Power units - drives by internal combustion engines

In practice, agreed, so-called internal combustion engine benchmarks are used for closer classification, assessment and application of internal combustion engines to drives, describing other properties and values needed to compare engines with each other regardless of size, power, type of construction, principle of operation, etc. The characteristics of internal combustion engines are used for quick and easy orientation in the basic operating values of internal combustion engines.

Several basic quantities are used to comprehensively assess the work and economy of an internal combustion engine. Such variables include, in particular, mean effective circulating pressure, engine power, specific consumables and engine efficiency.

Performances and their detection

Types of powers

The performance of an internal combustion engine is assessed from various perspectives. For the user of the internal combustion engine or the machine or car in which the engine is installed, the basic data in the technical documentation is power, either in numerical form as rated power at rated speed or in graphical form, most often as external speed (speed) characteristic. It is possible to read the values of power and other important operating variables from the graphic course depending on the speed at the maximum fuel supply.

These data represent parameters guaranteed by the manufacturer or supplier for the specified service life and normal operating conditions, while tolerance limits are also given in which these values may vary.

In professional practice, when assessing the processes related to the conversion of energy in the engine, we most often encounter the following types of performance:

Indicated power

The indicated power represents the value obtained by realizing the working cycle of the internal combustion engine inside the working space, without considering losses. For a single-cylinder internal combustion engine:

$$P_i = \frac{p_{is} \cdot V_z \cdot 2 \cdot n}{z}$$

Pi - indicated power,

pis - the mean value of the indicated pressure

Vz - cylinder displacement

n - crankshaft rotation frequency per second

z - number of strokes per cycle (for four-stroke engine z = 4, for two-stroke z = 2).

The mean indicated pressure represents the imaginary constant mean value of the pressure acting on the piston at which work would be performed in one stroke, equal to the magnitude of the indicated work of one circuit.

For preliminary and approximate calculations, the following mean indicated pressure values may be used for four-stroke engines at full load:

petrol, non-supercharged, petrol   pis = 0.8 to 1.2 MPa

- petrol, non-supercharged, gas   pis = 0.5 to 0.7 MPa

- diesel, non-supercharged   pis = 0.6 to 1.05 MPa

- diesel, supercharged   pis = 0.8 to 3 MPa

Effective power

Effective (useful) power is the power found at the power consumption point, at the output end of the main shaft of the engine, or at its coupling flange, if all auxiliary devices directly connected to the engine are working, absolutely necessary for its own, regular and continuous operation. It represents the value of engine power usable for driving appliances. Its size is reduced by the indicated power

• losses due to friction of moving engine components,

• losses due to the drive of engine auxiliaries,

• hydraulic losses,

• losses associated with changing the cylinder charge.

If the total power loss is denoted P_s, then the effective power on the crankshaft is equal to the difference between the indicated power P_i and the power loss P_s

$$P_e = P_i - P_s \,.$$

The effective engine power can also be expressed by the mechanical efficiency of the engine η_m.

$$P_e = P_i \cdot \eta_m$$

By analogy, the mean effective pressure can be expressed, which is one of the most important comparative indicators of internal combustion engines.

$$p_e = p_{is} \cdot \eta_m \,.$$

The mean effective pressure is the constant mean value of the pressure acting on the piston at which work would be performed in one stroke, equal to the magnitude of the effective work of one cycle.
Its values in current engines are within the limits for engines:

- 2-stroke petrol: 350 550 kPa,
- 4-stroke: 650 100 kPa,

- 2-stroke diesel: 300 650 kPa,
- 4 strokes: 550 750 kPa,

- supercharged 700 1800 kPa.

Power dissipation

The size of the power dissipation of the internal combustion engine includes the resistances against the movement of the engine components, and the power inputs of the auxiliary equipment of the internal combustion engine necessary for its operation, for example:

- equipment for fuel transport and mixture preparation,

- timing mechanism,

- cooling system equipment,

- lubrication system equipment,

- electrical equipment of the engine,

- supercharging system equipment,

- control and regulation systems.

These losses are also expressed in terms of mechanical efficiency η_m, which is the ratio of effective to indicated power.

The calculation of power dissipation is complicated because it depends on many factors, such as the type and temperature of the oil, the mechanical condition of the engine and its accessories, the speed, the thermal condition of the engine, etc. For the simplified calculation of P_s and comparison of different engines, the so-called mean loss pressure p_{ss} is defined, analogous to p_{is}. It represents a constant mean value of the pressure acting on the piston, but against its movement, which would require labor in one stroke, equal to the size of the loss-making work per circulation.

$$P_s = \frac{p_{ss} \cdot V_z \cdot 2 \cdot n}{z}$$

p_{ss} - mean loss pressure,

Experimental research has shown that the Pss value depends mainly on the mean piston speed. The results of the measured dependencies are processed and expressed in the form of empirical formulas or in graphical form in the professional literature.

