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THE STUDY OF THE DYNAMICS OF THE FREE PISTON AT THE SINGLE REGIME RUNNING ENGINES

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ABSTRACT: The single regime engine is a new concept. By functioning in single regime running the optimization of fuel consumption and emissions is easier to be done. Another particularity of this engine is that it has no idle functioning. It has also a start-stop system. Basically, the thermo-hydraulic generator is an alternative to the actual propulsion systems for cars.
KEYWORDS: single regime running, thermal energy, hydraulic energy, mechanical energy

INTRODUCTION

The single regime running thermal engine transforms the energy produced by the burning of a fuel (gasoline, diesel fuel, unconventional fuel) in mechanical energy, generally under the form of rotational movement. The main parts of the engine are: the single regime running thermo-hydraulic generator TM, the hydraulic accumulator AH and the hydraulic motor MH (Figure 1).

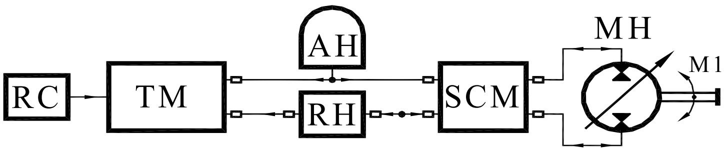


Figure 1. The constructive scheme of the thermal single regime running engines

The engine includes also the fuel tank RC and the tank for the hydraulic liquid RH. The monoregime thermo-hydraulic generator transforms the thermal energy produced by the burning of a fuel into hydrostatic energy, stoked in a hydraulic accumulator. The supply with energy of the hydraulic motor is done through the engine's command system SCM. The hydraulic engine transforms the hydrostatic energy into mechanical energy that's necessary to run the working machine.

The main part of the monoregime thermo-hydraulic generator is formed by two cylinders: the motor cylinder CM and the Hydraulic cylinder CH. The two cylinders are coaxially mounted and inside them the piston has rectilinear alternative movement (Figure 2).

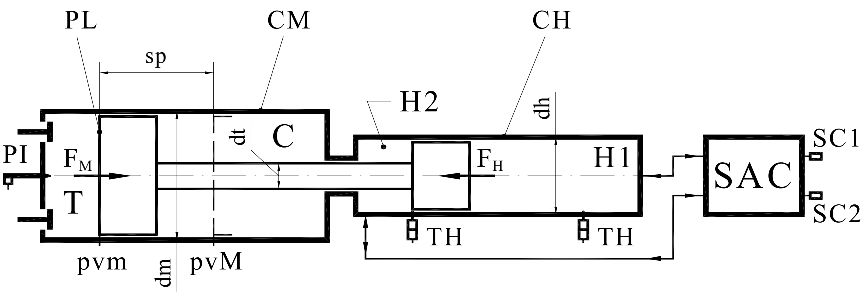


Figure 2. The functioning scheme of the thermal single regime running engines

The free piston is the only mobile part, without articulating elements. The start-stop of the piston at the end of the stroke is possible, without negatively affecting the engine functioning, because the speed (and the kinetic energy respectively) is zero in these points. Due to the piston movement, between the cylinder's walls and the piston four chambers with variable volume are formed: the thermal chamber T, the compression chamber T and the hydraulic chambers H1 and H2. The piston movement is done under the action of the pressure forces produced by the gases inside the thermal and the compression chambers and the pressure forces produced by hydraulic liquid in chambers H1 and H2. The cycle processes take part in the thermal chamber and in the compression *chamber the intake process for the boosting or an accumulation of pneumatic energy can be developed. In the hydraulic chambers are taking place the intake-delivery processes of the hydraulic liquid. The piston movement is coordinated by the automatic command system (SAC). The information regarding the piston position are furnished by the transducers TH. The piston stroke is between pvm (the point of minimum volume) and pvM (the point of maximum volume). The forces on the piston are* F_M *produced by the gas pressure inside the thermal and the compression chambers and force* F_H *produced by the pressure of the hydraulic liquid inside chambers H1 and H2. During a cycle* F_M *has great variation (exponentially) and force* F_H *is almost constant.*

The thermodynamic processes of the thermo-hydraulic generator cycle are identical with those from the internal combustion engines. Same as in ICE, the thermal cycle of the thermo-hydraulic generator can be done in two or in four strokes of the piston.

Single regime running engines are in the category of thermo-hydraulic hybrid engines. At these new engines two energy transformations are taking place: the thermal energy is transformed into hydrostatic energy (through the thermo-hydraulic generator) and the hydrostatic energy is transformed into mechanical energy (through hydraulic motors). The single regime running engines are characterized by the following particularities: single regime running (monoregime – the quantity of fuel introduced in the cylinder in every engine cycle is the same), no idle. It's estimated that following advantages can be obtained (regarding present engines): reduced fuel consumption and emissions (it's easier to optimize one regime instead of an infinity), constructive simplicity and greater reliability. Also, the single regime running engine can overtake, totally or partially, the transmission functions (the variation of the torque, rotational speed and rotational sense), including brake energy recovery.

