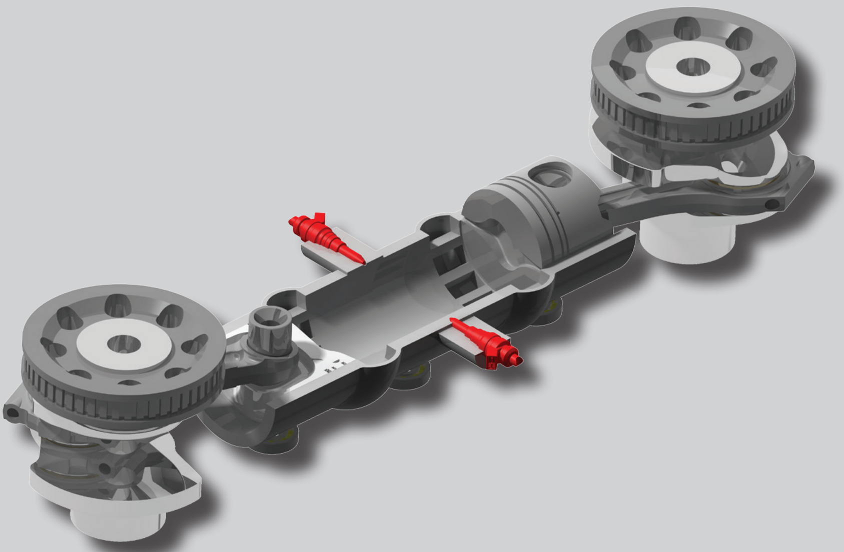
POLITECHNIKA LUBELSKA

Mirosław Wendeker

Aspects of internal combustion engines in machines



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Reviewers:
Dr. hab. inż. Paweł Droździel, prof. Politechniki Lubelskiej
Doc. Ing. Dalibor Bárta, PhD

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Introduction

Propulsion is a mechanism that sets a machine in motion. A given motion always occurs in an environment where there are conditions resisting that motion. The motion of blades in a lawn mower encounters resistance from the grass being cut, the motion of an automobile must overcome air resistance, friction from the road surface or resistance of an elevation on its way, while a horizontal motion of an aircraft must overcome air resistance as well as resist the force of gravity pulling it down. To produce a motion that will overcome resisting forces it is necessary to perform mechanical work. The mechanical work is performed by engines powered by sources of energy. Energy may come in the form of electric power from a storage cell, chemical energy from fuel, thermal energy resulting from nuclear fission, kinetic energy of wind or potential energy of a stream of water. Engines convert energy that has been accumulated or is directly harvested from the environment into mechanical work. Usually, the energy is transferred by a rotating shaft of the engine and then used as mechanical work or converted into electrical energy. There are also other kinds of engines, such as rocket engine or linear induction motor, which generate translational kinetic energy and have no shaft. Usually, rotational speed of an engine's shaft is either too slow or too fast relative to the motion of the machine. For this reason, power transmission systems in machines consist of propulsion motors and propulsion gears, oftentimes connected with the motors through clutches.

A combustion engine uses compression and expansion of a working gas to generate torque or force. The working gas is cooled during compression and heated during expansion. The amount of energy needed to compress a cooled gas is smaller than the amount of energy produced during expansion of the heated gas. Hence, energy produced by expansion is used to compress the gas and run the machine. If the heat needed to raise the temperature of a working gas is generated as a result of burning fuel, than an engine using the heat is called a combustion engine. A combustion engine is the basic element of any combustion propulsion systems.

This book is devoted to the subject of propulsion systems with internal combustion, as combustion engines are divided into internal combustion engines and external combustion engines. In internal combustion engines the chemical composition of the working gas is constant. The heat produced as a result of external combustion is delivered to the working gas while it expands. The transfer of combustion heat into the working gas occurs by heat exchange through convection and radiation. In internal combustion engines, the working gas contains fuel and oxygen, which are exhausted in the form of water vapor and/or carbon dioxide after combustion is finished. Internal combustion, the working gas is affected by the heat directly, and after the work process is completed it must be removed and replaced with a fresh load. The focus of this book is the kind of engines that work by that principle. The author must also indicate, however, that the book does not discuss propulsion gears or clutches, both of which, alongside with engines, make part of power transmission systems.

Internal combustion engines are categorized into three main groups:

- piston combustion engines, which have reciprocating or rotary pistons,
- turbine engines, which have rotors,
- reaction engines, in which a dynamic reaction of the jet-stream exhausted by the engine occurs.

For didactic purposes, combustion engines are classified according to different characteristics and properties. With regard to intended working environment, combustion engines are divided into:

- stationary: used in power plants, used in displacement pumps for liquids and gases, used on farms,
- portable: used as drive motors for automobiles, tractors, aircrafts, ships and vessels, locomotives, as well as other moving machines.

Limited natural resources and the risk of exceeding ecological limits have contributed to a considerable growth of interest in renewable sources of energy (Figure 1) and a vast expansion in the range of their applicability, restricted for political, legal and economic reasons or due to issues related to forms of natural occurrence and potential for practical use.

With regard to the kind of fuel used for combustion, one can distinguish engines running on:

- light liquid fuel (petrol, alcohols and naphtha),
- heavy liquid fuel (diesel oil, mazout, vegetable oil esters),
- gaseous fuel (hydrogen, propane, butane, natural gas and other gases),
- several kinds of fuel simultaneously, where the basic fuel is a gas, and liquid fuel is used in some of the working conditions of engine operation.

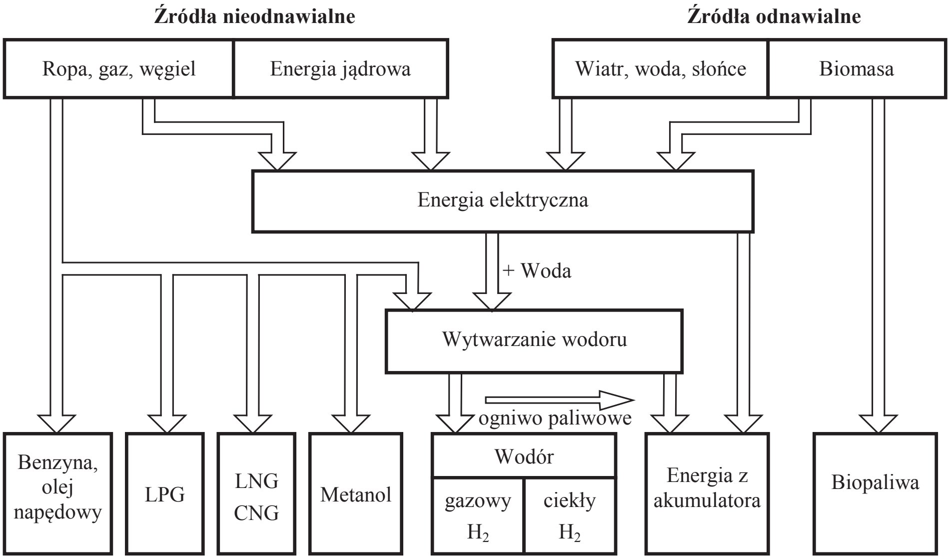


Fig. 1. *Alternative power sources for machines.*

Legend:

Źródła nieodnawialne – Non-renewable sources
Źródła odnawialne – Renewable sources
Ropa, gaz, węgiel – Oil, gas, coal
Energia jądrowa – Nuclear Energy
Wiatr, woda, słońce – Wind, water, sunlight
Biomasa – Biomass
Energia elektryczna – Electric power
+ Woda – + water
Wytwarzanie wodoru – Hydrogen production
Benzyna, olej napędowy – fuel cell, Petroleum, diesel oil
LPG – LPG
LNG, CNG – LNG, CNG
Wodór – Hydrogen
Gazowy H2, Ciekły H2 – gaseous H2, liquid H2
Energia z akumulatora – Energy from a **battery**
Biopaliwa – Biofuels

It should be noted that fuel is always burned as a gas, after it had evaporated. Usually, the boiling point of gaseous fuels is so low, that when used in engines (after being injected into an engine) they are already in the form of gas. Liquid fuels must evaporate prior to combustion and evaporation takes time to occur. Additionally, both liquid and gaseous fuels must mix with air after evaporation, so as to create an air-fuel mixture. When the mixture is being formed, different processes occur in engines, some of which are:

- formation and development of the jet of fuel being injected,
- deposition of fuel droplets on walls of the throttle, inlet pipe, inlet valve or piston,
- convection of fuel droplets by the airflow,
- fuel evaporation (from droplets and the fuel film),
- fuel leakage caused by gravity and pressure from airflow.

With regard to the method of forming the air-fuel mixture, engines are divided into:

- those with external mixture formation, whereby the mixture is formed in the carburetor, in the mixing tank of a gas-powered engine and in the inlet pipe of an engine, following injection of fuel into the inlet system,
- those with internal mixture formation. In this kind of engines, air flows in while the cylinder is being filled, then towards the end of compression process fuel is injected and the mixture is formed inside the cylinder. Engines belonging to that category are compression-ignition engines, spark-ignition engines with direct fuel injection into the cylinder, and gas engines in which liquid fuel or gas is let into the cylinder at the beginning of compression.

The process of burning fuel in combustion engines is designed so that the thermal energy produced by the chemical reaction of combustion (ignition) is sufficient to sustain the reaction and transfer the superfluous heat to neighboring part of the combustion chamber. With regard to the method of ignition of the mixture, one can distinguish the following kinds of engines:

- with ignition of the air-fuel mixture occurring from a spark (with spark-ignition),
- with self-ignition of the air-fuel mixture (compression-ignition engines, commonly known as Diesel engines),
- with a pre-chamber and a spark-ignition, in which the air-fuel mixture is ignited with a spark inside the precombustion chamber of a small volume, and then the process is continued in the primary combustion chamber,
- with an ignition of gaseous fuel from a small amount diesel oil itself ignited as a result of compression (the so called dual-fuel diesel engines).

High temperatures of combustion combined with a high ratio of load compression entail the necessity for cooling the components of an engine, especially those surrounding the combustion chamber. In case no cooling is installed, the temperature of vital components rises shortly, which in turn reduces lubrication efficiency (loss of lubricity and burning of the oil), causes premature ignition (self-ignition) of the mixture, and excessive thermal expansion of the piston in the cylinder, which usually leads to seizure. Working temperature that is too low causes worse conditions for fuel evaporation, which distorts the process of combustion and may result in an increased emission of harmful substances. A cooling system is responsible for keeping the temperature of the components in an optimal range.

A proper function of the cooling system is necessary for the engine to work with its indicated power. The types of engines with regard to the method of cooling, are:

- engines with liquid cooling,
- engines with air cooling.

The basic indicators of quality for internal combustion engines are:

- reliability of all the parts of the engine;
- efficiency of converting thermal energy into mechanical energy. It is a number representing efficiency of fuel, or fuel consumption in units, that is, the amount of fuel (in units of mass or volume) that is consumed in a unit of time, per a unit of engine power;
- Engine power in relation to a unit of working volume of the cylinder or to a unit of surface area of the piston cross section (the power-to-volume ratio);
- Mass of an engine per a unit of power (the power-to-mass ratio) and overall dimensions of the engine.
- Degree of exhaust toxicity and smoke opacity and the levels of acoustic emission and vibrations during engine operation;
- Simplicity of design, maintainability and operating comfort, costs of manufacturing, operation and servicing;
- Startability;
- Potential for upgrades and improvement of parameters in accordance with current technological trends;
- Dynamics of reaction to changing speeds and power loading, as they vary along with operating conditions;

1. Basics of piston combustion engines

Piston combustion engines convert chemical energy of fuel into mechanical energy, by taking advantage of the heat of combustion, which rises the heat content of the working gas, which in turn transforms into work of the rotating crankshaft, running in assembly with pistons and connecting rods. In internal combustion engines, the working gas is initially an air-fuel mixture, air being the source of oxygen needed for combustion. In the process of combustion, air and fuel gradually turn into exhaust gases. The history of burning fuel inside a cylinder enclosed with a movable piston on one side goes back to the year 1673, when Christiaan Huygens, a scholar on the court of Louis XIV, has been setting fire to gunpowder inside the cylinder of a machine pumping water to royal gardens. This was a single-cylinder, one-stroke engine. In 1860, Etienne Lenoir built a two-stroke engine of 8,8 kW, in which gas fuel ignition was induced electrically. One can arguably say that the first kind of fuel used for spark-ignited piston engines was gas – in that case it was town gas, a mixture of mainly hydrogen and carbon monoxide. In 1876, Nicolaus Otto succeeded in building a four-stroke engine with compression of air-fuel mixture. In 1884, Otto developed an innovative electrical ignition, and so using different kinds of liquid fuels, especially gasoline, became possible.

Thermodynamic processes that occur during conversion of chemical energy into thermal energy all take place in a working space that changes in each cycle of engine operation.

A piston engine has at least one piston, which constitutes a movable constraint of space available for the working medium. The piston works in a translational motion or in a resultant rotary motion and acts as a means of transmission of forces. Effective work of such engine involves movement of the piston under pressure of the working gas. The working gas is air, a mixture of air and fuel and exhaust gases produced as a result of burning fuel in air. The share of each gas changes over time, along with processes of filling, injection, combustion and emission of exhaust gases. As a result of interaction between crankshaft and pistons, the moving piston sets the crankshaft into rotary motion, through which the effective power of the engine is finally transmitted and received. Over the last few years, a different solution has also appeared, in which the energy of a piston moving along the axis of the cylinder is converted into electrical energy, induced within a winding encircling the cylinder. This approach eliminates forces of a piston pressing on the surface of the cylinder (related to the work of the mechanism).

The piston combustion engines may be divided according to the manner they complete a working cycle:

- four-stroke non-supercharged (a naturally aspirated engine – air intake from the atmosphere) and supercharged (compressed air intake).

- two-stroke unsupercharged and supercharged. A supercharger powered by the energy of exhaust gases may be used (turbocharging) or a supercharger powered by the engine or a couple of supercharges, of which one is powered by the engine and the other by a gas turbine.

With regard to piston position, piston engines are divided into:

- single piston engines (one piston and one working space in each cylinder),
- opposed-piston engines (the working space is situated between two pistons moving in opposite directions inside one cylinder),
- double-acting engines (with working spaces are on both sides of the free piston).

With regard to piston position, piston engines are categorized into:

- vertical in-line engines,
- horizontal in-line engines,
- v-type engines
- radial engines
- opposed-cylinder engines (the boxers).

A separate type of piston engines are engines with rotary pistons, known as Wankel engines. It is characterized by a cyclically changing volume of the working space, as a result of the piston revolving orbital in a cylinder with a trochoidal cross-section. The piston is triangle in shape, with each of the sides symmetrically curved and with tops touching to the cylinder at all times. As a result, three chambers that change their volume are formed at the same time. The volume of each of the chambers varies from minimal to maximal twice over a single revolution of the piston. The orbiting piston is connected to the shaft via a planetary gear. Engines with rotary pistons may be of three types:

- the piston (the rotor) is in a planetary motion inside the body of the cylinder,
- a chamber of a varying volume is formed during the revolving motion of the piston and the working cycle is performed inside it,
- the body of the cylinder is in a planetary motion, while the piston remains stationary.

The main advantages of a Wankel engine are its relatively small size and weight, a simple design (the number of parts is significantly smaller than in case of classic engines), low vibration and noise levels during operation, potential to work at large speeds (a high unit power) and high mechanical efficiency. Unfortunately, the disadvantages of such solution overweigh the benefits. The main flaw of this engine is the shape of the working chamber, which makes it difficult to be sealed correctly, translating into low compression ratio and low thermodynamic efficiency. Engines of this kind have a higher heat loss, as the combustion chamber is wide and flat in shape. A thin combustion chamber results in relatively high specific fuel consumption and high specific emission of hazardous waste gases.

Currently, the most prevalent type of engines used in machines is the four-stroke combustion engine, which comprises the following basic systems, units and mechanisms:

- *Engine block* – stationary parts supporting the movable elements of the crankshaft and piston mechanism: the lower cover of the crankcase, the block, the cylinders, the cylinder head(s);
- *Crank mechanism* – consists of movable parts which receive the pressure of gases, convert translational motion into rotary motion and transmit the forces onto the crankshaft. The crank mechanism comprises entire pistons, connecting rods, the crankshaft and the crank;
- *Valve gear* – maintains the specific order of releasing exhausts and allowing a fresh load of air into the cylinder. It comprises intake manifolds, exhaust manifolds, and elements that receive drive from the crankshaft: rods, valve lifters, levers, crankshaft and the camshafts;
- *Air supply system* – comprises a supercharger or a turbocharger, a driving elements, an air manifold, an air cooler, an air filter, a intake silencer;
- *Fuel supply system* – consists of elements and mechanisms that ensure fuel preparation and atomization as well as regulation of quantity or quality of a load inside a cylinder. The fuel supply system of a diesel engine comprises a fuel tank, low pressure pumps, filters, high pressure pumps, regulators, fuel lines, injectors. Fuel supply in engines with external mixture formation: carburettors or mixers and other;
- *Lubrication system* – independent devices and mechanisms providing oil to interacting surfaces and to elements that require cooling: oilers, oil pumps, filters, oil tanks, oil coolers etc.
- *Cooling system* – provides cooling to elements exposed to exhausts. The cooling may be done by water, a special liquid, air, oil and fuel (cooling pistons, injection pumps). Depending on the method of cooling a certain group of elements, the cooling system is made up of devices and mechanisms providing the working medium to the engine parts and heat exchangers;
- *Ignition system* – provides ignition to the air-fuel mixture inside a cylinder. It consists of electrical circuits of low and high voltage, a battery, coils and ignition distributor/electronic module, plug wires, sparking plugs.

Today, the main criteria by which engines are designed and built are meeting norms of emission for toxic substances being part of exhausts and reducing fuel consumption. The work of constructors is aimed at developments in two principal directions: adapting engines to meet new requirements (a classic mechanical system i.e a crankshaft-piston system + new components + electronic steering as well as integrating the engine with an automatic gear shaft) and dividing new generations of propulsion, which offer a healthy balance between toxic emissions, fuel consumption and performance. For this reason, constructors introduce new elements to the engine design, such as electromagnetic inlet and outlet valves, a turbocharger of variable geometry, electric supercharger, systems for storage of nitric oxides and particulate pollutants, etc. These engines offer a high degree of integration between technological and economic benefits. The very basic aspect of this integration is the use of direct fuel injection, so as to implement new conceptions for burning lean mixtures, which recently have been developing quickly due to extensive use of computer simulations.

2. Spark-ignition piston engine

Called after the inventor of the first four-stroke spark-ignition combustion engine, a comparative thermodynamic cycle, in which the processes of isentropic compression, isochoric heat addition, isentropic expansion and isochoric heat rejection, is known as the Otto cycle occur– shown in Figure 2.

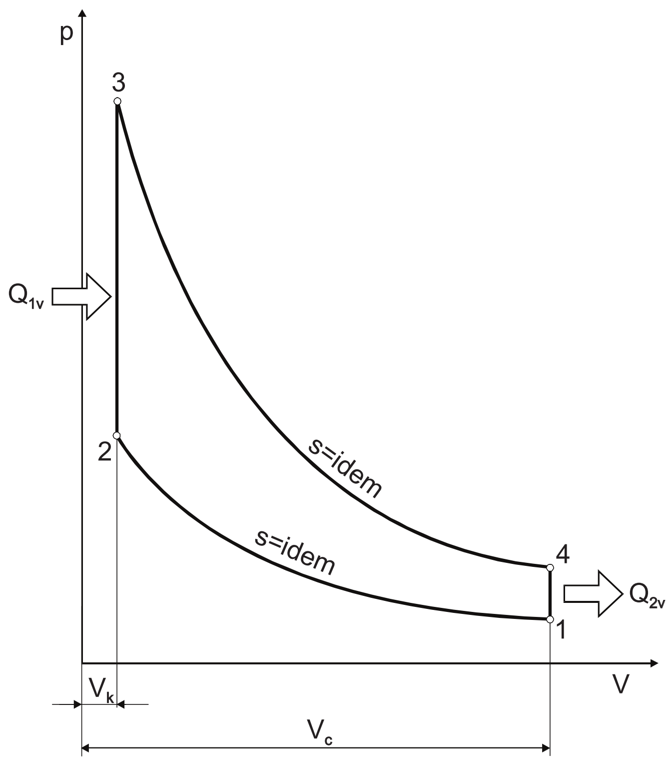


Fig. 2. *The Otto cycle*

Efficiency of the Otto cycle increases along with compression ratio and the isentropic exponent, and is not dependent on how much heat is added into the system. The efficiency of the Otto cycle is greater than the efficiency of a cycle which, apart from isochoric heat addition, also involves isobaric heat addition.

In four-stroke piston engines, performing a thermodynamic cycle requires two full turns of the crankshaft. During the first turn, the compression stroke and the working-stroke are performed while in the second turn the exhaust stroke and the inlet (intake) stroke take place.

In Figure 3, the basic principle of operation of a four-stroke combustion engine is demonstrated. The value of the crank angle varies from 0 to 720. For angle 0 (720) and angle 360, the piston is in the TDC position, for which the volume of combustion chamber is the smallest. After intake (segment 5 to 1 in Figure 3), compression occurs, (1-2) during which ignition of the mixture takes place (point c). Combustion begins, accompanied by pressure building up to point 3. Segment 3-4 represents the expansion (working) stroke, while segment 4-5, the exhaust. Outlet valve is opened between points d and e, whereas opening of intake valve occurs between points a and b.

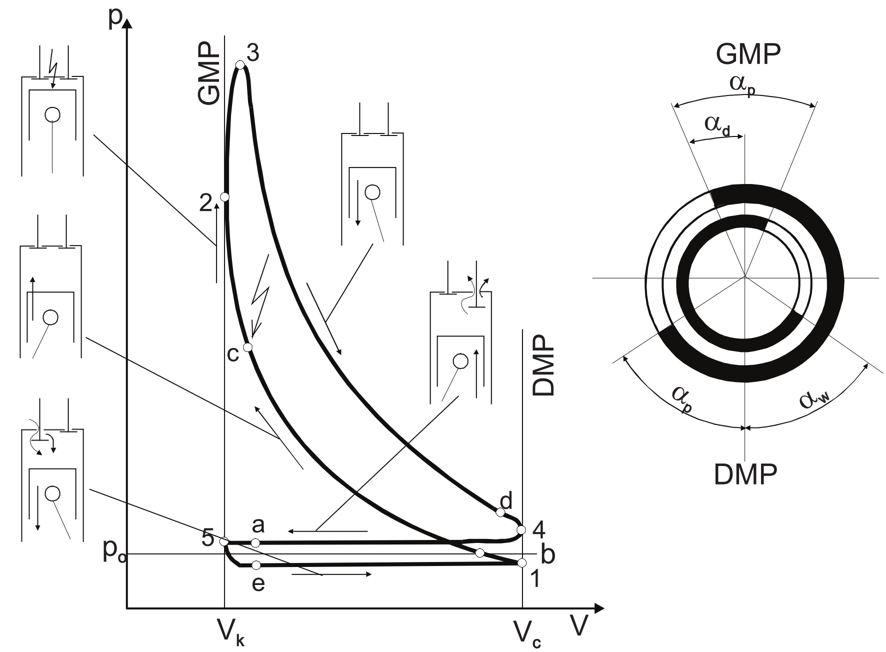


Fig. 3. *Operation diagram of a four-stroke piston engine with spark-ignition.*

GMP – TDC (Top Dead Centre), DMP – BDC (Bottom Dead Centre)

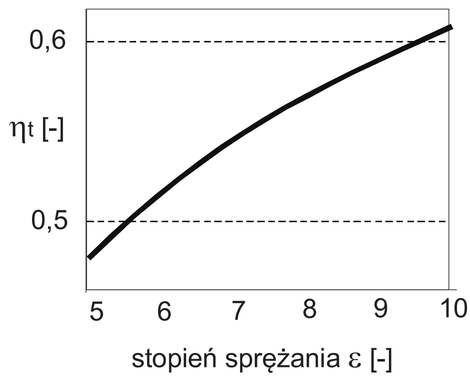
The process by which energy is extracted from fuel is burning hydrocarbons contained in gasoline in the presence of air. Both the fuel and the oxygen used should be present in the combustion chamber in precisely measured doses to form an air-fuel mixture. Out of the total amount of heat being released, only a part is converted into work of the working gas. Assuming that combustion is substituted with Q_1 heat transferred from outside and load exchange with Q_2 heat that is released out of the combustion chamber, theoretical efficiency of the cycle is expressed by the following formula:

$$\eta_t = \frac{Q_1 - Q_2}{Q_1} \qquad (1)$$

Theoretical efficiency of the Otto cycle (as well as other comparative cycles) depends predominantly on the compression ratio ε and the isentropic exponent *k*. By balancing heat added to the cycle and transforming the formula, one arrives at the following equation, representing theoretical efficiency of the Otto cycle:

$$\eta_t = 1 - \frac{1}{\varepsilon^{\kappa - 1}} \qquad (2)$$

Figure 4 demonstrates how compression ratio influences the theoretical efficiency of the Otto cycle for k value = 1,4.

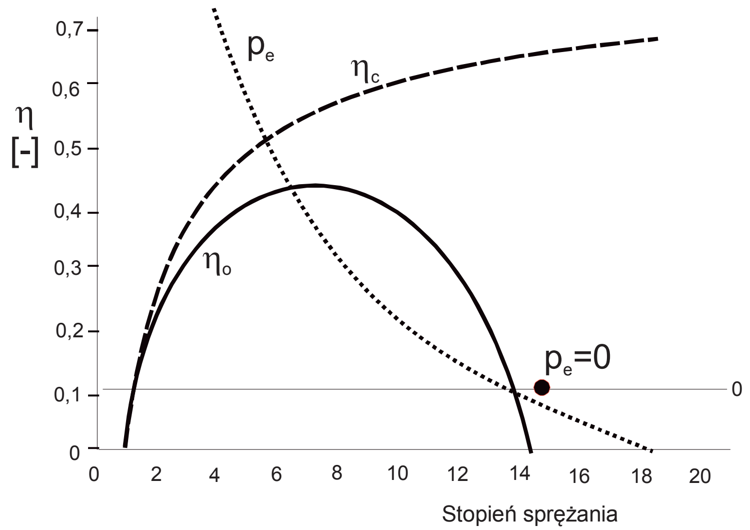


Legend:

Stopień sprężania – Compression ratio ε

Fig. 4. *Influence of compression ratio on theoretical efficiency of the Otto cycle.*

Increasing theoretical efficiency through increasing the level of compression has its limits. Given certain construction problems (increased compression involves increased pressure inside the cylinder and greater mechanical stresses) and imperfect combustion (increased compression causes more engine knocking), it must be noted that increase in ε leads to mechanical losses in a piston engine, as demonstrated in Figure 5. The figure requires explanation concerning designations used.



