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Dynamics of Buses - Part I

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Abstract: Dynamics, or dynamic processes, is the part of mechanics dealing with the study of processes trying to describe as real as possible the movement of a body, element, mechanism, car, etc., also taking into account the action of the forces on the respective system with their influence on the actual movement of system. The present paper aims to present the study of the dynamics of the vehicles, with particularization on the buses. Here are the main elements of the bus dynamics, taking into account all the elements that influence the dynamic operation of a bus, in general and in particular situations, with emphasis on the main systems and elements that act on the actual, dynamic, on a normal path or on an inclined with an alpha angle path.

Keywords: Mechanisms, Machines, Buses, Dynamics, Kinematics

Introduction

Transport is the movement of persons as well as goods, signals or information from one place to another. The term comes from Latin, from "transport", trans (over) and porting (meaning wearing or carrying).

Transport is an activity that arose with the existence of man. The physical limits of the human body in terms of walking distances and the quantity of goods that could be transported led, over time, to the discovery of a variety of ways and means of transport.

Transport facilitates access to natural resources and stimulates trade.

The transport sector has different aspects. Simplifying and generalizing can be discussed by three major branches: Infrastructure, vehicles, management:

Transport infrastructure, including the entire transport network (streets, motorways, railways, waterways, flight color, pipelines, etc.) and terminals (airports, railway stations, bus stations, etc.).

Vehicles of all types: Motor vehicles, trains, ships, airplanes, etc., together with all aspects related to vehicle design, construction, diagnosis and exploitation, road traffic, management.

Transport management is the responsibility of transport engineering and engineering for the design of transport networks and systems, aiming at optimizing transport systems, increasing transport safety, protecting the environment, etc.

Land transport is the most widespread form of transport. People can move by their own forces or using means of transport that use human power, such as a bicycle, or they can use animal traction to pull wagons or other types of carriages. The most widespread and efficient form of land transport uses vehicles equipped with liquid-fueled engines (Frăţilă *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).

Materials and Methods

Power, Moment and Forces at the Wheels of the Bus

From the engine 1 (Fig. 1), the power is transmitted by the clutch 2, the transaxle 3 of the longitudinal transmission 4 of the main planetary transmission 5 to the drive wheels 6. The effective power On the motor does not reach integral to the wheels of the bus due to the losses occurring in the transmission organs. The power at the wheel P_r (corresponding to all engine wheels) will be given by relation 1, where the mechanical transmission efficiency is denoted by η_t:

$$P_r = \eta_t \cdot P_e \tag{1}$$

If in relation 1 we introduce instead of the powers their expressions according to the moments and the corresponding angular velocities we obtain in the relational system 2 the moment expression according to the angular velocities, which can also be presented according to the speed, instead of the angular speeds of the motor and respectively of the motor wheels belonging to the bus. Where: M_e and M_r are the moments Motor and total wheel respectively in [*daN.m*], ω Angular speed of the motor (motor shaft), ω_r Angular speed of the drive wheels in [s^{-1}], n The speed of the spindle in [rpm] and the engine speed. The total transmission ratio, between the engine and the engine wheels, is dimensionless.

The $M_{r'}$ moment returns to a single engine wheel is given by relationship 3, in which i' is the number of the motor (driving) wheels of the bus:

$$\begin{cases} P_e = M_e \cdot \omega; P_r = M_r \cdot \omega_r; \\ \omega = \dfrac{2\pi \cdot n}{60} = \dfrac{\pi \cdot n}{30}; i_t = \dfrac{n}{n_r} \\ P_r = \eta_t \cdot P_e \Rightarrow M_r = M_e \cdot \eta_t \cdot \dfrac{\omega}{\omega_r} \\ M_r = M_e \cdot \eta_t \cdot \dfrac{n}{n_r} \\ M_r = M_e \cdot \eta_t \cdot i_t \end{cases} \tag{2}$$

$$M'_r = \frac{M_r}{i'} \tag{3}$$

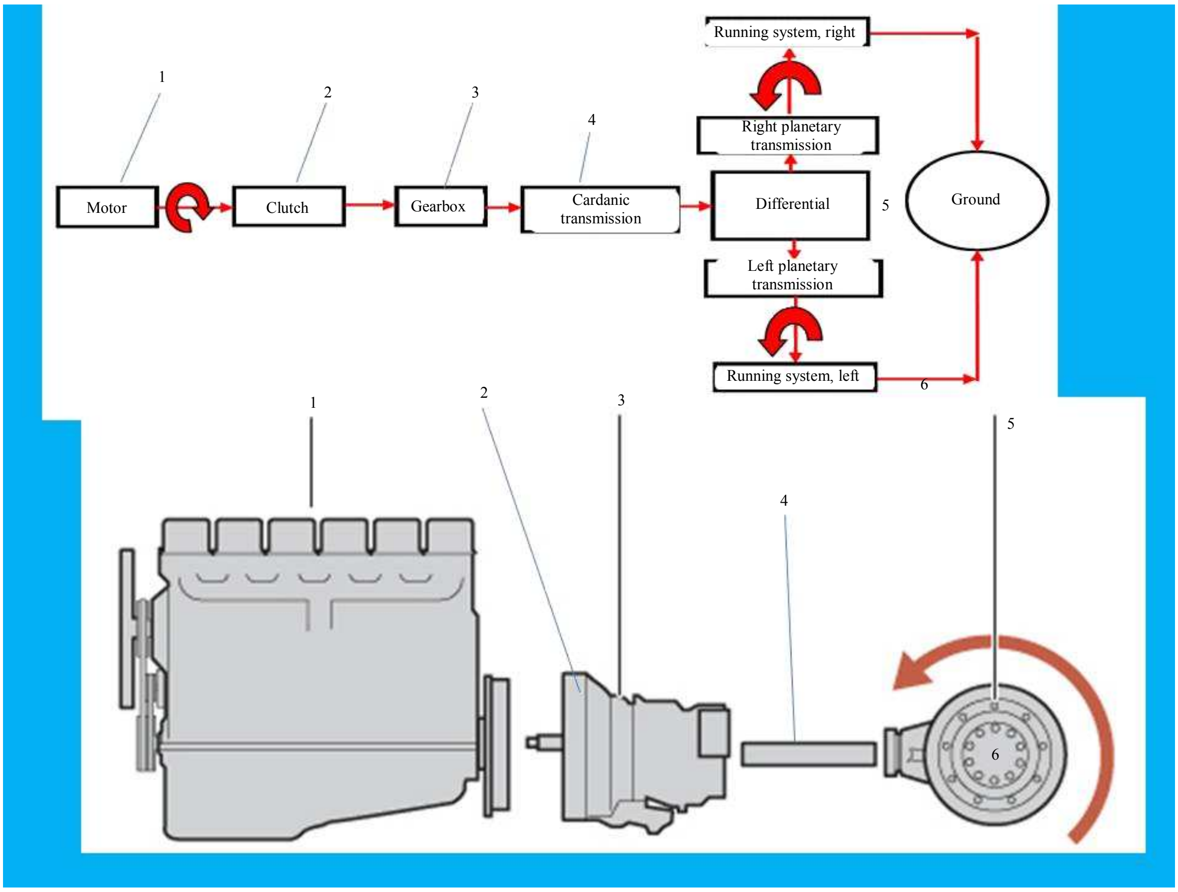


Fig. 1: Scheme of transmission of motion from the engine to the wheels of the bus

