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HYDRAULIC BRAKING ENERGY RECOVERY OF HEAVY AUTOMOTIVES

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This paper was written in the memory of mat. eng. Gabriel Rădulescu – INOE 2000- IHP Bucharest, ROMANIA with whom I had an exceptional scientific collaboration over 25 years.
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Abstract: The paper is presenting theoretical model and experimental solution for designing modular equipment used in order to recuperate braking energy of heavy automotives. The proposed solution reduces fuel consumption for heavy automotives of 8-th category by saving energy in the braking process and using it when stating off in order to overcome the system inertia. Tacking into account high level consumption for that type of heavy automotive, the paper is in line with international efforts in order to reduce fuel consumption with 35%.
Keywords: braking energy recovery, hydraulic drive, automotive

1. Introduction

In the transport sector and particularly that of the heavy automotives, pollution effects are visible. The number of cargo vehicles in the USA and EU is steadily growing and fuel consumption is also becoming higher. Evaluations made in the USA regarding fuel consumption in cargo vehicles sector show that they can be ranked according to vehicle mass, starting with 1st category vehicles with mass less than 2721 kg, up to the 8th category with a mass over 17686 kg. An analysis on actual fuel consumption for each given vehicle category – commercial trucks – shows that for the 8th category of heavy automotives any saving solution is welcomed. In recent years the global market of new medium and heavy cargo vehicles was estimated at more then1.4 million, concentrated in three areas: North America, Western Europe and Asia.

Given the fact that the highest fuel consumption is in the 8th category, a complex, large scale, structural and functional improvement action is observed of these vehicles, targeting a substantial fuel consumption reduction, with more than 35%, as shown in figure 1.

As shown in figure 1, we can observe specific objectives and courses of action, measured in energy units for each of the main components of an 8th category vehicle:

- Increase engine efficiency from 40% to 44% and decreasing energy losses;
- Decreasing losses due to rolling resistance, from 51kWh to 30,6 kWh;
- Drive systems from *9 kWh to 6,3 kWh;*
- Auxiliary charging from 15kWh to 7,5 kWh;
- Aerodynamic losses from 85kWh to 68kWh, figure 7 and 8.

Energy improvement actions of heavy automotives include other measures also, among which the use of technologically advanced materials: composite materials, carbon fibers, titanium, etc. Moreover, along with substantial energy reduction resulting from increasing the dynamic performances of heavy automotives, we can also implement some measures in order to recuperate braking energy, systems that proved to be extremely effective, at an acceptable average size, compatible with the dimensions of heavy automotives.

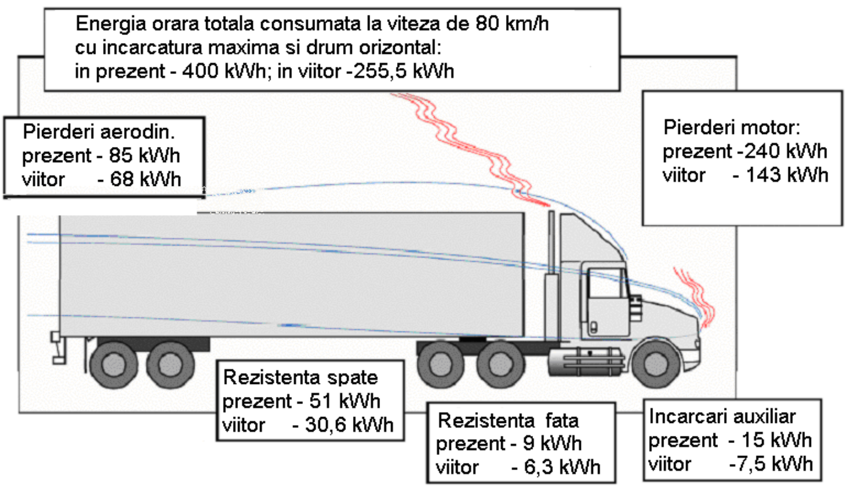

Fig. 1 Hourly energy consumption

(Total hourly energy consumption at a speed of 80km/h with maximum load and flat road: now – 400 kWh, targeted in the future: 255,5 kWh; Aerodynamic losses now-85kwh, in the future- 68kWh; Engine losses–now-240kwh, in the future-143 kWh;The rear endurance- now-51kWh, in the future-30,6 kWh; Front endurance–now 9kWh, in the future -6,3 kWh;Auxiliary loads– now-15kWh, in the future – 7,5 kWh)