When determining power by measurement, it is important to take into account the engine equipment, marked as engine accessories. These are elements directly connected to the engine, necessary for operation for a given drive, but not necessary for the actual operation of the engine (air cleaner, cooling fan, muffler, etc.). Therefore, a description of the engine equipment during the test is required, inter alia, in the performance measurement protocol. This issue was also reflected in the development of relevant international standards. For example, the German DIN standard considers a motor with accessories, ie with increased Ps, ie lower effective power Pe. In contrast, the SAE standard considers a motor without accessories when measuring, and thus has a higher effective power Pe than the DIN standard for the same motor. The relationship between these two power values is

$$P_{eSAE} = (1{,}1 \text{ to } 1{,}15)\ P_{eDIN}$$

In addition to the listed outputs, some other output values related to the operation, measurement and calculations of internal combustion engines are defined for the theoretical and experimental investigation of internal combustion engines.

Rated power - the maximum useful power guaranteed by the manufacturer and indicated on a label that the engine can develop for a specified time, t. j. permanently or intermittently, as appropriate for its intended use and loading, under the following conditions:

a) at rated speed,

b) at the nominal state of the air at a vacuum in the sledge not exceeding the specified limit value,

c) in case of overpressure in the exhaust (behind the exhaust manifold, or in case of turbocharging behind the turbine) not exceeding the specified limit value.

Maximum power - the maximum useful power that the engine can produce without harmful mechanical or thermal stress for at least 15 minutes, if it was before and will still be permanently loaded under the conditions applicable to rated power.

Continuous power - the maximum useful power that the engine can produce under continuous power conditions, in continuous operation, for 100 hours, with the possibility of overload, as specified in the following paragraph.

Overload (overload power) - power increased usually by 10% compared to permanent. This is the highest useful power available for uninterrupted operation. It may be developed by the engine under rated power conditions for a total of one hour (continuous or intermittent) in any period of six consecutive hours if it is subjected to continuous power in the meantime.

Reduced power - obtained by recalculating the measured power in the given conditions, to nominal conditions agreed by the standard. For example, it sets the following agreed benchmarks for car engines:

- barometric pressure

- air temperature

- relative humidity

- the temperature of the cooling water at the inlet to the charge air cooler.

Temperatures are given in Kelvin and barometric pressure in kPa. The reduced power for atmospheric conditions during the measurement (given by the pressure patm, the temperature Tatm) will be

$$P_{er} = P_e \,.\, \alpha$$

The coefficient α is realistically applicable in the range of 0.96 to 1.06.

Displacement power - the ratio of rated power to engine displacement expressed in liters, better known as liter output

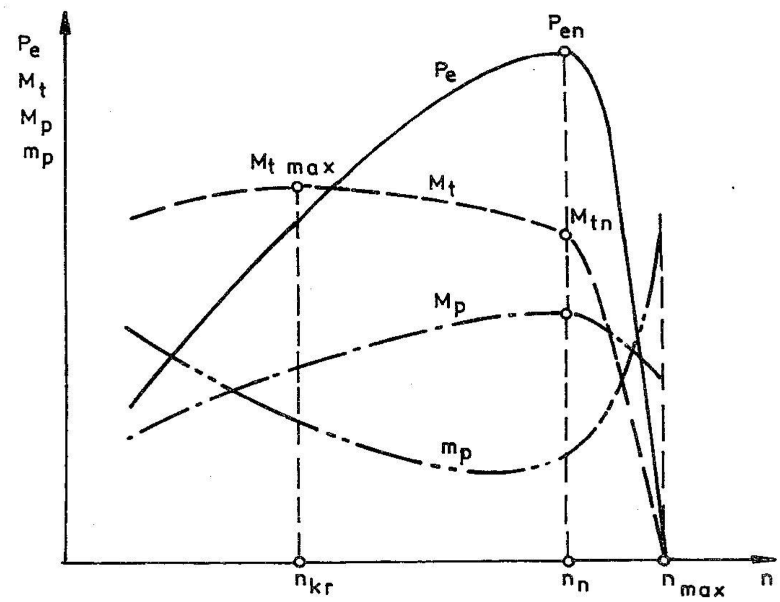
$$P_1 = \frac{P_m}{V_z} \qquad (\text{kW.dm}^{-3})$$

Power weight ration - the ratio of the dry engine weight to the corresponding rated power.

$$M_1 = \frac{M_m}{P_m} \qquad (\text{kg . kW}^{-1})$$

Engine evaluation according to speed characteristics

We follow the waveform of the diesel engine speed controller with the power controller from right to left. When the engine is fully relieved and the accelerator lever is set to the stop, the engine reaches a maximum speed of n_{max} and the controller adjusts the fuel supply to the amount needed to cover internal losses in the engine. Since the engine is not loaded on the crankshaft, the effective power P_e is zero, so the specific fuel consumption m_p is not defined (it is infinitely large). As the engine gradually loads, the speed starts to decrease and the regulator increases the fuel supply. It tries to keep the speed at n_{max}. With increasing fuel supply, performance increases sharply.



The speed at which the motor reaches the maximum effective power P_{en} is called the nominal engine speed n_n. At this speed, the controller allows the maximum fuel supply, which corresponds to the setting of the accelerator lever.

The torque M_{tn} in the nominal speed range has a similar course as the effective power P_e. It is rising sharply. It reaches the nominal value at low speed. The speed characteristic range between maximum and nominal engine speed is the control area in which the controller is active and adjusts the fuel supply according to the load.