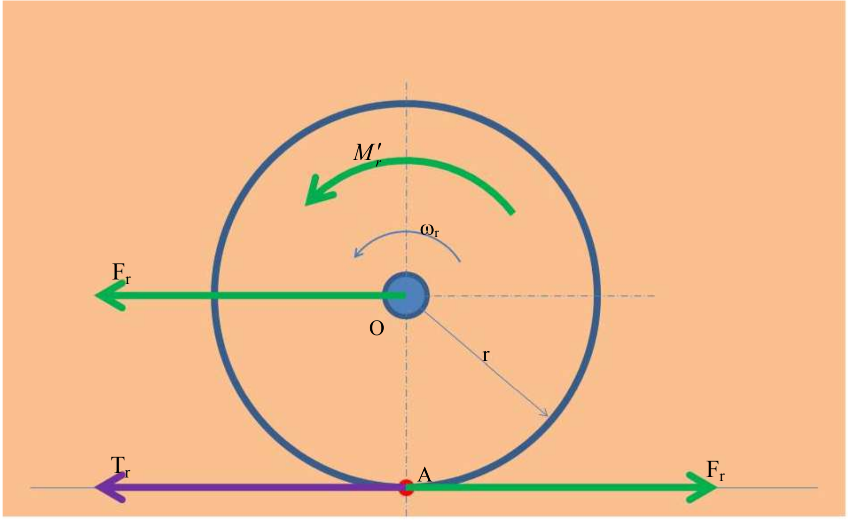


Fig. 2: Moment and forces on an engine (driving) wheel

The total bus transmission ratio is given by relation 4, where i_{cv} is the transmission ratio of the gearbox, i_0 is the transmission ratio of the main transmission, if the transmission ratio of the final transmission:

$$i_t = i_{cv} \cdot i_0 \cdot i_f \tag{4}$$

The torque M'_r which returns to a single drive wheel can be replaced by the torque of two forces F_r applied one in the center *O* of the bus engine (parallel to the tread) and the second applied at the point *A* between the wheel and a path tangent to *A* on the wheel circle and on the tread (Fig. 2, relationship 5), where r is the radius of the drive wheel:

$$F_r = \frac{M_r^{'}}{r} \tag{5}$$

The F_r force at the periphery of the drive wheel is called the wheel force. Under the action of force F_r, the T_r*'s* reaction of the rolling path, equal to and opposite to the force F_r, will be generated, thus preventing the wheel from sliding on the tread. The force F_r at the center of the wheel is transmitted (via the wheel bearing) to the bus, producing its displacement.

According to the definition, the total pulling force is given by the relation (6):

$$F_t = \sum F_r = \sum \frac{M_r^{'}}{r} = \frac{\sum M'_r}{r} = \frac{M'_r \cdot i'}{r} = \frac{M_r}{r} = \frac{M_e \cdot i_t}{r} \cdot \eta_t \tag{6}$$

The character of the bus movement is given by the force of traction in relation to the size of the resisting forces opposing its displacement.

Knowing the speed of the crankshaft *n*, i_t total transmission ratio and the rolling radius of the engine wheels r_r, can determine the speed of the bus 7:

$$\begin{aligned} v &= r_r \cdot \omega_r = r_r \cdot \frac{\omega}{i_0 \cdot i_{cv} \cdot i_f} = \frac{\pi \cdot n \cdot r_r}{30 \cdot i_0 \cdot i_{cv} \cdot i_f}\left[ms^{-1}\right] \\ &= 0{,}377 \frac{r_r \cdot n}{i_0 \cdot i_{cv} \cdot i_f}[km/h] \end{aligned} \tag{7}$$

Note: The relationship (7) is valid for as long as clutch slippage and/or some of the bus wheels do not occur.

Adhesion of Wheels

Under the action of a M'_r motor torque, a wheel advances only if a tangential reaction T_r (Fig. 3) is present in the contact point A with the tread that represents the action of the running wheel on the wheel of the bus.

The maximum value of the Trmax tangential response for which there is no slip or sliding of the wheel is called adhesion.

Strength of road, concrete, asphalt, etc. (hard surface) occurs almost entirely by the friction between the wheel tire and the tread. For deformable surfaces, the grip force is due to both the friction and the intersection between the tire protrusions and the track. The more tires are the more desirable the tire-to-rail tire is larger than the theoretical point in the drawing and the adhesion increases but is also resistant to advancing, thus producing greater energy and fuel consumption but also faster wear tires and a more difficult ride, with the clear benefits of increased adherence and enhanced bus comfort (lower vibrations and noise) along with increased dynamic stability.

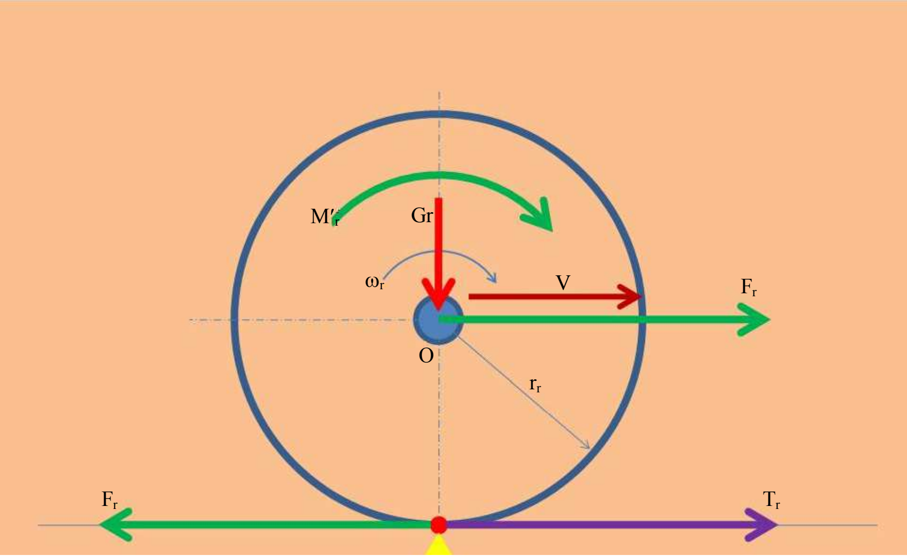


Fig. 3: Forces and reactions to the drive wheel

The ratio of the T_{rmax} adhesion value to the normal Z_r of the soil on the wheel is called the coefficient of adhesion to the wheel and is given by the relation (8), where φ is the coefficient of adhesion and Z_r is the vertical reaction of the rolling path equal to the mobile weight of the G_r (reversed than this):

$$\varphi = \frac{T_{r\max}}{Z_r} \tag{8}$$

The value of the F_{ad} adhesion force for all bus wheels of the bus shall be denoted by T_{max} *and* shall be equal to their sum (9):

$$T_{\max} = \sum T_{r\max} \equiv F_{ad} \tag{9}$$

For the entire bus, the tractive force F_t will not exceed the adhesion value for all the engine wheels (10), i.e.:

$$F_t \leq T_{\max}, or\, F_t \leq F_{ad} \tag{10}$$

The adhesion coefficient can now be written also according to the total adhesion force T_{max} corresponding to all the bus wheels of the bus (11), the G_{ad} being only the part of the weight of the bus allocated to the motor wheels, i.e., a sum of all weights G_r; of the relation (11) it is easy to observe that for a given road with the tires already chosen, i.e., for a φ imposed coefficient, it is possible to increase the total adhesion of the T_{max} bus only by increasing the G_{ad} force by increasing the number of engine wheels (motor bridges) (12):

$$\varphi = \frac{T_{r\max}}{G_r} = \frac{T_{\max}}{G_{ad}} \tag{11}$$

Table 1: Average coefficient of adhesion φ

	The state of the path --------------------------------	
Way of covering the path	Dry	Wet
Paved road	0,7-0,8	0,3-0,4
Cobbled road	0,5-0,6	0,3-0,4
Dirt road	0,5-0,6	0,3-0,4
Sand	0,5-0,6	0,4-0,5
Road covered with snow	0,2-0,4	
Ice Road	0,10-0,15	

$$T_{\max} = \varphi \cdot G_{ad} \tag{12}$$

The coefficient of adhesion φ depends mainly on the following factors: The nature and condition of the tread, the sliding of the tire and the track, the inner tire pressures, the shape of the tread, the speed of the bus, the total weight and the wheel of the bus, the type of tires and how they were manufactured (here an important role with the materials used).

Table 1 shows the mean coefficient of adhesion φ according to the nature and condition of the tread.

It can be noticed that, while the tread is wet, the coefficient of adhesion decreases drastically (almost half is halved, except for the sand that behaves better in the presence of liquid water). If we have snow, adhesion decreases even more and under the conditions of poll it diminishes dangerously long, having almost no importance of what the road is built, when it is covered by snow or ice.

Running Ray of the Bus Wheel

The wheels of the current buses (as well as those of the trucks) are equipped with tires that are elastically deformed in the radial, tangential and lateral direction more than for cars or other road vehicles due to the relatively high size of the bus tires but also because of the weight large buses that the wheel and the tire have to bear dynamically. For this reason, they are designed to have a higher elasticity and increased strength and stiffness. As with any other tire-covered vehicle and on buses, the tire's rolling radius and therefore the wheel of the wheel is not a constant one but a variable, dynamic one, the main variation due to the variable dynamic loads it carries permanent.

On a bus wheel the following radii are distinguished: Nominal radius, free radius, static radius, dynamic radius and rolling radius.

The nominal radius of one wheel is the radius of the outer circle of the tire, theoretically deducted by calculation or taken from the tables. The nominal diameters of the bus tires are given in the tables of the international manufacturers, or in the tire standard.

If these tables or chambers are missing, the nominal diameter can be deduced by calculation based on the notations marked on that tire. Based on the tire notation, the nominal radii of the wheels can be calculated. For high pressure tires used on buses, trolleybuses and trucks, the nominal (or external) diameter D_e (or D_u) is marked directly on the tire, the nominal radius being calculated simply by dividing it into two (Fig. 4, top right).

The free radius r_0 of a wheel is the radius of the outer circle of the tire tread at normal pressure, measured without any resting load (Fig. 5a).

The static radius rs of a wheel is defined as the distance between the center of the wheel and the support surface if the wheel is loaded with the normal load and is at rest. The static rays depend primarily on the G_r load and on the inner tire pressure (Fig 5b).

The dynamic radius of a wheel is defined as the distance between the center of the wheel and the support surface during motion under the action of normal load and torque or tangential force. Under the action of a moment M_r' will result in a tangential deformation of the tire which results in the center of the wheel being approached by the surface of the tread (Fig. 5c).

The dynamic range of a tire depends on a number of factors: Normal load, torque size, tire pressure, tire material elasticity, etc.

The rolling radius r_r of a wheel is the radius of a non-deformable conventional wheel running without sliding or skidding at the same rotation speed n_r and velocity of translation at the center of the wheel (13) where v is the linear displacement velocity of the center real wheel, ω_r is the angular velocity of the real wheel but also that of the conventional (calculation) wheel and n is the speed of the two wheels (real and conventional):

$$r_r = \frac{v}{\omega_r} = \frac{30 \cdot v}{\pi \cdot n_r} \tag{13}$$

The value of the tread depends on the following factors: The inner tire pressure, the angular speed of the drive wheels, the load on the drive wheel and the rigidity of the tire.

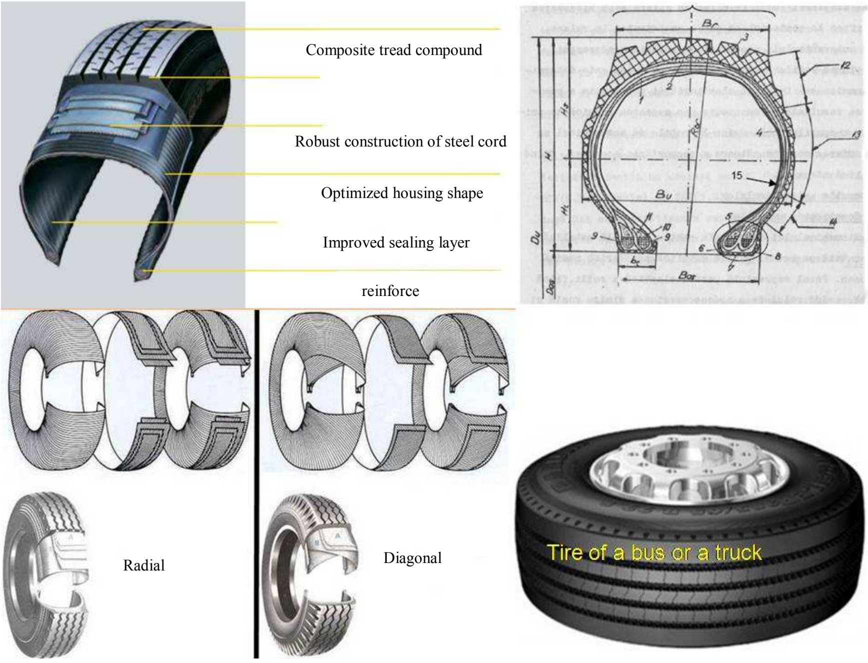


Fig. 4: Tires

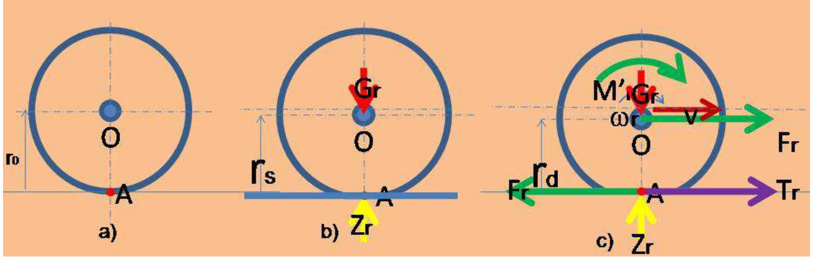


Fig. 5: The free, static and dynamic radius of a bus wheel

For simplification of calculations, the radius of r_r is usually calculated with relation (14), where r_0 is the free wheel radius, already known and λ is a tire deformation coefficient under the static and dynamic loads, which for tire buses and trolleybuses are between 0.945 and 0.950:

$$r_r = \lambda \cdot r_0 \tag{14}$$

Results and Discussion

Mechanical Yield of Transmission

The power of the engine is transmitted to the drive wheels through the motion transmitting organs. During the transmission of power, a part is spent to overcome the friction forces and other resistances that appear in the group of motion transmitting organs.

