (zero): $\alpha = 0$.

The motion equation is written as (37):

$$m \cdot \ddot{x} + C(\beta) \cdot x = C(\beta) \cdot h(\beta) \quad (37)$$

where, $h(\beta)$ is the motion law imposed by the cam.

The pressure angle, α, thus influences (38):

$$tg\alpha = \frac{1}{R(\beta)} \frac{dh}{d\beta} \quad (38)$$

where, $R(\beta)$ is the current radius, which gives the cam position (distance from the center of the cam to the cam contact point) and approximates by the mean radius $R_{1/2}$. The relation (38) can be put in the form (39); Where the average radius, $R_{1/2}$, is obtained with the formula (40):

$$tg\alpha = \frac{1}{R_{1/2}} \frac{\dot{h}}{\omega_s} \quad (39)$$

$$R_{1/2} = R_b + \frac{1}{2} h_m \quad (40)$$

R_b is the radius of the base circle and h_m is the maximum projected stroke of the tappet. This produces an average radius, which is used in the calculations for simplifications; ω_s = machine angle, constant, given by machine speed. The Equation (37) can now be written (41):

$$\ddot{x} = \frac{C_c.(h(t) - x)}{m.\left[1 + \frac{C_c}{C_{\tan}}\left(\frac{1}{R_{1/2}} \frac{\dot{h}}{\omega_s}\right)^2\right]} \quad (41)$$

The solution of Equation (41) is made for $\alpha = 0$, with the following notations.

The period of natural vibration is determined with relation (42):

$$T_c = 2\pi \sqrt{\frac{m}{C_c}} \quad (42)$$

The period of the natural vibration period is obtained by the formula (43):

$$\tau = \frac{T_c}{t_m} \quad (43)$$

The slope during the lifting of the cam (44) is:

$$tg\alpha_{mc} = \frac{h_m}{R_{1/2}.\beta_m} \quad (44)$$

The shaft stiffness factor is obtained by the formula (45):

$$F_a = \frac{C_c}{C_{\tan}} tg^2\alpha_{mc} \quad (45)$$

With dimensional parameters given by (46):

$$H = \frac{h}{h_m}; X = \frac{x}{h_m}; T = \frac{t}{t_m}; \dot{H} = \frac{h}{h_m} t_m; \ddot{X} = \frac{x}{h_m} t_m^2 \quad (46)$$

The motion equation is written in the form (47):

$$\ddot{X} = \left(\frac{2\pi}{\tau}\right)^2 . \frac{H - X}{1 + \dot{H}^2.F_a} \quad (47)$$

The nominal curve of the cam is known (48) and (49):

$$\dot{H} = \dot{H}(T) \quad (48)$$

$$H = H(T) \quad (49)$$

With (47), (48) and (49) the dynamic response is calculated by a numerical method.

The author of the paper (Koster, 1974) gives a numerical example for a motion law, corresponding to the cycloid cam (50):

$$H = T - \frac{1}{2\pi} \sin(2\pi.T) \quad (50)$$

The work is especially interesting in how it manages to transform the four degrees of freedom into one, ultimately using a single equation of motion along the main axis. The dynamic model presented can be used wholly or only partially, so that on another classical or new dynamic model, the idea of using deformations on different axes with their cumulative effect on a single axis is inserted.

Discussion

The development and diversification of road vehicles and vehicles, especially of cars, together with thermal engines, especially internal combustion engines (being more compact, robust, more independent, more reliable, stronger, more dynamic etc.)., has also forced the development of devices, mechanisms and component assemblies at an alert pace. The most studied are power and transmission trains.

The four-stroke internal combustion engine (four-stroke, Otto or Diesel) comprises in most cases (with the exception of rotary motors) and one or more camshafts, valves, valves and so on.

The classical distribution mechanisms are robust, reliable, dynamic, fast-response and although they functioned with very low mechanical efficiency, taking much of the engine power and effectively causing additional pollution and increased fuel consumption, they could not be abandoned until the present. Another problem was the low speed from which these mechanisms begin to produce vibrations and very high noises.

Regarding the situation realistically, the mechanisms of cam casting and sticking are those that could have produced more industrial, economic, social revolutions in the development of mankind. They have contributed substantially to the development of internal combustion engines and their spreading to the detriment of external combustion (Steam or Stirling) combustion engines.

The problem of very low yields, high emissions and very high power and fuel consumption has been greatly improved and regulated over the past 20-30 years by developing and introducing modern distribution mechanisms that, besides higher yields immediately deliver a high fuel economy) also performs optimal noise-free, vibration-free, no-smoky operation, as the maximum possible engine speed has increased from 6000 to 30000 [rpm].

The paper tries to provide additional support to the development of distribution mechanisms so that their performance and the engines they will be able to further enhance.

Particular performance is the further increase in the mechanical efficiency of distribution systems, up to unprecedented quotas so far, which will bring a major fuel economy.