Increasing the engine load even after reaching the nominal engine speed represents its overload. The overload corresponds to the area from the nominal engine speed to the left. Speed and efficient power are reduced due to a constant supply of fuel. Burning is prolonged and volumetric efficiency increases with decreasing speed. This causes the torque to rise up to the critical speed n_{kr}, where it reaches the maximum value M_{tmax}. To the left of this speed is an unstable area in which the engine can no longer run.

The best economic indicator of the efficiency of internal combustion engine operation is the specific fuel consumption m_p, which is in the range of 240 to 250 g.kW-l .h-1 for four-stroke diesel engines with direct fuel injection. You can see in the picture that it acquires the minimum value in the area of maximum performance. From an economic point of view, it is therefore important for the engine to operate in the area of minimum specific consumption, in the area of maximum performance. From the characteristics of the internal combustion engine we can deduct the necessary values to calculate other very important criteria for assessing the characteristics of the engine.

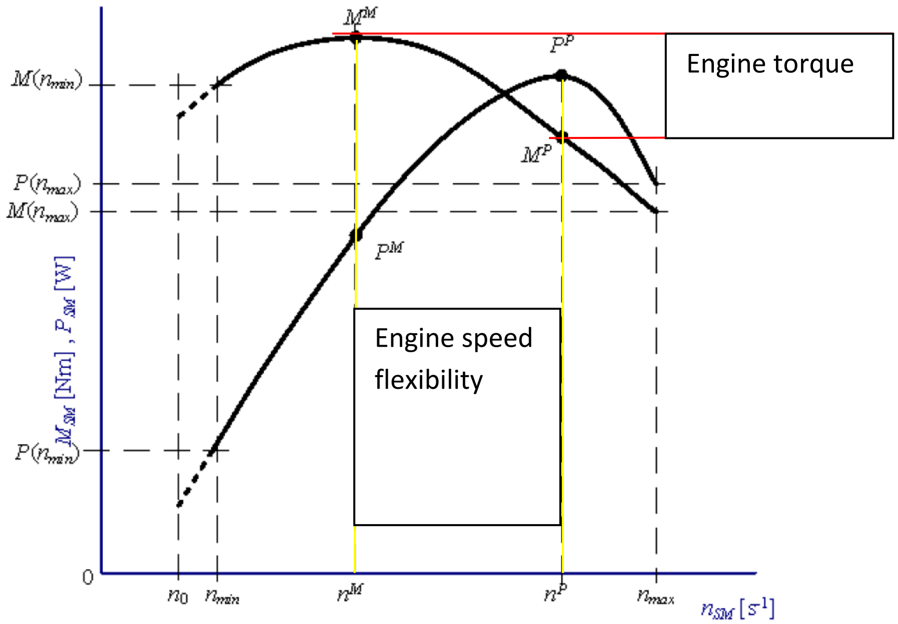
Engine flexibility

It expresses the adaptability of the torque of the internal combustion engine to its load in the part of the speed characteristic between the speeds n_m and n_p, which correspond to the maximum torque M_{max} and the maximum power P_{max}. The overall flexibility of the engine is defined by:

$E = e_m . e_n$

Engine torque: $e_m = M_{max}/M_p$

Engine speed flexibility: $e_n = n_p/n_m$



COMPARATIVE VALUES

The degree of unevenness of the controller σ is determined from the relationship

$$\sigma = \frac{2(n_{\max} - n_n)}{n_{\max} + n_n}$$

This value shall not exceed 0,038 to 0,1 for the tractor engine.

The coefficient of adaptability is given by the relation

$$\kappa = \frac{M_{t\max}}{M_{tn}}$$

M_{tmax} - maximum torque (N.m),

M_{tn} - nominal torque (N.m).

Individual types of internal combustion engines reach values

- compression ignition engines κ = 1,05 to 1,15,

- positive ignition engines κ = 1.15 to 1.20.

The speed non-uniformity factor ρ_n is given by the relation

$$\rho_n = \frac{n_{\max}}{n_n}$$

Critical speed to nominal ratio (should be 0.6 to 0.7)

