Structural solutions of the supercharged engine in the output and input system

Zakirjon Musabekov, *Oybek* Daminov*, and *Abdukhalil* Ismatov

Tashkent State Technical University named after Islam Karimov, 100095, Tashkent, Uzbekistan

Abstract. During the intake process, the working volume of the cylinder is filled with a fresh charge. As noted, a fresh charge consists of air in a diesel engine, and in engines with ignition from an electric spark (gasoline carburetor or gasoline with fuel injection, as well as gas) – from a mixture of air and vapors of light fuel or combustible gas.

1 Introduction

During the intake process, the working volume of the cylinder is filled with a fresh charge. In a diesel engine, as noted, a fresh charge consists of air, and in engines with ignition from an electric spark (gasoline carburetor or gasoline with fuel injection, as well as gas) – from a mixture of air and vapors of light fuel or combustible gas.

The air in the engine cylinder plays a dual role. Firstly, air and, mainly, air nitrogen, serves as a working fluid, that is, it is an accumulator of thermal energy, which is absolutely necessary to ensure that internal combustion engines can carry out mechanical work during the expansion of the working fluid. Secondly, atmospheric oxygen serves as a fuel oxidizer, that is, it is used in the combustion process to convert the latent chemical energy of the fuel into thermal energy.

The more air enters the engine cylinder, the more fuel can obviously be burned and the more work can be obtained, and therefore more power can be achieved.

Motor power

$$N = f\,(G...) \tag{1}$$

where G is the mass amount of fresh charge entering the engine cylinder. The combustion chamber (compression) before the intake process is filled with residual gases. A fresh charge can only fill the working volume of the cylinder. During the filling process, the pressure of the fresh charge decreases due to the vacuum in the cylinder, and the temperature increases compared to the pressure and temperature before the fresh charge entered the cylinder. This is due to its heating from hot walls [2, 4, 6].

The heating of the charge when mixed with the residual gases has practically no effect on the filling of the cylinder, since the residual gases are simultaneously cooled. As will be shown below, the expansion of the charge during heating is compensated by the reduction in the volume of residual gases during their cooling. A decrease in pressure and an increase in

* Corresponding author: oybek1980@mail.ru

the temperature of a fresh charge due to its heating from hot walls leads to a decrease in the mass amount of fresh charge G that enters the cylinder compared to the amount of fresh charge (theoretical) G_t that could fit in the working volume of the cylinder at pressure and temperature charge, which he possessed before entering the cylinder.

Thus, always

$$G < G_t. \tag{2}$$

Obviously, when designing intake systems, it is necessary to strive to ensure that the above inequality is expressed as weakly as possible, or the ratio of the first of these quantities to the second is as large as possible.

2 Methods

Influencing the hydraulic resistance of the intake system, the size of the flow sections, the number of elbows, the geometric profile of the flow path – these are engine design solutions in the output and input system.

From here we come to the concept, which is called the filling factor:

$$\eta_v = G/(G_t) \tag{3}$$

So, the filling factor is the ratio of the amount of fresh charge that actually entered the engine cylinder to the amount of charge that could fill the working volume of the cylinder at pressure and temperature of the ambient atmospheric air. The above definition of filling ratio applies to four-stroke naturally aspirated engines.

The data has been selected to cover a wide range of engines of various designs, sizes and applications. Analysis of these data shows that the criterion is relative.

The area limits the possibility of using a calculation method for them, for most engines it is valid, but for small two-stroke engines with a short exhaust process and for large engines with very long pipelines [7, 8].