Legend:
Stopień sprężania – Compression ratio ε

Fig. 5. *Efficiency of an Otto engine in relation to compression ratio*

Compression ratio

The quality of conversion of thermal energy Q, introduced into the cycle in the form of fuel on indicated work Li of the engine, is estimated on the basis of thermal efficiency nth:

$$\eta_c = \frac{L_i}{Q} \tag{3}$$

Thermal efficiency accounts for heat losses caused by gas exhaustion, heat released into the cooling system and heat lost in irreversible processes within the working gas. Indicated power is expressed by the formula for area of the cycle loop, made up as a graphical relationship of pressure p inside the cylinder and volume V of the combustion chamber.

$$L_i = \oint p(V) \cdot dV \qquad (4)$$

The ratio of indicated work Li to displacement volume V1 is called the indicated mean effective pressure of the cycle:

$$p_i = \frac{L_i}{V_s} \qquad (5)$$

Mechanical losses is another factor that diminishes engine efficiency and may have a considerable impact on the optimal ratio of compression. Relationships of general efficiency η_o and mean pressure p_o to compression ratio, presented in Figure 6, must first be supplemented with definitions for those values. Engine efficiency is a ratio of effective work L_e to heat which is added to the cycle with fuel:

$$\eta_o = \frac{L_e}{Q} \qquad (6)$$

True capacity of an engine to perform effective work is defined by mean effective pressure p_e:

$$p_e = \frac{L_e}{V_s} \qquad (7)$$

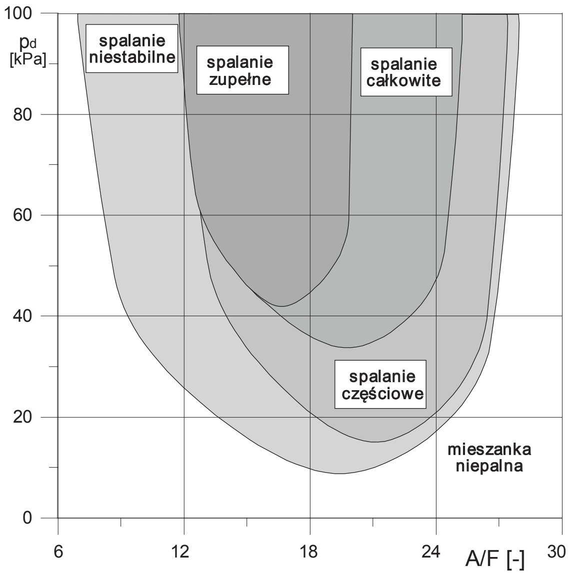
Heat losses resulting from load exchange (i.e. heat rejected along with exhaust) and heat released into the cooling system, cause thermal efficiency of a cycle increase as mass of fuel inside a cylinder drops, with mass of air unchanged. The ratio of air mass to fuel mass is the air/fuel ratio, also written as A/F or AFR:

$$A/F = \frac{m_{pow}}{m_{pal}} \qquad (8)$$

The air-fuel mixture is ignitable only to a certain degree, depending on engine load. In case of an ideal combustion, every particle of the fuel is oxidized. This sort of combustion is called complete or perfect. The parameter which may indicate a potential for a complete and perfect combustion is a fuel's demand for air.

An adequate amount of oxygen in the mixture relative to the amount of fuel allows for complete combustion (i.e. oxidizing the entire carbon contained in the fuel to carbon monoxide at the least) or perfect combustion (i.e. oxidizing the carbon to carbon dioxide, CO_2). This is a theoretical approach – in reality, some part of fuel which has not been burned completely and perfectly is always found in combustion chambers of modern engines. Boundary values for the factors A/F correspond to charges of fuel at which the engine does not gain power. Near combustibility limits, instability in engine work occurs, as a result of the process of combustion not being repetitive. Figure 6 shows five stages of combustibility of a mixture for different stresses expressed as values of pressure in the inlet manifold. In a real engine, the combustion process is exposed to numerous distortions, hence combustion is often incomplete and/or imperfect. This result from an uneven distribution of the mixture composition within the volume of the combustion chamber, distorted spark-ignition, flame extinction etc.

The amount of oxygen needed for combustion is calculated as follows. From notation for oxidization reaction one can assume, that in order to burn 1 kg of carbon, 8/3 kg of oxygen is needed. By analogy, this reasoning can also be applied for other ingredients, and so burning 1 kg of hydrogen requires 8 kg of oxygen and to burn 1 kg of sulphur, one needs 1 kg of oxygen. Oxygen contained in fuel will also be used for combustion, so there will be a smaller demand for oxygen from air:



Legend:

Spalanie niestabilne – Unstable combustion
Spalanie zupełne – perfect combustion
Spalanie całkowite – complete combustion
Spalanie częściowe – partial combustion
Mieszanka niepalna – incombustible mixture

Fig. 6. *Combustibility boundaries for different values of compression in the input manifold of a petrol engine*

The amount of air that is theoretically needed to completely burn fuel depends on its chemical composition. In order to find the demand for air needed for combustion, it is assumed that each of the ingredients of petrol (carbon C of mass fraction c, hydrogen H of mass fraction h, oxygen O of mass fraction o, sulphur S of mass fraction s) will be oxidized consecutively. Even though fuels are mixes of different hydrocarbons and contain impurities, for stechiometric calculations a model of fuel particle is assumed, in which mass fraction of each element is the same as in the actual fuel. For fuels that are not chemical compounds, a unit of matter being an arbitrary compound $\mathbf{C_\alpha\, H_\beta\, O_\gamma\, S_\delta}$**,** in which number of atoms in each element is selected in such a way that its molar mass corresponds to actual mass. The mass of air theoretically necessary to burn fuel is equal to:

$$L_t = \frac{1}{0{,}232}\left(\frac{8}{3}\cdot c + 8\cdot h + s - o\right)\frac{kg_{powietrza}}{kg_{paliwa}} \tag{9}$$

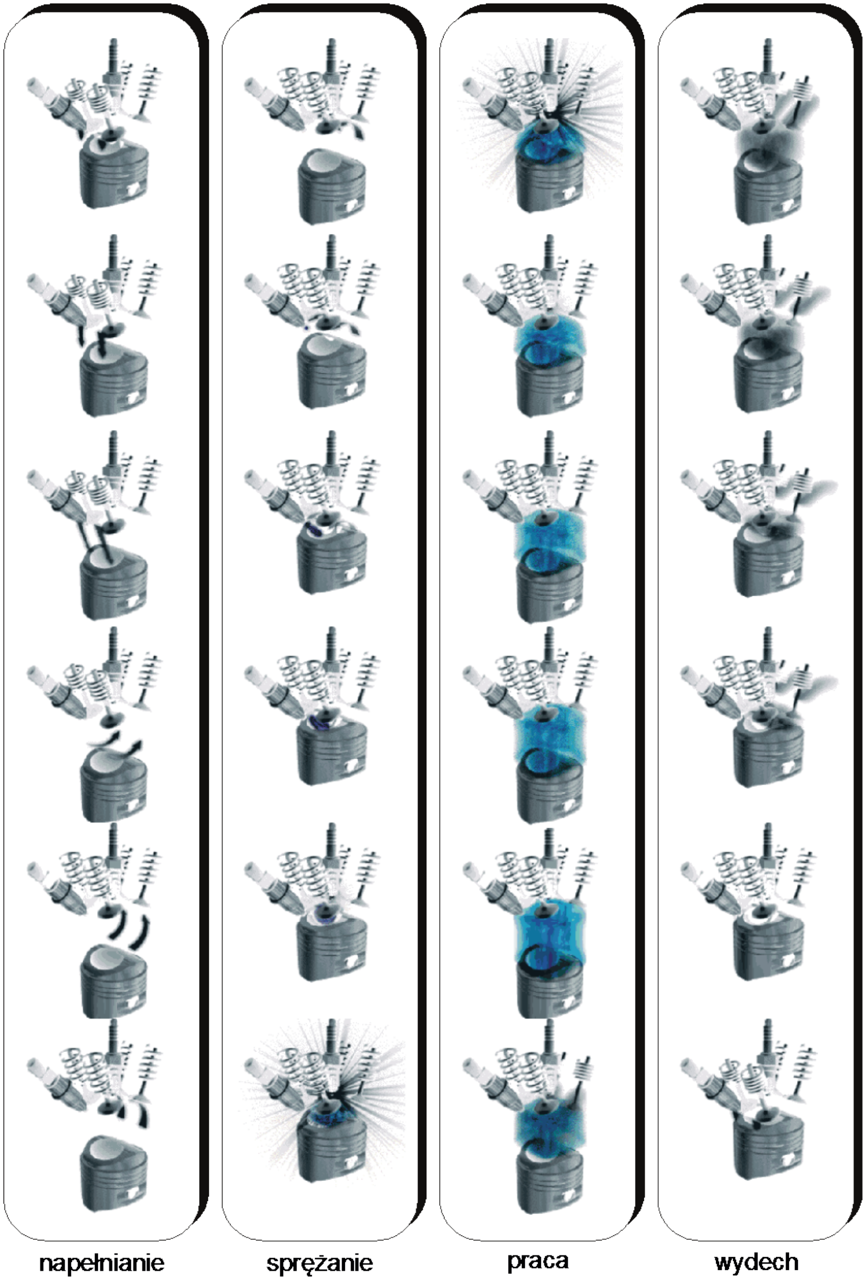
In order to quantify the difference between the amount of air that is theoretically needed and the amount that is actually necessary to burn a portion of fuel, the excess air number was introduced which is a ratio of A/F to theoretical demand:

$$\lambda = \frac{A/F}{L_t} \tag{10}$$

Another number used in order to describe composition of the mixture (predominantly in English language literature) is the fuel-air equivalence ratio, defined as the quotient of fuel-air ratio as it is in the mixture and the stoichiometric ratio:

$$\Phi = \frac{F/A}{(F/A)_{stech.}} \tag{11}$$

In spark-ignition engines which run on a homogenous mixture, fuel is delivered either before or during air intake through an intake valve, so that it is the air-fuel mixture that is subjected to compression inside the cylinder. In spark-ignition engines which burn layered mixtures, fuel is delivered to compressed air prior to combustion – Figure 7.



Legend

Napełnianie – Intake　　Sprężanie – Compression

Praca – Work　　Wydech – Exhaust

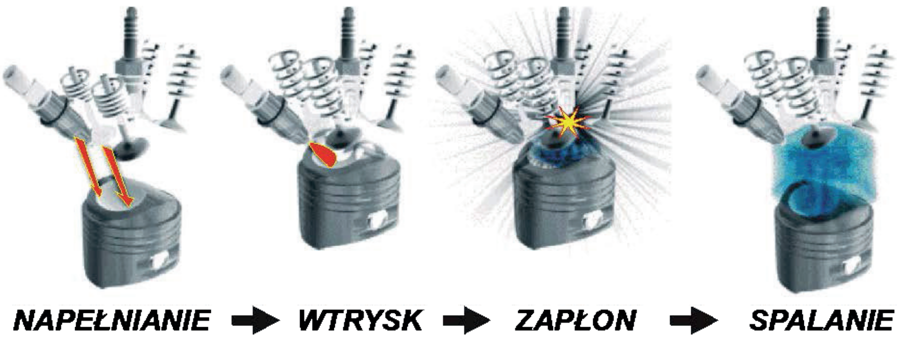
Fig. 7. *Illustration of a working cycle in a spark-ignition engine with direct fuel injection*

The main goal in development of combustion engines is to automatize engine work. This means that basic operations which have been consciously performed by a human operator (the driver) will be managed by a controller. Ignition control has been the first to become fully automatic, since particularly good timing is needed for initiating electric discharge in the combustion chamber.

Demands concerning mixture combustibility require adopting adequate compositions of air-fuel mixtures. Standards of environmental protection lead to full automation of fuel supply to the engine (until recently one could encounter a manually operated string for enriching the mixture during cold starts). Fuel dosage regulation is managed by electronic fuel injectors.

Recently, intake control function (adjusting the amount of air to engine power requirements) has been taken over by control systems.

These three basic functions: intake control injection control and ignition control are an indispensable part of contemporary control systems for automobile spark-ignition engines. Figure 8 presents basic processes leading to formation and combustion of air fuel mixture in a spark-ignition engine with direct fuel injection.



Legend:

Napełnianie – Intake　　Wtrysk – Injection

Zapłon – Ignition　　Spalanie – Combustion

Fig. 8. *Basic processes leading to formation and combustion of air fuel mixture in a spark-ignition engine*

2.1. Induction of the spark-ignition engine

Since the beginning of spark-ignition combustion engines, there has been a need to regulate engine power by changing the mass of working gas in the combustion chamber. The mass of a mixture contained in the cylinder affects the compression temperature and pressure, thereby influencing mixture's ignition capability. To this day, human has been the primary mixture mass regulator altering significantly the airflow resistance through the inlet tract by means of a throttle valve. Automatic filling control is carried out mainly with the throttle valve closed by operating the idle running elements (e.g. by pass valve put in motion by an electric motor). In recent years, filling control trends have been focusing on improving control algorithms which determine idle running work as well as on introducing innovative devices which control the engine filling in terms of rotational speed (e.g. electronically controlled throttle valve, electromechanical valves).

When the throttle is closed (in a number of cases, e.g. starting the engine, idle running, gear shifting and engine braking), it is necessary to regulate the air mass reaching the cylinders. Filling control system with a closed throttle valve is meant to fulfill the following tasks:

- keeping the desired speed during idle running,
- balancing the changes of engine load,
- preventing the engine from stopping due to overload,
- minimizing fuel consumption by keeping the idle running speed at the lowest possible level and by reducing the flow rate during vehicle braking (which also cuts down on fuel consumption),
- eliminating the need for periodical idle running regulation using algorithms compensating for the changes induced by the aging as well as wear and tear of the vehicle,
- reducing the emission of toxic components of exhaust gases and supplying additional air during braking with a throttle valve closed,
- improving (dynamic) driving qualities of the vehicle,
- fulfilling all the tasks in as much unnoticeable way for the driver as possible.

Over the last couple of years, the increasing demands for the engine and the drive power control function have given rise to an automatic throttle control system. First of all, it facilitates the calculation of air mass in the cylinder during future engine cycles (throttle movements are consistent with a previously assumed time scenario without the randomness introduced by the driver). Secondly, all the previously unavailable functions can be now carried out, i.e. gear shift support, mitigation of adverse effects of transient states induced by excessively rapid (unnecessary) reactions of the driver, protection of the catalyst from overheating and the possibility to warm it up in a much faster way after the start-up. Electronic throttle control is essential in systems with abrupt transitions between the power regulation using poor mixture components and the engine power regulation dependent on the amount of the mixture. Such a situation is common for engines burning the stratified mixture when the load and rotational speed are low and the homogenous mixture when the load and rotational speed are increased (engine with gasoline direct injection). The integration of control functions in one controller and a new approach to motor operator design aimed at mass production radically reduced the cost of the system.

One way of regulating the filling coefficient involves using electronic control of inlet and outlet valves which allows for the application of numerous new control algorithms improving the engine work:

- simplification of starting procedure,
- reduction of rotational speed during idle running,
- optimization of transient states,
- regulation of exhaust gas recirculation,
- expansion of cylinder deactivation method,
- effective engine braking,
- possibility of implementing complicated thermodynamic cycles,
- easier integration of load turbocharging.

The key task of an inlet system is to provide every cylinder of a multi-cylinder engine with as even a filling as possible. If the engine power regulation depends on the change of a filling coefficient (homogenous mixture burning system), the filling system is required to precisely regulate the air mass in every cylinder. It is also important to determine whether the filling system is capable of meeting the demands of a maximum engine filling under maximum load. The inlet system construction strongly affects the value of filling level. The maximum filling coefficient is influenced by the compression ratio, type of combustion chamber, intensity of engine cooling and geometry of crank and piston system.

Usually, the inlet system of spark-ignition engines has a more complex design than the inlet system of self-ignition engines. The key task of the inlet system of self-ignition engines is to evenly distribute air to every cylinder and to swirl it before the injection process. The tasks of the inlet system of spark-ignition engines are far more complex. As has been calculated above, alongside an even distribution of the mixture (or air in the case of GDI engines), it is also required to properly swirl the load in the cylinder and precisely measure the air mass which is responsible for charging the cylinders.

Older models of spark-ignition engines were equipped with carburetors which significantly affected the filling process. The current solutions for gasoline engines depend, with almost no exception, on synchronous multipoint or direct injection.

In developed inlet system designs, the air mass regulation in a cylinder is done by means of automatic inlet valve control. The key elements of a modern inlet system include (Fig. 9):

- air filter,
- throttle body with a throttling valve, air by pass valve,
- inlet manifold,
- individual pipes with inlet channel,
- measuring elements (air mass and pressure sensors in the inlet manifold),
- valve system.

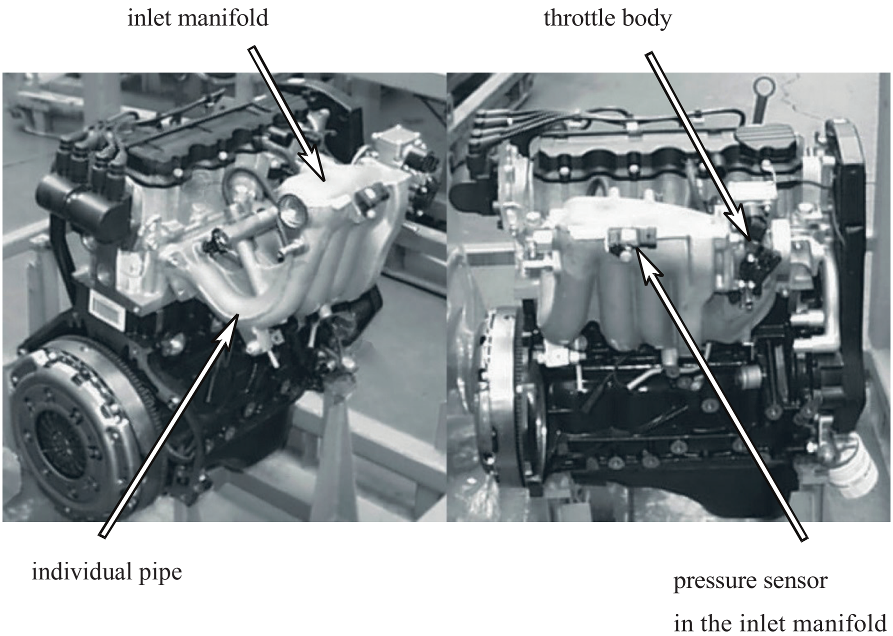


Fig. 9. *Photos of an inlet system of the Holden 2.2 MPFI engine in Lublin II car*

The configuration of the inlet system might be expanded with a charge system, variable length collector and variable valve timing systems. The way the collector pipe and the individual collectors are connected depends on the experience of a company designing the inlet system.

Timing gear system signifies a set of mechanisms controlling the inlet of fresh load to the cylinder and the outlet of exhaust gases. In four-stroked engines, the inflow of gas is controlled by means of lift valves, in two-stroked engines – by means of pistons or pistons and lift valves.

Requirements set before the timing system:

- Both inlet and outlet valves should open and close at the right time with maximum speed (the speed is limited by the pusher acceleration on the cam of the camshaft, permissible due to the risk that the pusher will break off of the surface of the cam).
- Inlet valve should open before TDC and close past BDC, whereas the outlet valve should open before BDC and close past TDC (simultaneous opening of the valves – valve overlap – allows for the cylinder to be flushed and the temperature of exhaust gases to be reduced).
- It is beneficial to open the inlet valve quickly (higher swirl coefficient reduces the self-ignition delay (self-ignition engine) and creates favorable conditions for the flame to spread from the spark (spark-ignition engine)).

- The beneficial course of valve lifting as a function of CA angle involves a high speed of valve lift in the initial stage of opening.
- The filling process is greatly influenced by the inlet valve closing angle (which creates the possibility of load outflow back to the inlet channel).
- The change of timing stages has a relatively small impact on the filling coefficient, it is influenced to a far greater extent by the flow resistance in the inlet system.

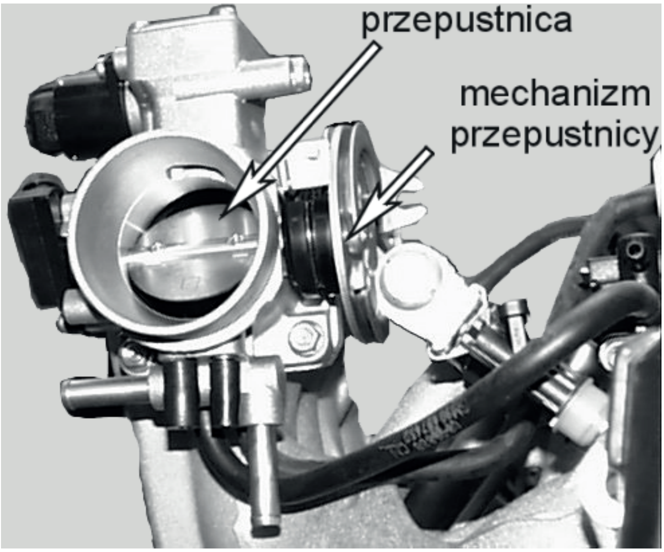
Higher angle values correspond to engines with higher rotational speed. The ultimate choice of timing setting is usually determined on the basis of prototype research results. Due to the technological aspects, in many engines, identical cams are used for both inlet and outlet valves. Actual angles differ from the theoretical ones because they are dependent on the clearances in the timing mechanism. The clearance compensates for thermal expansion as well as inaccurate handling of drive elements and the valve itself (if there is no clearance and the valves are not closed properly, the plates will burn out quickly and the cylinder will be leaking). In the case of high-speed engines, the clearance ranges between 0.1–1.0mm. In the case of low-speed high-powered engines, it can amount to 1.5–2.0mm. The overall timing design is strictly related to the assumed combustion chamber solution and is significantly influenced by the number and the arrangement of cylinders.

Depending on the type of mechanism drive, one can distinguish:

- direct timing (consisting of camshaft in head bearing, valve pusher equipped with clearance regulation device, cup and tool taper responsible for stabilizing the spring and the valve. Used for maximum speed of *22000* rotations/min).
- lever timing (equipped with second-order levers for cylinder heads with a flat or wedge-shaped crowns or first-order levers for semicircular and roof-shaped heads. Used for speed below *6000* rotations/min).
- indirect timing (with additional elements – pusher, valve rod and valve arm. Used for speed up to *5600* rotations/min).

Indirect valve drive is most commonly used despite higher costs of manufacturing, greater amount of components and bigger mass of the elements. This is due to the fact that valve clearance regulation is fairly easy.

In the case of the medium such as gas (air, exhaust gases, fuel vapors), the flow rate regulation is always conducted by means of valves. The basic valve in a gasoline engine is a throttle valve shaped like a round plate rotating around a diameter axis – Fig. 10.



Przepustnica – throttle valve, mechanizm przepustnicy – throttle valve mechanism

Fig. 10. *The key element regulating the filling process – throttle valve*

Modern ignition engines have to meet a great many demands, especially those connected to the necessity of complying with the environmental constraints. For this reason, the engines are equipped with numerous systems designed to alleviate the negative effects of motor operation. These systems cooperate with the inlet system, thereby changing the airflow through the inlet tract. From amongst the systems, special attention should be paid to the following:

- exhaust gas recirculation system,
- vapor fuel system,
- crankcase venting system.

Under the conditions of high temperature of working gas and oxygen excess, nitrogen oxide (NOx) is released during the combustion process. To reduce the amount of nitrogen oxide being released, the exhaust gas is transported through a special channel from the outlet collector into the inlet system.

The mixture of gases consisting of fresh air and exhaust gases is mixed with a reduced amount of fuel by the control module. The fuel is administered in dosages by the fuel system operating within the closed loop system. This process (along with water vapor in exhaust gas) reduces the combustion temperature and cuts the nitrogen oxide emission even by 30%. To decrease the amount of oxygen being delivered to every cylinder, it is necessary to reduce the amount of fuel being injected.

The fuel vapor disposal system is responsible for transferring the fuel vapors accumulated over the fuel mirror in a tank to the inlet collector where they are to be burnt. The exhaust gases are sucked through a container with active carbon to the collector. This way, the emission of hydrocarbons into the atmosphere is reduced. To increase the fuel vapors mass in the inlet collector, it is necessary to increase the amount of fuel being injected.

When the engine is running, a mixture of exhaust gases (which are purged from the combustion chamber into the crankcase through the rings) and an oil mist (formed during engine elements lubrication and cooling) appear in the crankcase. Crankcase venting system prevents this environmentally harmful mixture from being released into the atmosphere. The mixture is being transported to the inlet system and then is burnt in the cylinders.

The solutions and exhaust toxicity control systems listed above are used in various combinations depending on the engine cubic capacity and engine control system in use.

The inlet system also functions as a vacuum pressure supplier providing various systems which use pressure differences to move the movable components (braking system, membrane system for regulating the inlet length, etc.).

The presence of the components listed above affects the design and the functioning of the inlet system. Figure 11 presents an outline of a gasoline engine with systems supporting the inlet system. The outline of pneumatic connections presented in such a way complicates the calculations meant to determine the components of a working gas in the cylinders after the load exchange is finished. Additional complications arise from the necessity to take into account the fuel injection through the inlet valve and the effect the outlet system has on the process of filling.

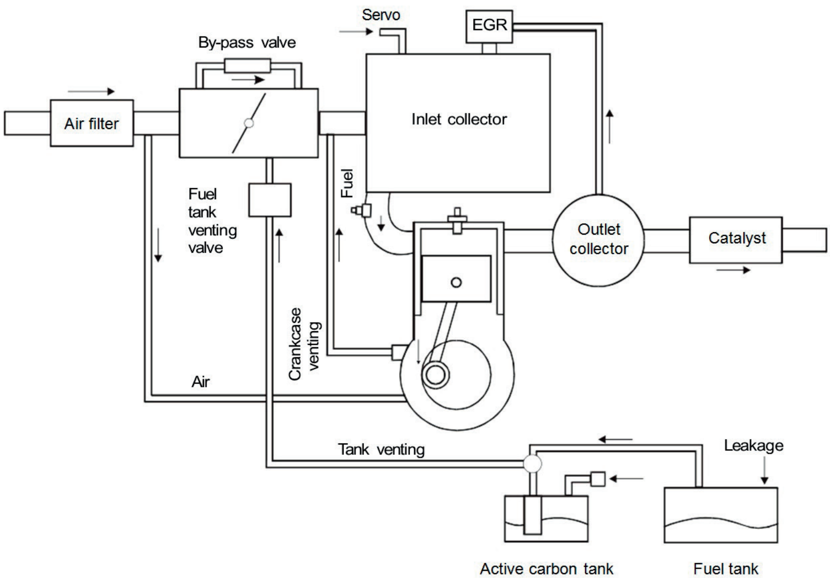


Fig. 11. *Gasoline engine outline with systems supporting the inlet system*

2.2. Powering the spark-ignition engine

To describe the mixture of petrol and air of spark-ignition engine, it is necessary to provide three key parameters: mass of the mixture in the cylinder before ignition, its average (in regard to the mass of the whole mixture) air excess coefficient and the degree to which the mixture is homogenized (we can distinguish homogenous and stratified mixtures). Mechanical energy generated by the engine and transferred by the crankshaft into the receiver at a constant rotational speed can be associated with the effective moment. From the ignition point of view, the effective moment can be influenced by changing mixture's mass or components. The increase of mixture mass in a cylinder leads to the increase of the effective moment. The mass of the air in a cylinder can be increased only to a certain extent depending on the engine design and the conditions under which it is running (rotational speed, thermal state, atmospheric air parameters). Further increase of the effective moment is possible by increasing the fuel mass in the cylinder (mixture enrichment) – Figure 12 – because the air/fuel ratio greatly affects the gasoline engine running parameters, e.g. cycle efficiency, average indicated pressure, maximum temperature of the working gas during combustion.

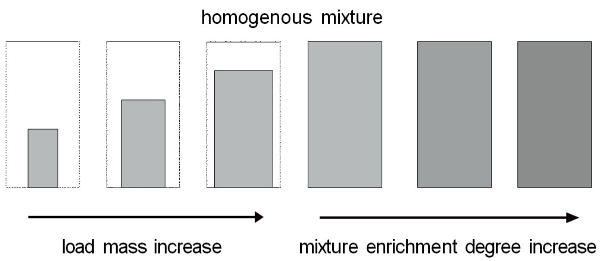


Fig. 12. *Outline depicting the process of increasing the effective moment by increasing the mass of homogenous mixture and, after the air mass ceiling is reached, by enriching the mixture (increasing the amount of fuel)*

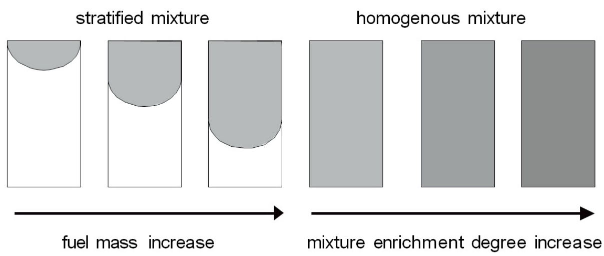


Fig. 13. *Outline depicting the process of increasing the effective moment by increasing fuel mass in the stratified mixture and, when the components of stoichiometric mixture are achieved, by enriching the homogenous mixture*

Low cycle efficiency during poor homogenous mixture combustion can be improved by mixture stratification. The air mass remains constant and is being supplemented with a decreasing fuel mass – Figure 13.

