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ADIABATIC BEHAVIOR OF A VARIABLE DISPLACEMENT STIRLING ENGINE

АДИАБАТИЧЕСКОЕ ДЕЙСТВИЕ ДВИГАТЕЛЯ СТИРЛИНГА С ИЗМЕНЯЮЩИМ РАБОЧИЙ ОБЪЕМ

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Abstract: *The construction of the Stirling engine with on load variable displacement is presented. The variable displacement Stirling engine behavior is analyzed using an adiabatic physico-mathematical model. Engine performance and power adjustment capability are analyzed.*
KEYWORDS: *STIRLING ENGINE, VARIABLE DISPLACEMENT, ADIABATIC MODEL, RHOMBIC DRIVE*

1. Introduction

Named as the man that patented them in 1816, the Stirling engines are external combustion reciprocating movement machines using heat regeneration. The Stirling engines run on a thermodynamic cycle made of two isothermal processes linked by two isochoric processes. After experiencing an important development around 1900, the Stirling engines have been overrun by the internal combustion engines and, in effect, forgotten.

In the last fifty years, due mostly to the impetuous development of science and technology, the interest for Stirling engines arose again as a new range of specific utilization fields came up for them [4], [5], [6], [7]. In this context, different configurations of Stirling engines were developed. The Stirling engine can be used to power electric generators (those used on space shuttles included), pumps and compressors, to motorize submarines, yachts or torpedoes, or in heat and power cogeneration units.

The Stirling engine presented in this paper features a motive drive allowing displacement modifications on load [1], [2] and thus continuous variation of the work produced.

The paper evaluates the performance of such an engine and the efficiency of the adjustment method proposed.

2. Variable Displacement Stirling Engine

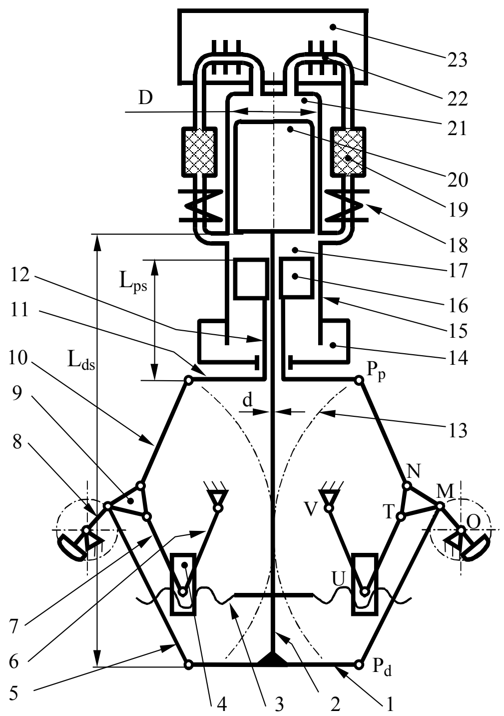

Fig. 1. VDSE schematic diagram

In the variable displacement Stirling engine (VDSE) schematic diagram (fig. 1) there are the following components: the cylinder 15, the power piston 16, the displacer 20 and three heat exchangers, 18 - the low temperature one (the cooler), 22 - the high temperature one (the heater) and the regenerator 19.

The low temperature heat exchanger is cooled with water (or air, the case of small Stirling engines) and the heater is placed in the burning chamber 23.

The expansion space 21 is found between the cylinder head and the displacer and the compression space 17 between the two pistons. The engine also includes the buffer space 14. The motive drive has two crankshafts 8 that spin in opposite directions. The crankshaft movements are synchronized by the gear wheels 13. The displacer 20 is equipped with a stem 2 that pierces through the power piston 16 in the middle. The stem 2 finishes at the lower yoke 1, at the ends of which the lower rods 5 are socketed. The power piston 16 is equipped with a cylindrical stem 12 which finishes at the upper yoke 11, at the ends of which the upper rods 10 are socketed. Between the upper rods 10 and the crankshafts 8 the sides MN of the triangular plates 9 have been introduced. The ends T of the triangular plates are socketed with the adjustment bars 7. The adjustment bars themselves are socketed in the leaning bars 6 which can oscillate in the fixed sockets V. The sockets U between the 6 and 7 bars are able to move along circle arcs thanks to the adjustment screw 3 which has two distinct regions of opposite threading. The screw 3 is spun from outside through the nuts 4 which stand for the U sockets. The spinning effect is that the ends T of the triangular plates 9 shift and thus the lengths of the "equivalent rods" MP are modified resulting in the displacement variation of the Stirling engine analyzed here.