FORCES AND MECHANICAL WORK

The piston PL has an alternative rectilinear movement inside two cylinders (engine and hydraulic), coaxially assembled (Figure 3). The engine's cylinder (with the bore d_m*) together with the piston is forming two chambers with variable volumes: the thermal chamber T (where the thermodynamically processes of the thermal cycle take place) and the compression chamber C (where intake processes of fresh charge can take place). The hydraulic cylinder (with the inside diameter* d_h*) together with the piston forms two variable volume chambers H1 and H2, called* $_{hydraulic}$ *chambers, where the suction-discharge of hydraulic liquid phenomena are produced. The pistons inside the two cylinders are connected by a rigid rod (with the diameter* d_t*). The distance between pvm and pvM is the stroke of the piston. The instantaneous position of the piston in respect with pvM is noted with x.*

During one cycle the piston realizes two strokes: the useful stroke (when the hydrostatic energy is accumulated) and the resistant stroke (hydrostatic energy is consumed). The movement of the piston between the two extreme points is made under the action of the following forces: F_m *[N] – the useful force developed in the engine's chambers (formed by the pressure force of the gases* F_{gm} *and the friction force* F_{fm}*);* F_h *[N] – the force developed in the hydraulic cylinder (formed by the pressure force of the hydraulic liquid* F_{lh} *and the friction forces* F_{fh}*). The intervals* $[x_k, x_{k+1}]$ *are chosen so that the functions of variation of the forces that action on the piston PL to be continuous inside one interval.*

The force of gases F_{gm} *inside the engine's cylinder that actions on the piston in an any point* $x \in [x_k \ x_{k+1}]$ *is given by the relation:*

$$F_{gm} = p_{gt} \cdot A_m - p_{gc} \cdot (A_m - A_t) \quad [N] \tag{1}$$

where: p_{gt} *[Pa] – the pressure inside the thermal chamber;* p_{gc} *[Pa] – the pressure of the gases inside the compression chamber;* A_m *[m*^{2}*] / the area of the cylinder inside section;* A_t *[m*^{2}*] – the area of the piston's rod section.*

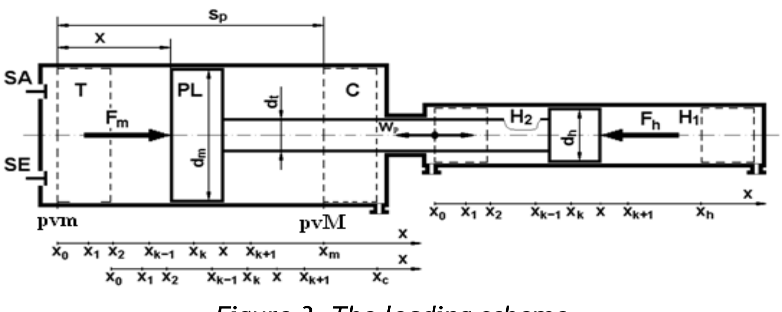


Figure 3. The loading scheme

The pressure p_{gt} *of the gases inside the thermal chamber T in position* $x \in [x_k, x_{k+1}]$ *of the piston PL is obtained with the relation:*

$$p_{gt} = p_{tk} \cdot \left(\frac{V_{Tk}}{V_T}\right)^{n_{tk}} = p_{tk} \cdot \left(\frac{V_{T0} + x_k \cdot A_m}{V_{T0} + x \cdot A_m}\right)^{n_{tk}} = p_{tk} \cdot \left(\frac{1 + b_t \cdot x_k}{1 + b_t \cdot x}\right)^{n_{tk}} \tag{2}$$

where: p_{tk} - the pressure inside the thermal chamber in point x_k; V_{Tk} - the instantaneous volume of the thermal chamber in point x_k; V_T - the instantaneous volume of the thermal chamber in point x; V_{T0} - the instantaneous volume of the thermal chamber in point x_0; $b_t=A_m/V_{T0}$ - the coefficient of the thermal chamber; n_{tk} - the polytrophic index of the gases inside the thermal chamber in the interval $x \in [x_k\ \ x_{k+1}]$.

The pressure p_{gc} of the fresh charge inside the compression chamber C in position $x \in [x_k, x_{k+1}]$ of the piston PL is:

$$p_{gc} = p_{ck} \cdot \left(\frac{V_{Ck}}{V_C} \right)^{n_{ck}} = p_{ck} \cdot \left[\frac{V_{C0} - x_k \cdot (A_m - A_t)}{V_{C0} - x \cdot (A_m - A_t)} \right]^{n_{ck}} = p_{ck} \cdot \left(\frac{1 - b_c \cdot x_k}{1 - b_c \cdot x} \right)^{n_{ck}} \quad (3)$$

where: p_{ck} - the pressure of the fresh charge inside the compression chamber in point x_k; V_{Ck} - the instantaneous volume of the compression chamber in point x_k; V_C - the instantaneous volume of the compression chamber in point x; V_{C0} - the instantaneous volume of the compression chamber in point x_0; $b_c=(A_m-A_t)/V_{C0}$ - the coefficient of the compression chamber; n_{tk} - the polytrophic index of the gases inside the compression chamber in the interval $x \in [x_k\ \ x_{k+1}]$.