Understanding the influence of mixture components on the rotational moment and the individual fuel consumption is facilitated by the following chain of thought. The energy released during combustion is transformed into working gas work, heat transferred into the cooling system and exhaust gas energy. For a certain amount of air in the cylinder (resulting from the load exchange and dependent on the geometry of inlet and outlet systems, location of the throttle valve and rotational speed), there is a specific fuel amount which guarantees mixture combustibility (very poor mixture) and the overcoming of the engine movement resistance (Figure 14, 1a). Theoretically, the combustion of a mixture consisting of these components should be total and complete. High values of *A/F* coefficient (relatively small fuel mass) are the reason for the maximum cycle temperature to be so low. Low working gas temperature generates small losses incurred as a result of heat exchange and small losses of the energy drifting with exhaust gases. This causes the cycle thermal efficiency to rise.

The key drawbacks of the working process involving poor mixture combustion include the difficulties with ignition as well as total and complete combustion of very poor mixture.

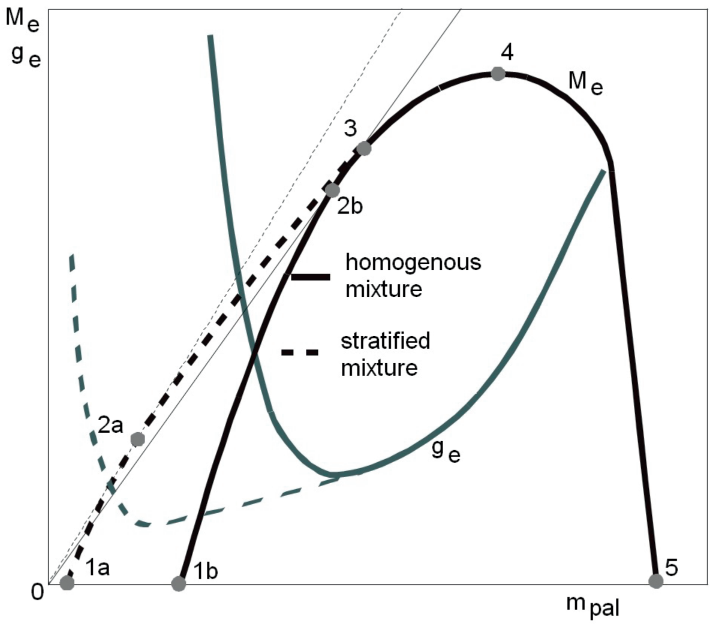


Fig. 14. *The relation of the engine moment Me and a unitary fuel consumption ge to the fuel mass mpal*

It is only the most recent achievements enabling a direct injection in gasoline engines that allowed for the use of combustion systems using the stratified load of variable *A/F* ratio within the combustion chamber. Very poor mixture combustion is characteristic of low intensity and significant angular length. This causes the percentage of the energy converted into gas work to be low, especially for the benefit of the heat released into the cooling system and, to a lesser extent, the energy used to heat up the working gas – the energy irretrievably lost with the exhaust gases. As a result of ignition misses, the amount of fuel necessary for the engine burning the homogenous mixture to run on idling is bigger – see 1b. The bigger the fuel dose (mixture enrichment), the higher the combustion intensity and the percentage of work share of gas in thermal balance. For a certain value of *A/F* coefficient, the increase of the effective moment reaches the maximum (2a and 2b) which constitutes for an optimal general efficiency (minimal unitary fuel consumption).

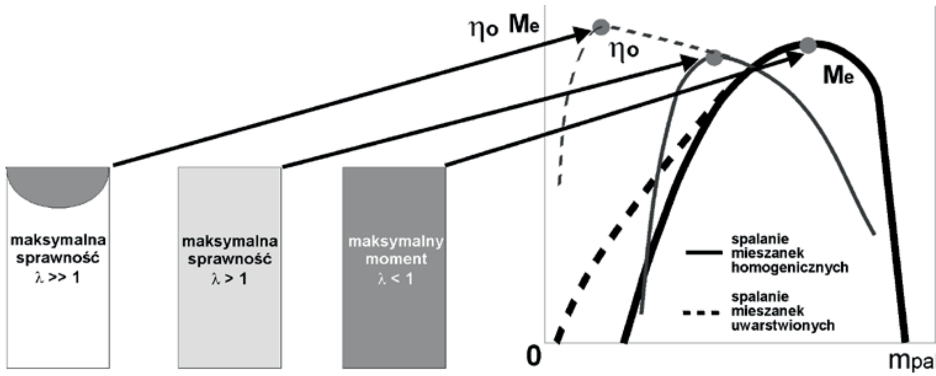
The mixture components corresponding to the optimal cycle thermal efficiency is used only with small loads and low rotational speed. The reason for using richer mixtures lies in the necessity to achieve a proper ratio of engine power and cylinder volume.

Mixture enrichment (reducing the air/fuel ratio) leads to the increase of the maximum cycle temperature, thereby increasing the maximum cycle pressure and the indicated pressure. Increasing the fuel dose does result in the increase of the rotational moment, however the percentage of fuel contained energy to be converted into gas work decreases as a result of the growing intensity of heat exchange with the walls surrounding the combustion chamber as well as the rising temperature of exhaust gases. For a certain combination of mixture components, the phenomenon of ignition misses is not important and both curves meet in point 3 (Fig. 14). Point 4 represents the maximum effective moment (rich mixture range). Thus, the maximum value of average indicated pressure is achieved for the enriched mixture. Further increase of fuel mass in the cylinder (reducing the air/fuel coefficient) causes the loses to rise in order for the fuel to be heated up and the maximum pressure in the cylinder to be reduced. Moreover, the combustion conditions worsen – the combustion intensity drops whereas the phenomenon of missed ignitions continually grows as a result of the excessively rich mixture being used. For a fuel dose indicated in point 5, the engine is still able to generate energy allowing for the internal resistance to be overcome. Further mixture enrichment leads to engine stalling.

Thus, a spark-ignition engine powered by the homogenous mixture reaches maximum rotational moment when the mixture components coefficient *A/F* ranges between 12.5–14, and maximum general efficiency (along with the minimal unitary fuel consumption) when the mixture components coefficient *A/F* fits within the range of 15.5–17. To sum up, the reasons for *Memax* and *gemin* not occurring simultaneously with these values are as follows:

- the components for *Memax* with fuel excess (rich mixture) require the air contained in the cylinder to be entirely used (complete combustion); the combustion occurs at a maximum speed; however, there is no maximum combustion efficiency (incomplete combustion);
- the components for *gemin* require a complete and total (if possible) combustion and the phenomenon of missed ignition is not yet discernible.

In a spark-ignition engine powered by the homogenous mixture, the optimal value of *A/F* coefficient ensuring the minimal unitary fuel consumption, *ge* depends on the engine load expressed, for instance, through the effective unitary pressure (*pe*). In the case of spark-ignition engines capable of producing and burning both stratified and homogenous mixtures, the unitary fuel consumption can be reduced under the conditions of low engine load, unlike in the case of the engine capable of producing and burning solely homogenous mixtures. Under the conditions of high engine load (maximum value of air mass in the cylinder), the demand for rotational moment imposes the necessity to use homogenous mixtures: slightly enriched, stoichiometric and slightly depleted. As the load is reduced (still under the conditions of maximum engine filling), the optimal general efficiency for homogenous mixture is achieved. Further depletion of homogenous mixture causes the general efficiency to drop. In such a case, it is advisable to change the way of producing the mixture from homogenous to stratified. Thanks to this procedure, provided that the cylinder is completely filled with air, reducing the engine load will result in the increase of the general efficiency – Fig. 15.

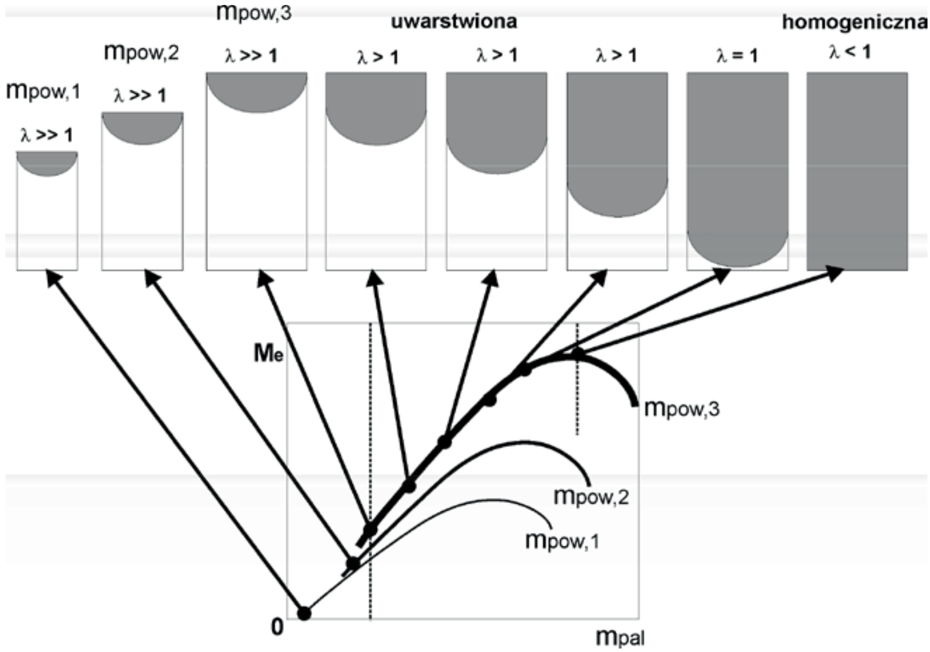


Legend:

Maksymalna sprawność – maximum efficiency
Maksymalny moment – maximum torque
Spalanie mieszanek homogenicznych – homogenous mixture combustion
Spalanie mieszanek uwartwionych – stratified mixture combustion

Fig. 15. *The way the effective moment (Me) is regulated by changing the fuel mass and the degree to which the mixture is homogenized provided that the cylinder is completely filled*

Reducing engine load (required torque) which causes constant decrease of fuel load, provided that air mass remains the same, results in reaching the maximum general efficiency for the stratified mixture. Further load decrease leads to the necessity of reducing the cylinder filling if the aim is to minimize fuel consumption. This will allow to meet the condition of the maximum general efficiency – Fig. 16.



Legend:

Uwarstwiona – stratified

Homogeniczna – homogenous

Fig. 16. *Outline depicting the process of implementing the required effective torque (Me) and meeting the condition of maximum general efficiency at the same time*

Ever since the spark-ignition engine was built, there has always been a problem with proper fuel dosage and mixture production. The first requirements towards the supply system were very liberal – after starting the engine (which, although troublesome, was possible), the only demand was the absence of "shots" (it was often the case for "shots" to occur in the carburetor when the mixture was very poor or in the tailpipe when some of the excessively rich mixture remained unburned). These requirements went quickly up and in one of the handbooks devoted to combustion engine construction from 1905, the attention was drawn to the quality of the mixture (fuel disintegration, homogeneity and relatively constant combination of fuel–air components regardless of rotational speed).

In mid 20th century, the requirements included:

- moderately complete vaporization of fuel and the mixing of fuel with air by means of a precise spray,
- fuel regulation in accordance with the engine load (fuel-efficient work in the case of low engine load and maximum power in the case of high engine load),
- dependable start-up and stable running when the engine load is low (idling).

In 1970s, the emphasis on the efficiency of fuel consumption increased and the problem concerning the influence of mixture components on the toxicity of exhaust gases was signaled.

It was not until 1980s, and especially 1990s, when the necessity to comply with the increasingly stringent toxicity norms was incurred. These applied to engine exploitation which took place for a very long period of time and concerned both the exhaust gases released from the tailpipe as well as the gases emitted by the whole supply system (leaking fuel supply system, etc.). Regardless of the achievements within the field of combustion systems using poor mixtures, the only practical solution of exhaust gases toxicity problem is their neutralization in the catalyst located in the outlet system of the engine. This entails the necessity to ensure that the value of excess air coefficient stays on the stoichiometric level with a very low tolerance of 0.5%. Large excess or deficit of air in the exhaust gases leads to a decreased catalyst reaction activity (oxidation of unburned hydrocarbons and carbon monoxide as well as reduction of nitrogen oxides).

The requirements were quickly followed by the development of fuel dosage technique. The first fuel dosing unit was a carburetor, already patented in1838 by William Bernett. In 1875, Marcuse built a brush carburetor (a rotating brush splashing the fuel into the inlet engine system), in 1880s, a surface carburetor was designed (predominantly, thanks to the work done by the motorization pioneers such as Gottlieb Daimler and Carl Benz). A diffuser type of carburetor was a turning point in meeting the first requirements, however the "elementary" carburetor had to be supplemented with additional devices. The first such devices appeared in 1905 (Krebs's regulator). In 1907, Zenith carburetor was built, and in 1910 – Solex carburetor. Until 1930s, about 2,500 patents related to a carburetor device were listed. However, never have the constructors managed to avoid the drawbacks of a carburetor. These include:

- uneven combination of fuel–air mixture components in various cylinders,
- fuel outdropping on the walls of outlet channels,
- poor characteristics of mixture components in the expenditure function of fuel flow,
- low precision of fuel dosing in the most common transient states of engine work,
- interruptions in the process of supplying the engine with fuel due to mass inertia forces resulting from braking, accelerating and driving in the mountainous terrain,

- significant sensitivity to variable weather conditions,
- commonly occurring vaporization of fuel which interrupts the fuel supply process,
- the risk of self-ignition with flame going back into the inlet pipe,
- excessive wear and tear of moving elements.

Despite the attempts at automating and adding additional systems to carburetors to improve their performance, no satisfactory improvement, adequate to the costs incurred, was achieved.

The breakthrough in developing the fuel dosage control technique came with the invention of mechatronic systems of gasoline injection in which the mass of the fuel being injected depends on the time in which the atomizer needle is lifted by the forces of electromagnet's magnetic field. The progress was made primarily thanks to the achievements of material engineering, making it possible to design a very light injector which reacts to control impulses in a fast a precise way. The control procedures, in the case of carburetors, carried out in a rather crippling (though brilliant) way by means of analogies of physical phenomena, in the case of injection system, can be stored in the microprocessor's memory and become incredibly complicated mathematical operations.

The process of fuel dosage control in gasoline injection systems involves influencing the time during which the injector's solenoid is open. To a certain extent, the opening time is in linear connection with the fuel mass (volume) being injected. One can influence both the time during which the injector is open as well as the initial stage of injection in terms of engine crankshaft location. Recent achievements in the field of constructing injectors capable of precise high-pressure injections allowed for the development of gasoline direct injection systems (GDI). High level of development was achieved by systems of gasoline injection into the inlet tract – single point injection (SPI) in the throttle valve and multi point injection (MPI) before the inlet valves. In each of the injection systems, a lambda probe is present allowing for the problems of stabilizing the mixture components to be solved by measuring the amount of oxygen in exhaust gases.

The injection control system is designed in such a way so as to provide the necessary fuel with a proper pressure in the fuel system (appropriate fuel mass and the degree of fuel disintegration). Subsequent injections are synchronized with the work process. The fuel can be injected once every process of mixture inlet (synchronically) or with a frequency varying according to the inlet and ignition frequency (asynchronous injection). The control signal (injector opening time) is an impulse with a specific length of time. Longer impulses indicate increased amount of injected fuel. The key task of the controller is to activate the fuel pump by means of a relay.

On the basis of the working conditions of the engine, the specified value of the *A/F* coefficient is chosen. Then, the algorithm estimates the mass of air being sucked on the basis of values indicated by various sets of measuring sensors as well as the model of processes taking place in the inlet collector. The time of the injection is adjusted depending on the voltages of the electricity network.

The fuel sucked from the tank is moved by an electric fuel pump to the fuel filter and then to the fuel injection unit. Fuel pressure regulator, connected to the injection unit, is responsible for maintaining a constant pressure in the fuel feed system. The excess of fuel is drained from the pressure regulator through the return pipe into the fuel tank. Depending on the type of the machine, the fuel tank might be equipped with preliminary filter. Between the fuel tank and the fuel pump or nearby the pump on the pressure side, there might be also a flow pulsation throttle installed. The fuel sucked through the pipe flows through the upper chamber of the throttle and, on the pressure side, it is moved through its bottom chamber. A classic outline of the feed system is presented in Fig. 17.

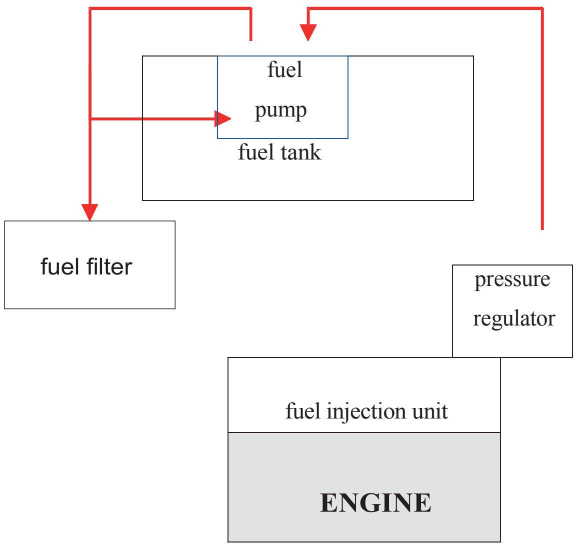
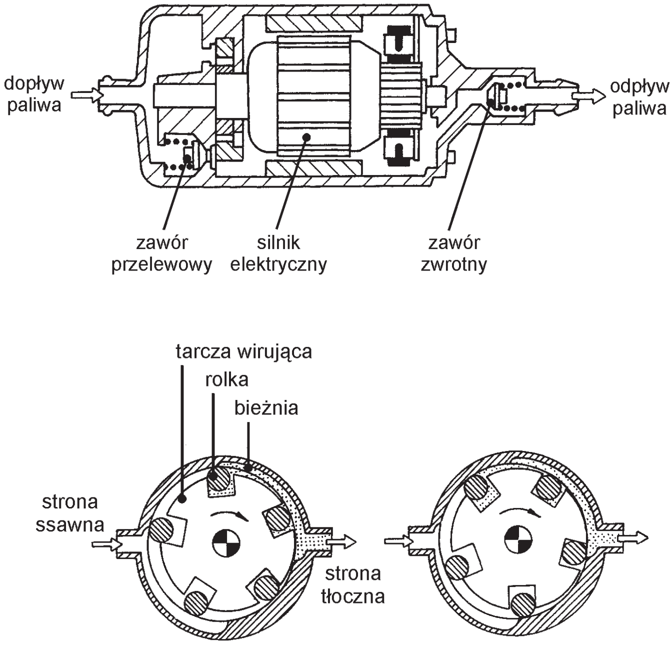


Fig. 17. *Outline of a feed system of engines with gasoline injection*

The fuel tank is responsible for delivering the fuel to the feed system, maintaining the proper fuel pressure which is at least equal to the pressure required in the system (pressure surplus is recommended for the pressure moved by the pump with respect to the pressure fixed by the regulator to ensure that the system is entirely filled) and maintaining the proper expenditure (higher than the maximum fuel consumption of the engine). There are two types of pumps used in low-pressure injection systems: rotary pump and impeller pump.

In a rotary pump (Fig. 18), the rotating driving plate is powered directly by the electric engine shaft of a constant excitation with a rotor being washed by the fuel. On the plate circumference, there are indentations where the working rollers are located. When the pump is working, the rollers are pressed against the axially fitted roller conveyor with the inertial force. The fuel located in the space between neighboring rollers is compressed and leaves the pump through the outlet flange at an elevated pressure.



Legend:

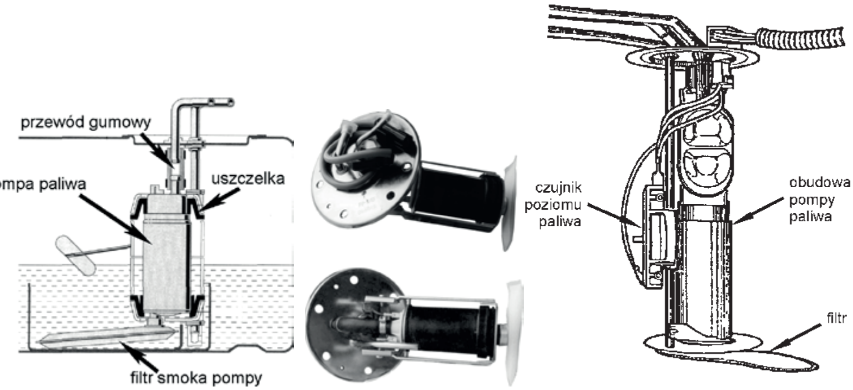
Dopływ paliwa – fuel inlet
Zawór przelewowy – overflow valve
Silnik elektryczny – electric engine
Zawór zwrotny – non-return valve
Odpływ paliwa – fuel outlet
Strona ssawna – inlet side
Strona tłoczna – pressure side

Fig. 18. *Example of a rotary pump and a driving plate*

The overflow valve located inside the pump prevents the pump pressure from rising too much in case of rising counter pressure on the pressure side. In such a case, the valve delivers the fuel to the inlet side creating a small fuel circulation inside the pump. The non-return valve on the pressure side is responsible for maintaining the residual pressure after the engine is shut down. This prevents the formation of fuel vapor bubbles.

The impeller pump is situated inside the fuel tank in a respective inlet rose (Fig. 19) where the fuel level sensors are and on the inlet side where the screen fuel filter is located. For the pump to work properly, it is necessary to immerse it entirely in the fuel being pumped. The fuel flowing through the pump cools the pump and cleans the brushes and the collector.

The pump is equipped with a relief valve which connects the pressure chamber with the inlet chamber when the pressure in the pressure chamber exceeds the permissible value. This protects the electric engine from overheating. The non-return valve situated in the pressure chamber prevents the feed system from being completely emptied when the pump is not working. The nominal pump capacity depends on the speed of the rotor which, in turn, depends on the voltage.



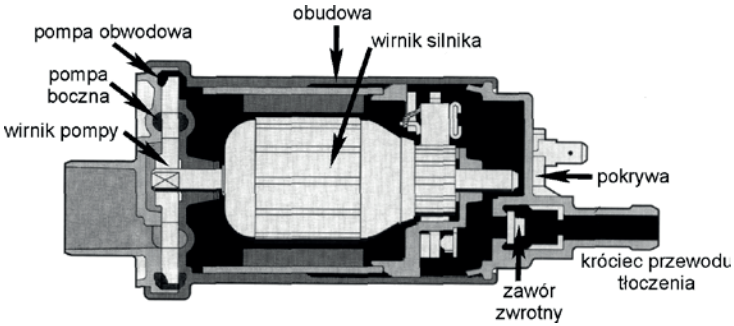
Legend:

Przewód gumowy – rubber pipe
Pompa paliwa – fuel pump
Uszczelka – gasket
Filtr smoka pompy – inlet rose filter
Czujnik poziomu paliwa – fuel level sensor
Obudowa pompy paliwa – fuel pump housing
Filtr – filter

Fig. 19. *Outline depicting the location of the impeller pump*

The impeller pump consists of two parts: electric part and mechanical part. The electric part includes a direct-current motor which powers the mechanical pump which supplies the fuel. To power the pumps, the electric motors with permanent magnets driven excitation and fuel washed rotors are used. This eliminates the risk of explosion because the conditions inside the pump do not allow for the formation of explosive mixture of air and fuel. Such a solution allowed to simplify the design and significantly reduce the thermal load and the power consumed by the pump. Using proper materials for the construction of the commutator and the brush, both highly sensitive to wear and tear, ensured a high level of reliability and durability of pumps.

The impeller pump is usually a two-stage pump, two stages of which are side-impeller pumps combined in one rotor – Fig. 20. The blade rim of the pump is surrounded by a side channel located in the cover housing on the inlet side and an analogical channel located in the cover on the inside of the pump. During the rotational movement of the rotor, the fuel in the blade rim gains kinetic energy which is then transformed into pressure energy in the side channels adjacent to the rotating rim.



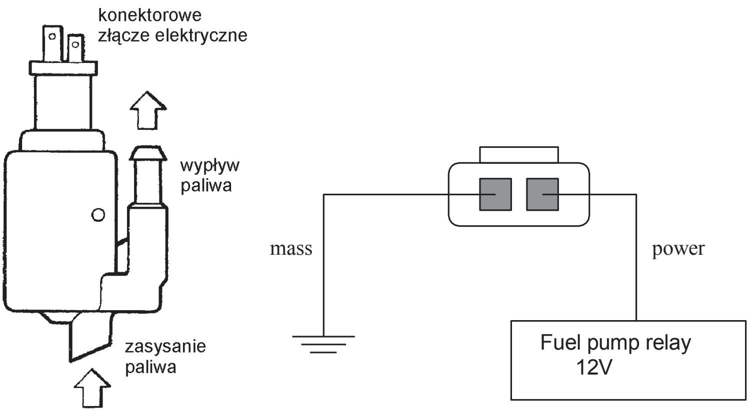
Legend:

Pompa obwodowa – peripheral pump
Pompa boczna – side pump
Wirnik pompy – pump rotor
Obudowa – housing
Wirnik silnika – engine rotor
Pokrywa – cover
Zawór zwrotny – non-return valve
Króciec przewodu tłoczenia – pressure flange

Fig. 20. *Outline of a two-stage impeller pump*

Low pressure pumps are supplied with electric power through a relay – Fig. 21. After the ignition is switched on, the pump is working for a couple of seconds and if the attempt to start the engine does not happen during that time, the ignition is switched off.

Submersible low-pressure impeller pump is also used for powering the high-pressure gasoline injection system. In this system, the pump is used to deliver the proper amount of fuel to a high-pressure pump. The electric power (controlled by the impeller pump) corresponds to that of low-pressure injection system.

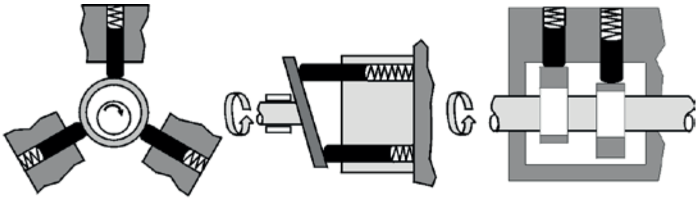


Legend:
Konektorowe złącze elektryczne – electric connector joint
Wypływ paliwa – fuel outflow
Zasysanie paliwa – fuel inlet

Fig. 21. *Outline of electric power in a fuel pump (PL 012)*

High-pressure pump is responsible for generating high pressure (amounting to tens of MPa) in the injection system. Because of that, the technological solutions which are sufficient in the case of low-pressure pumps (compressing fuel between the rollers, providing fuel with kinetic energy in the blade rim) have to be replaced with solutions basing on piston fuel pumping in the case of high-pressure pumps. Depending on the location of pump plunger axis against the driving shaft axis, there might be three solutions for high-pressure piston pumps: radial-flow pump, axial-flow pump and series-flow pump – Fig. 22.

Figure 22 shows that a radial-flow pump is the best solution. And indeed, this type of pump is most commonly used in high-pressure feed systems – Fig. 23 and Fig. 24. High-pressure pump is powered by the camshaft (flange connection on the side where the engine power consumption occurs).