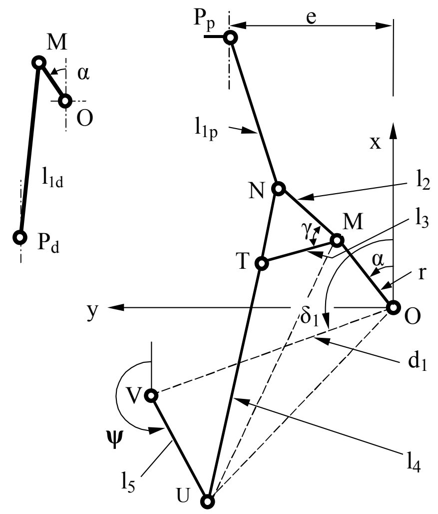

Fig. 2. The modified rhombic drive

On fig. 2. are presented the most important dimensions of the motive drive that allows the on-load displacement modifications.

3. Hypotheses

The isothermal and the adiabatic physico-mathematical models of the Stirling machines share the following hypotheses:

● the working agent is the ideal gas,
● the gas amount inside the machine is constant,
● at thermodynamic level all cycle functional processes are time independent,
● the metallic parts of the machine do not exchange heat among them,
● the processes inside heat regenerators are ideal ones (regeneration efficiencies are 100%); the agent temperature inside the regenerator is deemed constant, being taken as arithmetic or logarithmic mean,
● inside the cooler and heater isothermal processes take place only,
● the instantaneous pressure is identical in all the spaces occupied by the agent, its value varying along the cycle,
● the movement of the pistons is the real movement, given by the crankshaft.

The isothermal model uses the additional hypotheses:

● the temperature inside the expansion chamber is equal with the temperature inside the heater, constant and equal to the one of the displacer and that of the cylinder around its warm zone,
● the temperature inside the compression chamber is equal with the temperature inside the cooler, constant and equal to the one of the power piston and that of the cylinder around its cold zone.

The adiabatic model supplementarily assumes that

● inside the compression and expansion chambers adiabatic processes take place; so the temperature inside these chambers vary cyclically.

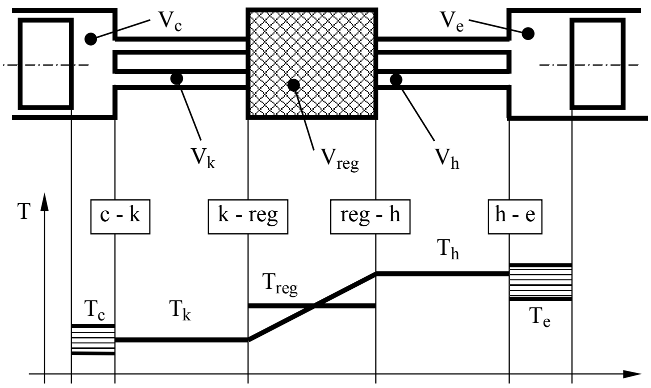

Fig. 3. The temperature-related hypotheses of the adiabatic physico-mathematical model of the Stirling engine

The hypotheses implying the temperatures inside the Stirling engine (presented in fig. 3) show that inside two of the machine chambers take place adiabatic processes and inside all other chambers isothermal processes take place only, thus confirming to the described physico-mathematical model the denomination of adiabatic model.

4. Physico-mathematical models

The performance of a Stirling engine and therefore of a VDSE also, can be analytically calculated with isothermal or adiabatic physico-mathematical models. The isothermal model obtains the pressure variation law from the equations of state applied for each chamber and from the agent total mass (m_T) conservation equation, as:

$$p(\alpha,\psi)=\frac{m_T\ R}{\dfrac{V_e(\alpha)+V_h}{T_h}+\dfrac{V_{reg}}{T_{reg}}+\dfrac{V_c(\alpha,\psi)+V_k}{T_k}}\ . \tag{1}$$

