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Development of External Combustion Engine

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Abstract The internal combustion engines are generally a major source of air pollution. However the spark ignition engines are recognized by their carbon monoxide and unburned hydrocarbon emission. One of the methods used to reduce the emission of these pollutants is the use of external combustion engine. This paper describes the development of an external combustion engine which uses air in tanks at pressures of 30 - 50 MPa and at ambient temperature as combustion energy carrier, and hydrogen, alcohols or traditional motor fuel from organic minerals as chemical energy carrier. Research workings out are in the field finished to level of practical use. The mathematical and simulation model has been developed, tested, and verified to simulate a 4-stroke cycle of a spark ignition engine fuelled with gasoline. The results obtained from the present study have shown the capability of the model to predict the performance satisfactorily.

Keywords: *external combustion engine, efficiency, temperature, pressure, mathematical model*

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1. Introduction

Due to the undesirable consequences of internal combustion engines burning fossil fuels, to their imminent exhaustion of these fuels and to their steadily rising prices, there has been a steady movement towards developing higher fuel efficiency engines and more alternative fuel vehicles for consumers [1].

The solutions to these problems are quite complicated. Of those, the external combustion engine is the most promising source of mechanical energy in the field of transportation. It has good technical characteristics: a very high efficiency, the ability to operate with different fuels, a simple structure, and low operating costs.

An external combustion engine is a heat engine where an (internal) working fluid is compressed and heated by combustion of an external fuel through the engine wall or a heat exchanger. The fluid then, by expanding and acting on the mechanism of the engine (piston or turbine), produces a shaft power. Steam engines and Stirling engines are the best known examples of external combustion engines.

The strength of external combustion engines compared to the internal combustion engines is their compatibility with a wide variety of renewable energy and fuel sources. They may use a supply of heat from any source such as biomass and biomass-derived products, municipal waste, nuclear, solar, geothermal, or exothermic reactions not involving combustion (in which. case they are then classed not strictly as external combustion engines, but as external heat engines). Other important advantages of external combustion engines are low emissions (due to continuous external combustion), and low noise (due to the elimination of exhaust of high-pressure combustion products). External combustion engines have a high starting torque. They are self-starting with the working fluid whereas in case of internal combustion engines, some additional equipment or device is used for starting the engines. They may be free from reciprocating parts, in which case there is complete freedom from vibration.

In the literature, the works on the external combustion engines are so rare, however, include some interesting works, like that produced by Yehuda B. and al. [2], using the Pontryagin maximum principle, they optimized the operating conditions of a model external-combustion engine to obtain maximal efficiency. Song, H. J. and al. [3] examined the optimal motion of a model of an external combustion engine with a piston fitted inside a cylinder containing an ideal gas. Kang Ma and al. [4] investigated a model of external combustion engine with a movable piston, and effects of heat transfer laws on the optimizations of the engine for maximum work output.

2. Experimental Setup

The principal schematic view of this heat engine on the basis of external combustion engine is presented in Figure

1. The air from the tank 1, under a pressure of 30-50 MPa and at ambient temperature, flows via main 2 to the reducer 3, then by main 4 to the surge tank 5 in which the pressure is kept at 2-5 MPa, as monitored by the pressure sensor 6. From the surge tank, the air by drain 7 arrives in the combustion chamber 8 which is equipped with a fuel supply system 9 (e.g.: an injector) and an ignition system 10 of the air-fuel mixture (e.g.: a spark plug). The combustion chamber 8 is connected by channels 11 and 12 to the inlet valves 13 and 14 of the cylinders 15 and 16. The valves 13 and 14 are actuated with a fast action device (e.g.: solenoid valves). The products of combustion from the zone located above the pistons of cylinders 15 and 16 are evacuated through the exhaust valves 19 and 20 and the exhaust mains 21 and 22 to the exhaust manifold 23 and then the atmosphere (this happens normally during the displacement of the piston towards top dead center (TDC)). The temperature of the products of combustion at the end of the exhaust manifold is controlled by the sensor 24. The engine power, for whatever external load, is established using the variation of the quantity of fuel injected into the combustion chamber 8, the duration of opening of the inlet valves 13 and 14, and the air pressure in the surge tank.

