THE MAIN PARAMETERS OF THE MONOREGIME THERMAL ENGINES

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Abstract: The monoregime thermal engine is a new concept. The thermodynamic processes that take place in a monoregime thermal engine are similarly with those in a classic reciprocating engine. The problem is to determine the instantaneous volume of the burning chamber, because the piston is free and has a linear movement imposed by the pressures in the thermal chamber and in the hydraulic chamber repectivelly.

1. GENERAL CONSIDERATIONS

The functioning scheme of the monoregime thermal engines is presented in fig. 1..

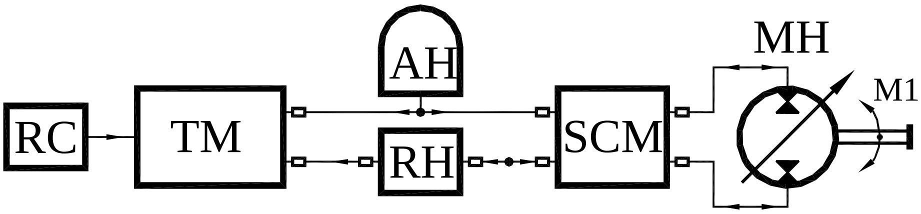

Figure 1. The functioning scheme of the monoregime thermal engines

The main parts are: the fuel tank RC, the thermo-hydraulic generator TM, the hydraulic accumulator AH, the tank for the hydraulic liquid RH, the automatic command system SCM and the hydraulic motor MH.

2. THE MONOREGIME THERMO_HYDRAULIC GENERATOR

The main component of the thermo-hydraulic monoregime generator is formed by two cylinders: the motor cylinder CM and the hydraulic cylinder CH, both mounted co-axially and inside them the free piston PL has an alternative rectilinear movement (fig. 2).

The free piston is the only mobile part, with no articulating elements. The stop- start of the piston at the end of the stroke does not affect negatively the functioning of the generator because the speed, the kinetically energy respectively is zero at the end of the stroke. Due to the piston movement, between the cylinder walls and the piston four variable volume chambers are formed: the thermal chamber T, the compression chamber C and hydraulic chambers H1 and H2. The piston displacement takes place under the action of the pressure forces produced by the gases from the thermal chamber and from the compression chamber and the pressure forces produced by the hydraulic liquid from the hydraulic chambers H1 and H2. In the thermal chamber take place the processes of the motor cycle, in the compression chambers take place the intake process necessary for the boosting of the engine or the accumulation of pneumatic energy necessary to produce the displacement of the piston.

In the hydraulic chambers take place the intake-discharged process of the hydraulic liquid. The piston movement is coordinated by the automatic command system SAC. The information regarding the piston position is provided by the transducers TH.

The piston stroke is between pvm (the point of minimum volume) and pvM (the point of maximum volume). The forces acting on the piston are: the force F_M developed by the gas pressure in the thermal and in the compression chamber and force F_H created by the hydraulic liquid pressure inside the hydraulic chambers H1 and H2. During the piston stroke force F_M has a great variation (exponentially) while force F_H is almost constant.

3. THE CALCULUS OF THE MAIN PARAMETERS

Further are presented *the calculus relations* for the design of a **thermo-hydraulic generator** with a simplified cycle formed by three thermodynamic processes: compression, burning and detension, considered theoretical processes (adiabatic and constant volume). The thermal cycle takes place in two strokes of the piston **PL** (two stroke cycles – fig. 2).