The adiabatic model assumes that the processes inside the compression and expansion chambers are adiabatic. It is also assumed that processes inside the heat exchangers are isothermal. Using the equation of state, the energy conservation equations written for each chamber and the agent total mass conservation equation, a differential equation system is obtained [2], [6]:

$$dp=\frac{-k\,p\left(\dfrac{dV_c}{T_{c-k}}+\dfrac{dV_e}{T_{h-e}}\right)}{\dfrac{V_c}{T_{c-k}}+\dfrac{V_e}{T_{h-e}}+k\left(\dfrac{V_k}{T_k}+\dfrac{V_{reg}}{T_{reg}}+\dfrac{V_h}{T_h}\right)}\ , \tag{2}$$

$$dm_c=\frac{1}{R\,T_{c-k}}\left(\frac{V_c}{k}dp+p\,dV_c\right), \tag{3}$$

$$dm_e=\frac{1}{R\,T_{h-e}}\left(\frac{V_e}{k}dp+p\,dV_e\right), \tag{4}$$

$$dT_c=T_c\left(\frac{dp}{p}+\frac{dV_c}{V_c}-\frac{dm_c}{m_c}\right), \tag{5}$$

$$dT_e=T_e\left(\frac{dp}{p}+\frac{dV_e}{V_e}-\frac{dm_e}{m_e}\right). \tag{6}$$

The unknown functions are the pressure p, the masses and temperatures inside the compression and the expansion chambers. In the system the conditional temperatures $T_{c\text{-}k}$ and $T_{h\text{-}e}$ of the surfaces that delimitate the adiabatic chambers and the neighboring heat exchangers appear also. The conditional temperatures depend on the sense of the mass flow inside the machine.

After the integration of the differential system (2) … (6), the pressure variation law is obtained in the shape of numerical value sets for the variate positions of the crankshaft and for variate instances of the adjustment angle ψ.

Inside relations (1) … (6) the notations were used as following: R = gas constant; k = isentropic exponent; m = agent mass; p = pressure; V = volume; T = temperature. The following subscripts were used: c, e = compression and expansion chambers; reg = regenerator, h = heater; k = cooler; T = total; d = displacer piston; p = power piston; s = stem.

The expansion and compression chamber volume variations are evaluated from kinematic considerations [3], in relation with fig. 2.

For both models the work is obtained as a sum of the works exchanged in the expansion and the compression chambers:

$$L=L_e+L_c=\int_0^{2\pi}p(\alpha)\,dV_e+\int_0^{2\pi}p(\alpha)\,dV_c\ . \tag{7}$$

The indicated power of the engine is

$$P=\frac{n}{60}L\ , \tag{8}$$

where n is the rate of the revolution, in rpm.

Thermal efficiency of the Stirling engine, also known as indicated efficiency, is defined as

$$\eta_t=\frac{L}{L_e}\ . \tag{9}$$

The brake horsepower and the brake thermal efficiency are obtained using the indicated efficiency and the mechanical efficiency. The indicated efficiency takes into account the mechanical losses occurring along the frictional flow of the working agent inside the machine as well as the losses due to the non-isothermal functioning of the heat exchangers.

5. Results and discussion

For an engine described by the following values: r = 0.0385 m; e = 1.6 r; l_{1d} = 3 r; l_{1p} = 2.5 r; $l_2 = l_3 = l_5$ = 2 r; l_4 = 3 r; d_1 = 2.5 r; γ = 50°; δ_1 = 100°; D = 0.073 m; d = 0.02 m; L_{ps} = 5 r; L_{ds} = 12.5 r; $V_h = V_k$ = 0.05 V_{emax}; V_{reg} = 1.2 V_{emax}, where V_{emax} = maximum volume of the expansion chamber; m = 0.0025 kg of hydrogen; T_h = 773 K; T_k = 310 K and n = 1500 rpm, for an adjustment angle ψ between 160° ... 205°, the performances presented in the following pictures were obtained. The subscript max stands for the maximum value. The curves drawn for the isothermal model are affected by the subscript "is" and the curves characteristic to the adiabatic model do not feature any subscripts.

In fig. 4 and fig. 5 are presented the cyclical temperature variations of the working gas inside the compression chamber T_c and inside the expansion chamber T_e for three adjustment angles. The mean temperature of the gas inside the adiabatic compression chamber $T_{c\ mean}$ is bigger than the isothermal temperature and raises with the adjustment angle ψ. In the adiabatic expansion chamber the mean temperature $T_{e\ mean}$ is lower than the isothermal one and decreases as the adjustment angle ψ increases (Table 1). These variations outline that the cycle takes place between narrower temperature limits, so that the adiabatic thermal efficiency becomes inferior to the isothermal efficiency.