Regarding the energy saving systems, first attempts consisted in modifying the classical structure of the vehicle. Thus, it was introduced full hydrostatic transmission associated with an energy storage system. The solution had limited implementation because the assimilation costs of this system were probably overcome by its benefits. The version in which vehicles are equipped with simple recovery systems, whose implementation in the vehicle structure is done operational and constructive with minimal changes and that are easily removed, has great chances of replication. Other method consists of fuel consumption limitation, such as local monitoring on vehicles through flow meters but is not agreed by carriers.

2. Mathematical model

The mathematical model is derived in order to provide data for designing the hydraulic system. The model is basically consisting in equations describing the kinematics and dynamics of the elements involved in the braking process and also in the equations describing the automotive braking process.

2.1 Kinematics and dynamics elements in braking process

Energy consumption mode of the automobile ensemble in the braking process is linked to mechanism participation and constructive characteristics of the kinematic gear chain transmission of motion. These elements are: brake friction in the specialized body, mechanical resistance, composed of wheel rolling endurance and loosed through transmission devices (gearbox, power distribution, etc); aerodynamic drag, wheel slip in the rolling process due to the power distribution flaw and contact imperfections between the wheels and the road.

2.1.1 Equation of braked wheel

The force and inertia moments are presented in figure 2:

$$F_{ir} = m_r \frac{dv}{dt} \text{ and } M_i = I_r \frac{dwr}{dt} \tag{1}$$

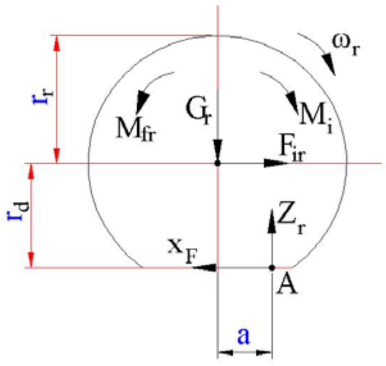

Fig.2 Work scheme for the wheel
(A - Point of application of the resilience at drag force point –Xf and Zr;
Ir – static resilient point; Mr – wheel mass)

The equation for forces projection horizontally is:

$$F_r + m_r \frac{dv}{dt} - X_f = 0 \tag{2}$$

The equation for forces projection vertically is:

$$Z_r - G_r = 0 \tag{3}$$

The equation of point is expressed through successive calculations, resulting X_f component:

$$X_f = \frac{Mfr}{r_d} + Zr\frac{a}{r_d} - \frac{I_r}{r_d}\cdot\frac{d\omega_r}{dt} = F_{fr} + R_r - \frac{I_r}{r_d}\cdot\frac{d\omega_r}{dt} \quad , (4)$$

$$X_F = F_{fr} + R_r - X_i \tag{5}$$

in which, $X_i = \frac{I_r}{r_d}\cdot\frac{d\omega_r}{dt}$, $w_r = \frac{V}{r_r}$ and finally $\frac{I_r}{r_d}\cdot\frac{d\omega_r}{dt} = \frac{I_r}{r_r r_d}\cdot\frac{dv}{dt}$.