The polytrophic indexes $n_{t,c}$ of the gases inside the thermal chamber or the compression chamber can take different values in function of the thermal processes that take place in these chambers: $n_{t,c}=0$ - constant pressure processes (p=ct.); $n_{t,c}=1{,}00$ - isothermal processes (T=ct.); $n_{t,c}=1{,}32\text{-}1{,}39$ - compression processes; $n_{t,c}=1{,}25\text{-}1{,}32$ - detention processes; $n_{t,c}=\infty$ - constant volume processes (V=ct.).

If relation (1) is correlated with relations (2) and (3) it can be obtained the calculus formula for the force F_{gm} developed by the pressure of the gases inside the engine's cylinder that actions on the piston in an any point $x \in [x_k\ \ x_{k+1}]$:

$$F_{gm} = \frac{b_t \cdot (n_{tk} - 1) \cdot e_{tk}}{(1 + b_t \cdot x)^{n_{tk}}} - \frac{b_c \cdot (n_{ck} - 1) \cdot e_{ck}}{(1 - b_c \cdot x)^{n_{ck}}} \quad (4)$$

where: e_{tk} - the coefficient of the force produced by the pressure of the gases inside the thermal chamber T; e_{ck} - the coefficient of the force produced by the pressure of the gases inside the thermal chamber C.

The values of the coefficients e_{tk} and e_{ck} are determined with the relations:

$$e_{tk} = \frac{p_{tk} \cdot A_m \cdot (1 + b_t \cdot x_k)^{n_{tk}}}{b_t \cdot (n_{tk} - 1)}; \quad e_{ck} = \frac{p_{ck} \cdot (A_m - A_t) \cdot (1 - b_c \cdot x_k)^{n_{ck}}}{b_c \cdot (n_{ck} - 1)}$$

Notation: p_{es} - the men elastic pressure of the rings; p_{si} - the radial pressure on the ring i. It's considered that $\sum_{i=1}^{n_{sm}} (p_{es} + p_{si}) \cong p_{gt}$.

The friction force F_{fm} between the engine's cylinder and the piston's rings can be established with the relation:

$$F_{fm} = \mu_{sm} \cdot \pi \cdot d_m \cdot b_{sm} \cdot \sum_{i=1}^{n_{sm}} (p_{es} + p_{si}) = \frac{r_{fm} \cdot b_t \cdot (n_{tk} - 1) \cdot e_{tk}}{(1 + b_t \cdot x)^{n_{tk}}} \quad [N] \quad (5)$$

where: μ_{sm} - the friction coefficient; n_{sm} - the number of rings in the engine's cylinder; b_{sm} - the height of the ring; $r_{fm} = (4 \cdot \mu_{sm} \cdot b_{sm})/d_m$ - the friction ratio in the engine's cylinder.

The total force developed by the pressure of the gases inside the engine's cylinder for $x \in [x_k, x_{k+1}]$ is obtained with the following relation:

$$F_m = F_{gm} \mp F_{fm} = \frac{(1 \mp r_{fm}) \cdot b_t \cdot (n_{tk} - 1) \cdot e_{tk}}{(1 + b_t \cdot x)^{n_{tk}}} - \frac{b_c \cdot (n_{ck} - 1) \cdot e_{ck}}{(1 - b_c \cdot x)^{n_{ck}}} \quad (6)$$

" - " - if the piston moves from pvm to pvM;
" + " - if the piston moves from pvM to pvm;

The mechanical work developed in the engine's cylinder in the distance (x - x_k), for $x \in [x_k, x_{k+1}]$, if the piston is moving from pvM to pvm, is:

$$L_m = \int_{x_k}^{x} F_m \cdot dx = (1 \mp r_{fm}) \cdot e_{tk} \cdot [f_{gt}(x_k) - f_{gt}(x)] + e_{ck} \cdot [f_{gc}(x_k) - f_{gc}(x)] \quad (7)$$

where: $f_{gt}(x) = \frac{1}{(1 + b_t \cdot x)^{n_{tk} - 1}}; \quad f_{gc}(x) = \frac{1}{(1 - b_c \cdot x)^{n_{ck} - 1}}$

The function $p_{gt}(x)$ of variation of the pressure in chamber T and the function $p_{gc}(x)$ of the variation of the pressure in chamber C are given by the relations:

$$p_{gt}(x) = \frac{b_t \cdot (n_{tk} - 1) \cdot e_{tk}}{A_m \cdot (1 + b_t \cdot x)} \cdot f_{gt}(x); \quad p_{gc}(x) = \frac{b_c \cdot (n_{ck} - 1) \cdot e_{ck}}{(A_m - A_t) \cdot (1 - b_c \cdot x)} \cdot f_{gc}(x)$$

The force F_{lh} in any point $x \in [x_k, x_{k+1}]$ in the hydraulic cylinder produced by the liquid's pressure is determined with the following relation:

$$F_{lh} = p_{h1} \cdot A_h - p_{h2} \cdot (A_h - A_t) \qquad (8)$$

where: p_{h1} [Pa] – the pressure inside chamber H1; p_{h2} [Pa] – the pressure inside chamber H2; A_h [m^2] – the area of the section of the hydraulic cylinder.