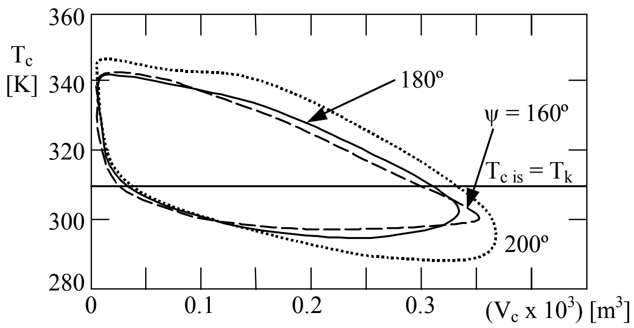

Fig. 4. Temperature inside the compression chamber

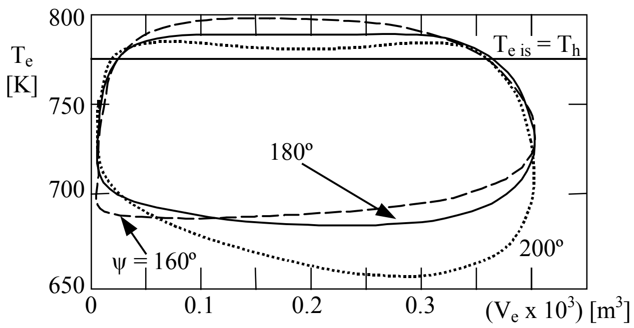

Fig. 5. Temperature inside the expansion chamber

Table 1. T_{mean} inside the compression and expansion chambers

	Unit	Adjustment angle ψ		
		160º	180º	200º
$T_{c\ mean}$	K	314.9	316.9	319.6
$T_{e\ mean}$	K	743.9	742.4	733.5

Fig. 6 presents the cyclical pressure variations calculated with isothermal and adiabatic physico-mathematical models, for an adjustment angle ψ = 180º. It can be seen that the maximum adiabatic pressure is greater than the maximum isothermal pressure. The minimum adiabatic pressure is less than the minimum isothermal pressure. The adiabatic cycle evolves between wider pressure limits as compared to the case of the isothermal cycle.

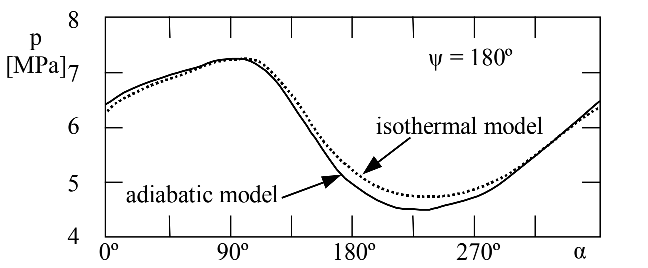

Fig. 6. Unfolded indicator diagram

In fig. 7, fig. 8 and fig. 9 are presented the p-V diagrams for the VDSE working with different values of the adjustment angle ψ. The diagrams show that the changes of the total volume occupied by the working gas are obtained mainly due to the movements of the power piston bottom dead point (BDP). The expansion chamber volume remains constant in spite of the adjustment angle variations. Although the total volume occupied by the gas increases with the adjustment angle ψ, the diagrams show that the modified rhombic drive is less efficient for adjustment angles ψ < 170º. It can be seen by comparing the two diagrams that when ψ is increasing the areas of the indicator diagram for both compression and expansion chambers are increasing too. The increase is more accentuate for the area of the indicator diagram of the expansion chamber. This conclusion is strengthened by the numerical data presented inside Table 2.