If it is denoted by P_e the effective power of the engine and with P_{tr} the power lost for overcoming all the resistances occurring in the mechanical transmission, then the mechanical efficiency of the transmission is given by relation 15:

$$\eta_t = \frac{P_e - P_{tr}}{P_e} = 1 - \frac{P_{tr}}{P_e} \tag{15}$$

The lost power P_{tr} is the sum of the partial losses occurring in the gearbox, in the longitudinal transmission, in the main transmission, in the differential bearings, in the final transmission and in the bearings of the drive wheels.

The mechanical efficiency of the transmission may also be expressed in terms of partial transmission efficiency (relation 16), where η_{cv} is the gearbox efficiency, η_c is the efficiency of the cardanic transmission, η_0 represents the efficiency of the main transmission (attack pinion plus toothed crown) and η_f is the efficiency corresponding to the final transmission (differential):

$$\eta_t = \eta_{cv} \cdot \eta_c \cdot \eta_0 \cdot \eta_f \tag{16}$$

Larger losses generally occur in gear units, but also those in longitudinal transmission (cardan transmission) can be very high if the angle γ between the gearbox output shaft and the drive shaft exceeds 10°. Gear losses are mainly of mechanical nature due to the mechanism itself, but they can also be amplified by a significant increase in friction in transmission couples if the lubricants used are not appropriate. In order to avoid operating problems and additional mechanical losses, the machining of the gears should be as appropriate as possible. In a normal execution, the operating efficiencies are the known classics and in special processing the gearing yields increase significantly. The exact way of determining the gearing yields will be done later in the study of each type of gear (gear coupling) in part.

Hydraulic losses depend on the amount and level of lubricant in the crankcase and its properties. By using a low viscosity lubricant but with good lubrication properties, the hydraulic losses are greatly reduced.

The performance of a gearbox depends primarily on the gear in which it operates, then on the torque and the input speed (engine speed, corresponding to the gearbox input). For a certain viscosity of the gearbox oil, gearbox output decreases when the rpm increases and increases as the torque increases (Fig. 6).

During the operation of the bus the yield tends to decrease in case of inappropriate exploitation. The transmission oils used should not only be appropriate but also be changed periodically, avoiding the prolonged use of an old lubricant, which is composed of more and more metal particles detachable in time from the transmission, plus rubbish and in addition old oils begin to lose their original lubrication capability. The timely change of the lubricant prolongs the life of the aggregate, whether it be an engine, a gearshift, or a differential and increases overall transmission efficiency while producing a reduction in fuel consumption. Extending to service an old lubricant used over normal life immediately results in the drastic increase in fuel consumption of the vehicle and the premature wear of the concerned assembly (engine, gearbox, differential).

Table 2 gives the values of the yields of the various transmission components according to the wheel teeth execution, where γ represents the angle of inclination of the shaft. In the calculations, it is recommended for the mechanical efficiency of the transmission to use the values η_t = 0,82 ... 0,90. The exact determination of its value can be made after studying separately the yields of each organ involved in the mechanical transmission.

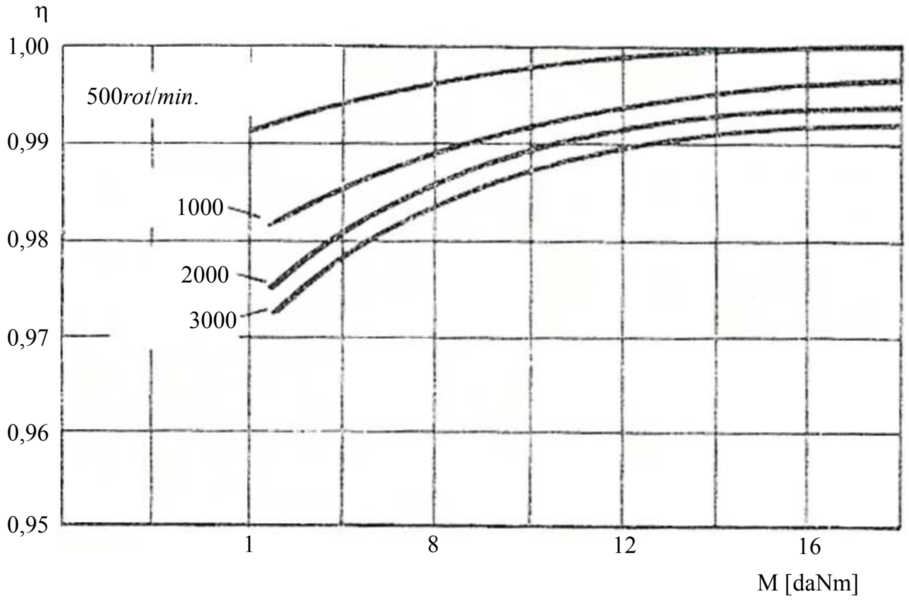


Fig. 6: Variation of gearbox efficiency η (or η_{cv}) depending on the transmitted M power and the spindle speed n

Table 2: Mechanical transmission efficiency

The element	Normal execution	Special execution
Gearbox		
Direct connection	0,98	0,99
Intermediate connections	0,92...0,94	0,94...0,98
Cardanic transmission		
γ= 0 [deg]	1	-
γ= 10 [deg]	0,988	-
γ= 18 [deg]	0,87	-
Main transmision		
Simple, conical, with straight teeth	0,95...0,96	0,96...0,98
Simple, conical, with curved teeth	0,90...0,92	0,93...0,95
Double	0,92...0,94	0,94...0,95

Resistances to the Bus

Resistance to the bus is a force that opposes its movement. The main resistances for the bus are as follows:

- Resistance to wheel running R_r
- Air resistance R_a
- Resistance to climbing an R_p slope
- Resistance to acceleration R_d

Resistance to Wheel Running R_r

Rolling resistance is the force that opposes the movement of the bus and is determined by the losses due to tire and road deformation, the superficial friction between the tire and the road, the friction in the wheel bearings and the wheels and the atmosphere (air) around them.

Figure 7a schematically depicts a deformed tire, under the action of the load G_r and the moment M', in the course of running on a road considered undeformable. The different elements of the tire (cc', bb', aa') as they enter the required area, suffer radial deformations (compression of the listed segments) and tangential (bending of the segments reached in the requested area).

Figure 7b shows the variation curve of the radial strain Δr depending on the load g that loads the element.

Because of the hysteresis phenomenon, the return curve does not coincide with the deformation curve. The hatched area is proportional to the mechanical work corresponding to the hysteresis losses (due to the internal friction of the material in the area of the deformed elements).