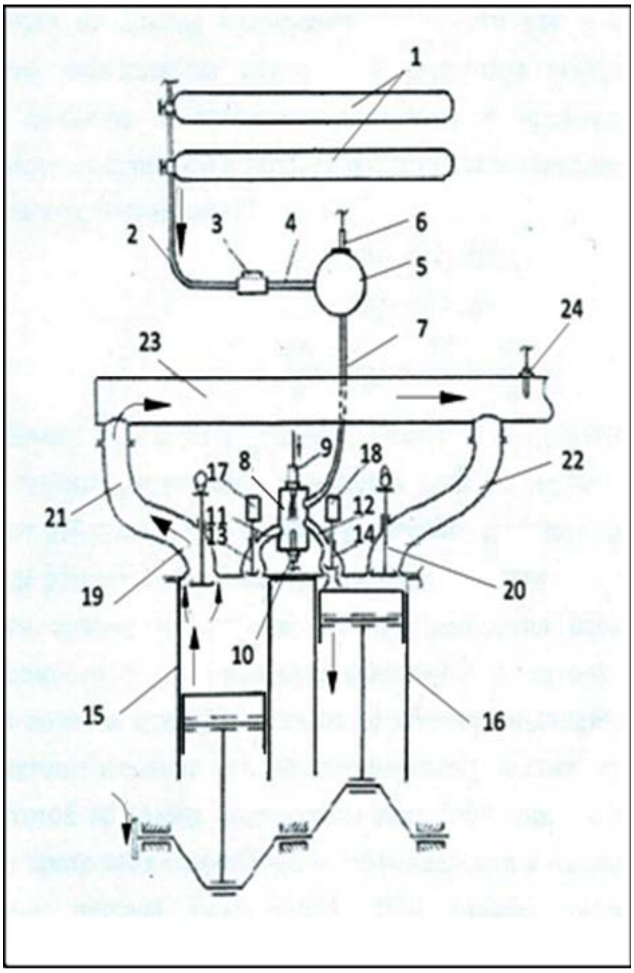

Figure 1. Schematic view of the external combustion heat engine

The durations of the mixture formation and combustion processes in external combustion engines are usually much longer than those for internal combustion engines. Moreover the maximum temperature of the products of combustion in the combustion chamber according to the loads is decreased by up to 800 – 1300 K for a coefficient of excess of air during combustion $\alpha > 2$. That defines the high ecological characteristics of this kind of engine, and the minimal losses of heat with the exhaust gases. During idling times or under partial loads, the temperature of exhaust gases will not be lower than the ambient temperature; that is ensured by the variation of the air pressure in the surge tank, by the fuel injection and by the duration of opening of the inlet valve.

3. Theoretical Model

The estimate of the influence of the pressure P_s and temperature T_s of the products of combustion in the combustion chamber on the performances of the real cycle of the external combustion engine is carried out using analytical calculations together with the use of certain data resulting from tests on an internal combustion engine (variation of the coefficients of discharge of the valves according to the height of the lifting of the valves, of the coefficients of heat transfer according to the dimensional parameters of the engine, of the parameters of the operating cycles, of the values of the temperatures of the wall surface of the zone located above the pistons, and of the temperatures of the combustion chamber). Since the volume of the surge tank is much larger than the volume of the compression chamber, into which arrive the products of combustion from the combustion chamber, the pressure in the combustion chamber is taken as constant. The temperature of the products of combustion is given by the equation for the thermal balance for the combustion chamber:

$$Q_H \cdot \eta_{K,C} \alpha \cdot M'_0 \cdot \mu C_{pm0} \cdot t_0 = M'_S \cdot \mu C_{pms} \cdot t_S + W_T \cdot Q_H \quad (1)$$

where, Q_H - lower calorific value of the fuel, kJ/kg;

$\eta_{K,C}$ - combustion chamber efficiency;

M'_0 - theoretical air quantity necessary for the combustion of 1 kg of fuel, kmole/kg;

M'_S - products of combustion quantity resulting from one kilogram of fuel, μC_{pm0}, μC_{pms} average molar heat capacity of air and of the combustion products, kJ/(kmole·K);

t_0, t_S - air temperature at the entrance of the combustion chamber and temperature of the combustion products at the outlet side of the combustion chamber;

W_T - relative loss of heat of the combustion products in the walls of the combustion chamber (W_T = 0.03 ÷ 0.05 according to the thermal insulation of the walls of the combustion chamber).