$$\nu = \frac{n_{kr}}{n_n}$$

The motor overload reserve is given by the relationship

$$R = \frac{M_{t\max} - M_{tn}}{M_{tn}}$$

The engine power utilization factor η_{vm} is given by the relationship

$$\eta_{vm} = \frac{P_e}{P_{en}}$$

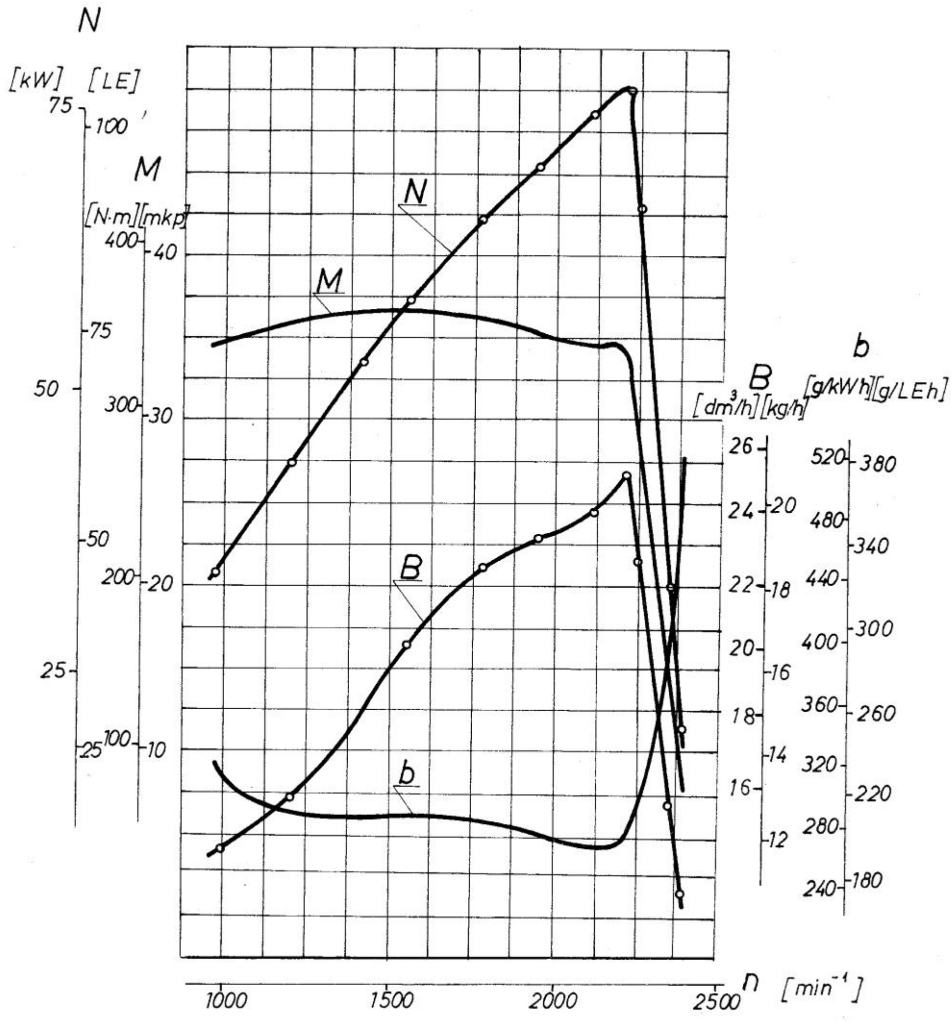
The picture shows the complete speed characteristic of the Zetor 8601 engine, which was fitted with the Z12045 and Z12011 tractors.

- M torque curve,

- N power curve,

- B hourly fuel consumption,

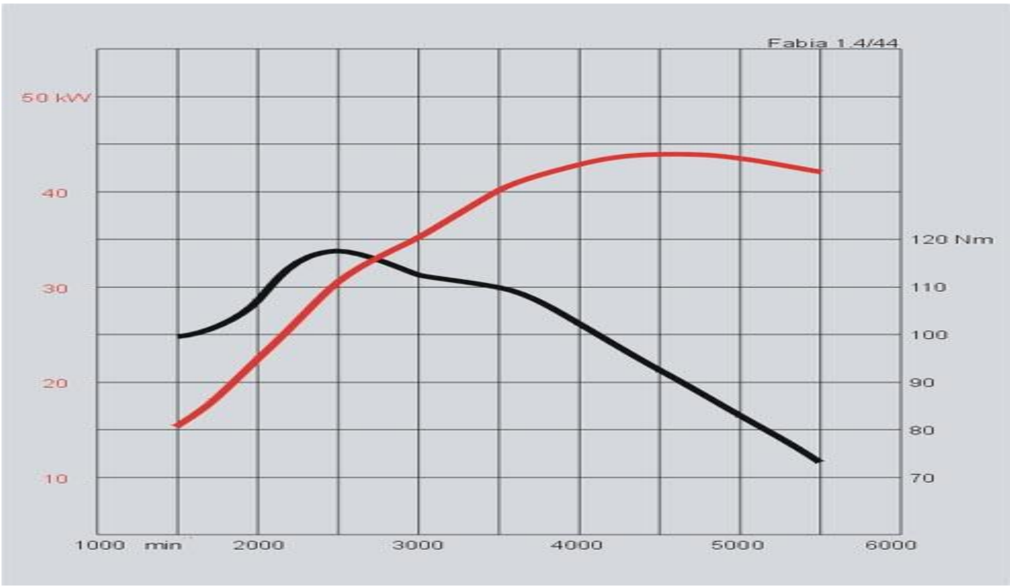
- b specific (specific) fuel consumption.