For a deforming element, the radial effort g_d is greater than the stress g_r corresponding to the same element in the equivalent but return position. Due to this, the vertical Z_r path reaction, considered for the entire contact surface, is displaced before point A with the distance a. By moving the force application point (load on the wheel), a moment of rolling resistance M_{RL} is generated in the opposite direction wheel rotation (relationship 17):

$$M_{rul} = Z_r \cdot a = G_r \cdot a \tag{17}$$

This torque can be replaced by the torque $R_r^{'}$ (Fig. 7c, relationship 18), where the force R_r' applied to the center of the wheel is the rolling resistance and the ratio *f* = *a*/*rr* is the coefficient rolling resistance:

$$R_r^{'} = \frac{Z_r \cdot a}{r_r} = G_r \cdot \frac{a}{r_r} = G_r \cdot f \qquad (18)$$

To maintain the movement, a compensating, pushing force parallel to the path equal to $R_r^{'}$ needs to be developed at the wheel axle.

Table 3 shows the average values of rolling resistance coefficient f depending on the type and condition of the tread (Frățilă *et al*., 2011).

The rolling resistance of the bus when traveling in a horizontal path is given by the rolling resistance of all the wheels of the bus (supporting both the motor and the non-motor) and can be calculated with relation 19, where G_r is the weight corresponding to a wheels and G_t represents the total weight of the bus:

$$R_r = f \cdot \sum G_r = f \cdot G_t \qquad (19)$$

When traveling on an inclined path with an angle *α* with respect to the horizontal, the relation for calculating the rolling resistance will take form 20 as the total weight of the bus will still press in the vertical direction and its component that will act perpendicularly to the plane of the rolling will be $Z_t = G_t.cos\,\alpha$ (Fig. 8):

$$R_r = f \cdot G_t \cdot \cos\alpha \qquad (20)$$

The power consumed to overcome rolling resistance is expressed by the relationship (21):

$$P_{rul} = R_r \cdot v = f \cdot G_t \cdot v \cdot \cos\alpha \qquad (21)$$

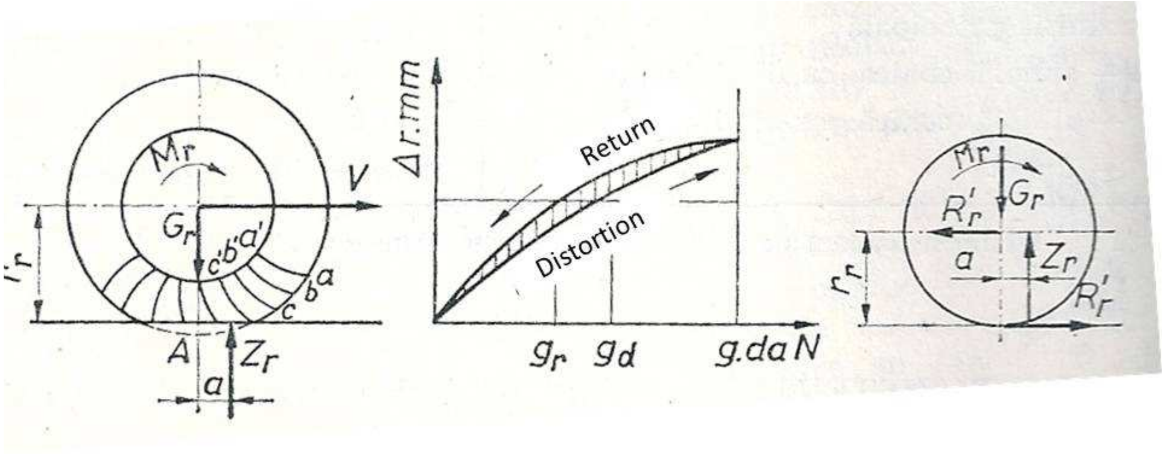
If G_t is expressed in [*daN*] and bus speed *v* in [m/s], then Prul will be expressed in [$daN.ms^{-1}$].

If we want to express the power in [W], 1W = 1/9,806 [*daN.m*/*s*] and one *daN.m*/*s* = 9,806 [W], so we can write the relation (22) in the international system, in W, the speed will also be introduced in *SI* i.e., in m/s, but for the weight of the G_t bus will continue to be used as the daN measurement unit instead of *N*:

$$P_{rul} = 9{,}806 \cdot R_r \cdot v = 9{,}806 \cdot f \cdot G_t \cdot v \cdot \cos\alpha \qquad (22)$$

In order to keep the same input and output units, but the velocity to enter in km/h instead of m/s, the relationship (22) must be amplified by 1/3.6, or else be divided to 3.6. Thus, the relation (23) is obtained in which the velocity V will be introduced in [km/h], the force G_t in [*daN*], but the power Prul will still be obtained in *SI*, i.e., in [W]

$$P_{rul} = 9{,}806 / 3{,}6 \cdot R_r \cdot V = 2{,}724 \cdot f \cdot G_t \cdot V \cdot \cos\alpha \qquad (23)$$



(a) (b) (c)

Fig. 7: Resistance to wheel running

Table 3: Average rolling resistance coefficients *f*

The way and the state of the runway	Coefficient of rolling resistance *f*
Road asphalt in perfect condition	0,012...0,018
The paved road in the proper condition	0,018...0,022
Cobbled road	0,020...0,025
Dirt road	0,025...0,150
Snowy road beaten	0,025...0,030
Sand	0,100...0,300

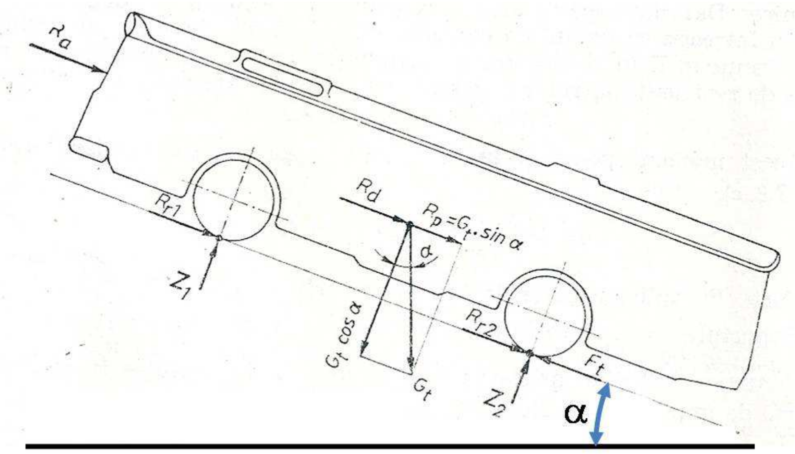


Fig. 8: Traveling the bus in an inclined way with an angle α to the horizontal

Conclusion

The present paper aims to present the study of the dynamics of the vehicles, with particularization on the buses. Here are the main elements of the bus dynamics, taking into account all the elements that influence the dynamic operation of a bus, in general and in particular situations, with emphasis on the main systems and elements that act on the actual, dynamic, on a normal path or on an inclined with an alpha angle path.

The paper presents the first part of the bus dynamics.

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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.

References

Antonescu, P., 2000. Mechanisms and Handlers. 1st Edn., Printech Publishing House. Bucharest.

Antonescu, P. and F. Petrescu, 1985. Analytical method of synthesis of cam mechanism and flat stick. Proceedings of the 4th International Symposium on Mechanism Theory and Practice, (MTP' 85), Bucharest.