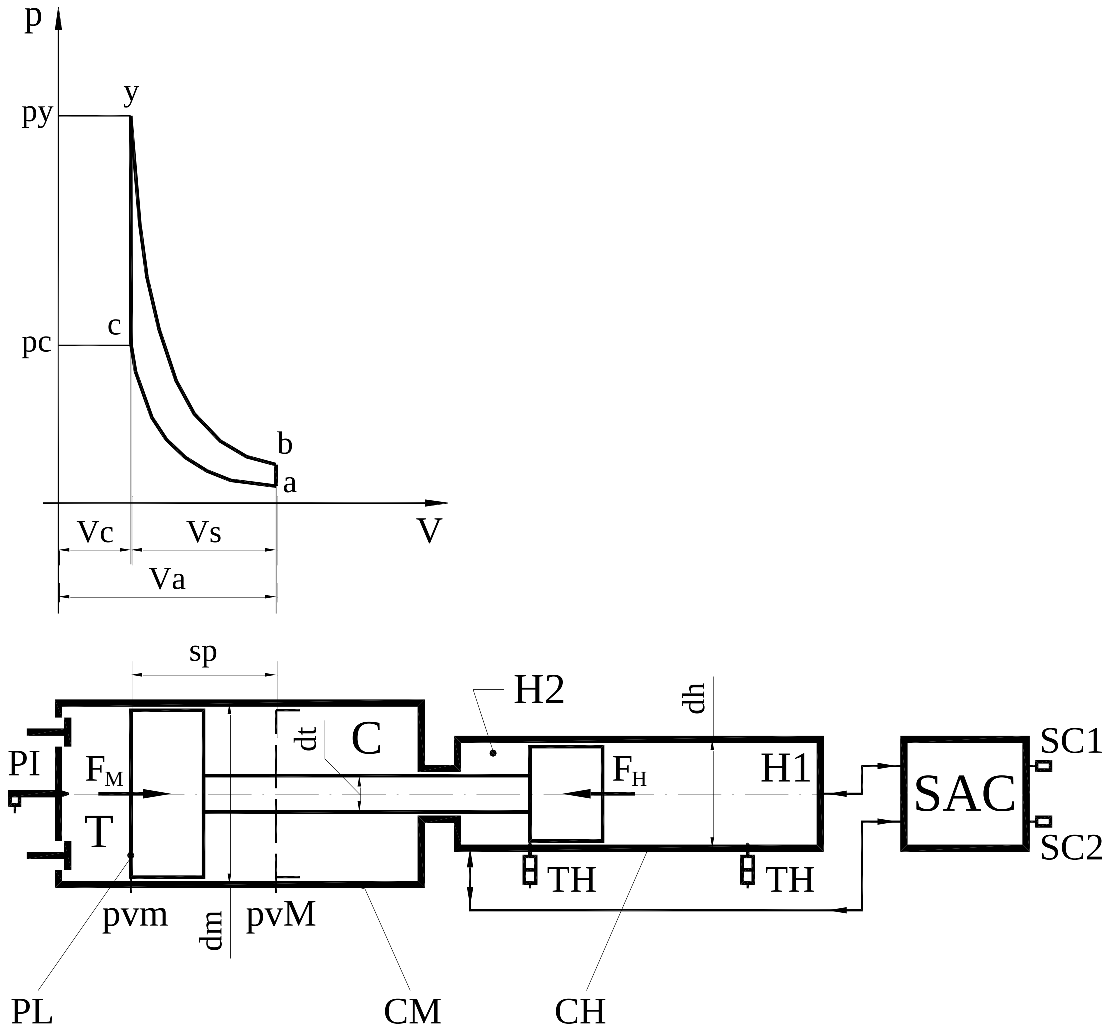

Figure 2. Dyagram p-V

<u>Time I</u> takes place when the piston moves from point **pvM** (the volume of the thermal chamber **T** is maximum) to point **pvm** (the volume of the thermal chamber **T** is minimum). If the pressure of the liquid inside the hydraulic accumulator **AH** is the maximum admitted, trough the automatic command system **SAC**, the piston is blocked in

pvm until the pressure drops under the minimum admitted value. In this moment, chambers **H1** and **H2** are connected with the accumulator. Due to the difference between sections of the piston and of the rod inside the hydraulic cylinder, the piston **PL** is moved towards **pvm**. The resistant stroke of the piston takes place. In the thermic chamber **T** the compression of the air takes place (**a-c**), this is considered an adiabatic process with the index $\boldsymbol{k_a}$.

When the piston is in **pvm** the injector **In** produces the fuel spraying in the compressed air. This autoignites and the burning process starts. The burning (**c-y**) is considered to be a constant volume process, with the pressure raising ratio $\lambda = p_y / p_c$.