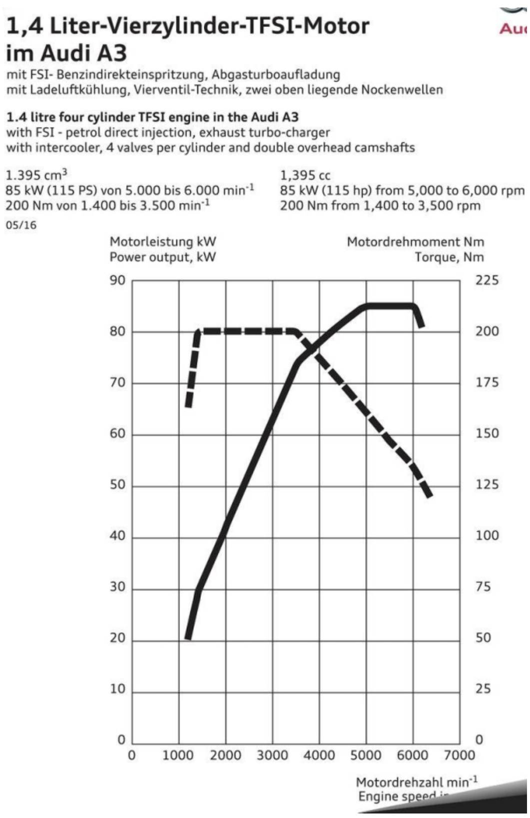
Internal combustion engine development trends

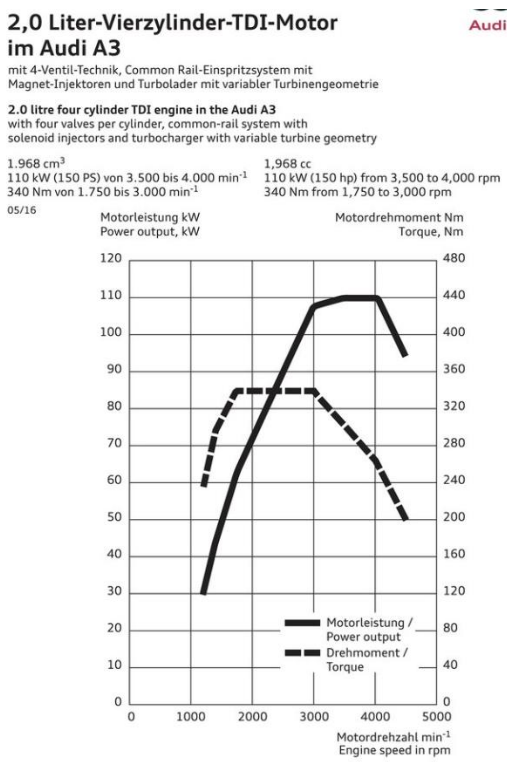
Spark ignition engines are usually characterized in that the minimum specific fuel consumption is in the speed range with a maximum Mk.

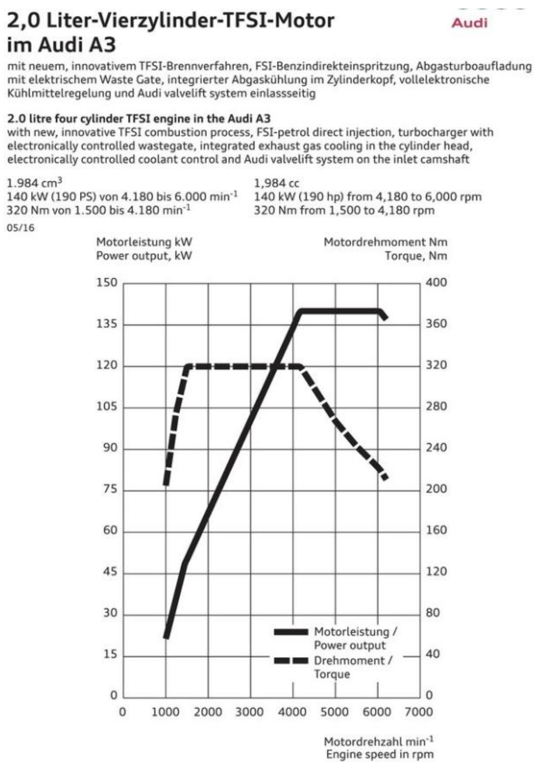
Comparison of engines of the same volume. - criterion year of manufacture



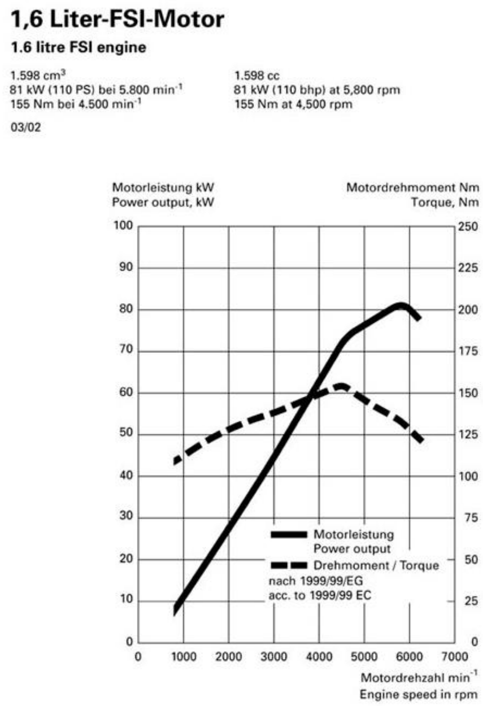
Courses of torque Mk and reduced effective power Pe for the Škoda 1.4 / 44kW MPI engine.