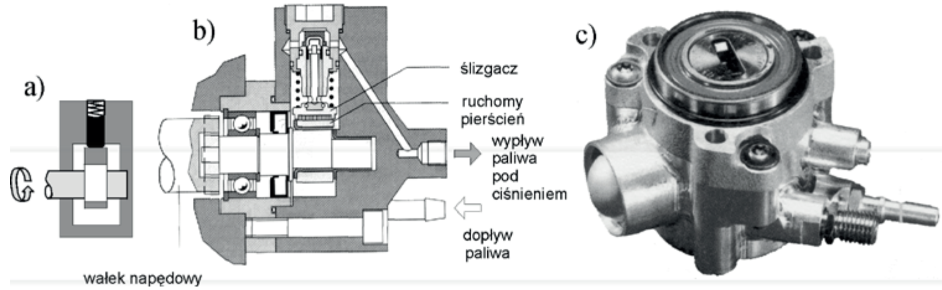


	promieniowa	osiowa	szeregowa
Żywotność	+	+/–	+/–
Skuteczność	+	+/–	+/–
Zwartość	+	+/–	–
Koszt	+	+	+/–

Legend:

Promieniowa – radial
Osiowa – axial
Szeregowa – in-line
Żywotność – durability
Skuteczność – efficiency
Zwartość- compactness
Koszt – cost

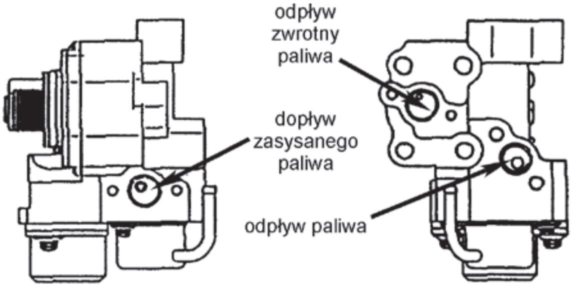
Fig. 22. *Types of high-pressure pumps and the assessment of their qualities*



Legend:

Ślizgacz – shoe
Ruchomy pierścień – movable ring
Wypływ paliwa pod ciśnieniem – fuel outflow under pressure
Dopływ paliwa – fuel inflow
Wałek napędowy – driving shaft

Fig. 23. *Outline (a), design (b) and the way the radial-flow pump looks (c) in a gasoline injection system*

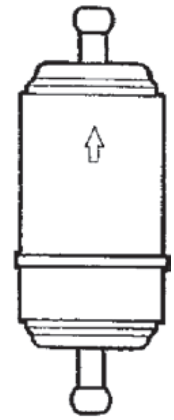


Legend:

Odpływ zwrotny paliwa – fuel return pipe
Dopływ zasysanego paliwa – fuel inlet
Odpływ paliwa – fuel outlet

Figure 24. *A structure of the radial pump of the fuel injection system.*

Fuel filter is placed in the fuel system on the pump (line) and serves the function of eliminating the possible contamination from the fuel pomp. The filter consists of metal cylindrical cartridge inside which there is a porous filter paper and woven sieve that are used to capture possible scraps of paper. Filtering elements are installed inside, fastened to the metal walls with a screw. At the one end of the filter a fuel inlet is connected, on the other – fuel outlet.



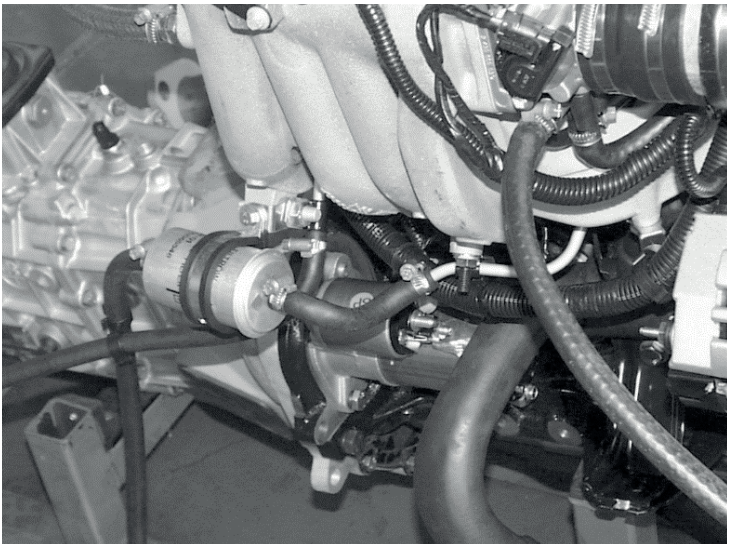
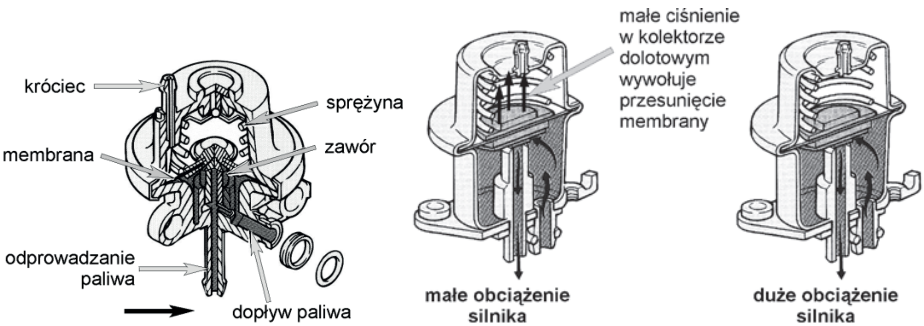


Figure 25. *How a fuel filter looks like and how it is installed in the engine.*

To insure correct installation, an arrow is situated on the body of the filter, indicating the direction of the fuel flow – Figure 25.

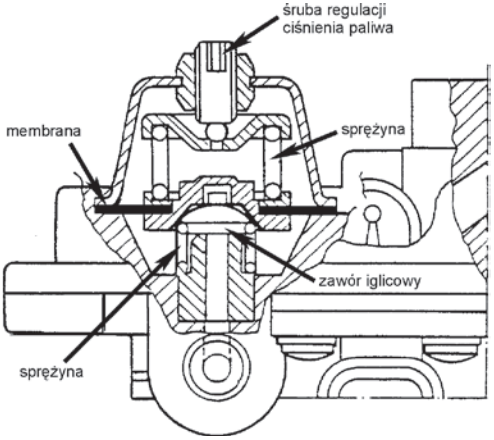
Fuel pressure regulator installed at the end of the injection unit regulates the pressure in the fuel system. It consist of a metal casing, divided into two chambers with a diaphragm. A compression spring pushes the diaphragm to the valve head. If the fuel pressure exceeds a certain value, the valve opens in order to allow the excess flow in to the tank. An elastic pipe connects the inlet manifold (pressure in the inlet circuit) with the spring chamber. If the pressure exceeds a certain value, a diaphragm bows and opens the regulators valve, which allows for fuel to return to the tank. This way the system maintains the constant pressure difference between the inlet manifold absolute pressure and the pressure in the feed circuit. It allows for fuel dosing completely independent from the momentary absolute pressure in the inlet manifold – Figure 26. Also in case of the lack of injection, the fuel moved by the pump is directed back into the tank.



Legend:

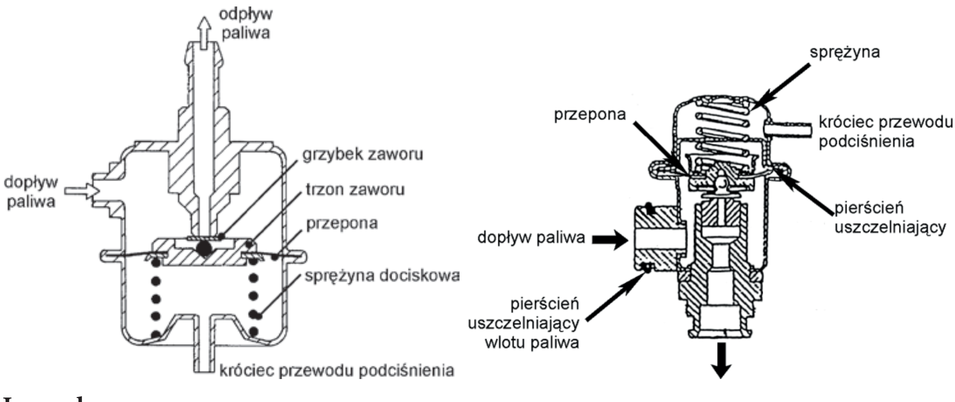
Króciec – nipple
Membrana – diaphragm
Sprężyna – spring
Zawór – valve
Odprowadzenie paliwa – fuel outlet
Odpływ paliwa – fuel outlet
Małe ciśnienie w kolektorze dolotowym powoduje przesunięcie membrany – low pressure in the intake manifold causes diaphragm to bow
Małe obciążenie silnika – low engine load
Duże obciążenie silnika – high engine load.

Figure 26. *Fuel pressure regulator – the structure and principles of operation.*



Legend:
Śruba regualcji ciśnienia paliwa – regulator's screw
Membrana – diaphragm
Sprężyna – spring
Zawór iglicowy – needle valve.

Figure 27. *The structure of the fuel pressure regulator in the in SPI injection system.*



Legend:
Odpływ paliwa – fuel outlet
Dopływ paliwa – fuel inlet
Grzybek zaworu – valve head
Trzon zaworu – valve stem
Sprężyna dociskowa – compression spring
Króciec przewodu podciśnienia – vacuum pipe nipple
Pierścień uszczelniający – gasket ring
Pierścień uszczelniający wlotu paliwa – fuel inlet gasket

Figure 28. *Examples of the structure of the fuel pressure regulator in the MPI injection systems.*

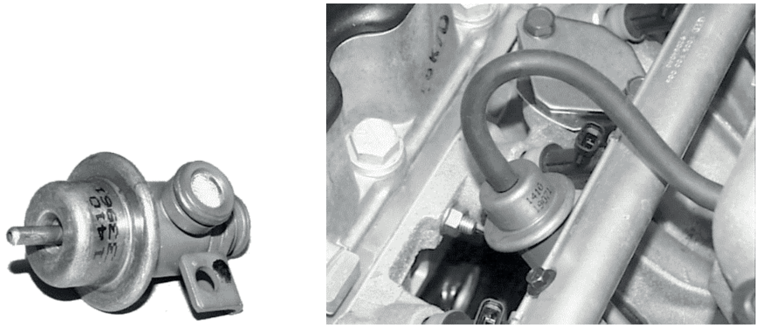
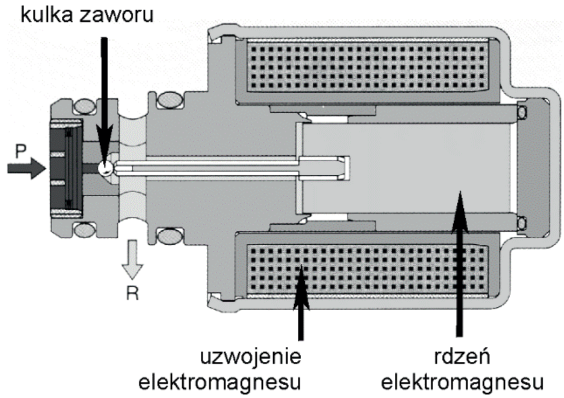


Figure 29. *The pressure regulator and the way it is installed in the engine.*

The pressure regulator might be integrated with the hydraulic part of the injection system in single point injection systems (Figure 27.) or be an element installed (added) to the injector rail in the multi-point injection systems (Figures 28. and 29.). With the moment the engine stops working, the fuel supply also stops. Return valve of the pump and pressure regulator valve close, maintaining the pressure in the system for a certain amount of time. It prevents from fuel stem bubbles that are created as a result of heating up the fuel pipe by the heat radiated by the engine. As a result, for a long time after stopping the engine there are conditions for its correct re-start.



Legend:

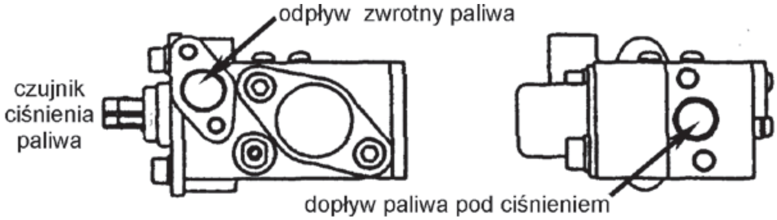
Kulka zaworu – The ball of the valve
Uzwojenie elektromagnesu – The winding of the electromagnet
Rdzeń elektromagnesu – The core of the electromagnet

Figure 30. *A scheme of an electromagnetic fuel pressure regulator.*

Precise regulation is not possible with a system using a diaphragm and spring. Mechanical fuel pressure regulator is replaced by the electromagnetic valve. Electromagnet via the mandrel presses the ball of the ball valve into its socket. The force of the thrust balances with the pressure force of the fuel. The ball takes over the function of the diaphragm and the electromagnet – the function of the pressure spring.

Because of the high values of the pressure in the high pressure fuel injection systems, the pressure oscillates during following injections and these changes are induced even by temperature.

Both reasons (precise pressure regulation and the correction of the changes of fuel features) enforce the necessity to use fuel pressure sensor as an element supporting the suitable precision of the injection – Figure 31.

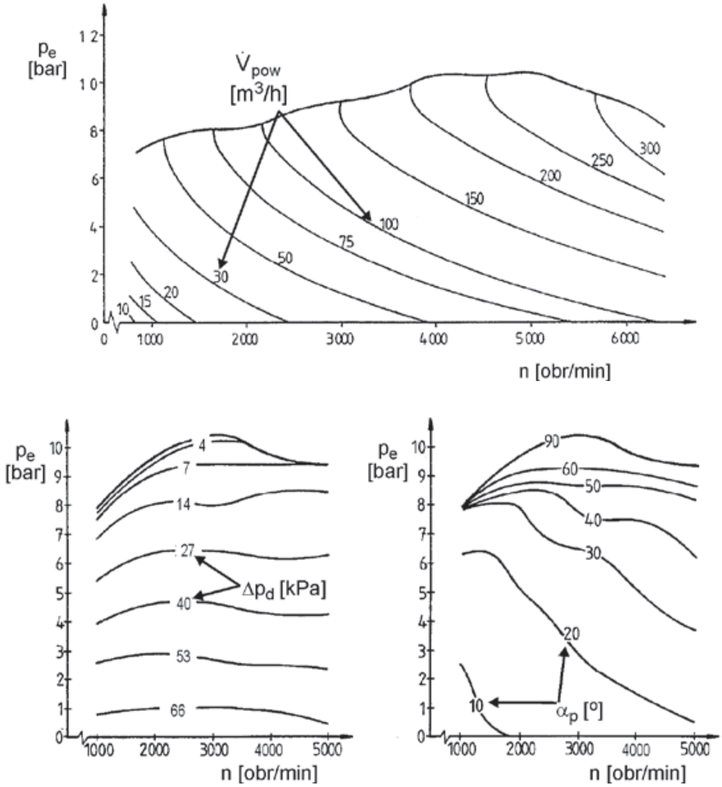


Legend:
Czujnik ciśnienia paliwa – Fuel pressure sesnor
Odpływ zwrotny paliwa – Return fuel pipe
Dopływ paliwa pod ciśnieniem – Inlet of the fuel under pressure.

Figure 31. *Fuel pressure regulator from the fuel injection system.*

Regulation of the force in the petrol engines supplied by the homogenic mixtures happens via changes of the mixture mass. The operator, changing the position of the throttle influences the vacuum in the inlet unit and thus influences the capacity of the air supplying the engine – Figure 32.

The main task – optimization of the supply system – … on the choice of the appropriate dose of the fuel in relation to the air supplying the engine, which means the choice of the appropriate mixture adjusted to the engine work. At the same time several optimization aims must be reconciled. Achieving the minimal emissions often stands in direct opposition to the minimization of the fuel consumption or maximization of the rotational speed – Figures 33 and 34.



Legend:

Obr/min – rpm.

Figure 32. *The Characteristics of the quantitative power regulation for homogenic mixture.*

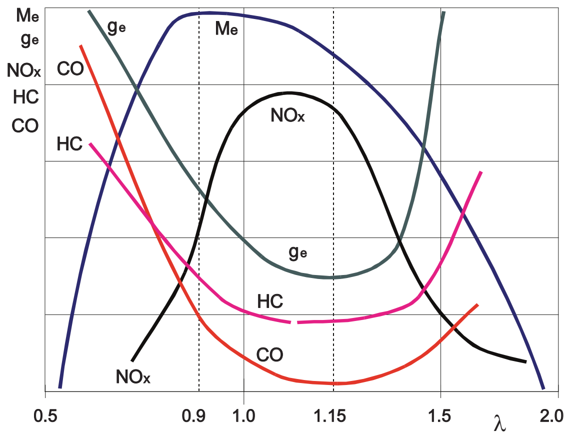
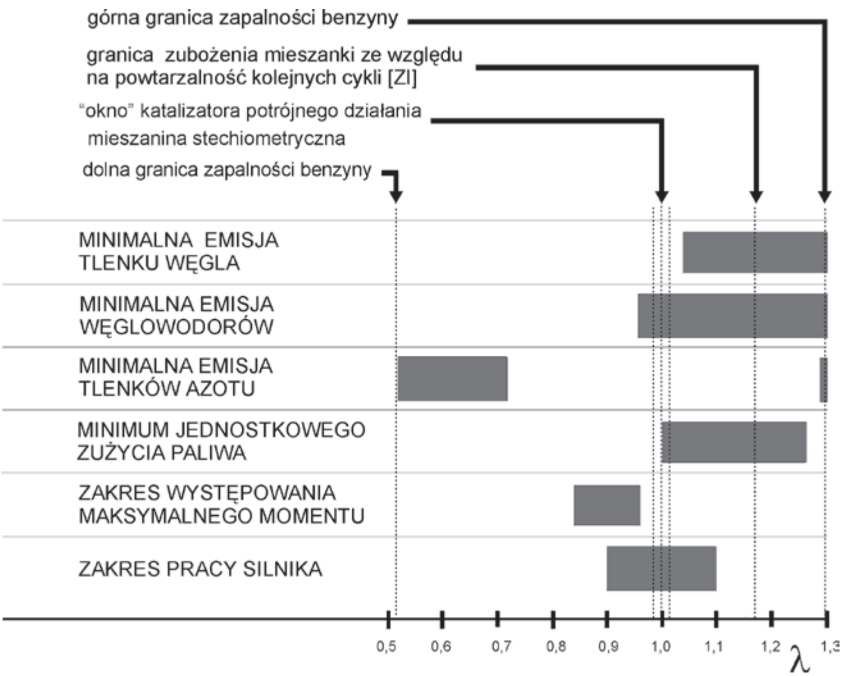


Figure 33. *Regulational characteristic of the homogenic mixture composition.*



Legend:

Górna granica zapalności benzyny – Top limit of the fuel flammability

Granica zubożenia mieszanki ze względu na powtarzalność kolejnych cykli [ZI] – The limit of the emasculation of the mixture because of the similarity of the subsequent cycles

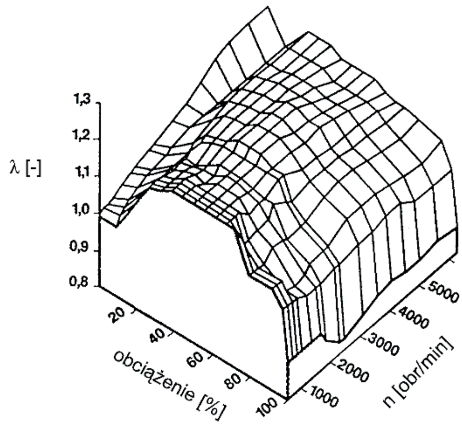
"Okno" katalizatora potrójnego działania – The "window" of the three-way catalyst operation

Mieszanina stechiometryczna – stoichiometric mixture

Dolna granica zapalności benzyny – Bottom limit of the fuel flammability.

Figure 34. *The ranges of the optimal values for the homogenic mixture composition.*

The final characteristic of the mixture composition in the function of the load and rotational speed emerges as a complicated, three dimensional surface – Figure 35. Introducing the emission norms and three-way catalyst enforced the usage of the stoichiometric mixtures in as many conditions of the engine work as possible. Regulation characteristic this way has been greatly simplified. The scheme of the realization of demanded effective moment had to be linked to the necessity of using the stoichiometric mixture, to the disadvantage of meeting the condition of the maximal general efficiency.

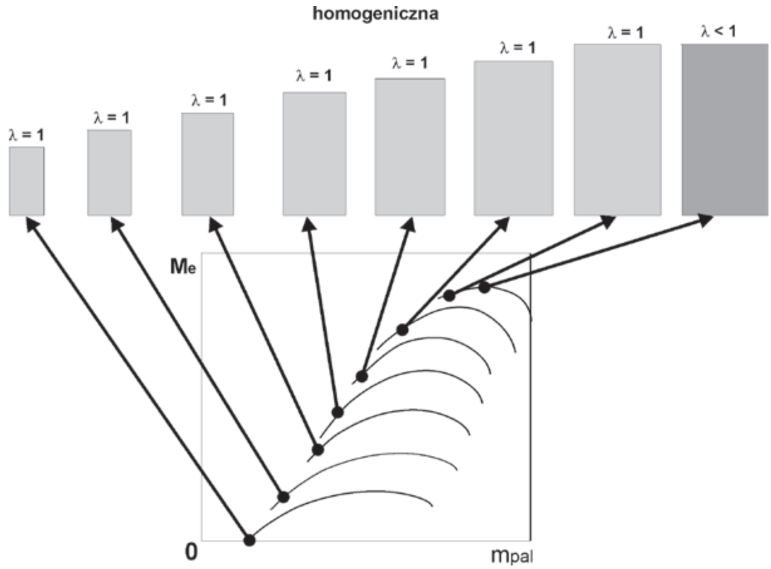


Legend:

Obciążenie – load, obr/min – rpm

Figure 35. *The characteristics of the homogenic mixture composition.*

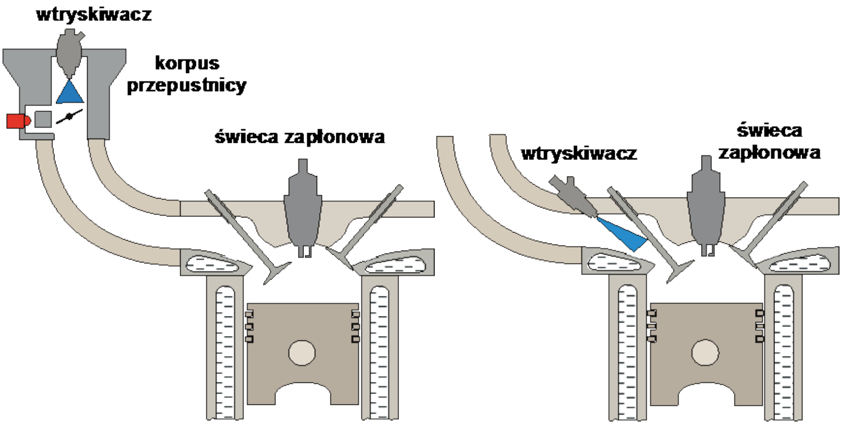
The scheme of the realization of the demanded effective moment is presented in Figure 36. To inject a dose of a fuel a single point injection or multi point injection can be used – Figure 37.



Legend:

Homogeniczna – Homogenic.

Figure 36. *A scheme of the realization of the demanded effective moment in engines supplied with homogenic mixture.*



Legend:
Wtryskiwacz – Injector
Korpus przepustnicy – Throttle body
Świeca zapłonowa – Spark plug.

Figure 37. *Scheme of the single point injection and multi point injection*

Single point injection units are equipped with one electronically controlled injector. It is installed in central position in relation to the cylinders, directly above the throttle. The unit remains in the same casing as the throttle. The injector supplies the fuel through the throttle, in order with the sequence of intake strokes of the individual cylinders. The beginning of the injection has practically no influence on the engine work. The basic drawback of this solution is the uneven mixture division into the cylinders. Because of the relatively big distance between the injector and intake valve, the mass of the fuel intended for any given cylinder might be increased or reduced (in favour of another cylinder).

Multi point injection units are equipped with injectors – installed inside the intake tubes – equal to the number of cylinders. The injectors are installed in the intake manifolds, nearby intake valves – Figure 38. Main advantages of multipoint injection units are:

- increasing the factor of filling caused by the smaller resistance to the mixture flow in intake tubes,
- increasing the average effective pressure because of the more balances delivery of the mixture into individual cylinders,
- increasing the ability to accelerate in the whole range of the rotational speed and load due to the faster unit reaction to the new work conditions.

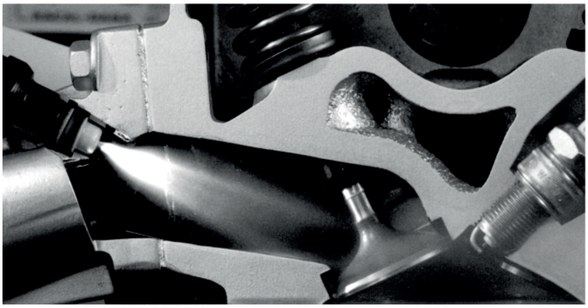


Figure 38. *The fuel injection through the intake valve in MPI units.*

2.3. Spark ignition

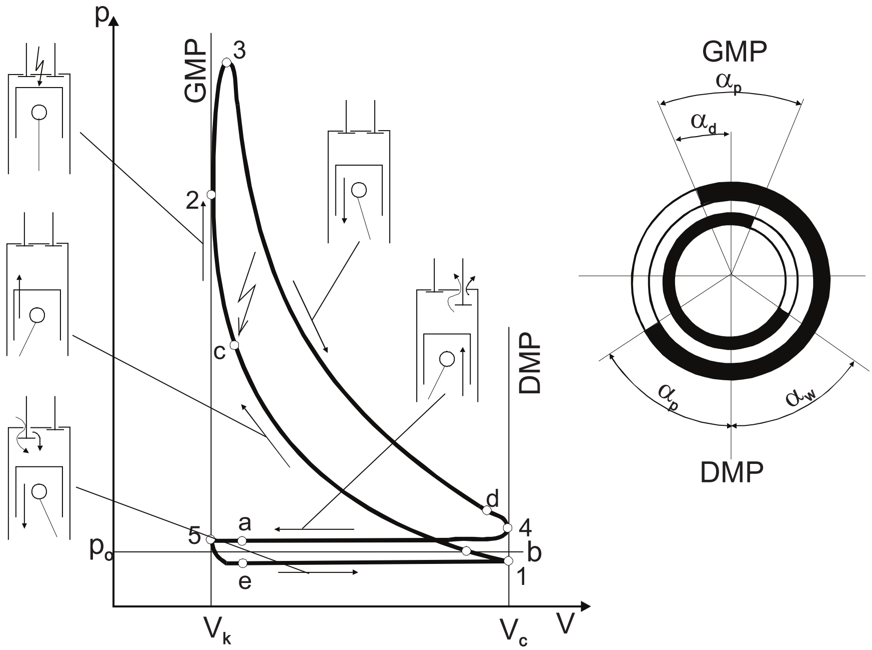
The history of the gasoline car engine from the beginning is closely related to the search for the effective, infallible method of igniting the fuel mix in the cylinder. As early as in 1801, Philippe d'Humbersin indicated the necessity to use the electric spark as the *energy* impulse that initiates the combustion in the petrol engine. In his times the realization of this idea in practice (was met with) the number of serious technical difficulties. This is why other methods, although worse from the safety and efficiency point of view, were used to ignite the fuel mix , e.g. ignition from an hot tube. In 1860 Etienne Lenoir built a gas engine with 2 kW output, in which he used a high-voltage output to ignite the illuminating gas. A breakthrough happened thanks to a precision mechanic Robert Bosch from Stuttgart who – after many years of research – constructed an ignition magneto. In 1886 in Benz and Daimler's cars ignition systems were installed that consisted of a galvanic battery, a spark inductor and a spark plug. The spark ignition has been invariably used in petrol engines for almost a hundred years. In favour of its use come the advantages like the simple principles of work and the ability to produce enough ignition energy with comparatively small requirements for a power source.

The energy that is required for a spark ignition depends primarily on the fuel and air mixture parameters: composition, pressure, temperature, the speed of the flow and the level of (turbulence). Especially vital are the composition of the mixture and the type of the fuel. The lowest required energy for an ignition is required for the mixture with a composition close to stoichiometric. Both the drop in the pressure and in temperature causes the need for the energy increase. Supplying the adequate energy is hindered by the technical capabilities of the ignition system. It leads to a conclusion that operating the ignition should lead to the rationalization of the time needed to charge the primary winding of the ignition coil.

Spark ignition system has undergone some changes throughout the years, but especially in the last twenty years. Mainly, these changes included the gradual elimination of mechanic elements and introduction of the electronic components in their place. In the 60s of the 20th century (Ford 1963 i Ford, General Motors 1964, Bosch 1969) stock transistorized ignition circuits appeared. Removing mechanic circuit breakers increased the durability and the infallibility of the whole system. What was a drawback was the voltage of the pulses of the inductive sensor which depended on the rotational speed of the magneto. Only the Hall effect sensor allowed to increase the spark energy and maintain the constant voltage regardless of the rotational speed of the engine. It also caused the significant miniaturisation of the ignition mechanism.