Time II takes place when the piston moves from the point of minimum volume **pvm** to the point of maximum volume **pvM**. The working fluid (burned gases) resulted after the burning actions uppon the piston with pressure *p*. The useful piston stroke takes place. The detention of the burned gases (y-d) is considered an adiabatic process with index k_a. Under the action of the moving piston, the hydraulic liquid inside the chamber **H1** is discharged in the accumulator **AH**, and in chamber **H2** hydraulic liquid is sucked from the tank **Rz**.

To determine the main parameters of the thermo-hydraulic generator, the following notations are used: V_s – the displacement of the engine [m^3]; p_a – the pressure of the intake fresh charge [Pa]; $\varepsilon = \boldsymbol{V_a} / \boldsymbol{V_c}$ the compression ratio; p_{ha} – the hydraulic liquis's pressure inside the accumulator [Pa]; $\boldsymbol{r_{cd}} = \boldsymbol{s_p} / \boldsymbol{d_m}$ -the stroke-bore ratio. The pressure of the liquid inside the tank is not taken in consideration.

The bore is determined with the following relation:

$$d_m = \sqrt[3]{\frac{4 \cdot V_S}{\pi \cdot r_{cd}}} \ [m] \tag{1}$$

The stroke s_p is given by the relation:

$$s_p = r_{cd} \cdot d_m \ [m] \tag{2}$$

If the relation $E_F = \eta_m \cdot L_M = \eta_m \cdot |L_F|$ is applied to the useful stroke of the engine, results the equation:

$$\eta_m \cdot L_{yd} = L_{ham} \tag{3}$$

where: L_{yd} is the mechanical work developed by the burning gases in the detention stroke [J]; L_{ham} is the hydrostatic energy accumulated in the useful stroke [J].

From equation (3) results the diameter of the hydraulic cylinder d_h:

$$d_h = d_m \cdot \sqrt{\frac{\eta_m \cdot \varepsilon \cdot \lambda \cdot (\varepsilon^{k_a - 1} - 1)}{(k_a - 1) \cdot (\varepsilon - 1)} \cdot \frac{p_a}{p_{ha}}} \quad [m] \tag{4}$$

If relation (3) is applied to the resistant stroke, results the equation:

$$\eta_m \cdot L_{har} = |L_{1-2}| \tag{5}$$

where: L_{har} is the hydrostatic energy used in the resistant stroke of the piston [J]; L_{1-2} is the mechanical work used in the compession of the fresh charge [J].

From equation (5) results the rod diameter d_t:

$$d_t = d_m \cdot \sqrt{\frac{\varepsilon \cdot (\varepsilon^{k_a-1} - 1)}{\eta_m \cdot (k_a - 1) \cdot (\varepsilon - 1)} \cdot \frac{p_a}{p_{ha}}} \quad [m] \tag{6}$$

If the theorem of kinetic energy is applied between the initial point (**pmv**) and anyother point "x" in the useful stroke, results:

$$\frac{m_p \cdot w_{px}^2}{2} = \eta_m \cdot L_{gx} - L_{hx} \tag{7}$$

where: m_p is the mass of the piston [kg]; w_{px} is the piston's speed in point "x".

The relation for the calculus of the indicated mechanical work L_{gx} developed until point "x" of the useful stroke is:

$$L_{gx} = \frac{\lambda \cdot \varepsilon^{k_a} \cdot p_a \cdot V_S}{(\varepsilon - 1) \cdot (k_a - 1)} \cdot \left[1 - \left(\frac{1}{1 + b_t \cdot x}\right)^{k_a - 1}\right] \quad [J] \tag{8}$$

where: $b_t = (\varepsilon - 1)/s_p$ is a constant value coefficient.

The function $p_d(x)$ of variation of the burning gases pressure inside the thermal chamber is given by the following relation:

$$p_d(x) = \lambda \cdot \left(\frac{\varepsilon}{1 + b_t \cdot x}\right)^{k_a} \cdot p_a \tag{9}$$

The relation for the calculus of the hydrostatic energy L_{hx} accumulated in the useful stroke of the piston until the point "x" is given by the following relation:

$$L_{hx} = \frac{\eta_m \cdot \lambda \cdot \varepsilon \cdot (\varepsilon^{k_a - 1} - 1)}{(\varepsilon - 1) \cdot (k_a - 1)} \cdot \frac{x}{s_p} \cdot p_a \cdot V_S \quad [J] \tag{10}$$

