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BASIC PRICIPLES FOR THE DESIGN OF THE MONOREGIME THERMAL ENGINES

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Abstract: The monoregime thermal engine is a new concept. The energy produced by the thermal engine is transformed into a hydraulic one and stored in a hydraulic accumulator. This hydraulic energy is then transformed in mechanical energy using a hydraulic motor. The advantage is that because the thermal is functioning in a single regime all the parameters can be optimized in order to achieve the best performances regarding fuel consumption and emissions.

1. GENERAL CONSIDERATIONS

The functioning of the monoregime thermal engines is based on a new concept. The concept involves the manufacturing of engines, generally composed of: energy generator, energy accumulator and motor machine in which two energy transformations are taking place. The primary transformation is realized by the energy generator that transforms thermal energy in other form of energy (hydraulic energy). The energy produced by the generator is stored in the hydraulic accumulator. The secondary transformation is realized by the motor machine that takes the energy stored in the accumulator and transforms it into mechanical energy, generally under form of rotational movement.

The main parts of the monoregime thermal engine are: the monoregime thermo-hydraulic generator TM, the hydraulic accumulator AH and the hydraulic motor MH (fig. 1).

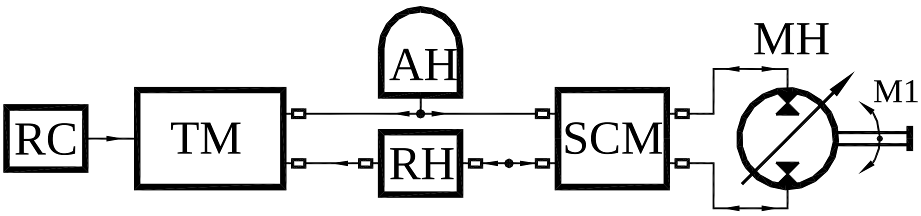

Figure 1. The functioning scheme of the monoregime thermal engines

The engine includes the fuel tank RC and the tank for the hydraulic liquid RH. The energy supply for the hydraulic motor is realized by the engine’s command system SAC. The hydraulic motor transforms the hydraulic energy into mechanical energy necessary to action the working machine.

2. MONOREGIME THERMO-HYDRAULIC GENERATOR

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

The piston is the only mobile part, without articulation elements. The start-stop at the end of the stroke doesn’t affect negativilly the functioning of the generator, because the speed, the kinetic energy respectively, is zero in that points. Because of the piston movement, four chambers with variabile volumes are formed betveen the cylinders walls and the piston: the thermal chamber **T**, the compression chamber **C** and the hydraulic chambers **H1** and **H2**. The piston movement is done under the action of the pressure forces produced by the gases from the thermal and the compression chamber and the pressure forces produced by the hydraulic liquid from the chambers **H1** and **H2**. The processes of the engine cycle take place in the thermal chamber, the intake of the air necessary for the boost or for the formation of an accumulation of pneumatical energy (pressurised air) necessary to displace the piston to the **pvm** (point of minimum volume) can take place in the compression chamber. In the hydraulic chambers the intake and the discharge of the hydraulic liquid take place. The piston movement is coordinated by the automatic command system (**SAC**). The informations regarding the piston's position are given by the transducers **TH**. The piston stroke is between **pvm** (point of minimum volume) and **pvM** (point of maximum volume). The forces that action on the piston are: $\mathbf{F_M}$ created by the gas pressure in the thermal and in the compression chamber and $\mathbf{F_H}$ created by the hydraulic liquid pressure in chambers **H1** and **H2**. During the piston stroke force $\mathbf{F_M}$ has a great variation (exponentially) and force $\mathbf{F_H}$ is almost constant.

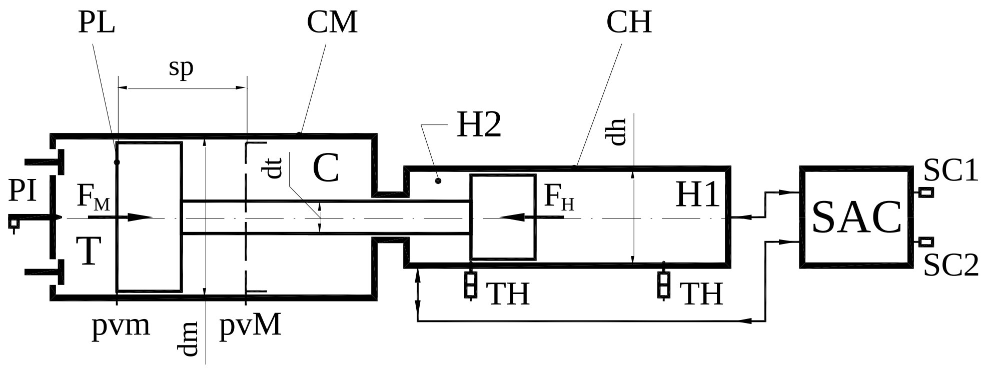

Figure 2. The functioning scheme of the monoregime thermo-hydraulic generator

The thermodynamic processes of the engine cycle from the thermo-hydraulic generator are identical with those of the engine cycle of the internal combustion engines. Same as in the case of the internal combustion engines, the cycle can be done in two strokes (two stroke thermo-hydraulic generator) or in four strokes of the piston (four stroke thermo-hydraulic generator). The hydraulic energy stocked in the hydraulic accumulator during one cycle, is determined with the following relation:

$$W_H = \eta_{tp} \cdot \oint \left[p_T(x) - p_C(x) \right] \cdot dV \qquad (1)$$

where: η_{tp} is the total efficiency of the generator; $\mathbf{p_T(x)}$ is the function of variation of the gas pressure in the thermal chamber; $\mathbf{p_C(x)}$ is the function of variation of the gas pressure in the compression chamber; **x** is the position of the piston from **pmv** (the point of minimum volume);

The value of the total efficiency of the thermo-hydraulic generator depends on the energy losses that are taking place during one cycle: energy losses produced by the friction forces, energy losses produced by the hydraulic fluid viscosity, the energy consumed for the piston movement in the intake stroke and the exhaust of the burned gases (in the case of the four stroke generator) and s.o.

The power of the monoregime thermo-hydraulic generator is given by the following relation:

$$P_H = f_{ct} \cdot W_H = p_{ha} \cdot Q_p \qquad (2)$$

where: $\mathbf{f_{ct}}$ is the frequency of the cycles; $\mathbf{p_{ha}}$ is the pressure of the hydraulic liquid in the accumulator; $\mathbf{Q_p}$ is the flow of the monoregime accumulator.

By solving the system of equation (1) and (2) (the thermodynamic calculus of the cycle) the following parameters are obtained: $\mathbf{d_m}$ the inside diameter of the engine's cylinder, the stroke $\mathbf{s_P}$ and the minimum volume of the thermal chamber.

In the compression stroke of the fresh charge, the hydraulic chambers are conected with the accumulator. The characteristic function $F_C(x)$ for the compression stroke is definied by the following relation:

$$F_c(x) = \frac{m_p \cdot w_{pc}^2(x)}{2} - L_C(x) \tag{3}$$

where: $\mathbf{m_P}$ is the mass of the piston; $w_{pc}(x)$ is the function of the variation of the piston speed during the compression stroke; $\mathbf{L_C(x)}$ is the function of the variation of the mechanical work developed in the compression stroke by the pressure of gases from the thermal and the compression chamber and by the pressure from the hydraulic chambers.

During the detention stroke of the gases inside the thermal chamber, the hydraulic chamber **H1** is conected with the accumulator and the hydraulic chamber **H2** is connected with the hydraulic liquid tank. The characteristic function $F_d(x)$ for the detention stroke is:

$$F_d(x) = \frac{m_p \cdot w_{pd}^2(x)}{2} - L_D(x) \tag{4}$$

where: $\mathbf{w_{pd}(x)}$ is the function of the variation of the piston speed during the detention stroke; $\mathbf{L_C(x)}$ is the function of the variation of the mechanical work developed in the detention stroke by the pressure of gases from the thermal and the compression chamber and by the pressure from the hydraulic chambers.

If the theorem of the kinetical energy is applied at the compression and the detention stroke, results the following system of equations:

$$\begin{cases} F_c(x_0 + s_p) - F_c(x_0) = 0 \\ F_d(x_0) - F_c(s_p + x_0) = 0 \end{cases} \tag{5}$$

where: $\mathbf{x_0}$ is the position of **pvm** in respect with the origin of the coordinate axle; $\mathbf{s_p}$ is the stroke.

The solutions for the system of equations (5) are the diameter of the hydraulic cylinder $\mathbf{d_h}$ and the diameter of the piston rod $\mathbf{d_t}$.

The following relation is definied for the variation of the piston acceleration in the detention stroke:

$$a_{pd}(x) = \frac{d}{d\tau} w_{pd}(x) = w_{pd}(x) \cdot \frac{d}{dx} w_{pd}(x) \tag{6}$$

The piston mass is determined from the condition of limiting the piston maximum speed in the detention stroke at a given value $\mathbf{v_{p\,max}}$. The maximum speed is registered in the point $\mathbf{x_a}$ where piston acceleration is zero:

$$a_{pd}(x_a) = 0 \tag{7}$$

The coordinate x_a is the solution of equation 7.