Notations: p_{ha}, p_{hr} – the pressure of the hydraulic liquid inside the accumulator and inside the tank, respectively; λ_{h1}, λ_{h2} – the hydraulic resistance coefficient in chambers H1 and H2, respectively; l_1, l_2, d_1, d_2 – the lengths and the diameters of the connecting pipes with chambers H1 and H2, respectively. Pressure p_{h1} is calculated with Bernoulli's relation:

$$p_{h1} = p_{1(a,r,z)} \pm \lambda_{h1} \cdot \frac{\rho_h}{2} \cdot \frac{l_1}{d_1} \cdot w_p^2 + l_1 \cdot \rho_h \cdot \frac{dw_p}{d\tau} \qquad (9)$$

$$p_{1(a,r,z)} = \begin{cases} p_{ha} & - if\ (H1)\ is\ connected\ with\ accumulator; \\ p_{rz} & - if\ (H1)\ is\ connected\ with\ tank. \end{cases}$$

Similarly, the pressure p_{h2} is:

$$p_{h2} = p_{2(a,r,z)} \pm \lambda_{h2} \cdot \frac{\rho_h}{2} \cdot \frac{l_2}{d_2} \cdot w_p^2 - l_2 \cdot \rho_h \cdot \frac{dw_p}{d\tau} \qquad (10)$$

$$p_{2(a,r,z)} = \begin{cases} p_{ha} & - if\ (H2)\ is\ connected\ with\ accumulator; \\ p_{rz} & - if\ (H2)\ is\ connected\ with\ tank. \end{cases}$$

By correlating relation (8) with relations (9) and (10) results the formula for the calculus of force F_{lh} produced in any point $x \in [x_k\ x_{k+1}]$ by the pressure of the liquid from the hydraulic cylinder:

$$F_{lh} = e_{hk} \pm e_v \cdot w_p^2 + e_m \cdot \frac{dw_p}{d\tau} \quad [N] \qquad (11)$$

Coefficients e_{hk}, e_v şi e_m are determined with the relations:

$$e_{hk} = p_{1(a,r,z)} \cdot A_h - p_{2(a,r,z)} \cdot (A_h - A_t)$$

$$e_v = \frac{\rho_h}{2} \cdot \left[\lambda_{h1} \cdot \frac{l_1}{d_1} \cdot A_h - \lambda_{h2} \cdot \frac{l_2}{d_2} \cdot (A_h - A_t) \right]$$

$$e_m = \rho_h \cdot \left[l_1 \cdot A_h + l_2 \cdot (A_m - A_t) \right]$$

The friction force F_{fh} in the hydraulic cylinder is produced by: the friction with the rings which have the height b_{sh}, by the viscous friction in the hydrostatic bearing which has the witness b_l and by the friction between the rod and the gasket with the witness b_{gr} (Figure 4).

The friction force $F_{f.sg}$ between the piston's rings and the interior surface of the hydraulic cylinder is directly proportional with the radial force that actions on the rings (produced by the mean elastic pressure p_{es} of the rings and by the pressure p_{si} of the hydraulic liquid. Experimentally, it was been established that a number n_{sh}=2 of rings are sufficient and that $\sum_{i=1}^{n_{sh}} (p_{es} + p_{si}) \cong p_{ha}$.

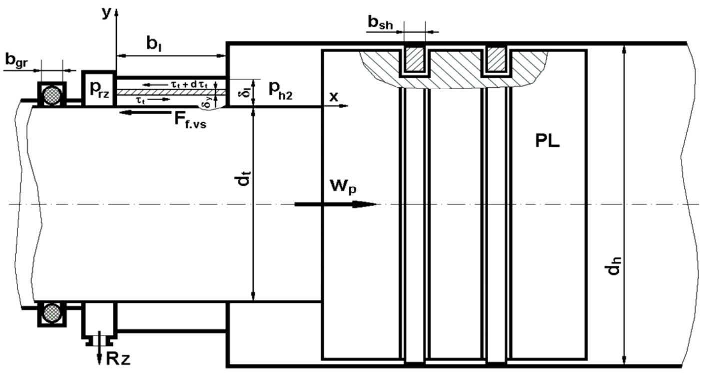


Figure 4. The scheme of the sealing inside the hydraulic cylinder

Notation: μ_{sh} – the friction coefficient between the rings and the hydraulic cylinder surface. The relation for the calculus of the friction force $F_{f.sg}$ [N] is:

$$F_{f.sg} = \mu_{sh} \cdot \pi \cdot d_h \cdot b_{sh} \cdot \sum_{i=1}^{n_{sh}} (p_{es} + p_{si}) = \mu_{sh} \cdot \pi \cdot d_h \cdot b_{sh} \cdot p_{ha} \qquad (12)$$