The current oil and energy reserves of mankind are limited. Until the implementation of new energy sources (to take real control over fossil fuels), a real alternative source of energy and fuel is even "the reduction in fuel consumption of a motor vehicle", whether we burn oil, gas and petroleum derivatives, whether we will implement biofuels first and later hydrogen (extracted from water).

The drop in fuel consumption for a given vehicle type over a hundred kilometers traveled has been consistently since 1980 and has continued to continue in the future.

Even if hybrids and electric motor cars are to be multiplied, let us not forget that they have to be charged with electricity, which is generally obtained by burning fossil fuels, especially oil and gas, in a current planetary proportion of about 60%. We burn oil in large heat plants to warm up, have domestic hot water and electricity to consume and some of that energy is extra and we add it to electric cars (electric vehicles), but the global energy problem is not resolved, the crisis even deepens. This was the case when we electrified the railroad for trains, when we generalized trams, trolleybuses and subways, consuming more electric power produced mainly from oil; oil consumption has grown a lot, its price has had a huge leap and we look at how the reserves disappear quickly.

Generally generalizing electric cars (though we are not really ready for this), we will give a new blow to oil and gas reserves.

Fortunately, biofuels, biomass and nuclear power have developed very much lately (currently based on the nuclear fission reaction). These together with the hydroelectric power plants have managed to produce about 40% of the total energy consumed globally. Only about 2-3% of global energy resources are produced by various other alternative methods (despite the efforts made so far).

This should not disarm us and abandon the implementation of solar, wind, etc.

However, as a first necessity to further reduce the share of global energy from oil and gas, the first vigorous measures that will need to be pursued will be to increase biomass and biofuels production along with the widening of the number of nuclear power plants (despite some undesirable events, which only show that nuclear fission power plants must be built with a high degree of safety and in no way eliminated from now on and they are still the one that has been so far "a bad evil").

Alternative sources will take them on an unprecedented scale, but we expect the energy they provide to be more consistent in global percentages so that we can rely on them in a real way (otherwise, we risk that all these alternative energies remain a sort of "fairy tale").

Hydrogen fuel energy "when it starts when it stops" so there is no real time now to save energy through them, so they can no longer be priority, but the trucks and buses could even be implemented now that the storage problems have been partially solved. The bigger problem with hydrogen is no longer the safe storage, but the high amount of energy needed to extract it and especially for its bottling. The huge amount of electricity consumed for bottling hydrogen will have to be obtained entirely through alternative energy sources, otherwise hydrogen programs will not be profitable for humanity at least for the time being. Personally, I think the immediate use of hydrogen extracted from the water with alternative energies would be more appropriate for seagoing vessels.

Maybe just to say that due to his energy crisis (and not just energy, from 1970 until today), the production of cars and cars has increased at an alert pace (but naturally) instead of falling and they have and were marketed and used. The world's energy crisis (in the 1970s) began to rise from around 200 million vehicles worldwide, to about 350 million in 1980 (when the world's energy and global fuel crisis was declared), about 500 million vehicles worldwide and in 1997 the number of world-registered vehicles exceeded 600 million.

Also in the work (Sava, 1970), the influence of the diameter of the rod, the lifting interval, the maximum length outside the tiller guides, the maximum lifting stroke and the various cam profiles on the A trajectory are analyzed.

It is noted that the reduction of the diameter of the rod of the tappet leads to the increase of the amplitude and the decrease of the average frequency of the transverse vibrations. Reducing the diameter of 1.35 times, leads to an increase in amplitude of almost three times and the average frequency decreases sensitively. Initial amplitudes are higher at the beginning of the interval, decreasing to the midpoint of the lifting interval, oscillation becoming insignificant and towards the end of the rise due to the reduction of the length a by decreasing the y stroke the frequency increases and consequently the amplitude decreases from double to simple the beginning of the interval. Increasing the stick length beyond its 2.2 to 3 cm guides leads to an increase in vibration amplitude of about 25 times.

Conclusions

Rigid memory mechanisms have played an important role in the history of mankind, contributing greatly to the industrial, economic, social changes in society, thus leading to a real evolution of mankind.

Used in automated tissue wars, in cars as distribution mechanisms, automated machines, mechanical transmissions, robots and mechatronics, precision devices and medical devices, these mechanisms have been real support for mankind along the time.

For this reason, one considered useful this paper, which presents some dynamic models that played an essential role in designing rigid memory mechanisms.

Acknowledgement

This text was acknowledged and appreciated by Dr. Veturia CHIROIU Honorific member of Technical Sciences Academy of Romania (ASTR) PhD supervisor in Mechanical Engineering.

Funding Information

Research contract: Contract number 36-5-4D/1986 from 24IV1985, beneficiary CNST RO (Romanian National Center for Science and Technology) Improving dynamic mechanisms internal combustion engines. All these matters are copyrighted. Copyrights: 548-cgiywDssin, from: 22-04-2010, 08:48:48.

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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