The variation of the parameters of the charge (combustion products) in the zone above the piston of the external combustion engine (estimated according to the volume of the space located above the piston, as is done for the internal combustion engine) is determined by the solution of the differential equations for mass and energy balances as well as the equation of state. By supposing that during an elementary time interval Δτ, the processes in the combustion chamber are stationary (quasi-stationary) and the load is in a state of equilibrium, the following equations result [5]:

$$dM = dM_s - dM_b \quad (2)$$

$$dQ = dI - V \cdot dp \quad (3)$$

$$\frac{dp}{p} + \frac{dV}{V} - \frac{dT}{T} = \frac{dM}{M} \quad (4)$$

where, dM - variation of the mass of the charge in the zone located above the piston conditioned by the inlet dM_s and the exit of the charge in an interval of the calculation time Δτ; heat supplied to the load or removed from the load (in an interval of the calculation time *Δτ*); p, V, T, M charge parameters at the beginning of the interval of the calculation time.

4. Simulation Results

Figure 2 shows the results of the calculation of the charge parameters variations in the zone above the piston as a function of crank angle rotation φ. On Figure 3, the variation of charge pressure in the zone above the piston is presented depending on its volume (P-V diagram). The calculations are performed for a cylinder automobile engine with a bore of 88 mm, a piston stroke of 82 mm, and a crankshaft speed n = 5000 min^{-1}, with the ratio of the maximum value of the effective square of flow areas of outlet valves to the piston square $2\mu_B\ f_B / F_B = 0.25$ and the inlet valve to piston $\mu_s\ f_s / F_p = 0.025$. At the given value of relation $\mu_s\ f_s / F_n$, the duration of opening of the inlet valve is approximately 20 °crankshaft rotation, pressure of products of combustion in the combustion chamber p_s = 5MPa, temperature $T_s \approx$ 900K, the indicated power will make approximately $N_i \approx$ 95.6 kW. The temperature of products of combustion in the zone above the piston in the beginning of opening of outlet valves (φ = 30° crankshaft rotation before BDC) does not exceed 320 K, i.e. the heat brought to a charge in the combustion chamber is used sufficiently effectively (Figure 2).

From the diagram P-φ (Figure 3), are defined: indicated work of gases for a cycle, J:

$$L_i = L_s + L_p - (L_b + L_c) \quad (5)$$

where, L_s - work on the section of admission of combustion products *in the zone* located above the piston; L_p - work of expansion of combustion products *in the zone* located above the piston; L_b - work spent for removal of combustion products from in *the zone* located above the piston; L_c - work of compression of combustion products remained *in the zone* located above the piston; indicated power, kW:

$$N_i = k \cdot L_i \cdot 10^{-3} \quad (6)$$

where, $k = n\dfrac{z}{60}$ – number of cycles in engine cylinders for 1s; indicated efficiency of the engine at filling of tanks by compressed air in the fuel station:

$$\eta_{i1} = N / (Q_T + L_{kn}) \quad (7)$$

where, $Q_T = G_S C_{PmS} t_S - G_B C_{Pm0} t_0$ - heat brought to air in a combustion chamber, kJ/s; C_{pms}, C_{pm0} - average specific mass thermal capacities of combustion products and air at constant pressure, kJ/(kg·K); L_{kn} - potential energy compressed air arriving in a combustion chamber, kJ/s:

$$L_{kn} = Ga \cdot \frac{1}{k_{CP} - 1} \cdot R \cdot T_0 \cdot \left[1 - (P_0 / P_S)^{(k_m - 1)/k_m}\right] \quad (8)$$

Ga - air consumption through the combustion chamber, kg/s; R - gas constant for 1 kg of air, kJ/(kg·K); k_m - mean value of an adiabatic exponent at expansion of air from pressure p_c(T=T_0) to pressure P_0, specific indicated fuel consumption:

$$g_{i1} = B_h / N_i \quad (9)$$

where, B_h hourly fuel consumption; indicated efficiency of the engine at tanks filling by the compressed air directly onboard a vehicle:

$$\eta_{i2} = N_i / (Q_T + N_c) \quad (10)$$

where, N_c - power of the compressor:

$$N_c = Ga \cdot m \cdot \frac{n}{n-1} \cdot R \cdot T_0 \cdot \left[(p_k / p_0)^{\frac{n-1}{n}} - 1 \right] \cdot 1/\eta_c \quad (11)$$

m=3 - number of steps of the compressor; n - mean value of polytropic exponent of compression of air in one step of compressor; p_c / p_0 rate of air pressure rise in one step of compressor; η_c =0.75 - compressor efficiency; specific indicated compressed air consumption:

$$g_{ai} = 3600 \cdot G_a / N_i \quad (12)$$

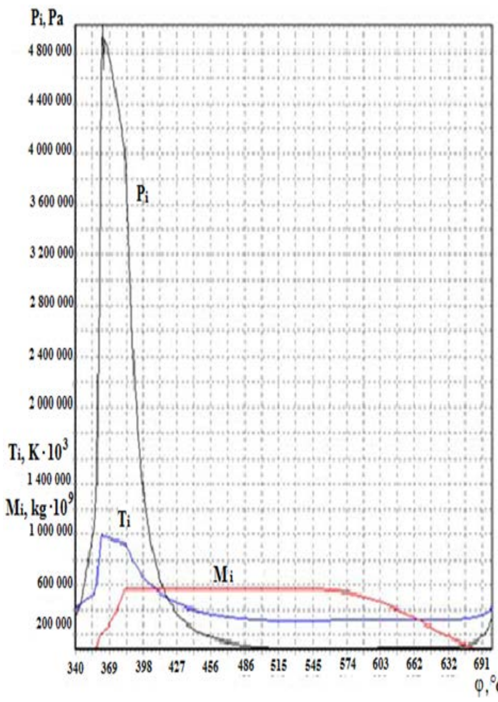

Figure 2. Variation in the charge parameters in the combustion chamber in the zone above the piston with crank angle

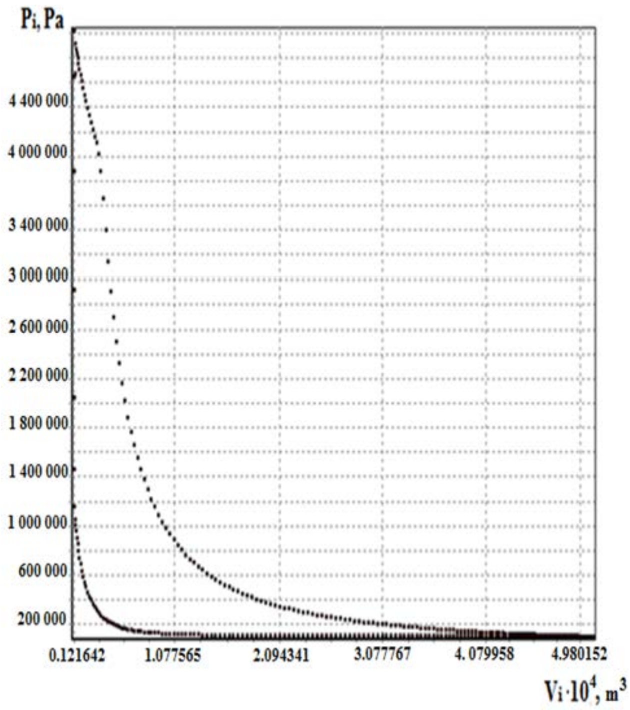

Figure 3. Cylinder pressure external combustion engine (n = 5000 min^{-1}, air pressure in the zone-buffer p_s = 5 MPa and the air to fuel ratio in the combustion chamber α = 3)

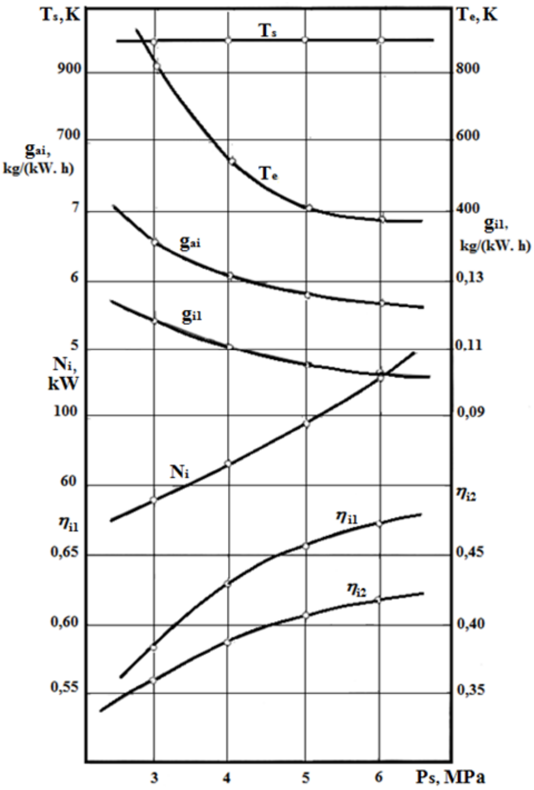

Figure 4. Influence of the pressure of combustion products p_s *in the zone* located above the piston on the indicated parameters of the external combustion engine at α = 3

The principal results of the simulation of the dependency of the various engine parameters on input conditions are shown in the next two figures: in Figure 4 as a function of piston pressure at a fixed α, and in Figure 5 as a function of α at a fixed piston pressure.