The real engine cycle differs from the theoretical Otto cycle in the following ways:

- The heat is created in the internal combustion and stays in the system through rather significant portion of the working cycle,
- The heat leaves the system in the form of exhaust. It means that in the four-stroke engines the full thermodynamic cycle equals two full rotations of the crankshaft. The first rotation causes the compression and power strokes, the second – is responsible for the exhaust and intake strokes,
- The gas used in the cycle is a real gas that exchanges the heat with the walls of the combustion chamber during the whole cycle.



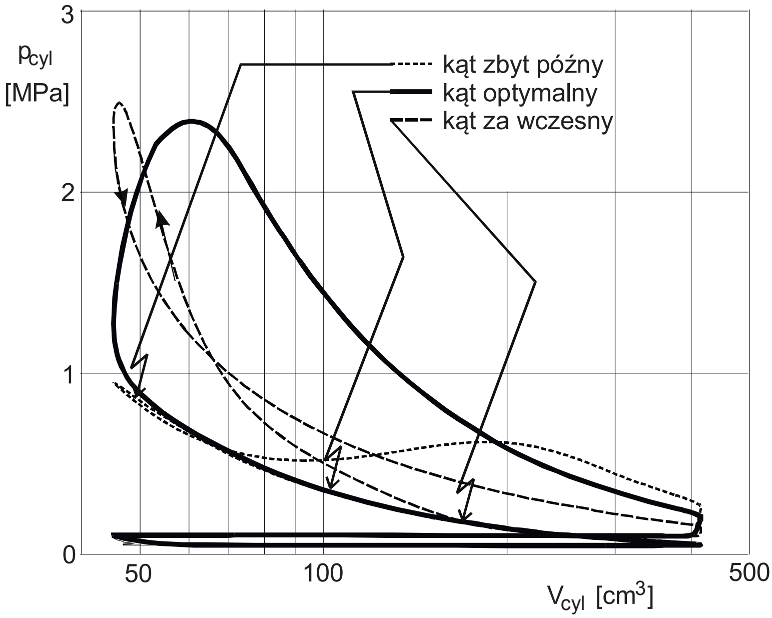
Legend:

GMP – TDC

DMP – BDC

Figure 39. *A scheme of the four stroke engine SI (spark ignition) work.*

Figure 39 shows a scheme explaining the principles of operation of the four-stroke cycle petrol engine. In point c the mixture is ignited, the burning begins. Because of the quite long time of the burning, there exists the optimal angle of spark advance, which results in the maximal work performed by the system. Too early ignition creates a loop of a negative work field on the closed pressure volume diagram. Too late ignition leads to the decrease in pressure through the whole process in the time of burning. Optimal angle of spark advance changes with the parameters of the engine work and its exploitation. The main task of the ignition system is controlling the angle of the spark advance.



Legend:

Kąt zbyt późny – Angle too advanced
Kąt optymalny – optimal angle
Kąt za wczesny – angle too delayed

Figure 40. *The influence of the spark advance angle on the closed pressure volume diagram.*

From the point of view of controlling the ignition, the transistor circuits still possess the faults inherited from the classical ignition system with mechanic regulators of spark advance angle – vacuum advance and centrifugal advance. The rising demands towards limiting the toxicity of the exhaust gases and decreasing the fuel intake forced in turn the introduction of the microprocessor technology also into the controlling the spark advance angle. Ignition moment influences in significant way the composition of the exhaust. Moreover, the too early ignition might be the cause of the phenomenon called 'knocking'.

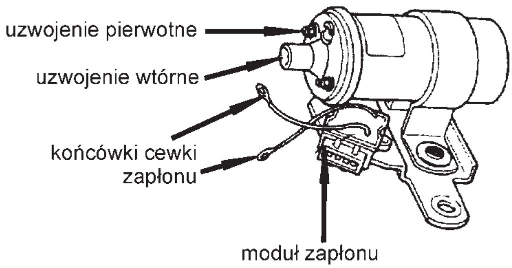
Microprocessor control technology provides better opportunities for precise regulation of spark advance angle through the whole range of the rotational speed of the engine and the engine load, while simultaneously taking into account a number of external factors. This technique also allows for the precise regulation of the dwell angle. In the systems of this kind the sophisticated controlling algorithms are used to control ignition. In 1977 first series microprocessor ignition system was installed in the Oldsmobile Tornado car produced by the General Motors.

The development of microprocessors technologies enabled individualized regulation of the angle of spark advance. Introducing the regulation of spark advance angle at borderline knock level made ignition systems adaptable, that means they are capable of adapting the work of the object to the condition in which it operates. Calling a system adaptable is in accordance with the definition of the notion if the system is equipped in the means that allow for the object identification and the constant change in the parameters caused by the installed indicator of the process quality. Building a system like that permits to take into account the differences between the copies of the same engine model and the changes of the characteristics of an engine caused by the exploitation.

The modern ignition system consists of microprocessor control unit and the group of individual ignition coils integrated with ignition plugs. Contemporary electronic ignition systems are completely maintenance-free, infallible and cheap.

The possibility to electronically regulate the spark advance undermined the point of using mechanical regulators. It has been possible for many years now to measure the parameters of the engine work by precise suitable electronic sensors (described in books dedicated to the intake and injection regulation written by the author). New parameters – not present in the mechanical ignition regulators (the mixture composition and the temperature of the coolant, to name just a few) – can be added to these already existing. The task of controlling spark advance angle and ignition coil dwell time is taken over by the electronic control unit. Systems like these can also take into account the signals of knocking combustion.

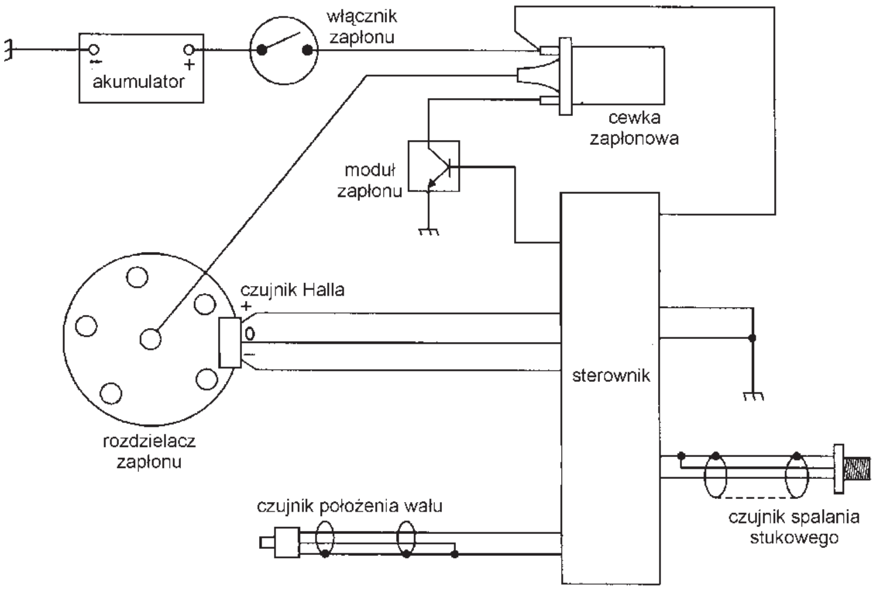
In the output (ignition amplifier module) of the ignition system the electronic signal is enhanced. Having taken into account that the time of the electronic current flow through the primary coil is specified by the microcomputer depending on the automotive battery voltage and the rotational speed, it was possible to delete conventional initial resistors without losing the energy of the spark ignition. Mentioned output also limits the maximal current flowing through the primary ignition coils. The output might be the Darlington transistor – figures 41. and 42.) and it might possess the features of the ignition module cooperating with Hall sensor or with the control unit.



Legend:

Uzwojenie pierwotne – Primary winding
Uzwojenie wtórne – Secondary winding
Końcówki cewki zapłonu – The ends of the ignition coil
Moduł zapłonu – Ignition module

Fig. 41. *Typical structure of the ignition amplifier module – final transistor.*

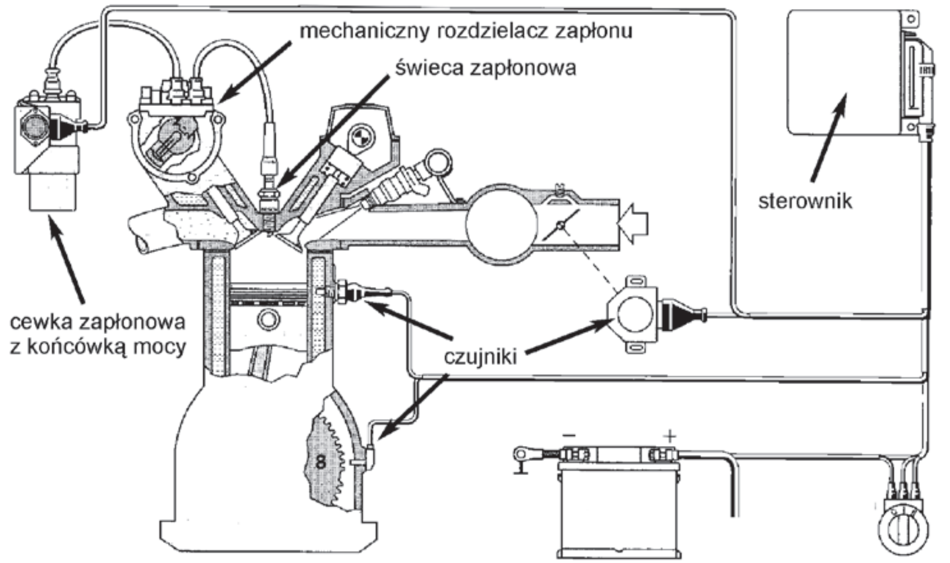


Legend:

Akumulator – Automotive battery
Włącznik zapłonu – Contactor
Cewka zapłonowa – Ignition coil
Moduł zapłonu – Ignition module
Czujnik Halla – Hall sensor
Rozdzielacz zapłonu – Ignition distributor
Sterownik – Control unit
Czujnik położenia wału – Crankshaft position sensor
Czujnik spalania stukowego – Knock sensor

Fig. 42. *A scheme of the ignition module connection – final transistor.*

The role of the ignition module output narrows down to the commutation of the current in the primary winding of the ignition coil. Figure 43. – ignition subsystem Motoronic – is presented as an example of the electronic ignition system with mechanic distributor and the output of the ignition coil.



Legend:

Mechaniczny rozdzielacz zapłonu – Mechanical ignition distributor
Świeca zapłonowa – spark plug
Cewka zapłonowa z końcówką mocy – ignition coil with power transistor unit
Sterownik – controller
Czujniki – sensors

Fig. 43. *A scheme of the electronic system controlling the ignition with the mechanical distributor in the Motronic system.*

Additional advantages of the electronic ignition system are following:

- accelerated start-up of the engine, enhanced engine work stability on idle and decreased fuel consumption,
- extended range of the control to various states of engine work (a strategy).

The ignition distributor (with caps and rotor arms) in this type of structure and principle of ignition system operation becomes just an anachronism. The next generation of the ignition systems – microprocessor controlled systems – for the task of distributing the ignition uses either high-voltage electronic distributors or low-voltage ones with individual coils designated for each spark plug.

In the distributorless systems the last mechanical element – high-voltage distributor – is replaced. It creates a number of advantages, among which the most important are:

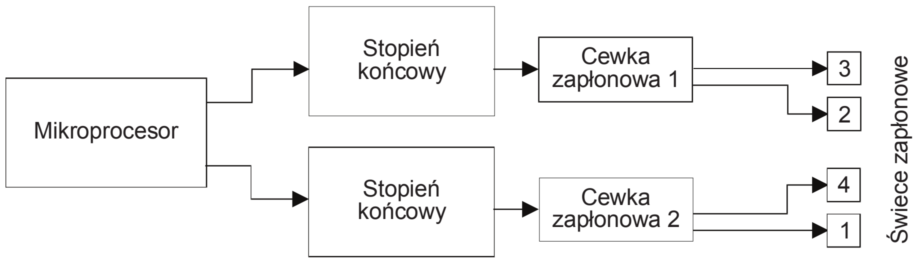
- eliminating internal discharge, due to which the level of electromagnetic disturbances is decreased,
- the lack of rotating elements, which means increasing the sustainability of the ignition system,
- the lower number of connection in the high voltage circuit
- decreasing noise levels,
- increasing the construction possibilities for the engine producers,
- the possibility of sequential ignition control (DI – Direct Ignition systems)..

Apart from the mentioned features, the system serves the same function as ignition distribution controlled electronically and has similar characteristics of power. These systems can come in two main types:

- with bipolar coils
- with individual coils for each cylinder (DI systems)

It is also possible to find systems with different coil configurations, for example with single quadripolar coil.

In relation to engines with even number of cylinders, in which one pair of pistons simultaneously reaches TDC (Top Dead Centre), elimination the distributor is possible with the use of spark plugs connected in series in each pair of cylinders, eg. 1–4 and 3–2 (see Figure 44.). In this arrangement the ends of the secondary winding are isolated from a mass potential and connected to spark plugs. The energy of the high-voltage impulse is distributed to the both cylinders: in one of them it causes the ignition of the mixture in the compression stroke and in the other it the spark happened during the exhaust stroke.



Legend:
Mikroprocesor – Microprocessor
Stopień końcowy – Power transistor
Cewka zapłonowa – Ignition coil
Świece zapłonowe – Spark plugs

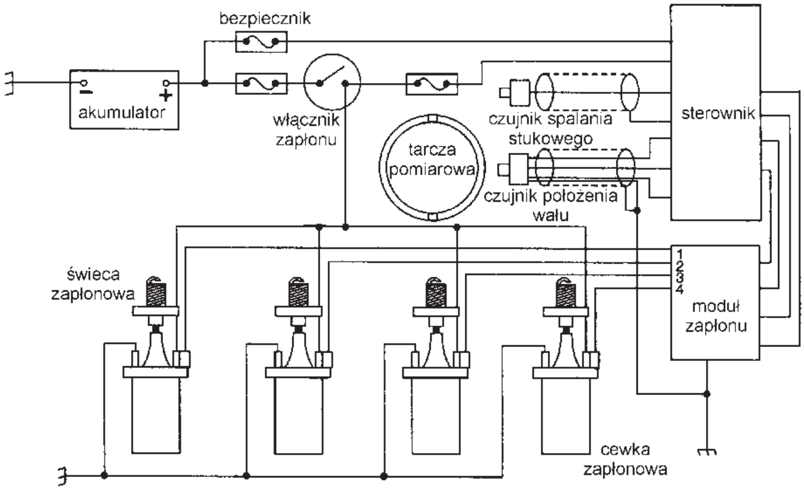
Figure 44. *The idea of the distributorless ignition system: a system with two bipolar coils.*

With using bipolar coils the demand regarding the value of high-voltage generated by the ignition system increases. The reason for that are different polarizations of the electrodes on the plugs. The doubled frequency of the spark discharge causes also faster deterioration of the electrodes and the damage during one of the final phases switches off two cylinders.

In electronic ignition systems of this type for two cylinders there is one bipolar coil. According to this rule every four-cylinder engine is equipped with two bipolar coils, six-cylinder engine – with 3 coils, and so on. In four-cylinder engine the operation of the second bipolar coil has to be shifted by 180 degrees of the CAD (Crank Angle Degree) in relation to the first one. It means that in the engine with four cylinders the spark is released at the same time in the pairs of the cylinders 1-4 and 3-2. The synchronisation of the individual pairs of cylinders happens in relation to the crankshaft position sensor, corresponding to the TDC position of the pistons in cylinders 1 and 4. The information about the TDC positon of the pistons in the pair 3-2 is the result of the analysis of the rotational speed of the crankshaft.

Older solutions regarding the system of controlling ignition with bipolar coils were limited by the speed at which microprocessor drive could work. The analysis of the signal from the crankshaft rotation sensor with the aim to designate next TDC requires suitably high computing power. The problem was solved by transferring the task of processing the signals of the crankshaft sensor to the external system – DIS (Distributorless Ignition System).

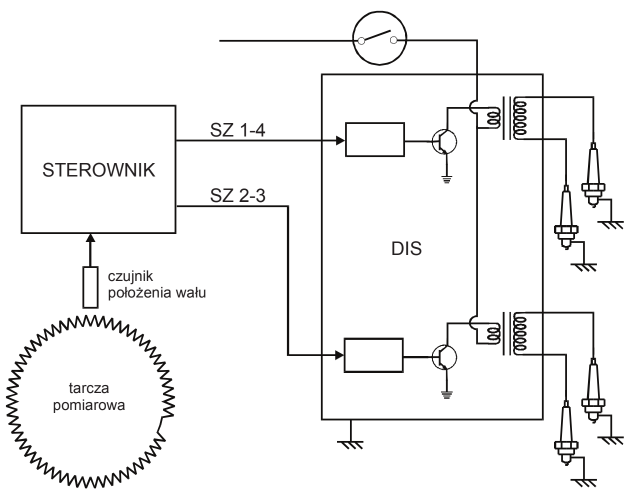
The solutions of the ignition control systems (in relation to the bipolar coils) transfer the function of the processing the crankshaft sensor signal and of designating dwell time to the central computer – Figures 45. and 46.



Legend:

Akumulator – Automotive battery
Bezpiecznik – Fuse
Włącznik zapłonu – Ignition switch
Tarcza pomiarowa – Measurement disk
Czujnik spalania stukowego – Knock sensor
Sterownik – Control unit
Czujnik położenia wału – Crankshaft position sensor
Moduł zapłonu – Ignition module
Świeca zapłonowa – Spark plug
Cewka zapłonowa – ignition coil

Figure 45. *A scheme of the connections in the distributorless ignition system.*



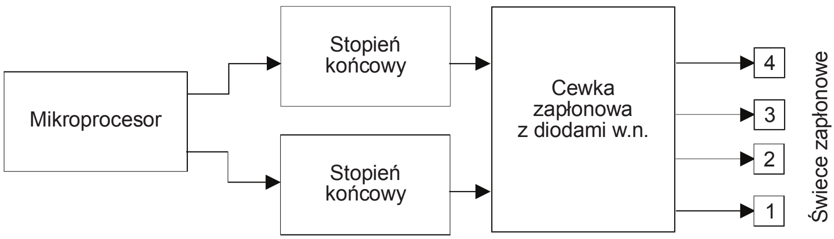
Legend:

Sterownik – Control unit
Czujnik położenia wału – Crankshaft position sensor
Tarcza pomiarowa – measuring disk

Figure 46. *A scheme of the connections between DIS and the control unit.*

An electronic control ignition system is placed inside the injection control unit creating an integrated system. It often includes output stages of power capable of simultaneous control over the two ignition coils. The primary winding of the coils is connected to the electronic control unit connector which connects and disconnects coil circuits to the mass in specific order.

Another possibility of the distributing high voltage into individual cylinders is using also a quadripolar ignition coil but that consists of two primary windings and one secondary. The coils are controlled alternately *via* two outputs assigned to the relevant primary winding. Secondary winding is equipped at the output with two diodes which are in circuit together also with high voltage wires and spark plugs – Figure 47. Arranging the spark plugs in suitable configuration enables the spark jump only between the given pair of spark plugs. What is somewhat problematic, is obtaining desirable infallibility of the high-voltage diodes in the wide range of temperatures.



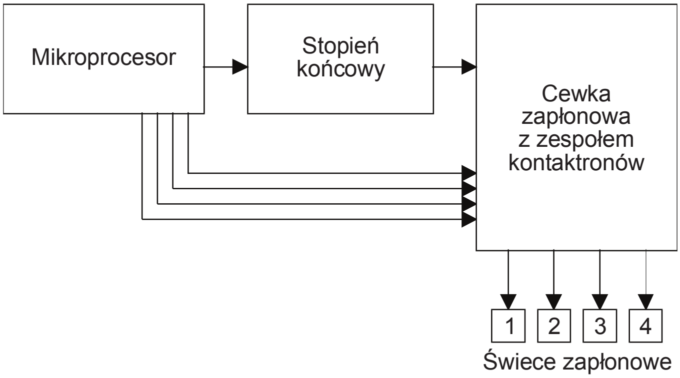
Legend:

Mikroprocesor – microprocessor
Stopień końcowy – Power transistor
Cewka zapłonowa z diodami w. n. – ignition coil with high voltage diodes
Świece zapłonowe – spark plugs

Figure 47. *A scheme of the distributionless ignition system with one coil and diodes.*

Figure 48. shows a directionless ignition system with single ignition coil. The elements connecting are hermetic high-voltage connections – high voltage reed switches. Their controlling coils – A, B, C and D – are switched on by the ignition system microprocessor according to the order of work of cylinders.

At the same time with the impulse starting the current through the primary winding of the ignition coil the microprocessor sends an impulse actuating the winding of the suitable reed switch. In the moment of generating the high voltage impulse the contacts of the reed switch are closed – energy is not wasted on the spark jump and the system does not emit electromagnetic disturbances. The predicted durability of the reed switch is 0.5 million km, and besides, in case of the damage of the reed switch, only one cylinder stops functioning.



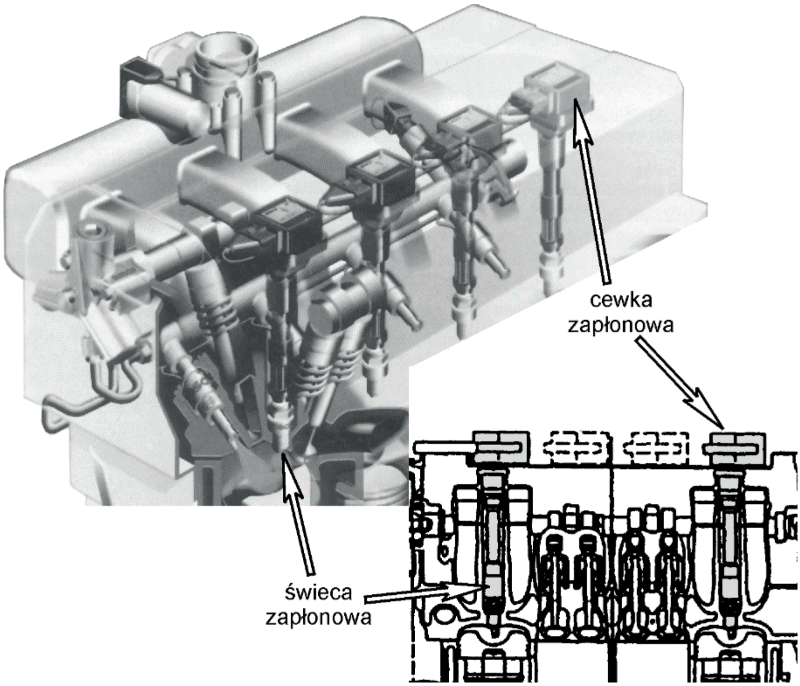
Legend:

Mikroprocesor – Microprocessor
Stopień końcowy – Power transistor
Cewka zapłonowa z zespołem kontaktronów – Ignition coil with reed switch unit
Świece zapłonowe – Spark plugs.

Figure 48. *A scheme of the distributionless ignition system with reed switch.*

In the engine with uneven number of cylinders, equipped with electronic ignition system, for technical reasons single ignition coils are used (they are of course also used in engines with the even number of cylinders). Proper voltage distribution to individual ignition coils takes place in the low voltage circuit in ignition module with sequential logic circuit. In this case the signal in the TDC position is not sufficient. In order to achieve the necessary synchronisation a camshaft position sensor must be used. A injection-ignition system from Bosch Motronic M5 is the example of such a solution for an ignition system with an individual coil for each cylinder. The pioneer in the field of the direct ignition (DI), that is the system with individual ignition coils integrated with spark plugs, was the SAAB company and its injection-ignition system Trionic. It is a capacitive system integrated inside one cassette, consisting of four coils and necessary electronic devices. Central device controls electronic system inside the cassette with a small current, which in turn controls the ignition coils. In the case of cold start-up this system provides ensures a multi spark ignition. After a failed start-up another series of sparks is discharged in all of the cylinders, which causes the burning of soot and fuel drops appearing on the plug electrodes. Because the DI system tolerates a greater space between the electrodes, the predicted work time of the plugs is ca. 60 000 km.

Ignition systems with individual coils are a standard in the engines with direct fuel injection – an example is presented in the Figure 49.



Legend:

Cewka zapłonowa – Ignition coil
Świeca zapłonowa – Spark plug.

Figure 49. *Direct ignition system of a Mitsubishi GDI engine.*

2.4. Limiting the emission of toxic elements of exhaust

Meeting the limits of the toxic substance emissions from the SI engines is possible mainly due to the vehicle emission control devices. The widest in use is three-way catalytic converter (TWC) using lambda sensor, which neutralizes three main toxic constituents of exhaust gases (CO, HC, NOx). Cooperation of the catalytic exhaust converter with lambda sensor is necessary in order to maintain high efficiency of the conversion of all three components, which is possible only with l ranging from 0.996 to 1.006. Therefore, using TWC converter demands more from the power supply, because it must work in closed loop feedback control system with the lambda sensor and (according to the received information adjusting the amount of supplied fuel.

In the engines powered by a bad fuel mix maintaining l in the range ensuring the high efficiency of the TWC is in principle impossible. Limiting the emission of NOx in this case demands the installation of an additional catalytic converter DeNOx (Decrease NOx).

Limiting the emission of nitric oxides can be also achieved by decreasing the maximum combustion temperatures. It is accomplished by using the exhaust gas recirculation unit EGR. Bringing a part of the exhaust again into the cylinder decreases the amount of oxygen accessible in the combustion chamber, and thus limits the amount of fuel that can be burned. Therefore, in the cylinder a smaller amount of heat is formed, and in consequence the maximum temperature of the circulation and emission of NOx decreases. The dilution of the load caused by introducing an amount of exhaust together with fresh load causes the lengthening of the burning time and the increase in the HC and CO emission because of the increase in the quenching of the fire on the cylinder walls. The flammability limit of the mixture restricts the degree of recirculation. Increasing the load temperature, EGR decreases the theoretical efficiency of the circuit and the efficiency of refilling. It can be avoided by cooling the recirculated exhaust gas.

The primary problem for limiting the emission of toxic compounds from SI engines is the start-up period and heating period. In this moment a lot of unfavourable events take place that are connected to the instability of the burning process (e.g. the lack of the possibility of correct production of the fuel/air mixture, misfires, cooling effect of the combustion chamber etc.) and the lack or limited efficiency of the mechanisms detoxifying the exhaust – the lowest temperature in which TWC systems work is 250-300oC. This is why the solution to this problem demands many technologies which are presented below.

The engine

- SPI ⇒ MPI ⇒ GDI,
- Increasing the flame speed,
- Modernised multi-valve technology,
- Multi-plug ignition systems,
- Optimalisation the dose of fuel during start-up,
- Better spraying of the fuel (homogenisation)
- Air support for the fuel injection,
- Delaying the spark advance angle,
- Strictly maintaining A/F ratio.

Fittings

- Heaters,
- External sources of heat for the engine,
- Hermetization and heating the engine chamber before the start-up,
- Isolated inlet manifolds,
- Materials with low thermal inertia,
- A high power starter integrated with alternator,
- Exchanging mechanical devices with electronic appliances that have variable parameters (switched off during the start-up).

Cooling system

- Separating the cooling units of the block and the head,
- The coolant pump with variable flow,
- Smaller mass of the coolant,
- Heating the liquid (with exhaust or electric),
- Heating the head with the exhaust energy from EGR system,
- Partial elimination of the liquid during the heating of the engine,
- Exchanger: coolant – lubricant, transferring the energy from one to the other.

Catalytic converters

- Placing the converter maximally near to the engine,
- Metal-based converters,
- EHC and EGI converters,
- DeNOx converters,
- Supplying the secondary air into the converters,
- Hydrocarbon absorbers LHES,
- Catalyst carrier based on the rhodium compounds,
- Heated λ sensors,
- Double λ sensors with feedback loop.