Antonescu, P. and F. Petrescu, 1989. Contributions to cinetoelastodynamic analysis of distribution mechanisms. Bucharest.

Antonescu, P., M. Oprean and F. Petrescu, 1985a. Contributions to the synthesis of oscillating cam mechanism and oscillating flat stick. Proceedings of the 4th International Symposium on Theory and Practice of Mechanisms, (TPM' 85), Bucharest.

Antonescu, P., M. Oprean and F. Petrescu, 1985b. At the projection of the oscillante cams, there are mechanisms and distribution variables. Proceedings of the 5th Conference for Engines, Automobiles, Tractors and Agricultural Machines, I-Engines and Automobiles, (MIA' 85), Brasov.

Antonescu, P., M. Oprean and F. Petrescu, 1986. Projection of the profile of the rotating camshaft acting on the oscillating plate with disengagement. Proceedings of the 3rd National Computer Assisted Designing Symposium in Mechanisms and Machine Bodies, (MMB' 86), Brasov.

Antonescu, P., M. Oprean and F. Petrescu, 1987. Dynamic analysis of the cam distribution mechanisms. Proceedings of the Seventh National Symposium of Industrial Robots and Spatial Mechanisms, (RSM' 87), Bucharest,

Antonescu, P., M. Oprean and F. Petrescu, 1988. Analytical synthesis of Kurz profile, rotating flat cam cam. Machine Build. Rev. Bucharest.

Antonescu, P., F. Petrescu and O. Antonescu, 1994. Contributions to the synthesis of the rotating cam mechanism and the tip of the balancing tip. Brasov.

Antonescu, P., F. Petrescu and D. Antonescu, 1997. Geometrical synthesis of the rotary cam and balance tappet mechanism. Bucharest.

Antonescu, P., F. Petrescu and O. Antonescu, 2000a. Contributions to the synthesis of the rotary disc-cam profile. Proceedings of the 8th International Conference on Theory of Machines and Mechanisms, (TMM' 00), Liberec, Czech Republic, pp: 51-56.

Antonescu, P., F. Petrescu and O. Antonescu, 2000b. Synthesis of the rotary cam profile with balance follower. Proceedings of the 8th Symposium on Mechanisms and Mechanical Transmissions, (MMT' 000), Timişoara, pp: 39-44.

Antonescu, P., F. Petrescu and O. Antonescu, 2001. Contributions to the synthesis of mechanisms with rotary disc-cam. Proceedings of the 8th IFToMM International Symposium on Theory of Machines and Mechanisms, (TMM' 01), Bucharest, ROMANIA, pp: 31-36.

Aversa, R., R.V. Petrescu, A. Apicella and F.I.T. Petrescu, 2017a. Nano-diamond hybrid materials for structural biomedical application. Am. J. Biochem. Biotechnol., 13: 34-41. DOI: 10.3844/ajbbsp.2017.34.41

Aversa, R., R.V. Petrescu, B. Akash, R.B. Bucinell and J.M. Corchado *et al*., 2017b. Kinematics and forces to a new model forging manipulator. Am. J. Applied Sci., 14: 60-80. DOI: 10.3844/ajassp.2017.60.80

Aversa, R., R.V. Petrescu, A. Apicella, F.I.T. Petrescu and J.K. Calautit *et al*., 2017c. Something about the V engines design. Am. J. Applied Sci., 14: 34-52. DOI: 10.3844/ajassp.2017.34.52

Aversa, R., D. Parcesepe, R.V. Petrescu, F. Berto and G. Chen *et al*., 2017d. Processability of bulk metallic glasses. Am. J. Applied Sci., 14: 294-301. DOI: 10.3844/ajassp.2017.294.301

Aversa, R., R.V. Petrescu, A. Apicella and F.I.T. Petrescu, 2017e. Modern transportation and photovoltaic energy for urban ecotourism. Transylvanian Rev. Administrative Sci., 13: 5-20. DOI: 10.24193/tras.SI2017.1

Aversa, R., F.I.T. Petrescu, R.V. Petrescu and A. Apicella, 2016a. Biomimetic FEA bone modeling for customized hybrid biological prostheses development. Am. J. Applied Sci., 13: 1060-1067. DOI: 10.3844/ajassp.2016.1060.1067

Aversa, R., D. Parcesepe, R.V. Petrescu, G. Chen and F.I.T. Petrescu *et al*., 2016b. Glassy amorphous metal injection molded induced morphological defects. Am. J. Applied Sci., 13: 1476-1482. DOI: 10.3844/ajassp.2016.1476.1482

Aversa, R., R.V. Petrescu, F.I.T. Petrescu and A. Apicella, 2016c. Smart-factory: Optimization and process control of composite centrifuged pipes. Am. J. Applied Sci., 13: 1330-1341. DOI: 10.3844/ajassp.2016.1330.1341

Aversa, R., F. Tamburrino, R.V. Petrescu, F.I.T. Petrescu and M. Artur *et al*., 2016d. Biomechanically inspired shape memory effect machines driven by muscle like acting NiTi alloys. Am. J. Applied Sci., 13: 1264-1271. DOI: 10.3844/ajassp.2016.1264.1271

Cao, W., H. Ding, Z. Bin and C. Ziming, 2013. New structural representation and digital-analysis platform for symmetrical parallel mechanisms. Int. J. Adv. Robot. Sys. DOI: 10.5772/56380

Comănescu, A., D. Comănescu, I. Dugăeşescu and A. Boureci, 2010. The Basics of Modeling Mechanisms. 1st Edn., Politehnica Press Publishing House, Bucharest, ISBN-10: 978-606-515-115-4, pp: 274.

Dong, H., N. Giakoumidis, N. Figueroa and N. Mavridis, 2013. Approaching behaviour monitor and vibration indication in developing a General Moving Object Alarm System (GMOAS). Int. J. Adv. Robot. Sys. DOI: 10.5772/56586

De Melo, L.F., R.A., S.F. Rosário and J.M., Rosário, 2012. Mobile robot navigation modelling, control and applications. Int. Rev. Modell. Simulat., 5: 1059-1068.

Frăţilă, G., M. Frăţilă and S. Samoilă, 2011. Automobiles, Construction, Exploitation, Reparation. 10th Edn., EDP, Bucharest, ISBN-10 978-973-30-2857-4.