Thrust force in the wheel bearings becomes:

$$F_r = F_{fr} + R_r - \left(m_r + \frac{I_r}{r_r r_d}\cdot\right)\frac{dv}{dt} \quad .(6)$$

Examining the relation (6) it is considered that the force responsible for deceleration is dominated by the resilience forces. The maximum limit of the horizontal component X_f is $X_{\max} \geq X_f$:

$$X_{\max} = \varphi Z_r \tag{7}$$

it results:

$$F_{fr} + R_r - \frac{I_r}{r_d}\cdot\frac{d\omega_r}{dt} \leq \varphi\cdot Z_r \tag{8}$$

So, the variation limits of the braking force at wheel level F_{fr} and braking point is:

$$0 \leq F_{fr} \leq (\varphi - f)Z_r - \frac{I_r}{r_d}\cdot\frac{d\omega_r}{dt} \tag{9}$$

$$0 \leq M_{fr} \leq (\varphi - f) r_d Z_r - I_r \frac{d\omega_r}{dt} \tag{10}$$

2.1.2 The equation of the braked vehicle

In figure 3 the ensemble of forces is schematized along with the moments when it acts on the braked vehicle. The forces are applied on the central of gravity Cg of the vehicle, represented on the wheelbase by *a* and *b* dimensions, and by the height in relation to the road surface *hg*. The ration between forces is performed at a rolling track considered inclined with α angle.

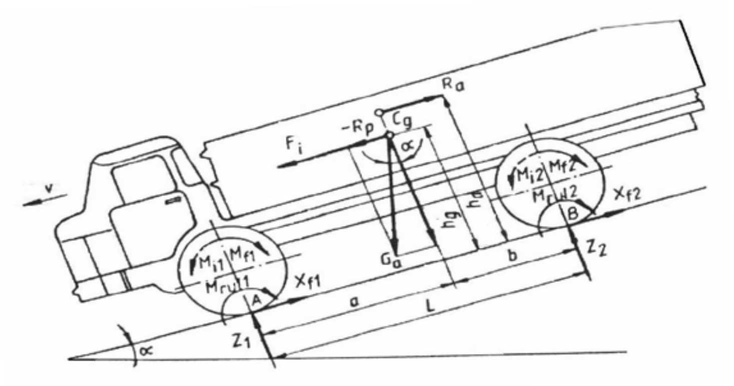

Fig. 3 Forces and moments for the vehicle

Ga – the weight of the vehicle applied in the center of gravity (Cg) , Cg (ha, a/b) $Ra = \frac{kA}{13} V^2$, the aerodynamic drag applied on the front pressure center at ha height, where A is the cross section, k drag coefficient, $R_p = G_a \sin\alpha$, is the resilience on descending path; M_{ru1}, M_{ru2} are the moments corresponding to the rolling endurance; M_{f1}, M_{f2} braking moments applied on wheels; $F_i = \frac{Ga}{g} \cdot \frac{dv}{dt}$ resilience force in translation; M_{i1}, M_{i2} – moment of wheels resilience; Z1, Z2- normal reactions on axles; X_{f1}, X_{f2} – tangential reactions on axles;

Considering the mechanical assumption that*:* vehicle ensemble is rigid, ignoring suspension. We apply the principle of d'Alembert:

$$F_i = X_{f1} + X_{f2} + R_a \pm R_p \,. \tag{11}$$

By replacing forces expressions we obtain the equation:

$$\frac{Ga}{g} \cdot \frac{dv}{dt} = X_{f1} + X_{f2} FGa \sin\alpha + \frac{kA}{13} V^2 \,, \tag{12}$$

but in general $X_f = F_f + R_r - X$.

According to the equation:

$$\gamma_F = \frac{F_{f1} + F_{f2}}{Ga} = \frac{F_f}{Ga} \tag{13}$$

called specific braking force:

$$\psi = \frac{R_{r1} + R_{r2} \pm R_p}{Ga} = \frac{R_r \pm R_p}{G} \tag{14}$$

called specific endurance of the road, we obtain:

$$\frac{1}{g}\cdot\frac{dv}{dt}=\gamma_F+\psi+\left(\frac{KA}{13Ga}\right)V^2 \tag{15}$$