The emission system of the VW Luo FSI engine (picture no. 50) consists of the following basic elements: the catalytic converter storing NOx (1), the close-coupled starter catalytic converter (2), the exhaust gas recirculation (3) with a valve (4) and NOx detector (5) in the exhaust gases along with a control device. The disposition of particular elements in the engine exhaust system was depicted in the picture no. 51.

The EGR level is adjusted by an electrical valve depending on the work conditions and it goes up to 35%. The NOx disposal under air excess conditions is provided by the catalytic converter storing NOx and co-operating with NOx detector in exhaust gases. The accumulated NOx are absolved when they can be reduced to N2, i.e. under stoichiometric mixture conditions. This converter has also the CO and HC burning ability with the efficiency of the classic TWC, both in homogeneous and stratified mode. The nitrogen oxides detector used in Lupo FSI determines their quantity stored in the converter. This information is transmitted to the control system which, after exceeding the specified value, starts the converter recovery mechanism to cause a temporal transition of the poor mixture into the stoichiometric mixture. The recovery process lasts about 2 seconds and is repeated approximately every minute of engine running in the stratified mode. The described exhaust gas recirculation system shows large susceptibility to sulfur in the gasoline. To remove sulfur compounds accumulated in the converter, the temperature of the exhaust gases must be over 650°C which is rarely seen while driving in urban areas.

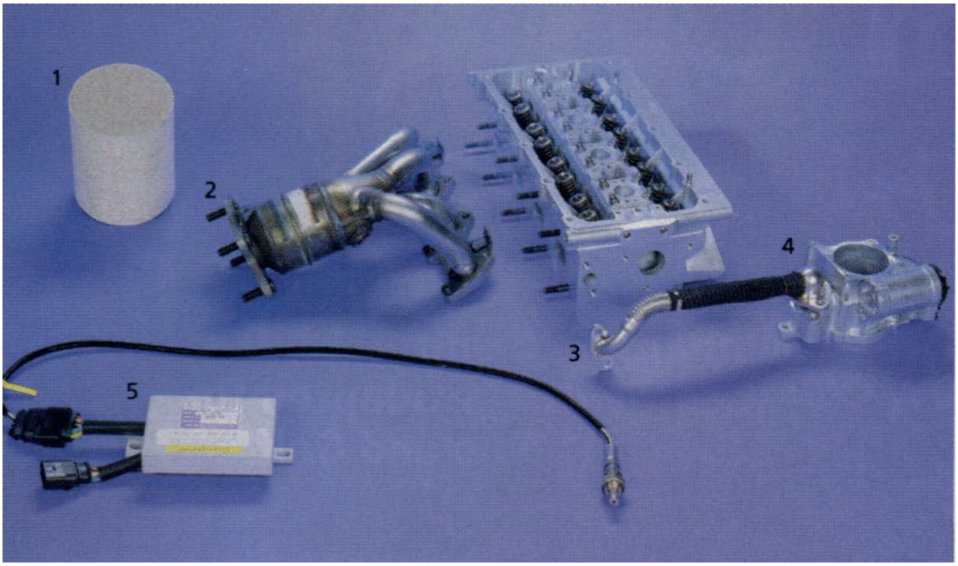


Fig. 50. *The elements of the emission control system of the VW Lupo FSI engine.*

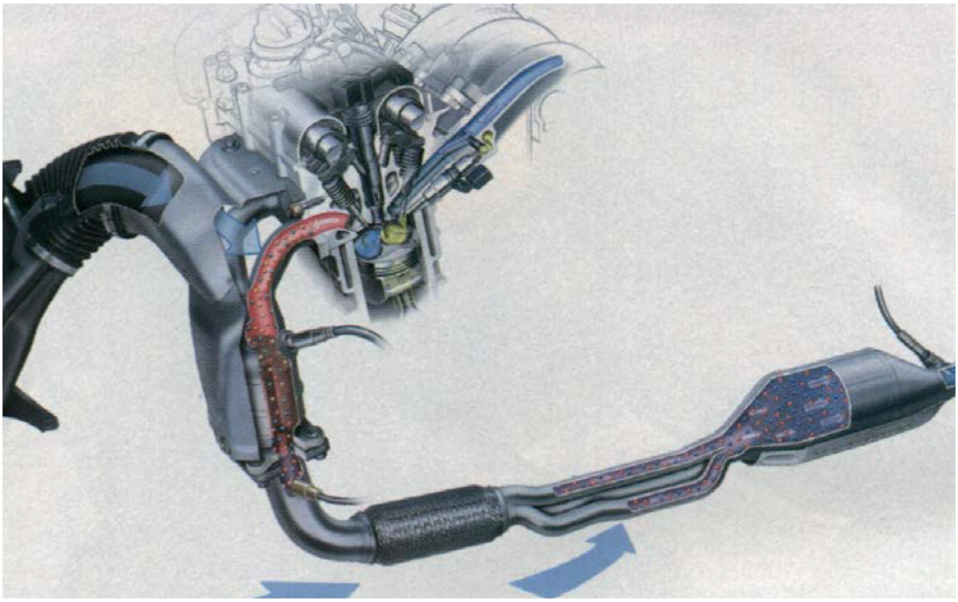


Fig. 51. *The exhaust system of the VW Lupo FSI engine.*

2.5. The development directions of gasoline engines

The engines with spark-ignition (SI) are a dominant source of drive for motor cars, they work with quantitative power supply adjustment, it means that depending on the power demand, various quantity of the combustible mixture with almost definite proportion is poured into the cylinder. It is achieved by choking the mixture or air supply to the cylinder. Yet, it involves charge exchange/circulation loss which is the reason for low efficiency of engines with spark-ignition at small loads. When it comes to the economy of the car exploitation, it would be the best if the engine ran at a variable rate of the air excess, the bigger the lesser power demand is. This solution would also conduce lowering the emission of intoxicants. However, acquiring the stable work of the regular engine powering it with the homogeneous mixture at the air excess rate l exceeding the value 1.3-1.4 is not possible and the unstable work causes the excessive emission of unburnt hydrocarbons (backfire phenomenon) and undoes the economic effects of the improvement of the engine efficiency running on a poor mixture. And additional profit of using poor mixtures are lower maximum temperatures in the combustion chamber, which decide about the quantity of nitrogen oxides. Cutting the quantity of nitrogen oxides created during the burning is not big enough to prevent an engine powered with such a mixture from meeting the toxicity requirements without exhaust gas recirculation systems.

The quick evolution of SI engine power systems lead to replacing the carb with a single point injection system (SPI), and then with a multipoint injection system (MPI) which prevails now. Since dozen of years, the fuel stratified injection (FSI) has been successfully applied and the offer of such type of engine powering is becoming more and more vast.

The development of the gasoline direct injection is lead in two directions. The first one, accepted mainly by the Japanese engineers, adopted from diesel engines is about injecting a dose of gasoline into the cylinder in the half of compression stroke when inlet and exhaust valves are closed. In order to make the preparation of combustion mixture better, the air sucked into the cylinder is propelled in the inlet channel in the rotary motion by the especially formed bottom of the piston. The advantage of this system is the possibility to burn very poor mixtures and all the benefits coming from this, the disadvantages – the necessity to use high injection pressures, problems with nitrogen oxides reduction, the developed exhaust system.

The second solution of the gasoline direct injection resembles well-known and commonly used multi point injection system (MPI) in which the gasoline is injected during the inlet stroke near the inlet valve, and the air going into the cylinder carries the gasoline and stirs with it.

The direct injection was solved similarly, but the injector was put in the combustion chamber. The gasoline injection starts in the half of the inlet stroke with the inlet valve being open. The spout of gasoline joins the air stream inside the cylinder. The work of the injectors ends in the moment of closing the valve. In this system, the mixture burnt in the engine is always homogeneous. The advantages of this solution are: low injection pressure, long injection time allowing to precisely dose the gasoline, very good stirring of gasoline and air, the possibility to use the regular three way catalyst (TWC) exhaust catalytic converter, lesser requirements of the gasoline quality. The disadvantages are: the necessity to burn the mixtures richer than in the previous described solution, the rotary motion of the mixture is required in the cylinder during the compression, bigger emission of carbohydrates, the complicated, multidetector work control system of the injectors.

The poor mixtures combustion is also possible with the classic ways of powering the SI engine; however, the achieved AFR, conditioned by the stable engine running are smaller than with the direct injection and usually reach values not exceeding 23.

The introduction of gas direct injection in SI engines was dictated by the necessity to fulfil another norms restricting the toxicity of gases, simultaneously decreasing the gasoline usage. Additional stimuli to introduce this kind of power supply were the modification of emission tests (UDC + EUDC → NEDC) and introducing new kinds of them (measurement of CO and HC emission in the ambient temperature of -7°C), implementation of the on-board diagnostics systems EOBD, or the requirements of the vehicle durability and reliability, especially when it comes to emission (the necessity to fulfil norms after the 80,000 km mileage or after 5 years). However, meeting the aforementioned requirements interferes with cutting the fuel consumption and CO2 emission. The Mitsubishi GDI engine which meets EURO II offered a 6-12% (6-9% according to independent sources) decline in fuel consumption. The PSA HPi engine meeting EURO III and EURO IV uses 10 % gas less than the version with MPI power supply system. In addition, it characterizes little NOx emission, owing to the DeNOx catalytic converter use, the reduction of the accumulated nitrogen oxides is performed by cyclical enrichment of the mixture for 3 seconds of every 60 seconds of the engine running.

Maintaining the complete efficiency of the DeNOx converter demands disposing of the accumulated sulfates in it – the recovery demands the mixture enrichment and increasing the temperature of gases. The frequency of this process depends on the quantity of sulfur in the gasoline: if it contains 150 ppm of sulfur (EURO III), the process is conducted every 500-700 km of the vehicle mileage, if it contains 50 ppm of sulfur (EURO IV), every 1500 km. The aforementioned 10% fuel consumption decrease by the HPi engine includes the periodic mixture enrichment essential for the proper working of the DeNOx converter. With partial loads to the rotational speed 3500 rpm, the engine is powered with the poor mixture of λ » 2, including EGR (the maximum EGR – 30% is on idle-running, at this point the throttle is already 20° ajar and ready to be fully opened at the slightest acceleration), the engine runs on the stratified mixture and as the load increases, the EGR decreases. The dynamics and elasticity in HPi engine were increased, owing to the restriction of the engine running with half-closed throttle, that is owing to decreasing the pump loss. In comparison with MPI version, the 3.6% elasticity increase was achieved. The strategy taken by Mitsubishi assumed high EGR, which did not provide better accelerations and decrease in fuel consumption in contrast to PSA HPi.

If the cylinder has a big capacity, increasing the EGR (according to the Japanese research up to 45-50%) in the wide scope of the motor action, enables to significantly decrease fuel consumption. There are other ways to reach this goal, for example: increasing l, improving the combustion of stratified mixtures etc. PSA HPi reaches l = 2,5 – 2,8 under some work conditions. Another way to decrease fuel consumption is to use a multiphase injection, owing to special injectors we get a favorable torque, both on high and low rotational speeds. The reduction of fuel consumption can be also acheived by decreasing the speed of idle-running – PSA HPi currently 700 rpm, target – 550 rpm.

The development of SI DI engines must also take into consideration the fact that the engine should be relatively cheap. The development of such engines takes place now mainly when it comes to the injection pump and catalytic converter. PSA HPi pump is small, dense and light (900 g) with three pistons. The main task of the catalytic converter is disposing of NOx, however, so far no NOx concentration detector has been used in gases and its content is estimated basing on a model.

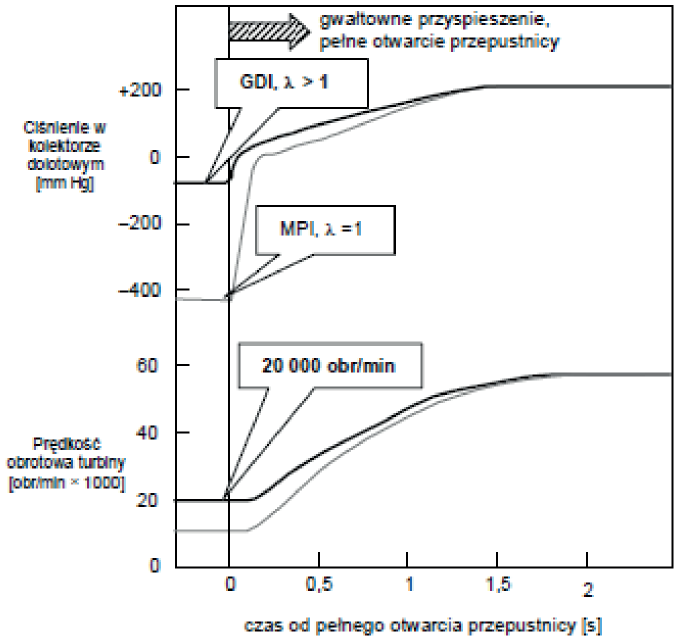
The injection system is meant to decrease the energy consumption. It can be achieved by using the electrical drive of the high pressure pump because then the pump will pump up the fuel only in the necessary quantity.

The electromagnetic valves are promising, especially when it comes to engines with small capacity. For big cubic capacities, the idea of side injection works out great. With the strong swirl generated by one of the inlet channels, we may reach AFR = 37. To cut the costs, we may use electromagnetic drive only from inlet valves, and the outlet valves can be traditionally driven. However, such a solution makes the use of internal exhaust recirculation or cylinder deactivation difficult. The estimated solutions allow to decrease the fuel consumption in NEDC cycle by 8-11 % (the first generation of SI DI engines) and 13-17 % (the second generation).

The latest generations of valves need less electric energy owing to the use of un-centric torsion spring cooperating with the axial spring.

Introducing the turbocharging of the SI DI engines, the same obstacles were encountered as when it comes to TDI engines, i.e. the problem with covering the whole area of the motor action. According to General Motors, turbocharging of SI DI engine allows for the additional decrease in fuel consumption by 8% while simultaneously decreasing the engine cubic capacity. The air excess in the mixture with small loads (bigger volume of gases) allows for the earlier work start of the compressor than in the power supply in MPI system. The course of the curve of the useful moment is more favorable for small and medium rotational speed. The information provided by Mitsubishi may be cited here: while driving at 30 km/h, the compressor turbine of the MPI engine revolves at 12,000 rpm, and of the GDI engine at 20,000 rpm (picture no. 52). After a rapid acceleration, the rotational speed of the turbines of both engines equalizes after only 1,5 second, so no wonder that the GDI engine vehicle characterizes better acceleration.

A matter worth consideration is, when it comes to SI DI engines, whether increasing λ is more favorable in charge conditions or whether it is better to increase the fuel dose and keep λ = 1. However, it seems that it is worth to broaden the motor action from λ > 1, which would provide decreasing the fuel consumption outside urban areas (picture no. 53). We can also increase engine effort to decrease Vss.



Legend:

Gwałtowne przyspieszenie, pełne otwarcie przepustnicy – Rapid, acceleration, full opening of the throttle

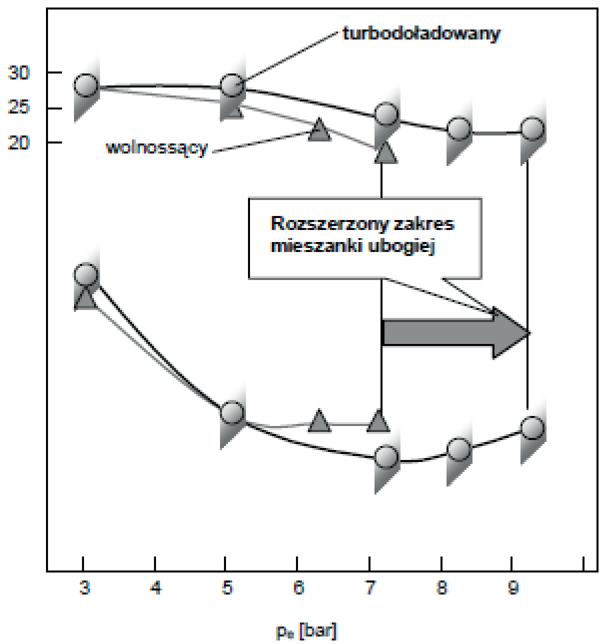
Ciśnienie w kolektorze dolotowym [mmHg] – Pressure in intake manifold [mmHg]

20 000 obr/min – 20,000 rpm

Prędkość obrotowa turbiny [obr/min x 1000] – Rotational speed of turbine [rpm x 1000]

time after the full opening of the throttle [s]

Fig. 52. *The comparison of the reactions to a rapid push on the accelerator pedal at 20 kph by cars with turbocharged MPI and GDI engines.*

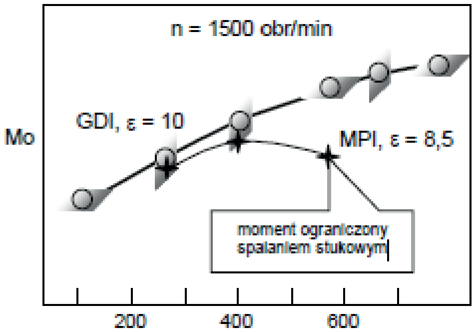


Legend:

Turbodoładowany – Turbocharged
Wolnossący – Naturally aspirated
Rozszerzony zakres mieszanki ubogiej – Extended scope of the poor mixture
pe [bar] – pe [bar]

Fig. 53. *The wider area of GDI motor action with λ > 1, owing to the use of turbocharging.*

It is worth noticing that the turbocharged DI engine can have significantly bigger λ compression level than turbocharged MPI, it comes from different conditions in which the mixture is made – less favoring the engine knocking (picture no. 54). For instance: four-cylinder sixteen-valve turbocharged SI engine can have λ = 10,5 in DI version, and only λ = 8,5 in MPI version. Mitsubishi states that naturally aspirated GDI engine can reach λ = 12,5, on the other hand, the turbocharged version should be on λ = 10 compression level. In Mitsubishi engines, there is a two-phase injection in acceleration conditions on small loads. In the turbocharged version – the air pumped into the combustion chamber cools it before the second phase of the injection. The fuel injection taking place in the intake stroke has a conical shape and creates a very poor mixture unable to ignite, the fuel evaporation causes the temperature to go down in the combustion chamber what prevents from engine knocking. Additionally, large quantity of air near the walls of the chamber (the swirl) helps to prevent the engine knocking, which is characteristic for GDI engines in λ >> 1 conditions.



Legend:

Mo – T (torque)

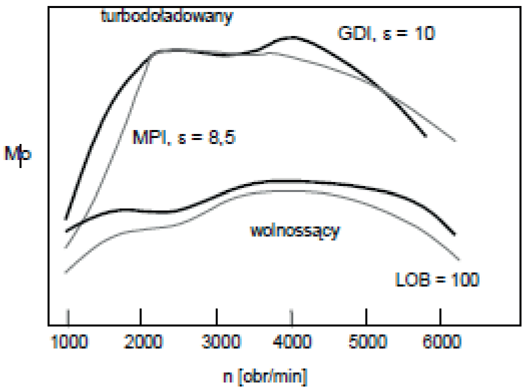
n = 1500 obr/min – n = 1500 rpm

GDI, e = 10 – GDI, e = 10

MPI, e = 8,5 – MPI, e = 8.5

Moment ograniczony spaleniem stukowym – The moment restricted by engine knocking

the pressure in the inlet collector [mm Hg]



Legend:

Mo – T (torque)

Turbodoładowany – Turbocharged

GDI, s = 10 – GDI, s =10

MPI, s = 8,5 – MPI, s = 8.5

Wolnossący – Naturally aspirated

LOB = 100

n [obr/min] – n [rpm]

Fig. 54. *The process of torque in turbocharged GDI and MPI engine.*

In turbocharged DI engines, the problem known in MPI engines stays unsolved, i.e. low efficiency of charging in small speed and load work conditions. The inlet channels are optimized to provide high efficiency of filling up in the maximum speed vicinity, and at small speed, the volumetric efficiency is evidently worse. The solution could be the use of inlet wires with variable length, providing the dynamic charging effect with low n values.

The works to solve the aforementioned problem have been ongoing since the beginning of the new millennium. The solution is ISAD (Integrated Starter Alternator Dumper), i.e. high power integrated alternator with a starter, which is built-in in the flywheel, it is also called the French turbodyne. This device can help the internal combustion engine when it has a great demand for power (pulling away, hill driving, accelerating), especially if the compressor is not working at that moment. It allows to use the internal combustion engine of the lesser power. Running in the alternator mode, ISAD provides 5-10 kW electric power, already reaching 80% efficiency on the very idle-running. This creates the possibility of using a small additional compressor driven electrically in the engine, which would charge the engine efficiently until the turbocharger starts running. Owing to the combination of the direct injection, turbocharging and ISAD, the SI engines characterize smaller cubic capacity (smaller weight) and fuel consumption with the utility parameters unchanged.

To sum up, as the development courses of the piston internal combustion engines with spark-ignition, we should enumerate:

- elimination of the carburetor power supply system and replacement of the single point injection system (SPI) with the multipoint one (MPI), and ultimately – with gasoline direct injection,
- supplying with the poor mixture, load stratification,
- increase of compression ratio ε (⇒ 11-12 and more),
- dropping the flow loss by the restriction of the throttle closure and ultimately – its elimination,

the new ECU introduction: Risc chip 32 bit Micro Processor controlling inter alia:

- the precise and adaptation mixture composition – l (with feedback), ultimately – individually in particular cylinders,
- the mixture failure (l ä) with the ignition retard system, in the process of the cold start and preheating including fuel parameters, limited mixture enrichment during rapid acceleration,
- predicted working conditions of the catalytic converters,
- the EGR dynamic system, with cooling as well,
- additional air (from the electric pump) delivered to the outlet system during the starting,
- OBD-II auto-diagnosing system,

- using multi-valve heads with VTEC system, they are connected to the effective and controlled dynamic or compressor charge (also with turbochargers with variable geometry),
- introduction of the electromagnetic timing belt,
- using the cylinder deactivation system with partial engine load,
- development and control over load whirling,
- using variable inlet systems,
- using technically advanced multi-spark ignition systems with engine knocking detection,
- using a separate injector for each inlet valve (in MPI injection system),
- using piezoelectric injectors, air-injection (better fuel fragmentation),
- realization of Miller's cycle,
- using heat accumulator,
- increasing the starting rotational speed,
- using the high power starter integrated with alternator,
- replacing the mechanical and hydraulic drive system of accessories with the electric one,
- using the heated UHEGO (Universal Heated Exhaust Gas Oxygen) mixture composition detector with one or two additional λ probes,
- using the advanced systems of combustion gas treatments:
- the TWC starter catalytic converter type EHC with short activation time,
- center – palladium converter,
- end system – hybrid catalytic converter (TWC converter + HC absorber),
- outlet system with small capacity and heat inertia,
- using high-quality reformulated lead-free gasoline.

3. Diesel engines

Although, the diesel engine was already developed by Rudolf Diesel in 1892, the research on the development of the engine construction, fuel supply systems, air delivery, combustion gas treatments and starter procedures in low temperatures is still being carried out.

Lesser unitary consumption of cheaper fuel in diesel engines and the ability to generate a bigger torque in comparison with spark-ignition ones contributes to the fact that diesel engines are mounted in motor vehicles (cars, commercial vehicles, trucks), tractors, stationary devices and ships and aircrafts. Machines with diesel engines are mostly used by companies, so the demand is dependent on the economy condition.

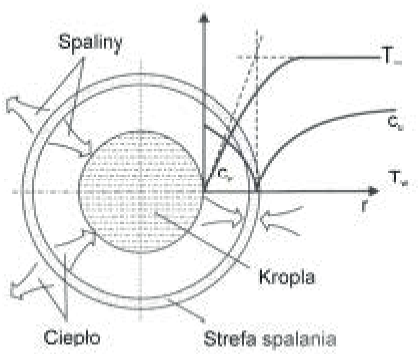
Air delivery system and the shape of the combustion chamber in Diesel engine are built in order to achieve turbulent flow. The turbulent flow is unstable, showing the fluctuation of consecutive work cycles. The parameters of turbulence (root mean square velocity of the fluctuation, turbulence scale) change irregularly and randomly. A high level of turbulence leads to flame extinction and dissipation of the energy in smaller whirls.

The turbulence in diesel engine is generated in the charge and compression stroke, and then supported by the process of the gasoline direct injection to the combustion chamber.

The factors above have influence on the turbulence level in combustion chamber:

- shape of the inlet system,
- combustion chamber construction,
- high-pressure process of fuel injection into the combustion
- chamber.

In the diesel engine, there is a system of burning the charge which consists of air and atomized fuel liquid as a set of vaporizing drops. As a result of the compression process, the air in the combustion chamber reaches 750-950°C, which is a lot higher than the temperature of the self-ignition of diesel fuel. The heat transferred to the drops makes its external layer vaporize while the fuel vapors are lifted to the air surrounding the drop, with which they mix (fig. 55). The mixture of fuel vapor and air with a huge gradient of local air excess factor is created around every drop. In the places where the mixture reached the incendiary level, self-ignition occurs and the heat released as the result of burning causes exhaust expansion.

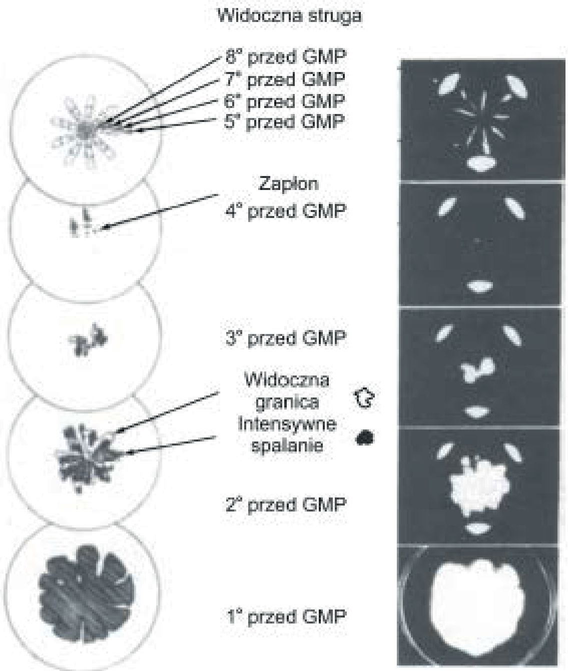


Legend:

Spaliny- Exhaust gases
Ciepło – Heat
Strefa spalania – The combustion zone
Kropla – A droplet

Fig.. 55. *The scheme of evaporation of the liquid fuel drops with marked temperature distribution and concentration of fuel droplets.*

The self-ignition sources are random and are simultaneously in many places creating the areas which start to combine with each other (fig. 56). The regional flame front is created around them, which heads to create a combustible mixture. In the whole area where the fuel was injected, one common, strongly developed on the surface burning front is created and it spreads until it encounters the combustible mixture. The whole process takes place in the combustion chamber with the high level of turbulence which is strengthened with the process of the fuel injection.



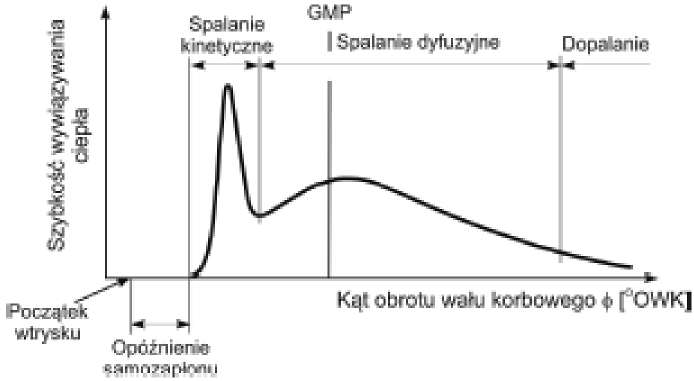
Legend:

Widoczna struga – Visible spout
8° przed GMP – 8° before TDC
7° przed GMP – 7° before TDC
6° przed GMP – 6° before TDC
5° przed GMP – 5° before TDC
Zapłon – Ignition
4° przed GMP – 4° before TDC
3° przed GMP – 3° before TDC
Widoczna granica – Visible border
Intensywne spalanie – Intensive burning
2° przed GMP – 2° before TDC
1° przed GMP – 1° before TDC

Fig. 56. *The process of fuel injection, evaporation and combining fuel vapors with air.*

The factors below influence the burning process in diesel engine supplied with liquid fuel:

- the course of the fuel injection process,
- air whirling in the combustion chamber,
- constantly changing composition of the charge in the combustion chamber.