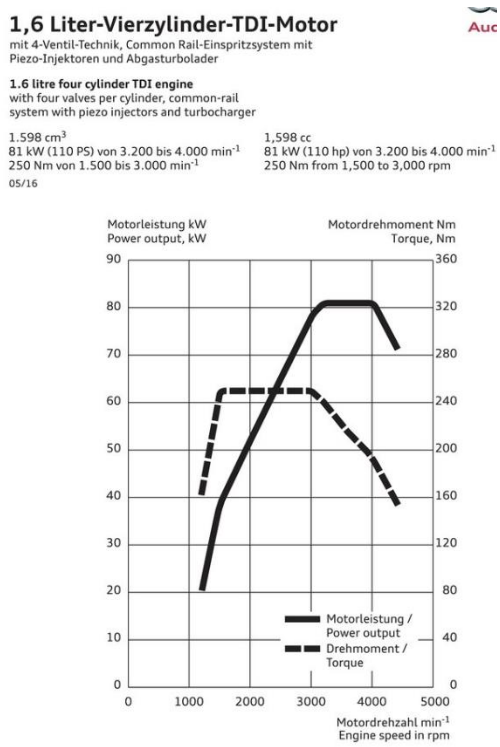






Comparison of engines of the same volume. - criterion year of manufacture and type of fuel



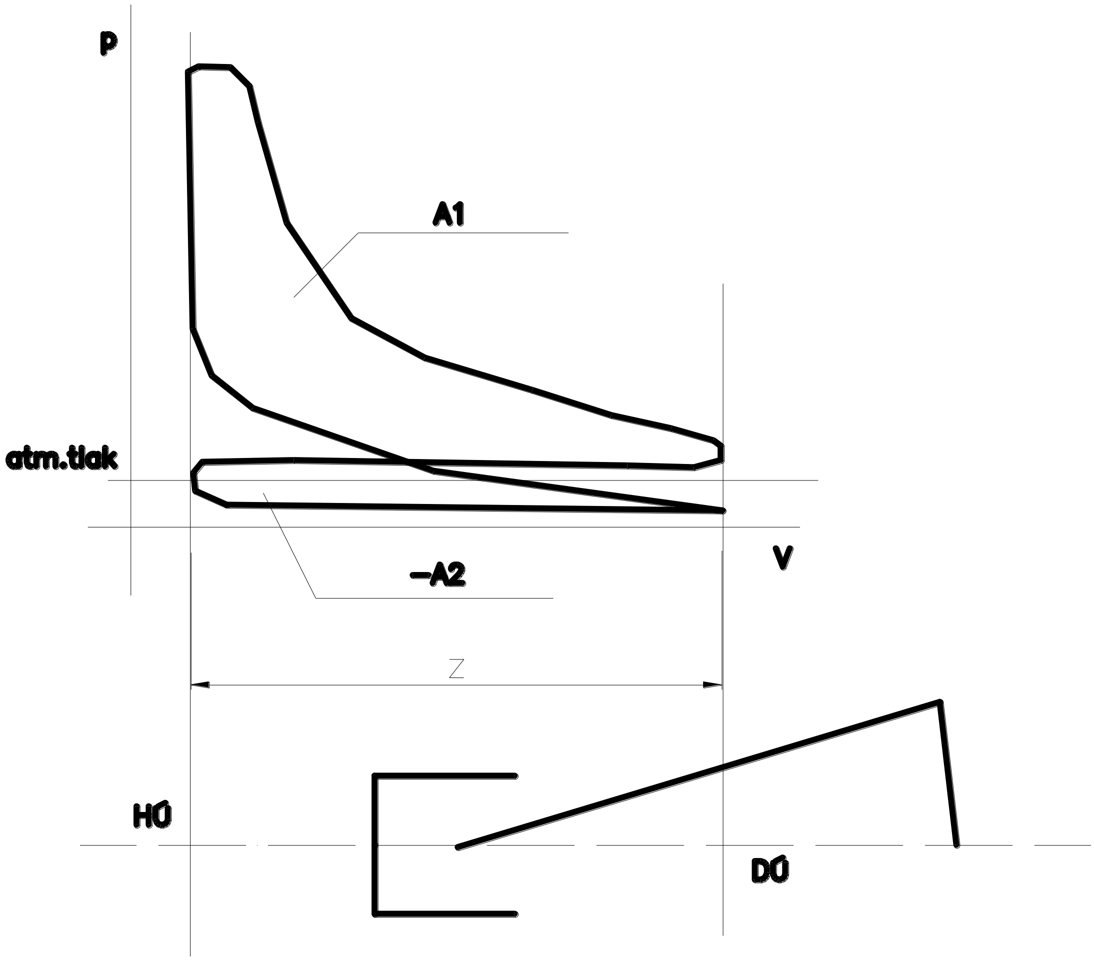


Defining engine power

The determination of engine power can only be made under a few conditions. The magnitude of the output depends on the amount of heat that is released per unit time by burning the mixture in the working space of the cylinder.

We can perform a separate calculation of engine power in various ways. The simple way is to use the average values that are achieved with an engine of the same type.