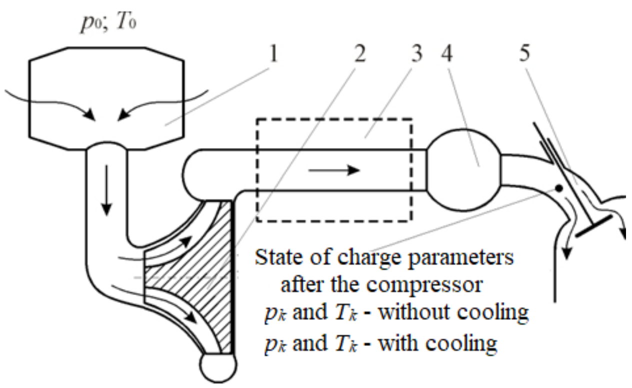

Fig. 1. Scheme of the intake system of a supercharged engine: 1 - air filter; 2 - compressor; 3 - charge air cooler; 4 - inlet pipeline; 5 - inlet valve or purge windows in two-stroke engines.

For two-stroke engines, the amount of fresh charge is taken that could fill the working volume of the cylinder at the pressure and temperature of the charge in front of the intake organs. In supercharged four-stroke engines, after the compressor or charge air cooler [12, 16].

The filling ratio allows you to objectively evaluate the design perfection of the intake systems of different engines.

From (3) for the amount of fresh charge entering the cylinder, we have

$$G = \eta_0 \cdot G_t, \tag{4}$$

where $G_t = \rho_0 \cdot V_h$ is the mass of fresh charge that could fill the cylinder at a pressure and temperature equal to the pressure and temperature in front of the inlet bodies;

$$\rho_0 = p_0/(R_g \cdot T_0) \tag{5}$$

fresh charge density at parameters corresponding to the conditions in front of the inlet organs;

p_0, T_0 are the pressure and temperature of the charge in front of the intake organs (it should be borne in mind that in the case of supercharging engines, the charge pressure and temperature in front of the intake organs are also denoted by p_k and T_k). R_g is the gas constant of air.

3 Results and discussion

Only certain values of thermodynamic parameters change, which depend on changes in pressure and temperature.

In a real engine, it must be taken into account that part of the power is spent on the compressor drive.

In gas turbine supercharging, a combined engine is formed, consisting of a piston part, a gas turbine and a piston compressor. High-pressure turbochargers are used in front of the turbine in automobile and tractor engines [1, 5].

Thus, for supercharged engines – see the intake system diagram in fig.1 - p_k and T_k - pressure and temperature after the compressor are taken as parameters of the state at the inlet to the cylinder. In the absence of pressurization, $p_c = p_0$, $t_c = t_0$, where p_0 and t_0 are, respectively, the pressure and temperature of atmospheric air, in other words, the parameters of the state of charge at the compressor inlet [10, 13, 15].

In naturally aspirated engines, ρ_0 is the fresh charge density at ambient conditions. Fluctuations in atmospheric air density depending on meteorological conditions are ± 15%. Consequently, the engine power will fluctuate approximately within the same limits.

As you know, with height, the air density decreases, and with it, the engine power also decreases. So, at an altitude of 6000 *m*, the engine loses about half of the indicated power that it develops at sea level. The state of atmospheric air has a similar effect on the power of four-stroke supercharged internal combustion engines and two-stroke engines [11, 14].

For the purpose of comparing the specific power of engines, it is customary to bring the experimentally measured power to the so-called normal state of the atmosphere, characterized by the value

$$p_0 = 0{,}1013\ MPa \quad and \quad t_0 = 20\ ^oC, \quad or \quad T_0 = 293\ K.$$

Thus, for p_v we get

$$\mu_v = G/(G_T \cdot V_0/V_h) \tag{6}$$

The numerical value of the filling factor is less than one.

The degree of pressure reduction at the end of the intake stroke p_a/p_0 is the main factor determining the value of the filling ratio η_v.

The pressure p_a depends mainly on the magnitude of the hydraulic resistances of the intake system: with their growth, p_a decreases, and vice versa.

The hydraulic resistance of any system is determined by design factors and the speed of the fluid.