Notations: η [kg·s/m^2] -the dynamic viscosity of the hydraulic liquid; τ_t [N/m^2] – the tangential tension between the layers of fluid; d_t [m] – the rod's diameter; w [m/s] – the liquid's speed through the annular slot of the hydrostatic bearing. From the balance equation of the elemental ring with the length b_l and thickness δ_y from the annular slot of the hydrostatic bearing results the relation for the calculus of the tangential tension between the layers of liquid τ_t:

$$(p_{rz} - p_{h2}) \cdot \pi \cdot d_t \cdot dy - d\tau_t \cdot \pi \cdot d_t \cdot b_l = 0 \Rightarrow d\tau_t = \frac{p_{rz} - p_{h2}}{b_l} \cdot dy \Rightarrow$$

$$\left.\begin{aligned}\tau &= \int d\tau_t = \frac{p_{rz}-p_{h2}}{b_l}\cdot y + C_1\\ \tau &= -\eta\cdot\frac{dw}{dy}\quad - Newton's\ law\end{aligned}\right\}\Rightarrow$$

$$dw = \frac{p_{h2}-p_{rz}}{\eta\cdot b_l}\cdot y\cdot dy - \frac{C_1}{\eta}\cdot dy \Rightarrow \quad w=\int dw = \frac{p_{h2}-p_{rz}}{2\cdot\eta\cdot b_l}\cdot y^2 - \frac{C_1}{\eta}\cdot y + C_2\ .$$

The boundary conditions are:

$$y=0;\quad w=\pm w_p \Rightarrow C_2=\pm w_p \quad y=\delta_l;\quad w=0\Rightarrow \frac{C_1}{\eta}=\frac{p_{h2}-p_{rz}}{2\cdot\eta\cdot b_l}\cdot\delta_l \pm \frac{w_p}{\delta_l}$$

$$w=\frac{p_{h2}-p_{rz}}{2\cdot\eta\cdot b_l}\cdot(y^2-\delta_l\cdot y)\pm w_p\cdot\left(1-\frac{y}{\delta_l}\right)\Rightarrow \quad \tau_t=-\eta\cdot\frac{dv}{dy}=-\frac{p_{h2}-p_{rz}}{2\cdot b_l}\cdot(2\cdot y-\delta_l)\pm\frac{\eta\cdot w_p}{\delta_l} \qquad (13)$$

It can be observed that the maximum tangential tension $\tau_{t\ max}$ is obtained from relation (13) for y=0. The pressure p_{h2} is approximated with the pressure $p_{2(a,rz)}$. The viscous friction force $F_{f.vs}$ in the hydrostatic bearing is:

$$F_{f.vs}=\pi\cdot d_t\cdot b_l\cdot\tau_{t\,max}=\pi\cdot d_t\cdot\left(\frac{p_{2(a,rz)}-p_{rz}}{2}\cdot\delta_l\pm\eta\cdot\frac{b_l}{\delta_l}\cdot w_p\right) \qquad (14)$$

Notation: μ_{gr} - the friction coefficient between the sealing gasket and the rod. The relation for the calculus of the friction force $F_{f.gr}$ is:

$$F_{f.gr}=\mu_{gr}\cdot\pi\cdot d_t\cdot b_{gr}\cdot p_{rz} \quad [N] \qquad (15)$$

Notation: υ [m^2/s] - the kinematic viscosity of the hydraulic fluid. By correlating relations (12), (14) and (15) one can obtain the formula for the calculus of the force F_{fh} in the hydraulic cylinder in any point $x \in [x_k\ \ x_{k+1}]$:

$$F_{fh}=F_{f.sg}+F_{f.vs}+F_{f.gr}=e_{fk}\pm e_{fv}\cdot w_p \quad [N] \qquad (16)$$

Coefficients e_{fk} şi e_{fv} are determined with the relations:

$$e_{fk}=\pi\cdot\left(\mu_{sh}\cdot d_h\cdot b_{sh}\cdot p_{ha}+\frac{p_{2(a,rz)}-p_{rz}}{2}\cdot\delta_l\cdot d_t+\mu_{gr}\cdot d_t\cdot b_{gr}\cdot p_{rz}\right)$$

$$e_{fv}=\pi\cdot\eta\cdot\frac{b_l}{\delta_l}\cdot d_t\ \ sau\ \ e_{fv}=\pi\cdot\rho_h\cdot\nu\cdot\frac{b_l}{\delta_l}\cdot d_t$$

The value of the force F_h for $x \in [x_k\ \ x_{k+1}]$ produced by the pressure of the liquid inside the hydraulic cylinder is calculated with the following relation:

$$F_h=F_{lh}\pm F_{fh}=(e_{hk}\pm e_{fk})\pm e_{fv}\cdot w_p\pm e_v\cdot w_p^2+e_m\cdot\frac{dw_p}{d\tau} \qquad (17)$$

Notation: w_{pmax} [m/s] - the piston's maximum speed during one stroke. In the point in which the acceleration is zero ($dw_p/d\tau=0$, $w_p=w_{pmax}$), relation (17) for the calculus of force F_h in the hydraulic cylinder is:

$$F_{h|w_p=w_{p\,max}}=(e_{hl}\pm e_{fk})\pm e_{fv}\cdot w_{p\,max}\pm e_v\cdot w_{p\,max}^2 \qquad (18)$$