From Figure 4: At constant value of the coefficient of air excess in the combustion chamber (α = 3), and the pressure varying between 3 and 6 MPa, one sees the following:

- The temperature of the products of combustion also does not change ($T_s \approx$ 1000 K).
- The temperature of the products of combustion T_e at the moment of the beginning of opening of outlet valves decreases markedly from 823 K at p_s = 3 MPa to 387 K at pressure p_s = 5 MPa, a drop of almost 50%.
- The specific indicated air rate g_{ai} decreases (to 5.7 kg/(kW·h)), and the specific indicated fuel rate g_{i1} (to 0.102 kg /(kW·h)) decreases also by some 10%.
- The Indicated power increases by some 167% from 50.5 kW at p_s = 3 MPa to 119 kW at p_s = 6 MPa, as do the two efficiencies accordingly indicated efficiency η_{i1} increases from 0.585 to 0.672.

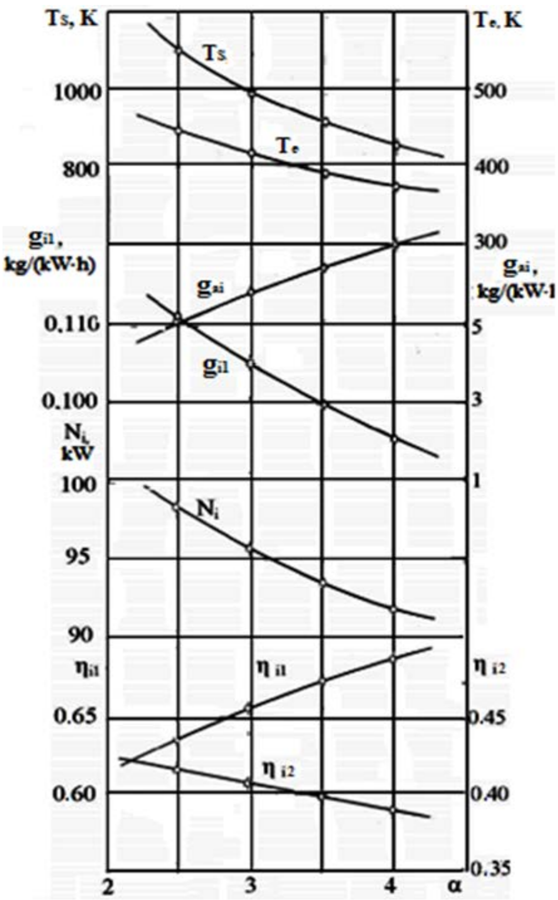

Figure 5. Influence of the variation of the air to fuel ratio α in the combustion chamber, on the indicated parameters of the external combustion engine at p_s = 5 MPa

At constant value of pressure of products of combustion ps, arriving in the combustion chamber, with air to fuel ratio rise, the temperature of products of combustion decreases from 1123 K at α = 2.5 to 837 K at α = 4, indicated efficiency ηi1 increases thus from 0.637 to 0.689 (Figure 5). If the filling of tanks is carried out directly onboard vehicle, indicated efficiency ηi2 with increase in the air to fuel ratio, decreases in consequence of the rise of the specific indicated rate of air gai, and accordingly the rise of losses of an indicated work of products of combustion at the compressor shaft. The engine indicated power decreases from 98.3 kW at α = 2.5 to 91.6 kW at α = 4.

5. Conclusion

The main conclusions obtained from the present study of external combustion engine are as follows:

In the external combustion engine with a filling of tanks by compressed air, the efficiency of transformation of chemical energy of fuel into mechanical work of gases essentially (on 60 - 70 %) surpasses the efficiency of traditional internal combustion engines with a spark ignition.

The external combustion engine, in consequence of the raised values of the air to fuel ratio and low values of the maximum combustion temperature (T_s <1300 K), possesses high ecological characteristics without the use of additional devices for neutralizing the exhaust gases.

In the external combustion engine, it is possible to use any kinds of fuel, both gaseous and liquid.

However practical use of the external combustion engine as power plant in vehicles, for example on buses for service conditions in cities, demands: the conducting of an expensive program of development of the engine and its systems; (among others: the manufacturing from polymeric materials of the tanks for compressed air capable to withstand operating pressures up to 50 MPa), and the creation of filling stations and car repair shops necessary to keep all this stock rolling.

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