In a single-cylinder engine, the work delivered to the crankshaft is determined by the pressure diagram of FIG. by the difference in the amount of work consumed and performed, which can be determined from the areas of diagrams A1, A2. The variable pressure in the cylinder can be replaced by the mean indicated pressure. The mean indicated pressure pi is a constant pressure that would do as much work in one cycle as the variable pressure in the cylinder.



We can then determine the work during one stroke as follows:

$$A = p_i . V$$

Where:

V- cylinder volume [m3]

The indicated engine power is the power inside the cylinder and is given by the amount of work per unit time as follows:

$$P_i = A . n . a$$

Where:

n - engine speed [s-1],

a - a constant expressing the number of expansions per revolution of the crankshaft. For two-stroke engines a = 1 for four-stroke engines a = 0.5.

The indicated power is transmitted to the crankshaft. The power we receive on the crankshaft is called the effective power Pe. This is power reduced by engine losses:

$$P_e = P_i \cdot \eta_m$$

Where:

η_m - mechanical efficiency of the engine

HCCI Engine

In HCCI mode of combustion, the fuel and air are mixed prior to the start of combustion and the mixture is auto-ignited spontaneously at multiple sites throughout the charge volume due to increase in temperature in the compression stroke. In this mode, the combustion process is arranged in such a way that the combustion takes place under very lean and dilute mixture conditions, which results in comparatively lower bulk temperature and localised combustion temperature, which therefore, considerably reduces the NOx emissions. Furthermore, unlike conventional CI combustion, in HCCI mode the fuel and air is well mixed (homogeneous). So, the absence of fuel rich regions in the combustion chamber results in considerable reduction in PM generation. Therefore, due to absence of locally high temperatures and a rich fuel-air mixture during combustion process, the simultaneous reduction of NOx and PM emissions is made possible.

Fuel	SI (gasoline-like fuel)	CIDI (diesel-like fuel)	HCCI (flexible fuel)
λ	≈1.0	=1.2-2.2	> 1.0
Mixture preparation	PFI, GDI	DI	DI, PFI & DI+PFI
Ignition	Spark ignition	Auto-ignition	Auto-ignition
Combustion form	Premixed	Diffusion	Diffusion but dominated by chemical kinetics
Combustion rate limitation	Flame propogation	Mixing rate	Multipoint or spontaneous
Flame front	Yes	Yes	Without
Combustion temperature	High	Partially high	Relatively low

Table 1 gives a brief comparison of the characteristic features of SI, CIDI and HCCI combustion modes. The advantages of HCCI combustion can be summarized as follows:

1. HCCI combustion uses a higher compression ratio, lacks a threshold value and has a shorter combustion period and faster combustion rate. Thus, it nearly achieves constant volume combustion and a higher thermal efficiency because of lower combustion temperature and lower radiation loss
2. There is no flame front or flame spread or local regions of excessively high temperature and rich mixture, thereby reducing soot and NOx generation to a very low level
3. Variety of fuel types can be used in this mode of combustion