The mechanical work developed by forces in the hydraulic cylinder on the distance($x-x_k$), for $x \in [x_k\ \ x_{k+1}]$ (if the piston moves to pvM or $x \in [x_{k-1}\ \ x_k]$ (if the piston moves to pvm) is:

$$L_h=\int_{x_k}^{x}F_h\cdot dx=(e_{hk}\pm e_{fk})\cdot(x-x_k)\pm e_{fv}\cdot\int_{x_k}^{x}w_p\cdot dx\pm e_v\cdot\int_{x_k}^{x}w_p^2\cdot dx+e_m\cdot\int_{x_k}^{x}\frac{dw_p}{d\tau}\cdot dx$$

Notation: w_{pm} the mean piston's speed on the interval [$x\ x_k$]. The formulas for the calculus of the integrals are the following:

$$\int_{x_k}^{x}w_p\cdot dx=w_{pm}\cdot(x-x_k);\quad \int_{x_k}^{x}w_p^2\cdot dx=\int_{x_k}^{x}w_p\cdot w_p\cdot dx=w_{pm}^2\cdot(x-x_k);$$

$$\int_{x_k}^{x}\frac{dw_p}{d\tau}\cdot dx=\int_{x_k}^{x}w_p\cdot dw_p=\left.\frac{w_p^2}{2}\right|_{x_k}^{x}=\frac{w_p^2(x)-w_p^2(x_k)}{2}$$

The relation for the calculus of the mechanical work L_h in the hydraulic cylinder is:

$$L_h=(e_{hk}\pm e_{fk})\cdot(x-x_k)\pm e_{fk}\cdot w_{pm}\cdot(x-x_k)\pm e_v\cdot w_{pm}^2\cdot(x-x_k)+e_m\cdot\frac{w_p^2(x)-w_p^2(x_k)}{2} \qquad (19)$$

If relations (7) and (19) are correlated, it is obtained the formula for the calculus of the total mechanical work L_{mt} developed by the pressure forces on the distance $(x\text{-}x_k)$ for $x \in [x_k \ \ x_{k+1}]$ if the piston PL is moving towards pmV and for $x \in [x_{k-1} \ \ x_k]$ if the piston PL is moving towards pmv.

$$L_{mt} = L_m - L_h = L_k \mp w_{pm}\cdot(e_{fv} + e_v \cdot w_{pm})\cdot(x - x_k) - e_m \cdot \frac{w_p^2(x) - w_p^2(x_k)}{2} \qquad (20)$$

where: $L_k = (1 \mp r_{fm})\cdot e_{tk}\cdot[f_{gt}(x_k) - f_{gt}(x)] + e_{ck}\cdot[f_{gc}(x_k) - f_{gc}(x)] - (e_{hk} \pm e_{fk})\cdot(x - x_k)$

The calculus of the total mechanical work is necessary to determine the dimensional parameters of the engine and the kinematic parameters of the piston.

THE CALCULUS OF THE DIMENSIONAL PARAMETERS

For the calculus of the dimensional parameters (the piston diameter dh, the rod diameter d_t and the piston's mass m_p) the theorem of kinetic energy for one stroke of the piston is applied. The variation of the kinetic energy for one stroke of the piston is zero.

The theorem of the kinetic energy in the useful stroke (pmv → (pmV) is applied and the viscous friction forces are neglected. Results: $L_{mt}=0 \Rightarrow$

$$(1-r_{fm})\cdot\sum_{k=0}^{m-1} e_{tk}\cdot[f_{gt}(x_k)-f_{gt}(x_{k+1})]+\sum_{k=0}^{c-1} e_{ck}[f_{gc}(x_k)-f_{gc}(x_{k+1})]-(e_{h0}+e_{f0})\cdot s_p=0 \qquad (21)$$

The theorem of the kinetic energy in the resistent stroke (pmV → (pmv) is applied and the viscous friction forces are neglected. Results: $L_{mt}=0 \Rightarrow$

$$(1+r_{fm})\cdot\sum_{k=1}^{m} e_{tk}\cdot[f_{gt}(x_k)-f_{gt}(x_{k-1})]+\sum_{k=1}^{c} e_{ck}[f_{gc}(x_k)-f_{gc}(x_{k-1})]+(e_{h1}+e_{f1})\cdot s_p=0 \qquad (22)$$

The solution for the system of equations (1.21) and (1.22) represents the value of the hydraulic cylinder diameter and the value of the rod's diameter d_t.