Legend:

Szybkość wywiązywania się ciepła – Heat release rate
Początek wtrysku – Injection start
Opóźnienie samozapłonu – Self-ignition delay
Spalanie kinetyczne – Kinetic combustion
GMP – TDC
Spalanie dyfuzyjne – Diffusion combustion
Dopalanie – afterburning
Kąt obrotu wału korbowego – Angle of the crankshaft rotation [°CR]

Fig. 57. *The course of heat release rate with combustion phases.*

The typical course of releasing the heat is depicted in the picture no. 57. On the basis of the diagram shape, we may single out four phases of burning. The first one of them, the self-ignition delay phase, covers the time from the beginning of the injection until the occurrence of the first self-ignition fires and it serves to prepare the fuel to burn. The time of the self-ignition delay lasts from 0.7 to 3 ms in engines. Depending on the injection system and rotational speed of the crankshaft, the whole dose of fuel may be injected in this time. The more fuel gets into at that time, the more rapid pressure increase will occur in the second phase of the burning, which results in the engine running loudly. The rapid heat release occurs as a result of many sources of self-ignition in the second phase. This phase is called kinetic burning, its burning speed depends on the speed of the chemical reactions of burning. The third phase is diffusive burning. The heat release speed depends on the speed of physical processes of preparing a combustible mixture and its burning. It starts with reaching maximum pressure in the combustion chamber and lasts practically until the end of the burning. In the last phase, finishing burning, the fuel not burnt until this phase is overreacted. When we look at the energy balance, this phase is unprofitable.

Burning takes place in two phases of fuel oxidation, both in kinetic and diffusive burning phase. The ability of the fuel to self-ignite is characterized by cetane number (CN). The more value CN has, the easier the fuel burns. To supply the diesel engine traditionally, diesel fuel with 40-60 CN (depending on the country of distribution) is mostly used. Increasing CN is possible by using additives such as acetone peroxide or ethyl nitrate.

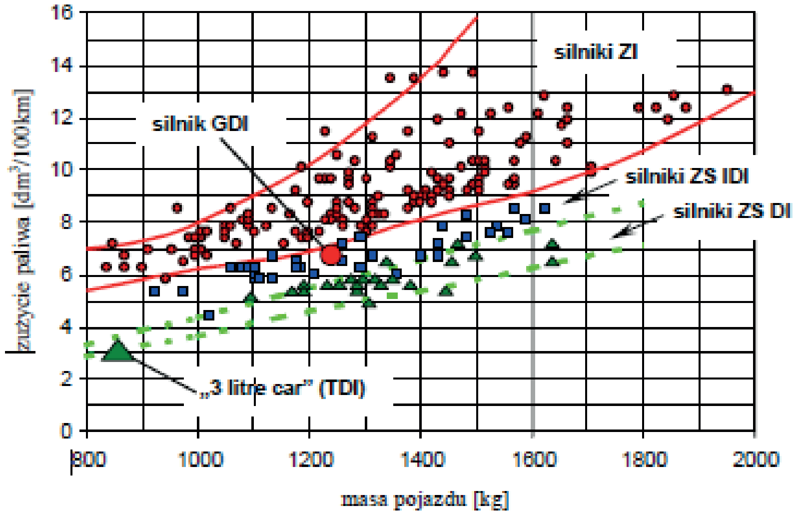
Diesel engines run at l air excess rate, changing in broad limits depending on the load. Because of the high compression ratio, which equals 14-15 for high power engines and 24 for high-speed engines, and small flow loss (lack of throttle), diesel engines are superior to SI engines when it comes to efficiency; however, achieving substantial efficiency with bigger loads is reliant upon creating favorable conditions for complete burning. Difficulties with creating these conditions grow with the increase of the engine rotational speed and are controlled by using specially constructed combustion chambers.

The development of combustion engines, including diesel engines, is now conditioned mainly by economic reasons. Constantly tightening up requirements for fuel consumption and CO2 emission impose the use of the diesel engines with direct injection (DI) as the most efficient drive source for motor vehicles. It also concerns cars in which diesel engines with indirect injection (IDI) were commonly used not long ago (picture no. 58).

In a burning system with direct injection to unitary combustion chamber put usually in the piston, the whole fuel dose is injected. Too slow burning process in DI engines causes difficulties in increasing the rotational speed ($n > 4000$ rpm). However, these difficulties were overcome in HSDI (High Speed Direct Injection) constructions. The tendency to build HSDI LDD (Light Duty Diesel) engines with a smaller and smaller capacity of a single cylinder (about 300 cm^3) and high effort level (60 kW/dm^3) is apparent.

The injection system is one of the main factors creating the burning process of the diesel engine. In direct injection engines, due to the way of creating the mixture, high requirements have been imposed on injection systems. The engine indicators depends on the following parameters:

- injection timing,
- the spray shape and the degree of fuel atomization,
- injection pressure,
- injection specifics.



Legend:

Zużycie paliwa [dm3/100 km] – Fuel consumption [dm3/100 km]
Silnik GDI – GDI engine
Silniki ZI – SI engines
Silniki ZS IDI
Silniki ZS DI – DI diesel engines
„3 litrowy samochód" TDI – "3 litre car" TDI
Masa pojazdu [kg] – Vehicle weight [kg]

Fig. 58. *Fuel consumption by cars from 1997-99 equipped with different kinds of engines.*

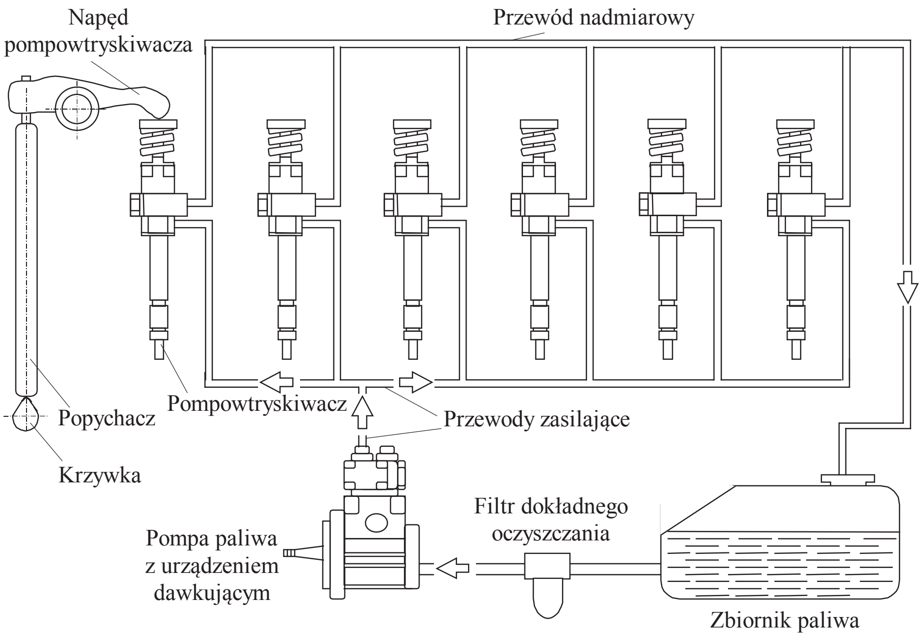
To turn the work volume down and decrease NOx emission, the fuel injection delay is sought. Owing to this, the maximum pressure and temperature in the combustion chamber is decreased, and analogously NOx emission, however simultaneously, the time for burning shortens, increasing the HC and PM emission. Thus, an improvement of the conditions of mixing fuel with air and shortening the self-ignition delay and burning time is necessary. It can be done by the proper forming of the combustion chamber or increasing the injection pressure. In the latter, lessening the spray jets diameter, because increasing the injection pressure alone cause better atomizing only on the edge of fuel spot, increasing the fuel penetration into the combustion chamber. But then the danger of reaching the walls of the combustion chamber by fuel droplets occurs and it increases the smoke level significantly.

Using the air whirling in the cylinder may decrease the level of PM emissions with lower injection pressure, however, then the emission NOx increases. The same level of PM emission may be achieved by increasing the injection pressure and decreasing the whirling at the same time, which does not make the NOx emission go up.

The improvement of mixing the fuel with air and decreasing the pressure rise is also realized by using multi-phase injection. This system is about putting so called pilot injection into the cylinder before injecting significant amounts of fuel. The initial amount of fuel shortens the self-ignition delay time and limits the speed of pressure rise in the cylinder. It cause a decrease of maximum temperature in the first phase of burning and limits the nitrogen oxides emission. Apart from that, using the burning system which uses the pilot injection contributes to decreasing the engine noisiness.

When it comes to the injection equipment, line injection pumps are replaced with rotational ones. The majority of the modern solutions of injection systems aims towards increasing the injection pressure, and owing to this, i.a. the bigger fuel injection delay is possible. However, while increasing the injection pressure, adverse wave phenomena occur in fuel wires and the injection pump. They are prevented by shortening the way to the cylinder for the high-pressure gasoline, e.g. by using pump injectors (picture no. 59) or Common Rail (CR) system (picture no. 60). This system is based on the idea of supplying all the cylinders by the mutual fuel main, similarly to SI engines with multi-point injection. Apart from mensuration elements and the central control unit, the system contains three main elements: high-pressure pump, pressure vessel and injector. The general rule how the system works is: the pressure is produced in the pump through the piston driven by the cam, just like in a regular line pump. The difference is primarily between the fact that pumping the fuel into the pressure collector does not occur during the injection but between consecutive injections. Owing to such a system, the pressure in front of the pump dispenser is constant the whole time of the injection and the amount of injected fuel depends on the time the injector opens and the pressure in the system. What is more, the CR system allows to pick individual amount of fuel for each cylinder and reacts on changes of the engine running conditions. High injection pressure, possible to achieve at small rotational speed, enables to increase the torque by 20 to 30%.

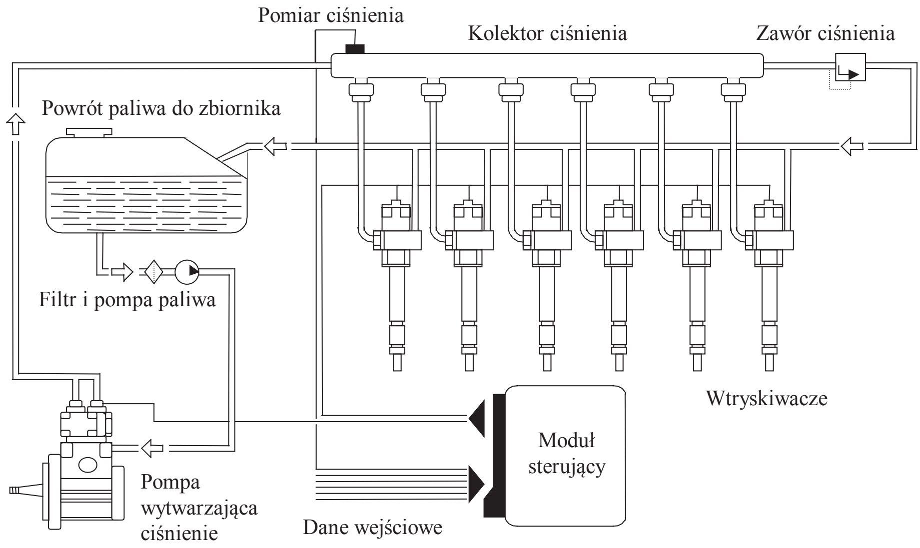
The elimination of mechanical and hydraulic restrictions of the conventional system allows to increase the maximum rotational speed. Using the CR system leads to decreasing the fuel consumption and exhaust toxicity, and increasing the effort of the drive unit.



Legend:

Napęd pompowtryskiwacza -injector drive
Popychacz – tapper
Pompowtryskiwacz – injection pump
Przewód nadmiarowy – overflow return line
Przewody zasilające – fuel lines
Krzywka – cam
Pompa paliwa z urządzeniem dawkującym – fuel pump
Filtr dokładnego oczyszczania – fuel filter
Zbiornik paliwa – fuel tank

Fig. 59. *Schematic injection system with injectors*

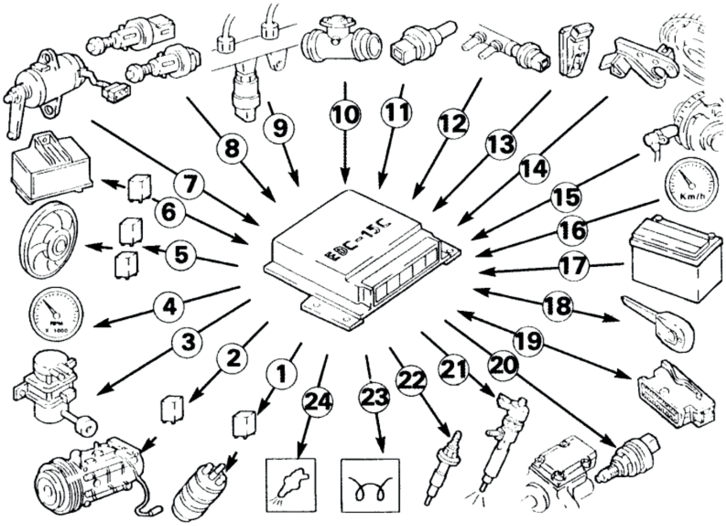


Legend:

Pomiar ciśnienia – pressure sensor
Kolektor ciśnienia – common rail
Zawór ciśnienia – pressure limiter
Powrót paliwa do zbiornika – return fuel tank
Filtr i pompa paliwa – Filter and low pressure pump
Wtryskiwacze – injectors
Pompa wytwarzająca ciśnienie – high pressure pump
Moduł sterujący – electronic control unit (ECU)
Dane wejściowe – input data

Fig. 60. *Common rail system diagram*

The electronic control system became more complicated in the CR system. The control processor collects, i.a. information (picture no. 61) about the rotational speed of the crankshaft, the location of the camshaft, accelerator pedal, temperature of the sucked air and cooling liquid, pressure in the fuel main and inlet collector, air mass flowing into the inlet system. The control system determines many other processes in the vehicle connected with, e.g. starting the glow plugs, exhaust gas recirculation, charging and diagnosing the engine. The described system is an example of the major integration of the fuel injection with electronic control unit.



Legend:

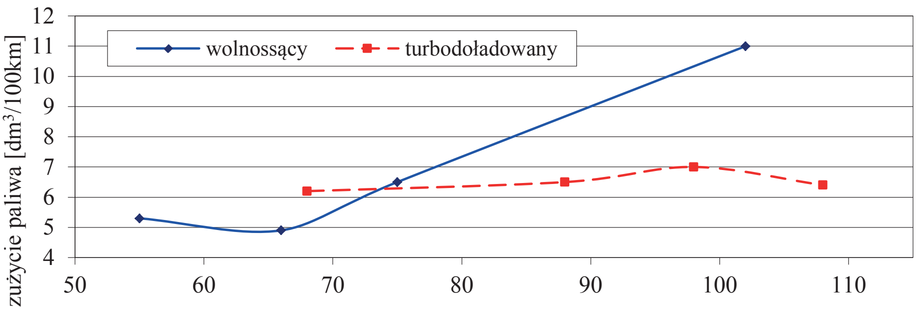
1 – low pressure fuel pump
2 – air conditioning compressor sensor,
3 – exhaust gas recirculation (EGR) valve,
4 – engine speed sensor,
5 – cooling fan relay,
6 – glow plug relay,
7 – accelerator pedal position sensor,
8 – the brake and clutch off sensor,
9 – fuel rail pressure sensor,
10 – mass (air) flow sensor (MAF),
11 – engine coolant temperature sensor,
12 – fuel temperature sensor,
13 – boost pressure sensor,
14 – the camshaft position sensor,
15 – crankshaft speed sensor,
16 – speed signal,
17 – the voltage at the battery terminals,
18 – immobilizer,
19 – diagnostic socket (OBD),
20 – the fuel pressure sensor,
21 – injectors,
22 – glow plugs,
23 – glow plugs indicator,
24 – injection system indicator.

Fig. 61. *Sensors and executive elements of the electronic control system of engine with the Common Rail power supply system:*

Apart from injection system, an electronic regulation of injection parameters (fuel dose, injection timing, injection time, course of injection and injection pressure) is introduced on broad scale depending on many variables, although there is a trend of formatting injection process in function of as many operating parameters of engine as possible.

Similarly to petrol engines, in Diesel engines also are used heads with 4 valves per cylinder, with electromagnetic lift system. The optimization of injector location is aimed to its central position in combustion chamber. There were released Diesel engines with regulated compression level depending on engine speed (grater at lower speed).

A wide spread solution for lower fuel consumption and emission of toxic substances and increase of volumetric power indicator in Diesel engines is turbocharging (diagram 62). Almost 100% of high power engines and over 75% of low power engines are turbocharged. The importance of turbocharged engines can be easily understood, if you take into consideration that during working at full load engine is running on mixture with AFR ≈ 16. In this case the possibility of reaching full power is conditioned by ability of delivering enough air to burn fuel, not by its injection amount. An increased bulk cargo and fall of temperature in combustion chamber is caused by charged air cooling, which promotes reduction of nitrogen oxides emission.



Legend:

Wolnossący – naturally aspirated
turbodoładowany – turbocharged
moc znamionowa [KM] – nominal power [Hp]

Fig. 62. *Comparison of fuel consumption by cars with naturally aspirated and turbocharged diesel engines.*

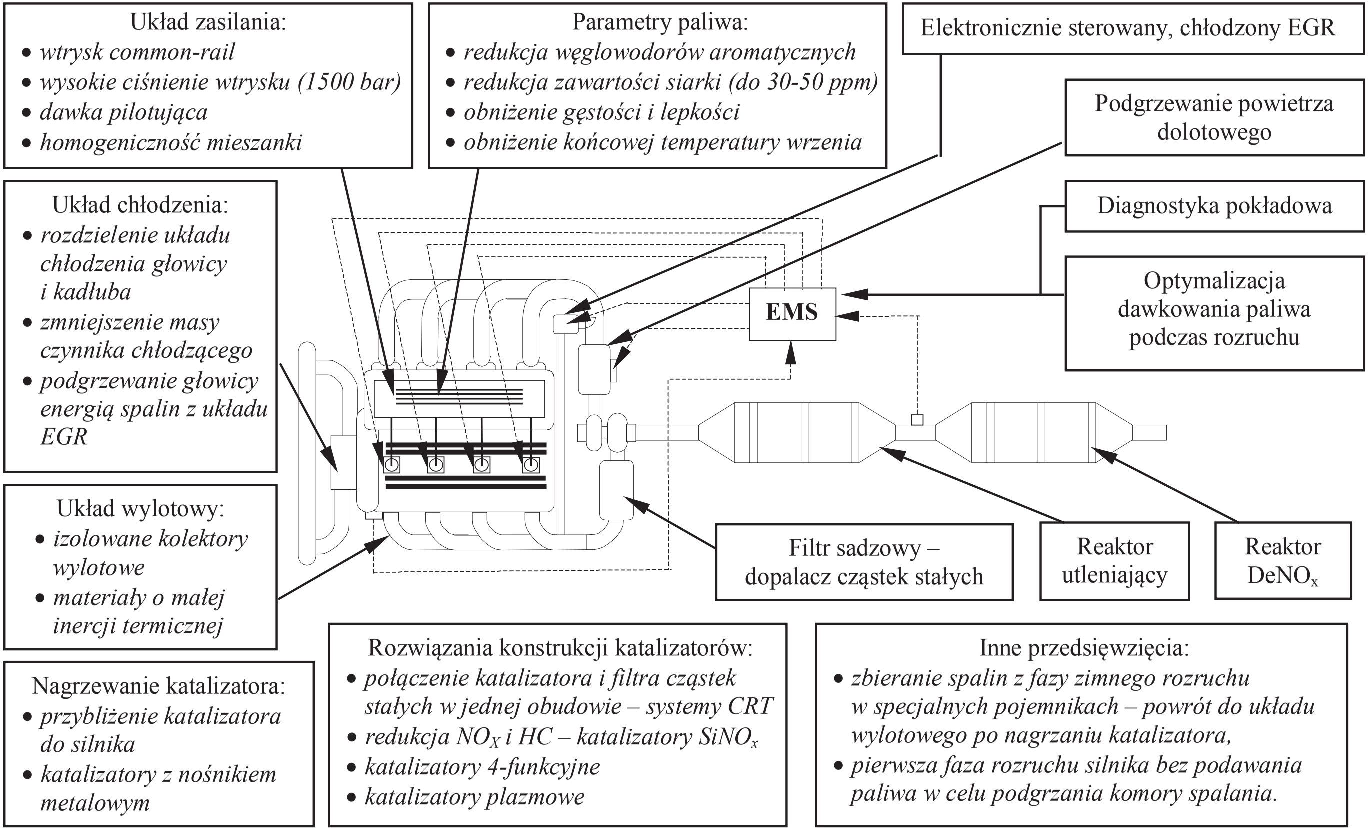
To ensure the full exploit of technical possibilities which result from the use of turbocharging, an assurance of its proper efficiency in engine speed and engine load is required. VGT (Variable Geometry Turbocharger) or VFT (Variable Flow

Turbocharger) turbocharges make possible to meet this condition. Thus the time of engine reaction on changing work conditions and torque throughout rotation speed range. Moreover, the NOx emission as well as fuel consumption in range of high rotation speed and low engine load can be reduced by adjustment of compressor rotation speed to temporary engine load.

The application of EGR including cooling exhaust from cylinder become a rule. Introduction of Exhaust Gas Recirculation only allows to decrease amount of NOx produced in engine, however, it causes the increase of emission of incomplete combustion product. The application of recirculated exhaust cooling allows to decrease NOx and solid particles emission.

The level of carbon monoxide and hydrocarbons emission in Diesel engines is similar to this observed in petrol engines equipped in three functions catalysis reactor, what is more its further reduction is not a problem. The most difficult problem is reduction nitrogen oxides and solid particles emission. For this reason, Diesel engines are equipped with Oxticat (Oxidation Catalyst), DeNOx catalytic reactor and solid particles filter.

The engine is equipped in systems reducing time of engine warm up and start work by exhaust after treatment systems, due to huge importance of limitation of emission during cold start and engine warm up (figure 63).



Układ zasilania:
• wtrysk common-rail
• wysokie ciśnienie wtrysku (1500 bar)
• dawka pilotująca
• homogeniczność mieszanki
Parametry paliwa:
• redukcja węglowodorów aromatycznych
• redukcja zawartości siarki (do 30-50 ppm)
• obniżenie gęstości i lepkości
• obniżenie końcowej temperatury wrzenia
Elektronicznie sterowany, chłodzony EGR
Podgrzewanie powietrza dolotowego
Diagnostyka pokładowa
Optymalizacja dawkowania paliwa podczas rozruchu
EMS
Układ chłodzenia:
• rozdzielenie układu chłodzenia głowicy i kadłuba
• zmniejszenie masy czynnika chłodzącego
• podgrzewanie głowicy energią spalin z układu EGR
Układ wylotowy:
• izolowane kolektory wylotowe
• materiały o małej inercji termicznej
Filtr sadzowy – dopalacz cząstek stałych
Reaktor utleniający
Reaktor $DeNO_x$
Nagrzewanie katalizatora:
• przybliżenie katalizatora do silnika
• katalizatory z nośnikiem metalowym
Rozwiązania konstrukcji katalizatorów:
• połączenie katalizatora i filtra cząstek stałych w jednej obudowie – systemy CRT
• redukcja NO_X i HC – katalizatory $SiNO_x$
• katalizatory 4-funkcyjne
• katalizatory plazmowe
Inne przedsięwzięcia:
• zbieranie spalin z fazy zimnego rozruchu w specjalnych pojemnikach – powrót do układu wylotowego po nagrzaniu katalizatora,
• pierwsza faza rozruchu silnika bez podawania paliwa w celu podgrzania komory spalania.

Legend:

Układ Zasilania – The power supply system:
Wtrysk common rail – Common-rail injector
Wysokie cięnienie wtrysku (1500 bar) – High pressure of injection (1500 bars)
Dawka pilotująca – Pilot dose
Homogeniczność mieszanki – Homogeneity of the mixture
Układ chłodzenia – Cooling system:
Rozdzielenie układu chłodzenia głowicy i kadłuba – Separation of cylinder head and hull cooling
Zmniejszenie masy czynnika chłodzącego – Coolant weight reduction
Podgrzewanie głowicy energią spalin z układu EGR – Heating cylinder head with EGR exhaust gas energy
Układ wylotowy – The exhaust system:
Izolowane kolektory wylotowe – Insulated exhaust manifolds
Materiały o małej inercji termicznej – Materials with low thermal inertia
Nagrzewanie katalizatora – Catalyser heating:
Przybliżenie katalizatora do silnika – An approximation of the catalyst into the engine
Katalizatory z nośnikiem metalowym – Metal carrier catalyser
Parametry paliwa – Fuel parameters:
Redukcja węglowodorów aromatycznych – Reduction of aromatic hydrocarbons
Redukcja zawartości siarki (do 30-50ppm) – Reduction in the sulfur content (up to 30-50 ppm)
Obniżenie gęstości i lepkości – Reduction of density and viscosity
Obniżenie końcowej temperatury wrzenia – Reduction of the final boiling point
Elektronicznie sterowany, chłodzony EGR – Electronically controlled, cooled EGR
Podgrzewanie powietrza dolotowego – The intake air heating
Diagnostyka pokładowa – OBD (On-board diagnostics)
Optymalizacja dawkowania paliwa podczas rozruchu – Optimization of the fuel dosage during starting
EMS – EMS
Filtr sadzowy – dopalacz cząstek stałych – The soot filter – particulates afterburner
Reaktor utleniający – Oxidation reactor
Reaktor DeNO2 – DeNOx reactor
Rozwiązania konstrukcji katalizatorów – Solutions in catalyser construction:
Połączenie katalizatora i filtra cząstek stałych w jednej obudowie – systemy CRT – Combination of catalyst and particulate filter in one case – CRT systems
Redukcja NOx i HC – katalizatory SiNO2 – Reduction of NOX i HC – SiNOx catalyser
Katalizatory 4-funkcyjne – 4-function catalyser
Katalizatory plazmowe – Plasma catalyser
Inne przedsięwzięcia – Other solutions:
Zbieranie spalin z fazy zimnego rozruchu w specjalnych pojemnikach – powrót do układu wylotowego po nagrzaniu katalizatora – Collecting exhaust gases from the cold start phase in special containers – back into the exhaust system when catalyser is hot.
Pierwsza faza rozcruchu silnika bez podawania paliwa w celu podgrzania komory spalania – The first phase of the starting the engine without fuel delivery in order to heat the combustion chamber.

The following directions of diesel engines can be distinguished:

- widespread application of indirect injection,
- use of two phase injection with pilot dose,
- optimization of combustion room and centrifugation (with taking into account interrelationships between inlet canal shape, configuration of combustion room and hydraulic system injection),
- single cylinder volume reduction, increase of rotation speed and effort (LDD engines only),
- multi-valve timing systems with regulated parameters,
- phase control timing – VTEC,
- dynamic charging + VGT or VFT turbocompressor with charged air cooling (intercooler),
- additional pulse charging during acceleration with compressed air from pressure tank,
- dynamic EGR with cooling,
- electronic regulation of injection parameters in function on many variables
- introduction of Common Rail injection system or high pressure injection pumps (160-200MPa),
- increase of air overate factor,
- application of open combustion chambers,
- application of quite combustion chamber (without vortex),
- vortex and turbulence control in cylinders,
- increasing the degree of compression at low speeds and decreasing at large,
- particle filter, Oxicat oxidizing catalytic reactors, reducing catalytic reactors type DeNOx (for large excess air factor) using plasma technology,
- cooling systems with higher coolant temperature and decreasing engine warm-up time,
- heat accumulator,
- turbo-compound,
- high-power starter integrated with alternator,
- replacing mechanical and hydraulic fittings drive with electric drive,
- OBD-II,
- Increased fuel "clean combustion" – reformulated diesel fuels.

Technological progress in the field of diesel engines has led to a significant increase of interest in this type of drive.