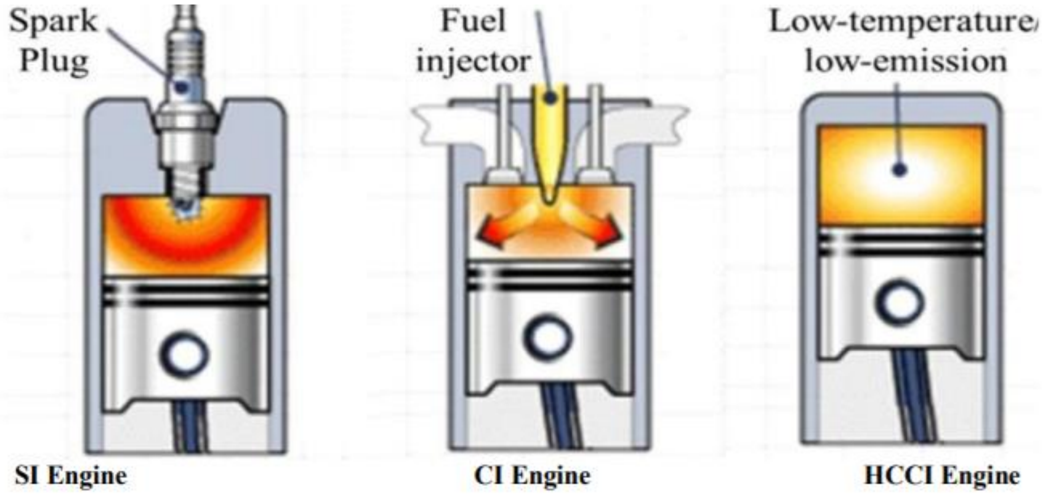


Fig. 1 represents the comparison among SI, CI and HCCI operations. In HCCI engines, a leanhomogeneous flammable mixture (fuel–air equivalence ratio $\Phi<1$) is formed, before the start of ignition and is auto ignited as a result of increase in temperature in the compression stroke. The HCCI operation is similar to SI engine as both use the homogeneous charge for combustion and similar to CI engine as both depend on the auto ignition of the mixture. Thus, HCCI combustion can be regarded as the hybrid of SI and CI combustion processes. In SI engines, three zones of combustion namely burnt zone, unburned zone and a thin flame reaction zone in-between are generated for turbulent flame propagation through the cylinder. In CI engines, fuel is diffused into the cylinder and a definite diffusion flame traverses within the cylinder. Whereas, in HCCI engine spontaneous auto-ignition of whole cylinder mixture at multiple sites occurs without any diffusion flame or flame front propagation. Theoretically, diesel fuelled HCCI combustion has the potential to reduce PM and NOx emissions to near-zero level by employing two basic processes: firstly, by forming a homogeneous mixture and secondly, by auto-igniting this mixture due to compression heat. However, these same features also lead to the main challenges. As diesel fuel possesses high viscosity, a wide range of boiling points, and a high cetane number. It means that the required mixing time scale for forming a homogenous mixture is very long but the chemical ignition time scale is very short. Furthermore, fuel-wetting of diesel-fuelled HCCI combustion is also an issue under consideration.

Challenges of HCCI Combustion

In spite of several inherent advantageous features of HCCI combustion, there are some unresolved issues that have kept the HCCI engine from being applied in commercial engines. The most difficult hurdle is the control of ignition. The control of ignition is principally more problematic as compared to the direct control mechanism such as spark plug or fuel injector used in SI and CI engines respectively to control the ignition timing. In HCCI mode, the ignition is controlled by the charge mixture composition and its temperature history. The main challenges of HCCI combustion are stated as follows:

The difficulty in combustion phasing control

Unlike conventional combustion mode as in SI and CI engines, the HCCI combustion lacks in direct method for controlling the combustion. In fact, in HCCI the start of combustion depends upon the auto-ignition chemistry of the mixture, which therefore is affected by the properties and the time-temperature history of the mixture. Therefore, the combustion phasing in HCCI engines is influenced by the several factors like autoignition properties of the fuel, fuel concentration, residual rate and possibly, reactivity of the residual, homogeneity of the mixture, compression ratio, intake temperature, latent heat of vaporization, engine temperature, heat transfer to the engine and other engine dependent parameters.

High levels of UHC & CO emissions

HCCI combustion generates inherently lower NOx and PM emissions at low loads but comparatively higher HC and CO emissions at low to medium loads as well as high NOx at high loads. In this mode a large fraction of the in-cylinder fuel is accumulated in the cylinder crevice region during the compression stroke and is therefore remains unburned. Furthermore, the larger part of this unburned fuel still remains unburned when it re-enters into the cylinder during the expansion stroke, as the temperature of the burned gas is too low. This leads to considerable increase in both HC and CO emissions as compared to the conventional combustion. Besides, the maximum temperature of the burned gas is not high enough (lower than 1400K or 1500K) to oxidize the CO to CO_2 at low loads and hence the combustion efficiency deteriorates.

Range of operation

HCCI combustion performs satisfactorily only in the limited operating range. In this mode, controlling the ignition timing over the full range of speed and load is a challenging issue. The range of operation depends mainly on the auto-ignition properties of the fuel, engine geometry and the operating parameters. Part /light load operation suffers from the lack of sufficient ignition energy to auto-ignite the lean mixture at the end of the compression stroke. In addition, UHC and CO emissions also increase at part load operation due to insufficient combustion efficiency. Furthermore, the high load operation is typically limited by very high rate of pressure rise during combustion and therefore resulting engine knock.

Cold start capacity

The HCCI ignition is very sensitive to the intake charge temperature and the small variations change the combustion phasing considerably. Furthermore, the initial temperature required to obtain auto-ignition condition changes with fuel properties and the operating conditions. HCCI engine will face a major problem in firing during cold start operations, as the temperatures are very low and the heat transfer to the cold combustion chamber walls is high. This problem can be overcome by starting the engine in a conventional mode and then switching over to the HCCI mode after a short warm-up period.

Homogeneous mixture preparation

Effective mixture preparation and avoiding fuel-wall wetting is the key to obtain high fuel efficiency, reduce HC and PM emissions and prevent oil dilution. Even for moderately volatile fuels like gasoline, wall wetting may adversely affect the HC emissions. This is specifically important for the poor volatile fuels like diesel. Mixture homogeneity affects the auto-ignition reactions, which control the combustion phasing. NOx emissions have found to be lower, even if there is some degree of inhomogeneity in the mixture

Abnormal pressure rise with noise

In HCCI mode, due to simultaneous auto-ignition of the whole homogeneous charge in the compression stroke, the heat release is instantaneous, which results in sudden rise in temperature followed by the abrupt rise in pressure leading to high levels of noise. Furthermore, at higher loads the rate of pressure rise can be so high that it may increase the engine noise considerably. If this condition is allowed to continue, then it may cause severe damage to the engine

Prompt response of cycle transient

HCCI suffers with a real-time, fast-response control system to tackle the challenges of maintaining required ignition timing during the transient operation, in which the engine speed and load fluctuates rapidly

https://www.youtube.com/watch?v=9nCr3h0-eV8&ab_channel=Ansys

https://www.youtube.com/watch?v=PT2Mt-tkJ_4&ab_channel=AlexonAutos

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