The piston's mass m_p is established at a value that should limit the maximum speed of the piston in the useful stroke at w_{pmax}. The acceleration of the piston in the point of maximum speed x_a is zero. In this point the forces developed in the engine's cylinder and in the hydraulic cylinder are equal. The following relation results:

$$m\cdot a = F_m(x) - F_h \Rightarrow F_m(x_a) = F_h\big|_{w_p=w_{p\,max}} \qquad (23)$$

The force $F_h\big|_{w_p=w_{p\,max}}$ is given by the relation (1.18). By replacing in the relation the equation for the calculus of distance $x_a \in [x_k \ \ x_{k+1}]$ is obtained:

$$\frac{(1-r_{fm})\cdot b_t\cdot(n_{tk}-1)\cdot e_{tk}}{(1+b_t\cdot x_a)^{n_{tk}}} - \frac{b_c\cdot(n_{ck}-1)\cdot e_{ck}}{(1-b_c\cdot x_a)^{n_{ck}}} = F_h\big|_{w_p=w_{p\,max}} \qquad (24)$$

The theorem of kinetic energy is applied between the initial position x=0 and the final position x_a of the piston and the relation for the calculus of the piston's mass m_p is obtained:

$$\frac{m_p\cdot w_{p\,max}^2}{2} = \sum L_{mt} \Rightarrow m_p = \frac{2}{w_{p\,max}^2}\cdot\sum L_{mt} \qquad (25)$$

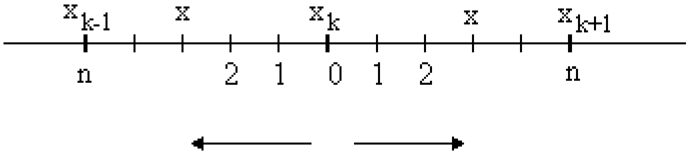
The piston mass increases direct proportional with the acceleration mechanical work and decreases inversely proportional with the square of piston's maximum speed.

THE CALCULUS OF THE KINEMATIC PARAMETERS

The relations for the calculus of the kinematic parameters are determined by applying the theorem of kinetic energy between two points of the piston's stroke. If the theorem of the kinematic energy id applied between an initial point x_k and any point $x \in [x_k \ \ x_{k+1}]$ (if the piston moves towards pmV) or $x \in [x_k \ \ x_{k-1}]$ (if the piston moves towards pmv) the following equation it's obtained:

$$\frac{m_p\cdot w_p^2(x)}{2} - \frac{m_p\cdot w_p^2(x_k)}{2} = L_{mt} \qquad (26)$$

For the calculus of equation (26) the method of the finite differences is applied. The interval $[x_k \ \ x_{k+1}]$ is divided in n subintervals with the length $\delta x_i = (x_{k+1} - x_k) / n = x_i - x_{i-1}$, $i = 1, 2,\ldots, n$, $x_0 = x_k$, $x_n = x_{k+1}$. In the same way, the interval $[x_k \ \ x_{k-1}]$ is divided in n subintervals with the length $\delta x_i = (x_{k-1} - x_k) / n = x_i - x_{i-1}$, $i = 1, 2,\ldots, n$, $x_0 = x_k$, $x_n = x_{k-1}$.



Notation: $\Delta w_i = w_{i+1} - w_i$ the difference between the final speed and the initial speed in subinterval δx_i. The mean speed w_{mi} in the subinterval δ_i is:

$$w_{mi} = \frac{w_{i+1} + w_i}{2} = \frac{\Delta w_i}{2} + w_i \qquad (27)$$

On the subinterval δx_i is applied the theorem of the kinetic energy, resulting the equation:

$$\frac{m_p \cdot w_{i+1}^2}{2} - \frac{m_p \cdot w_i^2}{2} = L_{ki} \mp w_{mi} \cdot (e_{fv} + e_v \cdot w_{mi}) \cdot \delta x_i - e_m \cdot \frac{w_{i+1}^2 - w_i^2}{2} \qquad (28)$$

where: $L_{ki} = (1 \mp r_{fm}) \cdot e_{tk} \cdot [f_{gt}(x_i) - f_{gt}(x_i + \delta x_i)] + e_{ck} \cdot [f_{gc}(x_i) - f_{gc}(x_i + \delta x_i)] - (e_{hk} \pm e_{fk}) \cdot \delta x_i$

The coefficient e_m represents the mass of the moving liquid. Notation: m_{in} - the inertial mass (the mass in movement). The mass m_{in} is obtained with the relation:

$$m_{in} = m_p + e_m \quad [kg] \qquad (29)$$

After processing, equation (28), has the form:

$$\Delta w_i^2 + 2 \cdot (w_i \mp r_{1i}) \cdot \Delta v_i \pm 4 \cdot r_{1i} \cdot w_i - r_{2i} = 0 \qquad (30)$$

where: $r_{1i} = \frac{e_{fv} + e_v \cdot w_i}{2 \cdot m_{in} \pm e_v \cdot \delta x_i} \cdot \delta x_i$; $r_{2i} = \frac{4 \cdot L_{ki}}{2 \cdot m_{in} \pm e_v \cdot \delta x_i}$.