Noting $\frac{KA}{13Ga}=V_0^2$; $\gamma_F+\psi=a$ we obtain in the end the equation and limits condition:

$$\begin{cases}\frac{V_0^2}{g}\cdot\frac{dv}{dt}=\left(V_0\sqrt{a}\right)^2+V^2\\ V(0)=V_{\max}\end{cases} \tag{16}$$

By integrating the equation and successively converting it we obtain:

$$V(t)=\frac{V_{\max}+V_0\sqrt{a}tg\left(\frac{g\sqrt{a}}{V_0}\right)t}{1-\frac{V_{\max}}{V_0\sqrt{a}}tg\left(\frac{g\sqrt{a}}{V_0}\right)t}\ ,\quad \text{with}\quad tg\left(\frac{g\sqrt{a}}{V_0}\right)t\le\frac{V_0\sqrt{a}}{V_{\max}} \tag{17}$$

Various expression forms of deceleration

The case of full braking:

$$\frac{dv}{dt}=\begin{cases}g\left(\varphi\cos\alpha\pm\sin\alpha+\frac{KA}{13Ga}V^2\right), & \text{for}\quad \mathrm{v}\ge 80\ \mathrm{km/h}\\ g(\varphi\cos\alpha\pm\sin\alpha)\quad , & \text{for}\quad \mathrm{v}\le 80\ \mathrm{km/h}\\ g\varphi\quad (\text{drum plan})\ \alpha=0\ , & \text{for}\quad \mathrm{v}\le 80\ \mathrm{km/h}\end{cases} \tag{18}$$

The case of back axle braking:

Considering: $\left(\frac{g}{Ga}\right)\cdot\left(\frac{2Ir}{r_r^2}\right)<<1$ for $\mathrm{V}<80\ \mathrm{km/h}$

$$\frac{dv}{dt}=\begin{cases}g\left(\varphi\frac{\frac{a}{L}}{1+\varphi\frac{h_g}{L}}\cos\alpha\pm\sin\alpha\right), & \alpha\ne 0\\ \mathrm{g}\varphi\frac{\frac{\mathrm{a}}{\mathrm{L}}}{1+\varphi\frac{\mathrm{h_g}}{\mathrm{L}}}\quad , & \alpha=0\end{cases} \tag{19}$$

3. Braking energy recovery installation hydraulic scheme

Resulting from the above calculations, the parameters for which we determine the functional measurements for the components of the braking energy recovery installation result as presented in figure 4.