By processing relation (7) results the variation of the function of the piston's speed in the useful stroke $\mathbf{w_{pm}(x)}$:

$$w_{pm}(x) = \sqrt{e_m \cdot \left\{1 - \left[\frac{1}{(1 + b_t \cdot x)^{k_a - 1}} + f_t \cdot x\right]\right\}} \tag{11}$$

where: $e_m = \frac{2 \cdot \eta_m \cdot \lambda \cdot \varepsilon^{k_a}}{(\varepsilon - 1) \cdot (k_a - 1)} \cdot \frac{p_a \cdot V_S}{m_p}$ şi $f_t = \frac{\varepsilon^{k_a - 1} - 1}{\varepsilon^{k_a - 1}} \cdot \frac{1}{s_p}$ are constant value coefficients. By derivating the function in equation (11) the function of the variation of piston's acceleration in the useful stroke is obtained:

$a_{pm}(x) = dw_{pm}(x)/d\tau = (dw_{pm}(x)/dx) \cdot (dx/d\tau) = (dw_{pm}(x)/dx) \cdot w_{pm}$.

The function of the variation of piston's acceleration in the useful stroke $\boldsymbol{a_{pm}}\mathbf{(x)}$ is:

$$a_{pm}(x) = \frac{1}{2} \cdot e_m \cdot \left[\frac{(k_a - 1) \cdot b_t}{(1 + b_t \cdot x)^{k_a}} - f_t\right] \tag{12}$$

The piston's speed is maximum $\boldsymbol{w_{pmax}}$ in the point where acceleration is zero $\boldsymbol{x_a}$. The distance x_a results from relation (2.12) for $\boldsymbol{a_{pm} = 0}$:

$$x_a = \frac{\left[\frac{(k_a - 1)\cdot b_t}{f_t}\right]^{\frac{1}{k_a}} - 1}{b_t} \tag{13}$$

If in equation (11) $\boldsymbol{w_{pm}}$ is replaced with $\boldsymbol{w_{pmax}}$ and $\boldsymbol{x}$ with $\boldsymbol{x_a}$, results the relation for the the calculus of the piston's mass $\boldsymbol{m_p}$ [kg]:

$$m_p = \frac{2\cdot\eta_m\cdot\lambda\cdot\varepsilon^{k_a}}{(\varepsilon-1)\cdot(k_a-1)}\cdot\left\{1+\frac{f_t}{b_t}-k_a\cdot\left[\frac{f_t}{(k_a-1)\cdot b_t}\right]^{\frac{k_a-1}{k_a}}\right\}\cdot\frac{p_a\cdot V_S}{w_{pmax}^2} \tag{14}$$

The mean speed of the piston $\boldsymbol{w_{mm}}$ in the usrful stroke is calculated with the relation:

$$w_{mm} = \frac{\int_0^{s_p} w_p(x)\cdot dx}{s_p} = \frac{\Delta x\cdot\sum_{i=0}^{n-1} w_p(\Delta x/2+\Delta x\cdot i)}{s_p} \tag{15}$$

The time $\boldsymbol{\tau_{pm}}$ for the movement of the piston from **pvm** to **pvM** in the useful stroke is obtained with the relation:

$$\tau_{pm} = \frac{s_p}{w_{mm}}\quad [s] \tag{16}$$

If the theorem of kinetic energy is applied between the initial point (**pmV**) and anyother point "x" in the resistent stroke, results the equation:

$$\frac{m_p\cdot w_{px}^2}{2} = \eta_m\cdot L_{hx} - \left|L_{gx}\right| \tag{17}$$

The relation for the calculus of the hydrostatic energy L_{hx} necessary until poin "x" to compress the fresh charge is:

$$L_{hx} = \frac{\varepsilon\cdot(\varepsilon^{k_a-1}-1)}{\eta_m\cdot(\varepsilon-1)\cdot(k_a-1)}\cdot\left(1-\frac{x}{s_p}\right)\cdot p_a\cdot V_S \quad [J] \tag{18}$$

The relation for the calculus of the indicated mechanical work L_{gx} developed until point "x" in the resistant stroke of the piston is:

$$L_{gx} = \frac{\varepsilon\cdot p_a\cdot V_S}{(\varepsilon-1)\cdot(k_a-1)}\cdot\left[1-\left(\frac{\varepsilon}{1+b_t\cdot x}\right)^{k_a-1}\right] \tag{19}$$