It is also known that the hydraulic resistance varies in proportion to the square of the fluid velocity (droplet or elastic). Based on this and assuming that the difference (1 - η_v) characterizes only hydraulic resistances (Fig. 2), we can write:

$$\eta_v = 1 - k \cdot w^2 \quad (7)$$

where $k \cdot w^2_{mid}$ – is the loss in the value of the filling factor due to hydraulic resistance of the intake system;
$k \cdot w^2_{mid}$ – is the average speed of the fresh charge at the inlet to the engine cylinder;
k – coefficient of proportionality, depending on the design factors affecting the hydraulic resistance of the intake system, the size of the passage sections, the number of elbows, the geometric profile of the flow path, the features of the air cleaner by decreasing k, hydraulic losses decrease and η_v increases. If we take into account the final speed of opening and closing the valves (more precisely, the phases of the valves operation), the time-section should be understood as a definite integral of the form, Fig. 2.

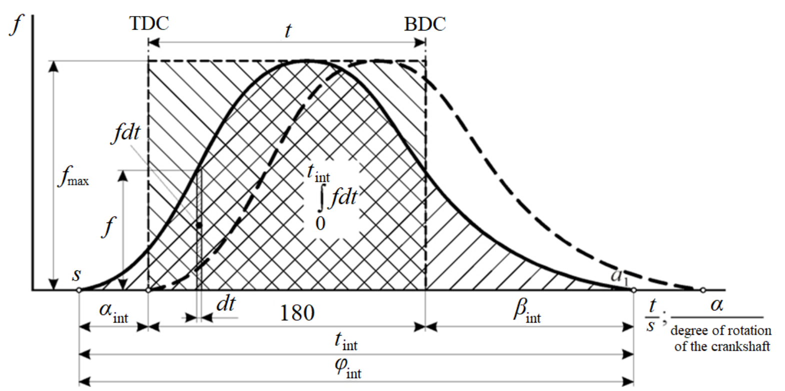

Fig. 2. To the concept of time-section of the valve.

$$\int_0^{t_{B\Pi}} f(t)\, dt, \text{ m}^2/\text{s}, \quad (8)$$

where $f(t)$ is the variable area of the passage slot of the inlet body. It should be noted that the value of f_{max} is limited by design factors

With valve timing

$$f_{max} = F_1 (D_v; h_v; i_v), \quad (9)$$

where D_v - is the valve diameter; h_v - valve lift height; i_v - is the number of valves.

With spool valve timing

$$f_{max} = F_2 (B_w; h_w; i_w), \quad (10)$$

where B_w is the width of the window; h_w – window height; i_w – number of windows.

The duration of the intake process is limited by the engine speed and the largest intake phase according to the ratio

$$t_{int} = \Phi_{int} / \omega \quad (11)$$

where Φ_{int} is the phase (duration) of the intake process, *rad*;

$\omega=\pi\cdot n/30$ – shaft angular speed, s^{-1}.

For the current moment of time, if φ is defined in *rad*, the expression is true

$$t=\Phi/\omega. \tag{12}$$

The time-section at the constancy of the law of change in the area of the intake passage and the phase of the intake process is inversely proportional to the engine speed, that is, with an increase in the shaft speed, it decreases.

The relationship between times in seconds, crankshaft rotation angle in degrees and its rotational speed is determined taking into account the following: a time interval of 60 *s* corresponds to 360º degrees of rotation of the crankshaft [3, 8].

The numerical values of the opening and closing phases of the intake valves are given in Table 1.

Automotive engines are characterized by large phase values gas distribution, for tractors, on the contrary, are smaller. At the same time, passenger car engines, in turn, are of great importance.

α_{int} and β_{int} compared to truck engines.

Table 1. The opening and closing phases of the intake valves.

Engine type	**α_{int}, degree of rotation of the crankshaft to TDC**	**β_{int}, degree of rotation of the crankshaft after TDC**
Tractor	10…20	30…40
Automobile	20…30	50…70

4 Conclusion

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