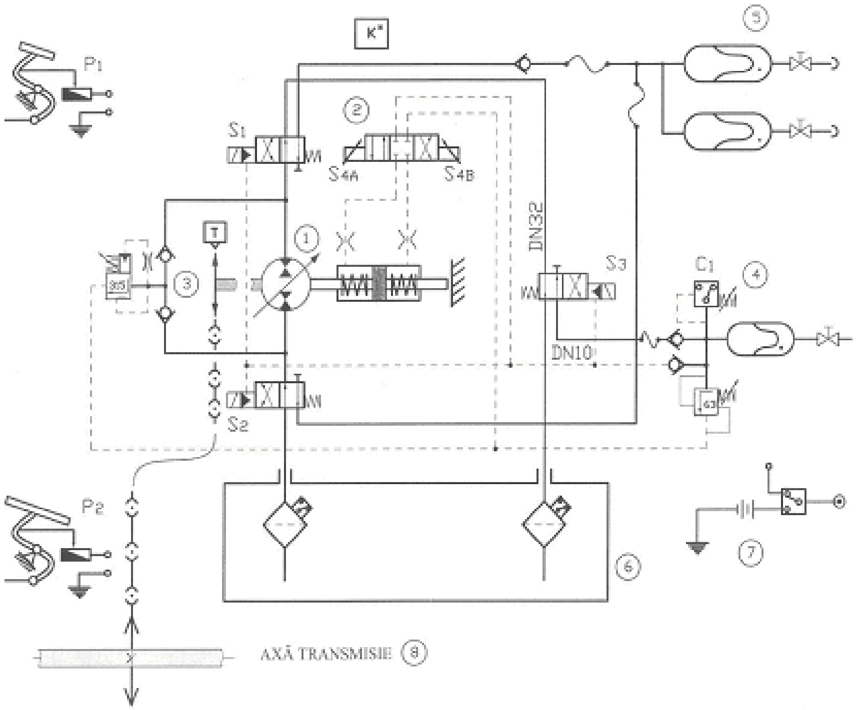

Fig. 4 The hydraulic scheme of installation equipment

The placement of the equipment on the vehicle is presented in two versions in fig. 5 a,b. The 2 schemes differ by the conception of the mechanical transmission from the driveshaft to the hydraulic unit – the first figure is considering conic gear transmission and the second a cylindrical gear transmission, depending on the mounting and drive conditions and possibilities.

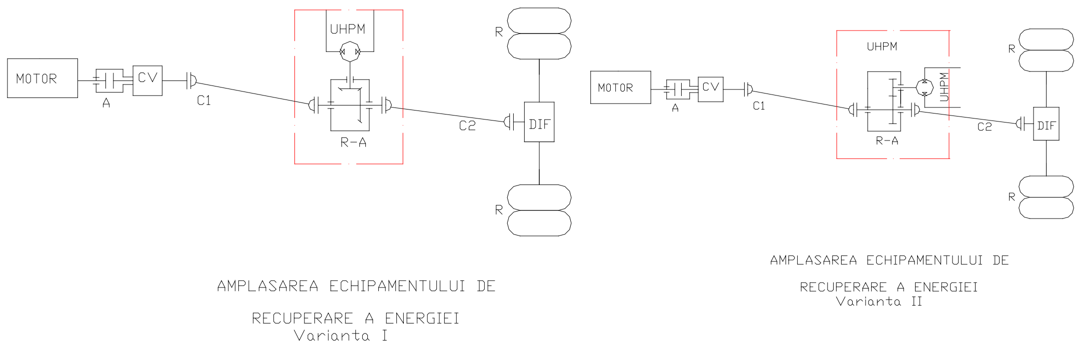

Fig. 5 Equipment setting

Equipment structure:

Engine assembly:

- Closed circle variable pump with bearing (*lagar*) hardener, with hydraulic servo control 1;
- Distribution system composed of electro hydraulic distributor (dispenser) S1,S2, S3 and proportional control (adjustment) distributor S4A,S4B;
- Protection system 3 composed of a manned valve and two check valves;
- The mechanically driven assembly from the gear axle 8 and monitored by the speed transducer (T)

Storage system:

- Storage batteries - (5)

Auxiliary elements:

- Storage tank (6);
- Pressure source (4) which include battery, filling and divider valves, pressure switch, pressure valve;
- Autonomous power supply (7) composed of battery and circuit breaker, powered by the vehicle system.

Main technical parameters of the braking energy recovery equipment

Modular equipment conceived for braking energy recovery has the following technical parameters:

- Maximum mechanical power	160kw/210 CP
- Maximum rotation speed at propshaft axle	500-600 rot/min
- Working fluid	hydraulic fluid
- Maximum operating pressure	315 bar
- Geometric volume of the pump	125-250 cm^3/rot
- Voltage	24 Vcc,

tested by installing it on a truck of approx. 5 tons.

4. Conclusions

Generally, we have two methods to recover energy in driving systems:

- On the one hand, recovering technologies and equipments have been developed for driving systems based on electric engines;
- On the other hand, a very high interest can be noticed in technological developments regarding energy recovery systems for heavy automotives, based on thermic engine drive, especially with diesel engines.

Regarding the braking energy recovery equipments for heavy automotives, the following conclusions can be drawn off: the existence of these equipments represents an innovative domain in the last 5 years, with a very small distribution in the world, and the hybrid drive, hydro-mechanical, that can reduce fuel consumption for heavy automotives, with all the positive outcomes arising from this, up to 20, 25 and even 40%. Also, these equipements will prolong the life of the braking equipment of the heavy automotives and thus will lead to the reduction of transportation costs.

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