The solutions for the equation (30) are:

1. If the piston PL is moving from pmv to pmV:

$$\rightarrow \Delta w_i = -(w_i + r_{1i}) + \sqrt{(w_i - r_{1i})^2 + r_{2i}} \qquad (31)$$

2. If the piston PL is moving from pmV to pmv:

$$\leftarrow \Delta w_i = -(w_i - r_{1i}) - \sqrt{(w_i + r_{1i})^2 + r_{2i}} \qquad (32)$$

The movement of the piston in the interval δx_i is approximated with a uniformly accelerated movement and the duration of the movement $\Delta\tau_i$ on this interval is:

$$\Delta\tau_i = \tau_{i+1} - \tau_i = \frac{\delta x_i}{w_{mi}} \quad [s] \qquad (33)$$

The acceleration a_i of the piston on the interval δx_i (considered to be constant) is:

$$a_i = \frac{\Delta w_i}{\Delta\tau_i} \quad [m/s^2] \qquad (34)$$

CONCLUSIONS

In conclusion, the kinematic parameters of the piston have been determined also in function of the viscosity of the hydraulic fluid. In this way, it's possible to study the influence of the viscosity of the hydraulic fluid on the characteristic points of the piston's stroke that influences the functioning of the thermo-pump.

The results of this research permitted the development of the design methodology of the single regime running engines. In project financed by the Hungarian-Romanian Cross-border Programme HURO 2007-2013 the design of an experimental model was developed, necessary to validate the results.

In Figure 5 is presented the constructive scheme of the thermo-hydraulic generator.

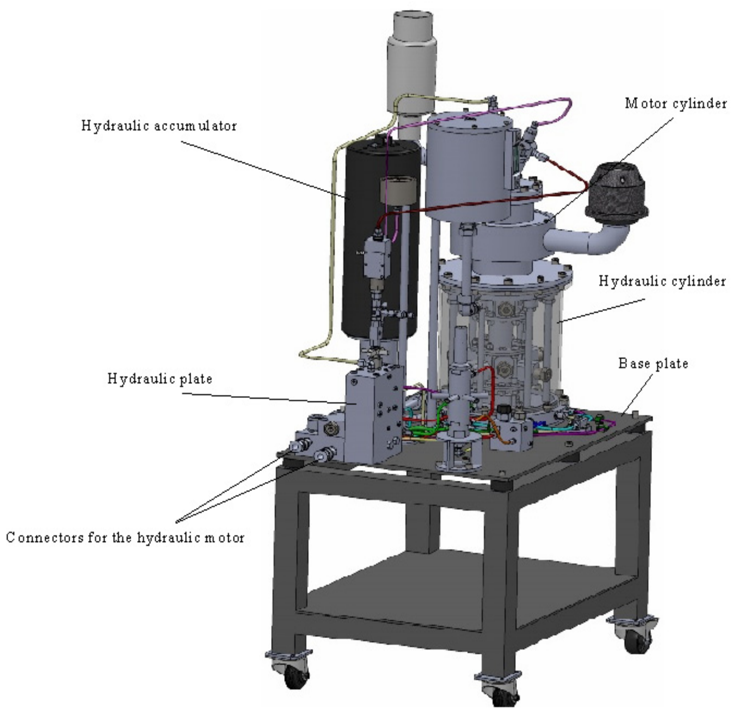


Figure 5. The constructive design of the thermo-hydraulic generator

The hydraulic plate is practically the command and control system. It includes valves and distributors. The hydraulic plate commands the start-stop system for the thermal engine and the supply of the hydraulic motor.

For reducing vibrations generated during the functioning, the base plate is mounted on dumpers.

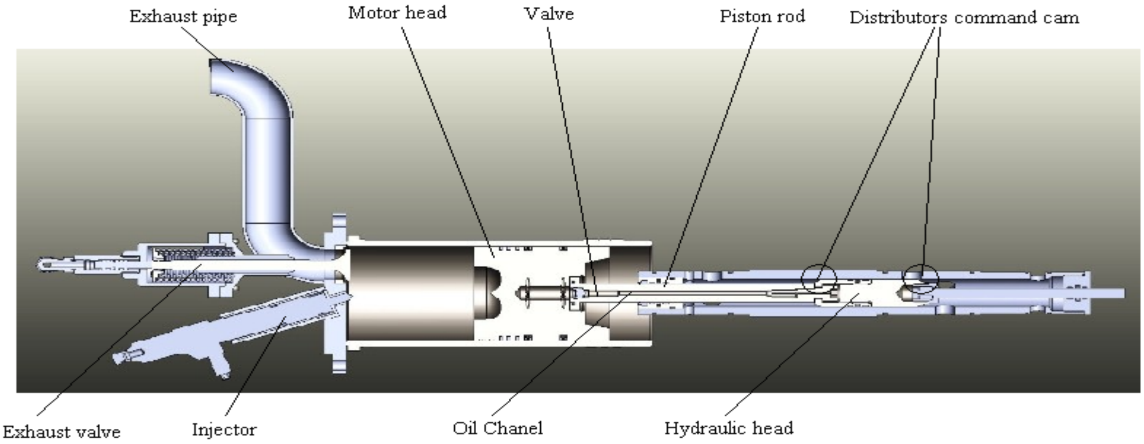


Figure 6. The free piston design

The design of the free piston is presented in Figure 6. The piston rings lubrication is assured through a channel manufactured along the piston, oil pressure being controlled by a valve. The start-stop system is controlled through the pressure inside the hydraulic accumulator. If the pressure reaches a maximum value, the piston automatically stops at the end of the stroke. As the pressure drops, when the minimum value is reached, the automatic control system SAC commands the start of the functioning of the thermal engine.

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