By processing relation (17) results the variation of the function of the piston's speed in the resistent stroke $\mathbf{w_{pr}(x)}$:

$$w_{pr} = -\sqrt{e_r\cdot\left\{1-\left[\frac{1}{(1+b_t\cdot x)^{k_a-1}}+f_t\cdot x\right]\right\}} \tag{20}$$

where: $e_r = \frac{2\cdot\varepsilon^{k_a}}{(\varepsilon-1)\cdot(k_a-1)}\cdot\frac{p_a\cdot V_S}{m_p}$ is a constant value coefficient.

By derivating the function in equation (20) the function of the variation of piston's acceleration in the resistent stroke $\boldsymbol{a_{pr}}\mathbf{(x)}$ is obtained:

$$a_{pr}(x) = dw_{pr}(x)/d\tau = (dw_{pr}(x)/dx)\cdot(dx/d\tau) = (dw_{pr}(x)/dx)\cdot w_{pr}\,.$$

The function of the variation of piston's acceleration in the useful stroke $\boldsymbol{a_{pr}(x)}$ is:

$$a_{pr}(x) = \frac{1}{2}\cdot e_r \cdot \left[\frac{(k_a - 1)\cdot b_t}{(1 + b_t \cdot x)^{k_a}} - f_t\right] \tag{21}$$

The function of variation of the burned gases in the piston's resistant stroke $\boldsymbol{p_c(x)}$ is given by the following relation:

$$p_c(x) = \left(\frac{\varepsilon}{1 + b_t \cdot x}\right)^{k_a} \cdot p_a \tag{22}$$

The mean speed $\boldsymbol{w_{mr}}$ of the piston in the resistant stroke is calculated with the relation:

$$w_{mr} = \frac{\int_0^{s_p} w_p(x)\cdot dx}{s_p} = \frac{\Delta x \cdot \sum_{i=0}^{n-1} w_p(\Delta x/2 + \Delta x \cdot i)}{s_p} \tag{23}$$

The time $\boldsymbol{\tau_{pr}}$ for the movement of the piston from **pvM** to **pvm** in the resistent stroke is:

$$\tau_{pr} = \frac{s_p}{w_{mr}} \quad [s] \tag{24}$$

The mean piston's speed for one cycle $\boldsymbol{w_{pm}}$ results from the relation:

$$w_{pm} = \frac{2\cdot w_{mm}\cdot w_{mr}}{w_{mm} + w_{mr}} \tag{25}$$

The time in which one cycle takes place $\boldsymbol{\tau_p}$ is:

$$\tau_p = \frac{2\cdot s_p}{w_{pm}} \quad [s] \tag{26}$$

The frequency of the cycles $\boldsymbol{f_c}$ is given by the relation:

$$f_c = \frac{1}{\tau_p} \quad [s^{-1}] \tag{27}$$

The flow of the thermo-hydraulic generator Q_p is calculated with the relation:

$$Q_p = \frac{\pi}{8}\cdot(d_h^2 - d_t^2)\cdot w_{pm} \quad \left[\frac{m^3}{s}\right] \tag{28}$$

The power of the thermo-hydraulic generator P_p is calculated with the relation:

$$P_p = p_h \cdot Q_p \quad [W] \tag{29}$$

5. CONCLUSIONS

The monoregime engines are hybrid ones of thermo-hydraulic type. They are an absolute novelty. There are hybrid engines of thermo-electrical type at whish three energy transformations take place: thermal energy is transformed in mechanical energy (through the thermal engine); mechanical energy is transformed in electrical energy (through a electrical energy generator) electrical energy is re-transformed in mechanical energy (through the electrical motor) or is stoked in electrical accumulators. At the new thermal engines only two energy transformation take place> the thermal energy is transformed in hydrostatic energy and the hydrostatic energy is transformed in mechanical energy (through a hydraulic motor). The new concept regarding the functioning of the thermal engines in a single regime (monoregime thermal engines) represents a new direction for research, because it's obvious that the optimization of a single regime is much more easier to be realized than an infinity as in the present engines

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