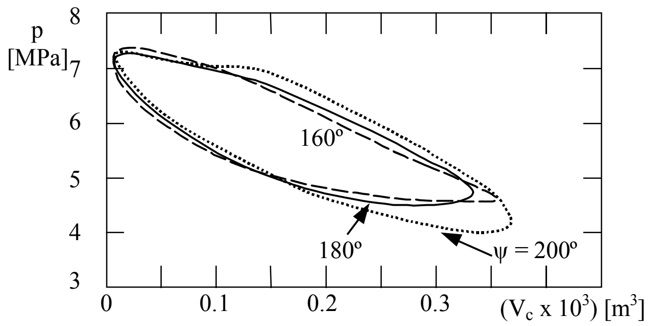

Fig. 7. Indicator diagram for the compression chamber

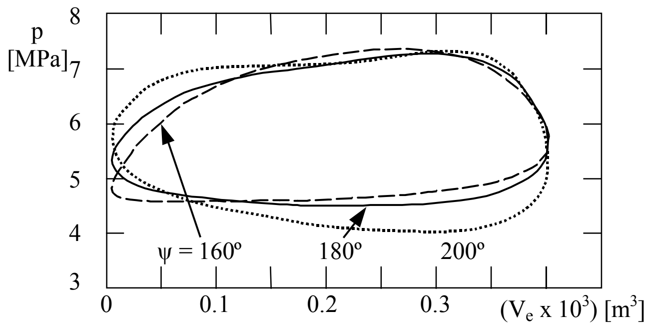

Fig. 8. Indicator diagram for the expansion chamber

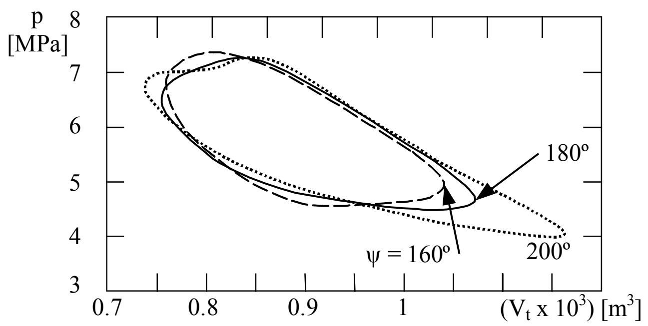

Fig. 9. Indicator diagram for the whole engine

Table 2. Work exchanged inside VDSE chambers

	Unit	Adjustment angle ψ		
		160º	180º	200º
work in expansion chamber	J	827.0	864.5	1045.4
work in compression chamber	J	-371.4	-390.5	-488.9
total work	J	455.6	474	556.5

Calculated with both methods, i.e. isothermal and adiabatic, the work yielded during a single cycle provided values close to one another. Greater differences appeared in which concerns the thermal efficiencies (fig. 10). Calculated with the isothermal model, the thermal efficiency was found equal to the Carnot cycle efficiency (working between the same extreme temperatures). The thermal efficiency calculated for the adiabatic model was obtained as less than the Carnot efficiency and varying with the adjustment angle ψ (the load in effect). The adiabatic thermal efficiency of VDSE

decreases with the load, approximately with 5% of its maximum value [4].
The VDSE can regulate the load (for adjustment angle ψ between 160° and 205°) within the ratio

$$\frac{P_{max}}{P_{min}} = \frac{14862}{11390} = 1.3\ . \qquad (10)$$

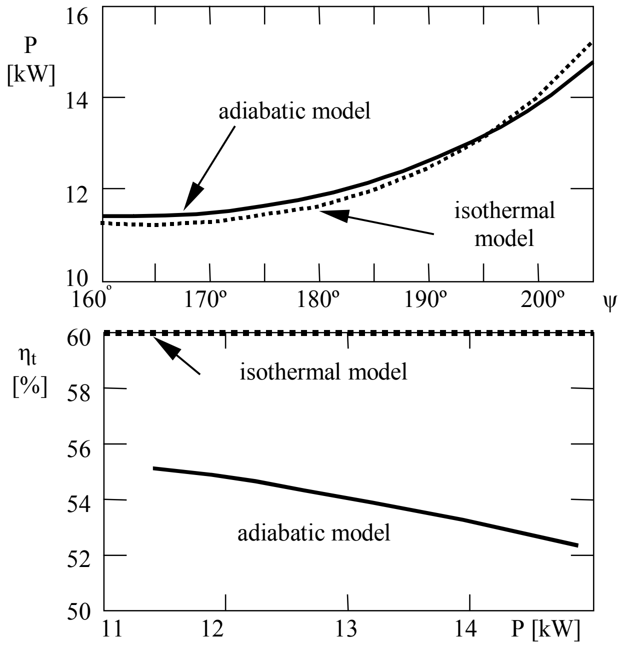

Fig. 10. VDSE performances

6. Conclusions

The displacement variation was made possible by gearing the engine with a modified rhombic drive [1].
Although both isothermal and adiabatic models have theoretical character, they allow to obtain a coherent image of the VDSE functioning. The adiabatic physico-mathematical model introduces variable temperatures inside the compression and expansion chambers and by this leads to a thermal efficiency less than the isothermal efficiency. The isothermal efficiency is always equal to the efficiency of the Carnot cycle evolving between the same extreme temperatures.
The application of the adiabatic model on the VDSE shows that when the adjustment angle increases the work produced in the expansion chamber increases quicker than the increasing of the work consumed in the compression chamber (the work consumed is taken in absolute value). The raising of the total gas volume with the adjustment angle leads to a raising of the work produced. The maximum pressure of the gas decreases when the adjustment angle increases.
The thermal calculations show that the VDSE can regulate the load within the ratio of 1.3 . By optimizing the dimensions of the VDSE motive drive, a raise of the maximum/minimum power ratio up to 1.5 was estimated.
The adiabatic thermal efficiency of VDSE decreases with the load, approximately with 5% of its maximum value.
VDSE can be employed with good results in heat and power cogeneration plants.

7. References

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