If the theorem of the kinetic energy is applied between points **pvM** and $\mathbf{x_a}$, the relation for the calculus of the piston mass is obtained:

$$m_p = \frac{2}{w_{p\max}^2} \cdot L_D(x_a) \tag{8}$$

The time in which one stroke **i** take place is determined with the relation:

$$\tau_{pi} = \frac{s_p}{\overline{w}_{pi}} = \frac{s_p^2}{\int_{x_0}^{x_0+s_p} w_{pi}(x) \cdot dx} \tag{9}$$

where: $w_{pi}(x)$ is the function of variation of the piston speed in stroke **i**; $\overline{w}_{pi}$ is the mean piston speed in stroke **i**.

The frequency of the engine cycles is calculated with the relation:

$$f_{ct} = \frac{1}{\sum_{i=1}^{\nu} \tau_{pi}} \tag{10}$$

where τ is the number of strokes made in one cycle, $\nu = 2, 4$.

3. THE HYDRAULIC ACCUMULATOR

The hydraulic accumulator is of hydro-pneumatical type. The accumulator has the role to accumulate the energy produced by the thermo-hydraulic generator that's necessary to run the resistant strokes of the engine cycle and the hydraulic motor. Also, the accumulator dumpes the flow pulsations and hydraulic shocks. The pressure in the accumulator is almost constant, it variates between two limits: the maximum pressure $\mathbf{p_{hmax}}$ and the minimum pressure $\mathbf{p_{hmin}}$. The nominal pressure in the accumulator is $p_{ha} = (p_{h\min} + p_{h\max})/2$. The accumulator has an optimum functioning if the rate of between the maximum and the minimum pressure is $p_{h\max}/p_{h\min} = 1{,}1 \div 1{,}2$. The accumulator parameters are: the total volume $\mathbf{V_{ta}}$ and the nominal pressure $\mathbf{p_{ha}}$. The useful accumulator volume is determined with the relation:

$$V_{ua} = V_{H1} + \frac{Q_p}{f_c} \tag{11}$$

where: v_{HI} is the displacement of the hydraulic chamber **H1**.

If the variation of the pressure $(p_{h\max} - p_{h\min})$ is small, the accumulator's total volume will be smaller and the frequency of the start-stop operation of the generator will increase. The total volume of the accumulator should be big enough to dump the flow pulsation and the hydraulic shocks. Because of it's conception mode, the frequency of start-stop operation doesn't affect in any way the functioning of the generator.

4. THE HYDRAULIC MOTOR

The hydraulic motor transforms the hydraulic energy in mechanical energy under the form of rotational movement. The displacement $\mathbf{V_M}$ of the motor is variabile, so the rotational speed can be adjusted. Also, the motor is a reversible machine, so the braking mechanical energy can be recovered through it's reprocessing into hydostatic energy.

$$M_1 = \frac{1}{2\pi} \cdot \Delta p \cdot V_M \cdot \eta_{mh} \tag{12}$$

where: $\Delta p = p_{ha} - p_{rz}$ is the difference between the hydraulic liquid inside the accumulator and inside the tank, respectively; V_M is the displacement of the hydraulic motor; η_{mh} is the mechano-hydraulic efficiency of the motor.

The movement stability, the setting and the change of the direction of rotation of the hydraulic motor's shaft is realized by *the motor command system* **SCM**.

5. CONCLUSIONS

The monoregime engines are characterized by the following features:

- single regime running;

- no idle functioning;

The following advantages are estimated (comparing with present thermal engines): reduced fuel consumption and emissions (is much more easier to optimize the functioning of a thermal engine for a single regime instead of an infinity of regimes, as happened to present engines), simple construction and greater reliability. Also, the monoregime thermal engine can retrieve, totally or partially the function of the transmission (the variation of the torque and of the rotational speed and reversing the direction of rotation), including braking energy recovery.

The main directions for the improvement of the thermal engines are the reduction of fuel consumption and of emissions [1], [5], [8].

At the monoregime engines, the application of measures for the reduction of fuel consumption and of emissions can be made easier and more efficient, without special adjusting devices (automatic devices). The thermal part (the thermo-hydraulic generator) is functioning in a single regime with maximum efficiency and has no idle functioning. Also, fuel consumption reduction is realized through the recovery of braking energy. At present engines the prolonged expansion by rising the displacement cannot be applied because of some great disadvantage (the outline dimensions are rising and the power weight ratio decreases) By a prolonged expansion the thermal efficiency rises and the noise level decreases. In conclusion, the experimental research of a monoregime engine is eased by the functioning of the thermal part in only one regime.

Because of their conception the new thermal engines have a simple construction. The thermal part has no reciprocating mechanism, no parameter adjustment systems during the functioning and no start system. Because of that a better reliability of these engines is expected.

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