Garcia, E., M.A. Jimenez, P.G. De Santos and M. Armada, 2007. The evolution of robotics research. IEEE Robot. Autom. Magaz., 14: 90-103. DOI: 10.1109/MRA.2007.339608

Garcia-Murillo, M., J. Gallardo-Alvarado and E. Castillo-Castaneda, 2013. Finding the generalized forces of a series-parallel manipulator. IJARS. DOI: 10.5772/53824

He, B., Z. Wang, Q. Li, H. Xie and R. Shen, 2013. An analytic method for the kinematics and dynamics of a multiple-backbone continuum robot. IJARS. DOI: 10.5772/54051

Lee, B.J., 2013. Geometrical derivation of differential kinematics to calibrate model parameters of flexible manipulator. Int. J. Adv. Robot. Sys. DOI: 10.5772/55592

Lin, W., B. Li, X. Yang and D. Zhang, 2013. Modelling and control of inverse dynamics for a 5-DOF parallel kinematic polishing machine. Int. J. Adv. Robot. Sys. DOI: 10.5772/54966

Liu, H., W. Zhou, X. Lai and S. Zhu, 2013. An efficient inverse kinematic algorithm for a PUMA560-structured robot manipulator. IJARS. DOI: 10.5772/56403

Mirsayar, M.M., V.A. Joneidi, R.V. Petrescu, F.I.T. Petrescu and F. Berto, 2017. Extended MTSN criterion for fracture analysis of soda lime glass. Eng. Fracture Mechan., 178: 50-59. DOI: 10.1016/j.engfracmech.2017.04.018

Padula, F. and V. Perdereau, 2013. An on-line path planner for industrial manipulators. Int. J. Adv. Robot. Sys. DOI: 10.5772/55063

Pelecudi, C., 1967. The Basics of Mechanism Analysis. Publishing house: Academy of the People's Republic of Romania.

Perumaal, S. and N. Jawahar, 2013. Automated trajectory planner of industrial robot for pick-and-place task. IJARS. DOI: 10.5772/53940

Petrescu, F. and R. Petrescu, 1995a. Contributions to optimization of the polynomial motion laws of the stick from the internal combustion engine distribution mechanism. Bucharest.

Petrescu, F. and R. Petrescu, 1995b. Contributions to the synthesis of internal combustion engine distribution mechanisms. Bucharest.

Petrescu, F. and R. Petrescu, 1997a. Dynamics of cam mechanisms (exemplified on the classic distribution mechanism). Bucharest.

Petrescu, F. and R. Petrescu, 1997b. Contributions to the synthesis of the distribution mechanisms of internal combustion engines with Cartesian coordinate method. Bucharest.

Petrescu, F. and R. Petrescu, 1997c. Contributions to maximizing polynomial laws for the active stroke of the distribution mechanism from internal combustion engines. Bucharest.

Petrescu, F. and R. Petrescu, 2000a. Synthesis of distribution mechanisms by the rectangular (cartesian) coordinate method. University of Craiova, Craiova.

Petrescu, F. and R. Petrescu, 2000b. The design (synthesis) of cams using the polar coordinate method (the triangle method). University of Craiova, Craiova.

Petrescu, F. and R. Petrescu, 2002a. Motion laws for cams. Proceedings of the 7th National Symposium with International Participation Computer Assisted Design, (PAC' 02), Braşov, pp: 321-326.

Petrescu, F. and R. Petrescu, 2002b. Camshaft dynamics elements. Proceedings of the 7th National Symposium with International Participation Computer Assisted Design, (PAC' 02), Braşov, pp: 327-332.

Petrescu, F. and R. Petrescu, 2003. Some elements regarding the improvement of the engine design. Proceedings of the 8th National Symposium, Descriptive Geometry, Technical Graphics and Design, (GTD' 03), Braşov, pp: 353-358.

Petrescu, F. and R. Petrescu, 2005a. The cam design for a better efficiency. Proceedings of the International Conference on Engineering Graphics and Design, (EGD' 05), Bucharest, pp: 245-248.

Petrescu, F. and R. Petrescu, 2005b. Contributions at the dynamics of cams. Proceedings of the 9th IFToMM International Symposium on Theory of Machines and Mechanisms, (TMM' 05), Bucharest, Romania, pp: 123-128.

Petrescu, F. and R. Petrescu, 2005c. Determining the dynamic efficiency of cams. Proceedings of the 9th IFToMM International Symposium on Theory of Machines and Mechanisms, (TMM' 05), Bucharest, Romania, pp: 129-134.

Petrescu, F. and R. Petrescu, 2005d. An original internal combustion engine. Proceedings of the 9th IFToMM International Symposium on Theory of Machines and Mechanisms, (TMM' 05), Bucharest, Romania, pp: 135-140.

Petrescu, F. and R. Petrescu, 2005e. Determining the mechanical efficiency of Otto engine's mechanism. Proceedings of the 9th IFToMM International Symposium on Theory of Machines and Mechanisms, (TMM' 05), Bucharest, Romania, pp: 141-146.

Petrescu, F.I. and R.V. Petrescu, 2013. Cinematics of the 3R Dyad. Engevista, 15: 118-124.

Petrescu, F.I. and R.V. Petrescu, 2012a. Kinematics of the planar quadrilateral mechanism. Engevista, 14: 345-348.

Petrescu, F.I. and R.V. Petrescu, 2012b. Mecatronica-Sisteme Seriale si Paralele. Create Space Publisher, USA, ISBN-10: 978-1-4750-6613-5, pp: 128.

Petrescu, F.I. and R.V. Petrescu, 2011. Mechanical Systems, Serial and Parallel-Course (in Romanian). LULU Publisher, London, UK, ISBN-10: 978-1-4466-0039-9, pp: 124.

Petrescu, F.I. and R.V. Petrescu, 2016a. Parallel moving mechanical systems kinematics, Engevista, 18: 455-491.

Petrescu, F.I. and R.V. Petrescu, 2016b. Direct and inverse kinematics to the Anthropomorphic Robots, Engevista, 18: 109-124.

Petrescu, F. and R. Petrescu, 2016c. An otto engine dynamic model. IJM&P, 7: 038-048.

Petrescu, F.I. and R.V. Petrescu, 2016d. Otto motor dynamics. Geintec, 6: 3392-3406.

Petrescu, F.I. and R.V. Petrescu, 2016e. Dynamic cinematic to a structure 2R. GEINTEC, 6: 3143-3154.

Petrescu, F.I., B. Grecu, A. Comanescu and R.V. Petrescu, 2009. Some mechanical design elements. Proceeding of the International Conference on Computational Mechanics and Virtual Engineering, (MEC' 09), Braşov, pp: 520-525.

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and F.I.T. Petrescu, 2016a About the gear efficiency to a simple planetary train. Am. J. Applied Sci., 13: 1428-1436.

Petrescu, R.V., R. Aversa, A. Apicella, S. Li and G. Chen *et al*., 2016b. Something about electron dimension. Am. J. Applied Sci., 13: 1272-1276.

Petrescu, F.I.T., A. Apicella, R. Aversa, R.V. Petrescu and J.K. Calautit *et al*., 2016c. Something about the mechanical moment of inertia. Am. J. Applied Sci., 13: 1085-1090.

Petrescu, R.V., R. Aversa, A. Apicella, F. Berto and S. Li *et al*., 2016d. Ecosphere protection through green energy. Am. J. Applied Sci., 13: 1027-1032.

Petrescu, F.I.T., A. Apicella, R.V. Petrescu, S.P. Kozaitis and R.B. Bucinell *et al*., 2016e. Environmental protection through nuclear energy. Am. J. Applied Sci., 13: 941-946.

Petrescu, F.I.T. and J.K. Calautit, 2016a. About nano fusion and dynamic fusion. Am. J. Applied Sci., 13: 261-266.

Petrescu, F.I.T. and J.K. Calautit, 2016b. About the light dimensions. Am. J. Applied Sci., 13: 321-325.