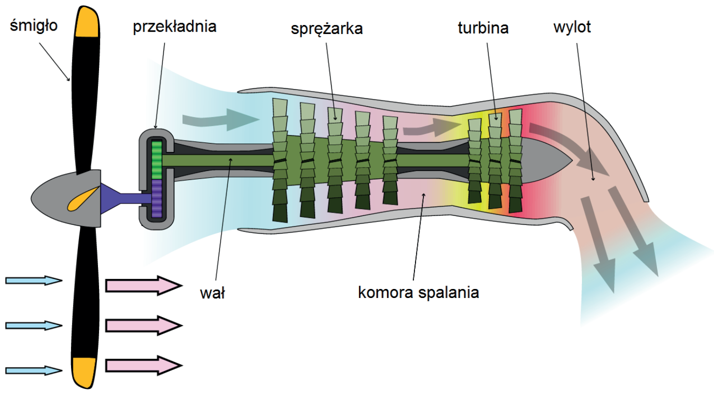
4. Turbojet engines

Turbine is a flow machine. Air is collected thought inlet duct and then compressed by compressor. In combustion chamber the fuel is injected, it mixes with the air and the mixture burns. Warm exhaust flows along turbine, it increases in volume and it transfers its energy to compressor driver and through the gearbox to propelled machine. In this kind of engines the basic fuel is kerosene. The efficiency of gas turbine based on Joule cycle is given by the same formula as efficiency of Otto cycle, for this reason, if the level of compression is the same, the efficiency of theoretical cycles of both engines are equal. This condition requires that the temperature of both cycles is the same, that is impossible with nowadays used materials due to a constant thermal load and unfavorable work conditions of turbine elements. Due to thermal load of alloy steel the permissible temperature before turbine is up to 850 Celsius degrees. Although the application of ceramic materials in production of turbine rotor and combustion chamber incandescent pipes can increase the opportunity of development of this kind of engines.

The turbine engine is characterized by mass power indicator that qualifies it as machine that requires light drive. It is used as a drive of flying machines like a plane or helicopter. Power output is received from engine main shaft, which is conjugated with prop. In turbine engine, the exhaust has low energy and its recoil does not affect flying machine driving significantly. The turbine coupled with compressor and with machine by reduction gearbox, which changes low in moment and high rotational speed into high torque and low rational speed.

The engine consists of listed below basic elements (Fig. 64 and 65):

- air inlet duct,
- compressor,
- combustion chamber,
- turbine,
- exhaust nozzle.



Legend:

Śmigło – Prop Przekładnia – Gearbox Sprężarka – Compressor
Turbina – Turbine Wylot – Exhaust Wał – Shaft
Komora spalania – Combustion chamber

Fig. 64. *The scheme of turbojet engine*

The basic element of the engine is compressor. There are applied centrifugal compressors (also called radial compressors) and axial compressor. The advantages of axial compressor in comparison with centrifugal are: high levels of compression, simple construction and small axial dimension. The number of compressions reaches 4,5 at such compressors (in axial compressors the maximum reaches up to 1,2-1,3 per level).

The gas turbine is the source of power necessary to compressor drive (the compressor absorbs 2/3 of turbine top power) and plant use which is transmitted to engine out shaft. In case when the distributed increase of enthalpy crosses the value of 320÷330 kJ/kg, the multistage turbine in which the overall decrease of enthalpy is distributed among the stages it consist of. The advantage of multistage turbines are: lesser energy loss associated with lower rate of flow speed. Moreover, there is lower peripheral speed of rotor that beside the unchanged rotational speed causes about double decrease its diameter and makes it possible to increase the length of rotor blades. That decreases the loss caused by leaks in radial clearance. What is more, if the diameter does not change, it is possible to decline the rotational speed and that makes the construction of reduction shaft simpler. To following advantages can be added to those above:

- in subsequent stages of the turbine the kinetic energy of flow at the outlet of the preceding steps is used,
- the sum of the disposable enthalpy decreases in turbine stages of the multistage turbine exceeds the disposable enthalpy decline across the turbine due to the recovery a part of heat of internal friction in the subsequent steps.

In contemporary gas turbine engines only multistage turbines (2 or 4 degrees) in the two-shaft systems are used, in which between the turbine unit driving the compressor and the unit producing engine power output lacks stiff coupling (only gas-dynamic coupling occurs).

Disadvantages of multistage turbine are complex (expensive) construction and higher work temperatures in first stages of turbine caused by smaller enthalpy falls and need of cooling blades. The most widest spread is air cooling of turbine components. The air can be delivered by engine compressor (it can be taken form particular stage of compressor, out of compressor or it can by secondary air for combustion chamber). Air delivery to combustion chamber requires several times more air than its theoretical requirement is due to need of exhaust, elements of combustion chamber construction and turbine cooling and often the use of air from combustion chamber for other needs.

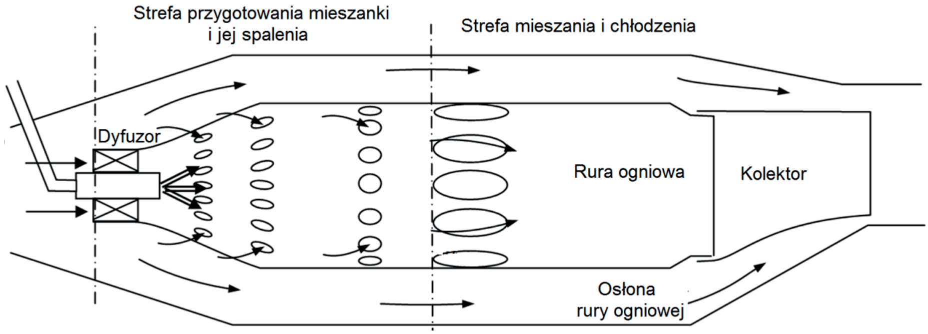
While the excess air ratio for combustion generally is in the range $\lambda = 0.9 \div 1.2$, the total value of excess air ratio is in the range $\lambda = 3.5 \div 5.5$.

The most stable combustions are given by rich mixtures $0.85 \div 0.95$, for which a turbine engine is less sensitive to automatically switch off in case of unstable operation of the compressor. The limit values of mixture composition are not critical and depend on many external factors, the engine operating range, the fuel spray quality and its type, specific properties of the combustion chamber, and others.

A turbine engine combustion chamber should provide conditions and space for the process of combustion and allows the engine load. The combustion chamber determines the axial and radial size of engine and strongly affects the stability, reliability and efficiency of the engine. In all kinds of the combustion chamber, the working volume consist of a flame tube placed within the combustion casing. The geometry of the two main units of the combustion chambers is basis for the classification. There are several types of construction of combustion chambers:

- tubular combustion chamber
- annular combustion chamber
- tubular-annular combustion chamber

The first basic component of the combustion chamber is a diffuser that reduces air flow rate to a value of $50 \div 80$ m / s with a simultaneous increase of the static pressure. The basic work-space creates a flame tube attached to combustion casing. A collector is responsible for adjustment of the exhaust stream to the turbine inlet diameter (figure 66). Functionally, the volume of the flame tube can be divided into the primary zone and dilution zone. There are no clear boundaries of these two zones.

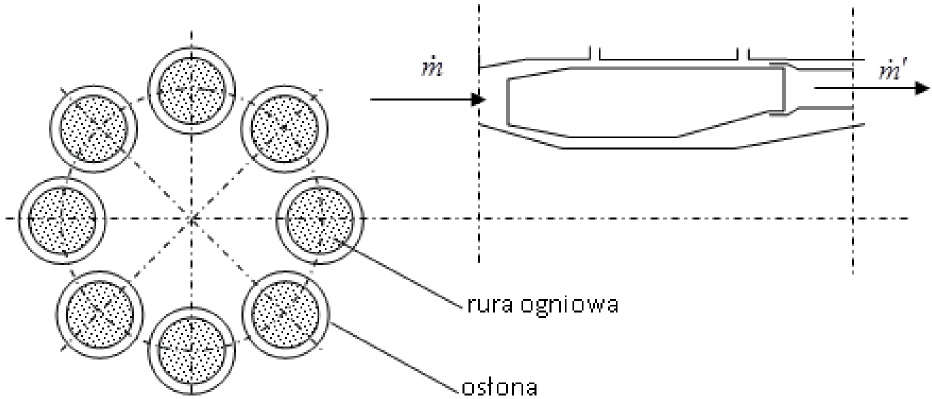


Legend:

Dyfuzor – Diffuser
Strefa przygotowania mieszanki i jej spalenia – Primary zone
Strefa mieszania i chłodzenia – Dilution zone
Rura ogniowa – Flame tube
Kolektor – Exhaust manifold
Osłona rury ogniowej – Combustion casing

Fig. 66. *Diagram of combustion chamber of turboprop engine*

Tubular combustion chamber is used in engines with radial compressors and small thrust. It is created by a group of (1 ÷ 18) individual tubes (figure 67). The processes are carried out separately in each tube and averaging of these processes (mainly the free passage of flame and pressure stabilization) is obtained by telescopic fittings tube connection.



Legend:

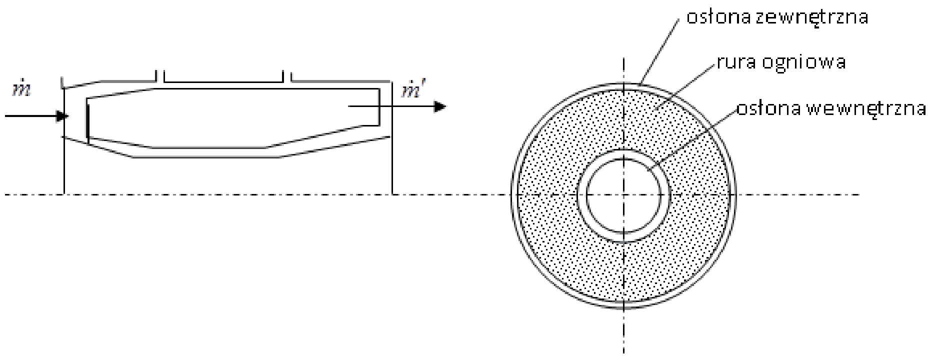
Rura ogniowa – Flame tube
Osłona – Combustion casing

Fig. 67. *Diagram of tubular combustion chamber of the turbine engine*

The main advantage of tubular combustion chamber is its easy disassembly.

The advantage of the tubular combustion chamber is easy disassembly and experimental validationof intra-chamber processes. The disadvantages include high flow resistance and high peripheral inequality of the temperature field at the outlet, increasing the radial dimensions of engines and overall increase of engine weight.

An annular combustion chamber is formed by flame tube with annular cross-section, located within the combustion casing (figure 68). In order to obtain sufficient and equal distribution of the fuel within the flame tube a high number fuel injectors (20 to 30) is used.



Legend:

Osłona zewnetrzna – Combustion outer casing

Rura ogniowa – Flame tube   Osłona wewnetrzna – Combustion inner casing

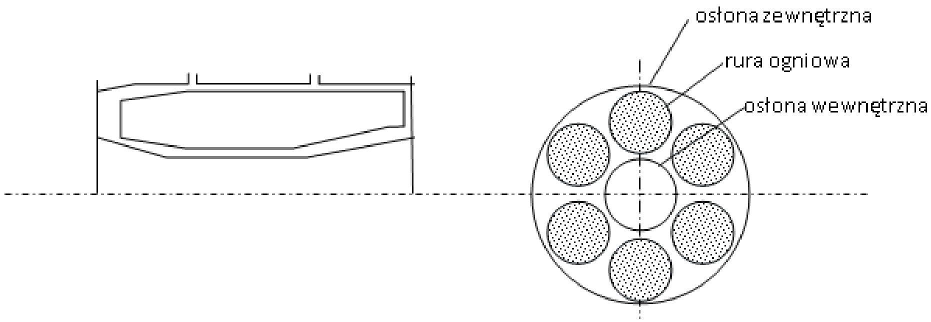
Fig. 68. *Scheme of annular combustion chamber of the turbine engine*

Advantages of annular combustion chamber:

- high content of construction , small size of radial
- the possibility of including casing in load-bearing frame of engine,
- low weight,
- low flow resistance,
- high equality of temperature outlet field.

The disadvantage of the annular chamber is complex assembly and disassembly (practically results in complete dismantling of engine), which means difficulty of carrying out service and research. Thus the accumulated experience of producers and new methods of checking the technical condition the defect is not bothersome.

Tubular-annular combustion chamber is an intermediate solution between the tubular and annular combustion chamber. This solution simplifies experimental improvement of intrachamber processes while including casing into the strength frame. Chamber consist of a few (6 to 9) tubes closed in an annual combustion casing (figure 69).



Legend:

Osłona zewnetrzna – Combustion outer casing
Rura ogniowa – Flame tube
Osłona wewnętrzna – Combustion inner casing

Fig. 69. *Scheme of tubular-annular combustion chamber of turbine engine*

Inside the combustion chamber there are undergoing processes of commutation and evaporation of fuel, mixing fuel (in both the liquid phase and gas) and air, combustion of so formed mixture and cooling the combustion products to the allowed limits before turbine.

Overgrinding of the fuel is obtained by the impact force accompanying the flow of fuel through the injector nozzles – vortex (in the main chambers) and plow (afterburner chambers). The diameter of the droplets depends on fuel flow rate through a injector and the pressure drop on the injector. To obtain small droplet diameters requires low flow rates and large pressure drop. The main combustion chambers of today engines use from a few to several injectors.

Within fuel overgrinding facilitates, the heating of the drops by warmth uploaded to areas of their location from the combustion zone by directed stationary swirls of return currents has a crucial role in this process. The evaporation process is also intensified by relative movement of the fuel droplets and air.

The process of air and fuel mixing begins from mechanical spread of droplets in volume of primary and dilution zone. A proper combustion mixture is formed as a result of blending fuel gases and air. At this stage the most important is turbulent mixing of flow microparticles.

Only mixture of fuel (gas) and air takes part in combustion process. The burning temperature strongly depends on air excess ratio and stability of process depends on mixture move conditions. The speed of mixture movement cannot be larger that propagation of flame front.

The process of cooling the products of combustion takes place in the primary zone and is the result of mixing of the hot exhaust with a large mass of cold secondary air. This process has a significant influence on the outlet temperature field.

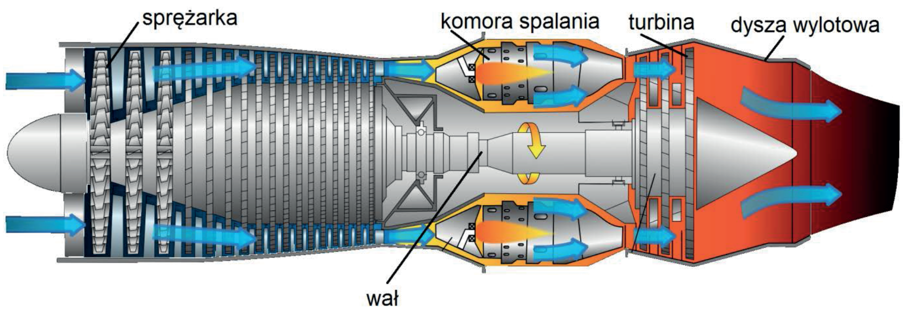
An intracombustion process runs at variable of excess air ratio and at variable and large flow rates, larger than the propagation speed of the flame front. In order to ensure proper combustion process, introduction of air into the flame tube must be graded in order to control the excess air ratio, which ensures stable combustion and to obtain at the outlet of the exhaust gas at a temperature acceptable for the turbine blades. Nevertheless, it should be locally created a reverse flow zone and reduced the flow velocity in order to ensure stable the conditions of combustion and improve fuel evaporation. Graduation of air introduction is carried out by dividing the input stream into the combustion chamber at the primary and secondary stream.

A small part of the primary flow (about 4 to 12% of mass) flows into the flame tube through an inlet next to the fuel injector and is swirled. The major part of the initial flow is introduced into the flame tube along primary zone. The primary air is approx. 20 to 50% of mass of air supplied to the combustion chamber. A method for introducing primary air is essential in the formation of mixture composition variation in the combustion zone. At the end of the combustion zone $\lambda = 1.7 \div 1.8$ is achieved, resulting in a exhaust gas temperature of about $1800 \div 2000$ K.

Secondary stream forms required λ distribution factor along the primary zone and forms a protective layer for the flame tube wall from exposure to high temperatures. The secondary air is injected in small quantities by a large number of openings or slots. The edges of the outlet and exhaust manifolds are cooled by passing a small amount of air through the annular gap between the fire tube and the inlet to the collector. In order to reduce the temperature of combustion products and allow for suitable distribution of the temperature field in outlet, air is introduced by large holes at high speeds (60 to 100 m/s). The end of the cooling process should take place at the end of the flame tube length.

Radial and peripheral non-uniformity of the temperature field in the combustion chamber is caused by variations in the quality of prepared fuel-air mixture, different course of combustion process, the impact of local factors and others. The peripheral non-uniformity is undesirable. The radial non-uniformity can be beneficial, because of the heat load of the turbine blades, when the highest temperature occurs in 50 to 70% of the height of the blade.

The simplest is single thrust turbine turbojet engine (Figure 70). As in the case of a turboprop engine, air is sucked and compressed by the compressor (axial or radial, single or multi-stage). Then it flows into the combustion chamber (or several chambers arranged circumstantially around the axis of the motor) where the injectors upload fuel (eg. kerosene). Fuel evaporates, heats up and lights from the hot exhaust. Cold start requires a spark plug. Combustion of the fuel raises the temperature of the working gas in the combustion chamber. The hot combustion gases drive the turbine and this drives a shaft coupled to the compressor. Turbine does not provide power to the driven machine, it serves only to keep the engine running. Increased pressure and temperature of the working gas results in its outflow through the outlet nozzle and the rise of engine thrust.

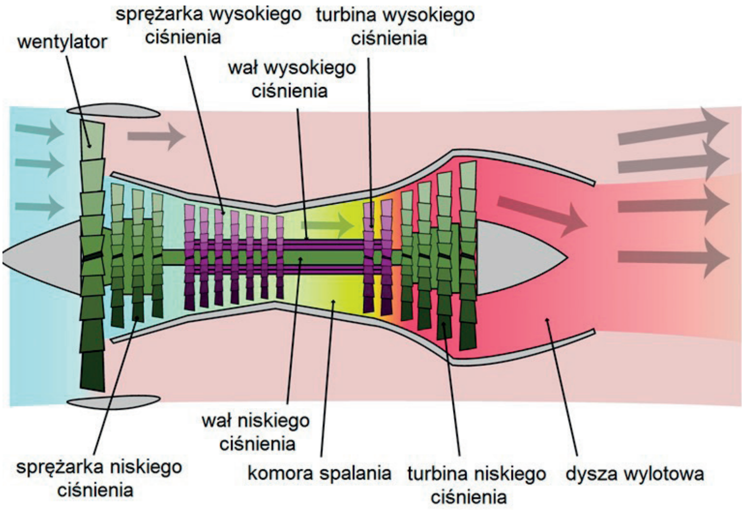


Legend:
Sprężarka – Compressor
Komora spalania – Combustion chamber
Turbina – Turbine
Dysza wylotowa – Nozzle
Wał – Shaft

Fig. 70. *Diagram of turbojet*

The reduction fuel consumption per thrust unit in comparison to turbojet allows the separation of the primary air flow to the internal and external flow after the first stage of the engine compressor (or first steps, if there are several). An external flow bypasses further compressor steps following the channel along the entire engine directly towards the outlet nozzle of the engine. An internal flow is directed to all stages of the compressor and takes part in the fuel combustion. It transmits the gas energy through multilevel turbine to the compressor shaft, and then flows to the nozzle to generate thrust. Turbofan engine driving efficiency is greater than the efficiency of turbojet singlethrust drive due to higher output and lower air flow ratio and as result lower the kinetic energy of the stream leaving the engine, thus the lower the energy loss.

A turbofan engine is characterized by the smallest fuel consumption. It is equipped with a large diameter fan (first part of compressor) that pre-compresses the air. The fan produces a thrust in a manner similar to a propeller driven by eg. piston engine. Turbofan engines tend to have a twin-shaft structure (coaxial shafts) connecting suitably the degrees of the low pressure turbine with fan and the high pressure turbine stages with high pressure compressor (figure 71).



Legend:

Wentylator – Fan
Sprężarka wysokiego ciśnienia – Low pressure compressor
Wał wysokiego ciśnienia – High pressure shaft
Turbina wysokiego ciśnienia – High pressure turbine
Sprężarka niskiego ciśnienia – High pressure compressor
Wał niskiego cisnienia – Low pressure shaft
Komora splania – Combustion chamber
Turbina niskiego cisnienia – Low pressure turbine
Dysza wylotowa – Nozzle

Fig. 71. *Diagram of turbofan engine*

Fig. 72. *View of turbofan engine*

The main drawback of the turbine engine is reduction of its efficiency with load decrease. This disadvantage can be eliminated by the adjustment of blades angles.

An effective way of improving the economy of the turbine engine, especially at partial loads, is the application of a heat exchanger, which recovers some of the heat from the exhaust gas flowing into the environment in order to raise the intake air temperature. The possibility of competition of turbine engine with reciprocating engines depends on improvement of design of heat exchangers. The application of a suitable heat exchanger is a simpler way than eg. increasing the efficiency of the compressor and turbine, at this stage of engine development . Due to the way the heat exchange the heat exchanger should be devised into recuperators (without heat accumulation) and regenerators (with heat accumulation). The recuperator is a membrane heat exchanger, in which air and exhaust gas are separated by a fixed diaphragm in the form of flat or cylindrical wall. The most common in machines are plate recuperators. The efficiency of heat recovery of recuperative heat exchangers is 0.75-0.82.

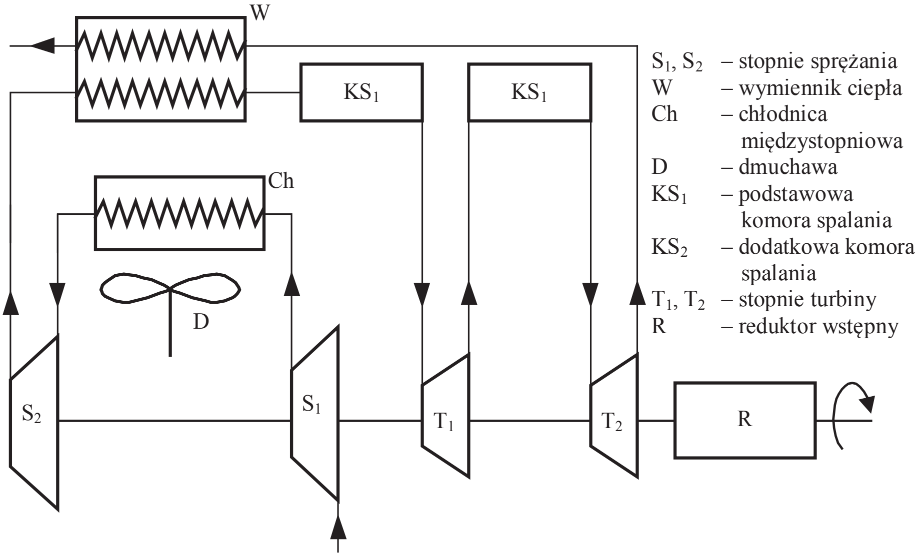
The regenerator in a turbine engine is heat exchanger, in which exhaust gases and the air flow alternately through the same part of a rotating cylinder with a high heat capacity. Part of the tank is located during circulation in exhaust gas flow channel heat accumulates from them, and then heat is given to colder air flow in flow channel. As a principle the flow of these two factors is counterflow. The regenerative heat exchangers can achieve the highest degree of heat recovery up to 0.92, while maintaining the lowest overall dimensions of all types of heat exchangers.

Rotational speeds of turbine drive rotors of driving machines are within the range of 25.000-50.000 rpm. Such high speeds require the use of pre-reduction gears. The gear ratios is selected so that the output shaft speeds are 3000-6000 rpm. Further reduction ratio is obtained in the gearbox of the machine.

Turbine engines are equipped with automatic controls for the rotational speed of the turbines. In the case of achieving the maximum permissible speed of the turbine fuel flow is decreased by the controller, allowing you to maintain a predetermined constant speed. If reaching the maximum speed refers to the driver turbine, then it is due to the reduction of fuel flow speed and turbine power driven compressor should be reduced, while maintaining a constant speed driver turbine.

The maximum speed of a double-shaft engine exceeds 20-30% the value of the rotational speed equivalent the highest power output of the engine. Double-shaft engine torque is more useful throughout whole engine speed range than for a reciprocating engine or turbine single shaft. Working time is roughly rectilinear, although the maximum value is almost 2.5 times greater than a corresponding value of the maximum engine power.

Thanks to using the latest technology, in recent years there has been a sharp drop in specific consumption of turbine engines fuel. The use of air cooling (supplied to the compressor) and a heat exchanger (for heating air supplied from the compressor to the combustion chamber) lead to an increase in the efficiency of the engine cycle (figure 73). Compact turbine engine for automotive applications should be characterized by: ceramic, catalytic, multi-fuel combustion chamber, composite compressor, ceramic turbine, air bearings, ceramic heat exchanger. Heat exchanges can be a ceramic recuperator or regenerator. Turbine equipped with air-cooled heat recovery units (ICR – intercooled-recuperated) characterized by 30-40 percent reduction in unit fuel consumption in comparison with turbines without air cooling and heat recovery from the exhaust outlet.



Legend:

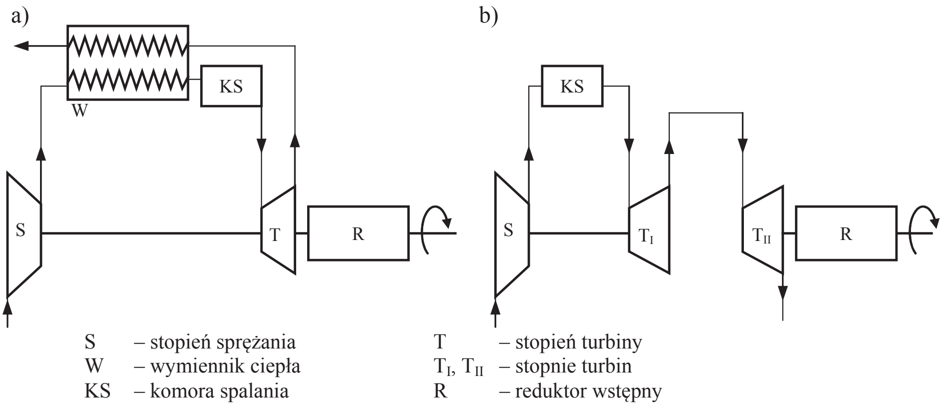
S 1 , S 2 – Compression levels
W – Cheat exchanger
Ch – Intercooler
D – Blower/superchanger
KS 1 – Primary combustion chamber
KS 2 – Additional combustion chamber
T 1 , T 2 – Turbine stages
R – Primary regulator

Fig. 73. *A block diagram of turbine engine with cooling and combustion intercooling*

As it has been already indicated, gas turbine engines can be divided, in terms of the drive system construction, into two types (Figure 74):
- single-shaft,
- two-shaft.

Torque generated in the rotor of the turbine (Figure 74a) is partially transmitted to the drive compressor, the rest is transferred to the machine drive. In the second case (Figure 74b) in the turbine TI exhaust gas partially expands, resulting in a torque value equal to the torque necessary to drive the compressor and aggregate units. The overall expansion of the exhaust occurs in the turbine T II (driving), which transmits torque to machine drive. Introduction of a separate power turbine and the application of two rotors rotating independently from each other complicates the structure of the engine. However, a much more favorable course of useful torque to the change in the rotational speed the engine drive shaft compensate these disadvantages. In the double-shaft system turbine TI rotor rotates in the whole range of engine run at a rotational speed close to the nominal.

The use of two-shaft construction allows to achieve better acceleration of machine. The disadvantage of this system is the small engine braking torque, which, can be increased by the application of adjustable power turbine vanes.



Legend:
S – Compression level
W – Cheat exchanger
KS – Combustion chamber
T – Turbine stage
T I , T II – Turbine stages
R – Primary regulator

Fig. 74. *The drive transmission schemes of: a) single-shaft system, b) a two-shaft system*