Petrescu, R.V., R. Aversa, B. Akash, R. Bucinell and J. Corchado *et al*., 2017a. Modern propulsions for aerospace-a review. J. Aircraft Spacecraft Technol., 1: 1-8. DOI: 10.3844/jastsp.2017.1.8

Petrescu, R.V., R. Aversa, B. Akash, R. Bucinell and J. Corchado *et al*., 2017b. Modern propulsions for aerospace-part II. J. Aircraft Spacecraft Technol., 1: 9-17. DOI: 10.3844/jastsp.2017.9.17

Petrescu, R.V., R. Aversa, B. Akash, R. Bucinell and J. Corchado *et al*., 2017c. History of aviation-a short review. J. Aircraft Spacecraft Technol., 1: 30-49. DOI: 10.3844/jastsp.2017.30.49

Petrescu, R.V., R. Aversa, B. Akash, R. Bucinell and J. Corchado *et al*., 2017d. Lockheed martin-a short review. J. Aircraft Spacecraft Technol., 1: 50-68. DOI: 10.3844/jastsp.2017.50.68

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017e. Our universe. J. Aircraft Spacecraft Technol., 1: 69-79. DOI: 10.3844/jastsp.2017.69.79

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017f. What is a UFO? J. Aircraft Spacecraft Technol., 1: 80-90. DOI: 10.3844/jastsp.2017.80.90

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017g. About bell helicopter FCX-001 concept aircraft-a short review. J. Aircraft Spacecraft Technol., 1: 91-96. DOI: 10.3844/jastsp.2017.91.96

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017h. Home at airbus. J. Aircraft Spacecraft Technol., 1: 97-118. DOI: 10.3844/jastsp.2017.97.118

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017i. Airlander. J. Aircraft Spacecraft Technol., 1: 119-148. DOI: 10.3844/jastsp.2017.119.148

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017j. When boeing is dreaming-a review. J. Aircraft Spacecraft Technol., 1: 149-161. DOI: 10.3844/jastsp.2017.149.161

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017k. About Northrop Grumman. J. Aircraft Spacecraft Technol., 1: 162-185. DOI: 10.3844/jastsp.2017.162.185

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017l. Some special aircraft. J. Aircraft Spacecraft Technol., 1: 186-203. DOI: 10.3844/jastsp.2017.186.203

Petrescu, R.V., R. Aversa, B. Akash, J. Corchado and F. Berto *et al*., 2017m. About helicopters. J. Aircraft Spacecraft Technol., 1: 204-223. DOI: 10.3844/jastsp.2017.204.223

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017n. The modern flight. J. Aircraft Spacecraft Technol., 1: 224-233. DOI: 10.3844/jastsp.2017.224.233

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017o. Sustainable energy for aerospace vessels. J. Aircraft Spacecraft Technol., 1: 234-240. DOI: 10.3844/jastsp.2017.234.240

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017p. Unmanned helicopters. J. Aircraft Spacecraft Technol., 1: 241-248. DOI: 10.3844/jastsp.2017.241.248

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017q. Project HARP. J. Aircraft Spacecraft Technol., 1: 249-257. DOI: 10.3844/jastsp.2017.249.257

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017r. Presentation of romanian engineers who contributed to the development of global aeronautics-part I. J. Aircraft Spacecraft Technol., 1: 258-271. DOI: 10.3844/jastsp.2017.258.271

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017s. A first-class ticket to the planet mars, please. J. Aircraft Spacecraft Technol., 1: 272-281. DOI: 10.3844/jastsp.2017.272.281

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017t. Forces of a 3R robot. J. Mechatron. Robot., 1: 1-14. DOI: 10.3844/jmrsp.2017.1.14

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017u. Direct geometry and cinematic to the MP-3R systems. J. Mechatron. Robot., 1: 15-23. DOI: 10.3844/jmrsp.2017.15.23

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017v. Dynamic elements at MP3R. J. Mechatron. Robot., 1: 24-37. DOI: 10.3844/jmrsp.2017.24.37

Petrescu, R.V., R. Aversa, B. Akash, F. Berto and A. Apicella *et al*., 2017w. Geometry and direct kinematics to MP3R with 4×4 operators. J. Mechatron. Robot., 1: 38-46. DOI: 10.3844/jmrsp.2017.38.46

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and S. Kozaitis *et al*., 2017x. Current stage in the field of mechanisms with gears and rods. J. Mechatron. Robot., 1: 47-57. DOI: 10.3844/jmrsp.2017.47.57

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and S. Kozaitis *et al*., 2017y. Geometry and inverse kinematic at the MP3R mobile systems. J. Mechatron. Robot., 1: 58-65. DOI: 10.3844/jmrsp.2017.58.65

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and S. Kozaitis *et al*., 2017z. Synthesis of optimal trajectories with functions control at the level of the kinematic drive couplings. J. Mechatron. Robot., 1: 66-74. DOI: 10.3844/jmrsp.2017.66.74

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and S. Kozaitis *et al*., 2017aa. The inverse kinematics of the plane system 2-3 in a mechatronic MP2R system, by a trigonometric method. J. Mechatron. Robot., 1: 75-87. DOI: 10.3844/jmrsp.2017.75.87

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and S. Kozaitis *et al*., 2017ab. Serial, anthropomorphic, spatial, mechatronic systems can be studied more simply in a plan. J. Mechatron. Robot., 1: 88-97. DOI: 10.3844/jmrsp.2017.88.97

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and S. Kozaitis *et al*., 2017ac. Analysis and synthesis of mechanisms with bars and gears used in robots and manipulators. J. Mechatron. Robot., 1: 98-108. DOI: 10.3844/jmrsp.2017.98.108

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and S. Kozaitis *et al*., 2017ad. Speeds and accelerations in direct kinematics to the MP3R systems. J. Mechatron. Robot., 1: 109-117. DOI: 10.3844/jmrsp.2017.109.117

Petrescu, R.V., R. Aversa, A. Apicella, M.M. Mirsayar and S. Kozaitis *et al*., 2017ae. Geometry and determining the positions of a plan transporter manipulator. J. Mechatron. Robot., 1: 118-126. DOI: 10.3844/jmrsp.2017.118.126

Reddy, P., K.V. Shihabudheen and J. Jacob, 2012. Precise non linear modeling of flexible link flexible joint manipulator. IReMoS, 5: 1368-1374

Tabaković, S., M. Zeljković, R. Gatalo and A. Živković, 2013. Program suite for conceptual designing of parallel mechanism-based robots and machine tools. Int. J. Adv. Robot Sys. DOI: 10.5772/56633

Tang, X., D. Sun and Z. Shao, 2013. The structure and dimensional design of a reconfigurable PKM. IJARS. DOI: 10.5772/54696

Tong, G., J. Gu and W. Xie, 2013. Virtual entity-based rapid prototype for design and simulation of humanoid robots. Int. J. Adv. Robot. Sys. DOI: 10.5772/55936

Wang, K., M. Luo, T. Mei, J. Zhao and Y. Cao, 2013. Dynamics analysis of a three-DOF planar serial-parallel mechanism for active dynamic balancing with respect to a given trajectory. Int. J. Adv. Robotic Sys. DOI: 10.5772/54201

Wen, S., J. Zhu, X. Li, A. Rad and X. Chen, 2012. End-point contact force control with quantitative feedback theory for mobile robots. IJARS. DOI: 10.5772/53742